THE

MECHANICAL ENGINEERING

OF

STEAM POWER PLANTS

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PREFACE

The Power Plant lies at the basis of the comfort and the life of a modern community, and has an interest for nearly every one. There are, however, six groups whom it concerns specially.

The first is the general public, who are supplied with light and power from it, for street transportation, in business and domestic economy, and in the railway, steamer, and motor vehicle; in the waterworks and the factory. To such, the one requisite is reliability; the stoppage or shutdown of the plant must not invade their comfort or convenience in activities or pleasures, in their wage-earning or their safety.

The second group is the business man of the investor or stock-holding class, who has put capital into the power plant, or into the factory which rests upon it. His object is the sale of his product, either in the form of energy or horse power to users of power, or to the other departments of the works which it serves. He also is mainly concerned with the reliability of the plant, both because a stoppage means a loss of income directly, or a future loss in good-will towards his investment; but in addition he must demand that the design and combinations in the plant are in accordance with advanced applications of science in order that efficiency and economy shall be attained, and these to the degree that he can still produce at a profit when the market price of his power commodity shall be low. He will not be a designer himself, but will get his designing from others whom he pays.

The third class will be the power-house operators, or engineers-in-charge. Their interests are the same as those of the investors as respects reliability and economy, with the added one of convenience, since these must stand by and maintain the efficiency of the plant as the object of their daily toil. They will not be designers or creators as a rule, but are often charged with the responsibility of critical selection from competing apparatus, and to do the duty of the consulting engineer for the second class. Of all the classes, these perhaps have the keenest personal interest and touch with the problem.

The fourth class may be called the contractors. These are not primarily users of the plant, but are business men who make a specialty of the design, erection, and sale of complete installations. They do not usually create the details which they combine into a harmonious whole, although they may manufacture certain elements. They will have engineers of the highest grade working with and under them, as coordinators of design, and may be engineers themselves. In this class are district managers of large producing concerns, certain sales-managers and agents. Their interests are in reliability, economy, and acceptability as in the previous group, since the continuity of their business is involved in satisfying their customers and the consequent reputation. The consulting engineers in this fourth class will be representatives of the seller of the plant.

In a fifth class will be the consulting engineers representing the buyers. These will rarely be creative designers of detail, but as office practitioners will write specifications and draw general plans, for the contractor class or productive manufacturers to meet. Their interests and requirements are the same as with the contractors, and for the same reasons.

Finally, in the sixth class are the creative designers of the details in the plant, who are usually also producers or manufacturers, or who are employed by such as engineers or draftsmen. They are required to meet all the foregoing demands, and in addition to be consulting experts scientifically and professionally, and both as scientific men and shop executives. The demand for such men is continually narrowing the scope of the fifth group; and the commercial effectiveness of business done by the fourth group is as steadily broadening their field, and bringing the producer, who is also the creator through the engineers in his employ, into the first rank and securing increasing consideration for him.

The student of engineering may become a member of any group of the foregoing. The broadest field for him on graduation and where he is best paid is in the growing fourth group. His study of the power plant may therefore wisely be directed to fit him for usefulness in that class, leaving his later specialization in design or creation to be the result of circumstances in his life.

The view-points for power-house problems are two. The one is the functional requirement: what is to be accomplished, and what apparatus is needed to do this work. The other is quantitative: how large must the apparatus be to do a certain quantity of such required work. Plainly the first three groups are concerned only with the first; and only the engineers of the fourth group with the second. The engineers and designers of the sixth class are mainly concerned with proportions or quantities; but they also must begin the study of their specialty at the functional end, which will be the basis for quantitative design in any case. Hence the object of this book is largely the study of function and purpose of power-plant apparatus. It is not to be regarded as a treatment of design of the elements of the power plant, but only of the power plant as an aggregate of elements. To have included machine design would have rendered its real contents hard to get at in the multiplicity of topics, and would have made the conditions prohibitory as respects size and price. Furthermore, power-plant design should be taken up with a more mature knowledge than is supposed to belong to the beginner, and should cover many matters difficult to treat fairly in a text-book. The last chapter is intended as an open door to some of these questions, and for more advanced students.

The basis of theoretical mechanics under some elements of power-plant practice has been given in smaller type than the main trend, so that those to whom it is of secondary interest may skip it without sacrifice of continuity.

A former edition of this book, issued in 1897, embodied the study and experience of the author gathered during the previous twenty years and brought together for teaching purposes. The years since then have been a period of great and rapid progress in the power plant and in all engineering departments contributory thereto; and while the old edition was modernized here and there and year by year, the time had come with the opening decade of the twentieth century that it be rewritten entirely. The present edition is the result of such rewriting.

It is a new book so much enlarged that the old plates could not be used, but the size of page has been increased, new illustrations chosen, and many new topics and treatments have been introduced. While the former approved analytical view-point is retained and amplified, there has also been introduced a discussion in many chapters of the principles and data of applied mechanics attaching to the subject in hand. This has been done to enable teachers who desire to enliven the drill in the mathematical classes to find practical problems and applications of interest and future meaning, and to encourage teachers of the applications of theory to find easily the links and bases for such sound applications. The distinction between the applied thermal principles and those derivable from other departments of theory should tend also to clearness and benefit.

The new treatments which are specially noteworthy are those of the analysis of the power plant and its diagram, and the separation between the simple and the complex phases of this problem; the treatment of the steam pipe as an element of co-ordinate importance in the plant with the boiler and the engine; the chapters on the auxiliaries as distinguished from the essentials; the steam turbine chapter, the engine mechanism chapter, and the establishment of the philosophy of the expansion of the elastic medium as the basis for the valve gear, the governor, the condensing and the compound engine. This is new, and it is believed that it will be helpful and illuminating. Some data and tables have been introduced, but only sparingly. The author prefers that students should acquire the habit of going to the "Engineers' Pocket Books" for statistical or quantitative information, using this book to give a perspective or setting to make clear the meaning and interpretation of such data and constants as these excellent books are prepared to furnish.

The book has been written with other objects as well as for its use as a text-book in schools of engineering. But the recognition of the basal and fundamental importance of the power plant in all kinds of engineering enterprises has necessarily given this possible use a great share of consideration. It may not be amiss therefore to suggest some methods for such use which have been tried out in the author's experience, and have been before his mind in preparing the text.

1. The primary purpose of the educational process should be to make the learner *think*. While it is true that knowledge is power, such knowledge and power are only really possessed for any time by him whose faculties have exercised themselves in the process of deducing principle from fact. Hence the recitation process is far to be preferred to the lecture method of instruction in such a subject as this.

2. The recitation method at the blackboard by sketch or upon the drawing-pad is far the best. The sketch cannot be vague or indefinite, and hence the knowledge which it reveals must be equally precise and definite. This in itself has an educational value.

3. The reproduction of the illustration in outline should be so done that the principle is laid bare, and the accidental or accessory detail is relegated to second place or omitted. The illustrations have been chosen to favor this procedure; and old cuts have been retained even when newer practice differs from the old, where the principle was clearer in the old than it would have been in the new. This explains

further the copiousness of illustration; it justifies the use of figures from commercial sources as giving living and present interest to what might else be dry and unattractive.

4. The instructor should realize that in any class the number is small who will be called on to design and construct any detail of the power plant; but that all are sure to be called on to *buy* some or all of these elements, or to design power plants as a whole, in which such elements are to function. Hence the training in the process of selection, of critically weighing arguments for and against an apparatus, and evaluating advantages and disadvantages, becomes a most valuable discipline. The treatments designed to this end should be diligently so used.

5. It would be impossible to make the treatment and particularly the illustrations cover every embodiment of principle which is now in use. The book would exceed the intelligent limits of size and price with even current practice; and the progress of science, invention, and design is continually bringing new material. The instructor is urged to bring such new illustrations into class in the form of photographs, lantern-slides, trade-catalog literature, and the articles from the technical press and society proceedings; and to use them in the same way as in Nos. 2, 3, and 4 above in supplement to what the text covers. The country is so large that standard practice in one section may not be well known in another, but a broad view requires consideration of all practical solutions.

6. For the more advanced student whose instruction should be more personal, and in seminars or conferences, there are no better text-books to suggest topics for discussion than the technical newspapers and magazines. These offer up-to-date and live issues, new problems and their solutions, accidents and failures, queries and opinion. But to utilize such material by partly trained students or those having limited experience only, recourse should be had to fundamental principles, accepted standards, and the teachings of the past and of experience. For such elements the treatments of this book may be conveniently used, and such advanced students will find more material upon a re-study than they found in the first contact with the text.

7. It is an admirable drill in the thought-compelling process and for its impress on the student, to compel him to formulate the questions upon the sketch or illustration. This can be done most effectively by the lantern-slide, exhibited by an electric projection lantern to a whole class in a room not wholly darkened. If one student can ask the questions for another to answer, not only is the answerer forced to think, but the questioner even more. The instructor can reach the

perspective and view-point of the class for a guiding and stimulation of the discussion and for an effective summing up.

8. Finally, there is no engineering literature of greater teaching value than the specifications which have been used in successful practice. To use and digest these, however, a certain previous knowledge is requisite. If the student can have these specifications given to him, and be compelled to defend the requirements therein set forth on the basis of discussions in such a text-book as this, he receives an unequaled training for his future work. Advanced students can be drilled with specifications in evaluating alternate requirements, on the basis of what they can find in the fundamental treatments in the text.

The critical student of power-house problems may propound two questions: Is not the development of the steam turbine to render valueless a treatise of this sort in a year or two, in so far as it relates to the reciprocating engine? and will not the spread and development of the internal combustion motor or gas engine render valueless the space devoted to both boiler and steam engine?

There are several categories to the answer to the first question:

1. The turbine will not displace the reciprocating engine in cases where the starting resistance is high at necessarily low speed of turning.

2. The turbine is not so much better in economy or convenience than a good economical reciprocating engine that owners and users are going to displace the latter. The turbine may be installed for new developments, but the reciprocating engines will run for years yet.

3. The turbine is at its best for electrical transmission of the power: where some other medium is better the reciprocating engine will remain.

4. The turbine is at its best for large units; for small and medium conditions the reciprocating engine still has a field of its own.

5. Everything in this treatise concerning the boiler, the piping, and the auxiliaries remains untouched by such a change if it occurs. Perhaps the auxiliaries receive enhanced importance in a turbine plant, and what attaches to the details of the main motor herein will be equally true for the greater part of the enlarged auxiliaries.

6. The turbine will not displace the reciprocating engine where the resistance must be slow-moving, calling for a gearing down of the speed of the motor element.

7. The most complete disappearance of the reciprocating steam engine will occur, if at all, when all industry is carried on by transmitting electrical energy distributed from central generating stations. The driving engines in these central stations will be turbines or gas engines,

and there will be no isolated or independent steam engines at all, in the plants of individual parties.

The second question, concerning the vogue of the gas engine, is also one which is not to be answered by a single sentence.

1. There is no doubt that enormous increase is to be foreseen in the use of the small internal combustion engine using carbureted air from liquid fuel. This will be in motor boats, in motor vehicles, in railway branch service, and in stationary practice, and it will displace the small steam engine and be used in places where the reciprocating engine will not go.

2. But for large installations, the cost of liquid fuel compared with coal becomes prohibitory under economic conditions, and the gas producer must be provided. As soon as this happens, many advantages of the gas engine begin to balance up with those of the steam plant, although the avoidance of the auxiliaries of the steam plant leaves a heavy balance in favor of the other system.

3. But when the largest size of generating unit is called for, the big gas engine meets its limiting condition, just when the steam engine is reaching its best. The problem of cooling the big engine with a large diameter cylinder, and disposing of the great quantity of heat released in the combustion of masses of gas, introduces difficulties with the gas engine which are not yet solved, and perhaps are not going to be. The big gas engine becomes costly, bulky, and occupies great space. Repairs and maintenance mount up; auxiliaries for starting appear, and the advantages of the big steam plant begin to emerge anew.

4. Finally, in any industry where the exhaust from the steam engine is required in the industrial process for heating solutions, or for drying as in the paper and textile industries, or for the heating of buildings in winter as in the office-buildings of the congested cities, the steam engine as source of needed power will have the balance in its favor.

The gas turbine, if it becomes realizable by discovery of new laws or methods to avoid the prohibitive action of known old ones, will of course revolutionize many departments. But until these steps are taken there will be some time during which the steam engine will justify the study this book encourages and provides for.

The writer of to-day is under profound obligations to those who have preceded him, and this edition in particular owes much to writers whose work has appeared since the earlier publication was made. Besides repeating the acknowledgments of the earlier edition, special recognition should be made for valuable co-operation from Mr. Charles L. Hubbard,

author of "Power Heating and Ventilation;" from Mr. Lester G. French and Prof. C. C. Thomas in the discussion of the steam turbine; from Mr. Henry C. Meyer, author of "Power Plants," and to their publishers for permission to make use of cuts in convenient form. The help of manufacturers and designers is also gratefully acknowledged in the matter of illustrations and special designs of such producers. Prof. John Goodman of Leeds, and Prof. John Perry of London, England, have been strong influences upon the author in his more mathematical discussions of topics, and he is glad to recognize these obligations. He would heartily thank those instructors and other users of the text-book for their appreciative comment and helpful criticism.

Attention should finally be called to the re-arrangement of topics, whereby the boiler and its accessories has been made to precede the engine and its problems. This will be found to offer a teaching advantage in that the phenomena of the boiler and combustion belong to the simpler problems of chemistry, physics, and mechanics, and their study can begin early in a progressive course. The moving engine using transformed heat in the form of pressure and changing pressure into work compels an outlook upon the principles and data of thermodynamics, and is conveniently placed later where greater maturity and a wider range of study will permit the theory and the practice to be more effectively co-ordinated than with the older arrangement upon a different logical basis. Hence also much of the book can be made to precede the study and practice of the practical or experimental engineering laboratories of the schools, as well as courses in engineering design on the drawing board; and with manifest advantage.

It is the wish and hope of the author that the material here brought together and the methods of treatment may be found serviceable in the work of engineering education to which he has given so many happy years.

F. R. HUTTON.

New York City,
September, 1908.

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THE MECHANICAL ENGINEERING OF STEAM POWER PLANTS.

PART I. INTRODUCTORY.

1. The Function of the Power Plant. A power plant is an installation of the necessary devices and machinery to make available a supply of mechanical energy or to generate or liberate such energy from the material in which it is stored. Such installation will cost money, and the necessary capital will, therefore, be invested or planted at the selected point with a view to a financial return from the sale of the power. In some cases the owner of the power plant and the consuming user of its product are the same parties; in others, the power is sold outside to others. The accounting or financing should be able in either case to give the exact cost of the unit of power to the producer in dollars per year, that the plant may be self-supporting and yield a profit on the capital invested. Hence the function of the power plant is to furnish the maximum power at the least cost, such cost being made up of the elements of interest on the investment, depreciation, and maintenance, together with the aggregate of the operating expenses properly so called.

There will therefore be usually two differing standards to be borne in mind. One is that where the first or investment and the up-keep costs are to be kept low, with a view to short life of the plant and its sale at a sacrifice when its function is discharged. The other is the larger and more permanent plant, with an expected longer or perhaps continuous life during the existence of the same industrial or civic conditions. In the former the first cost is less regarded because usually smaller in amount: in the latter, large investment is justifiable to save fuel expense and diminish the repair and maintenance expenditure to keep the plant in prime condition for economic production of power. Solutions of engineering problems in the power plant have, therefore, always the financial factors present in them, since nearly every technical difficulty can be overcome if it is worth the cost in dollars to solve it.

2. Sources of Motor Energy for the Power Plant. Internal and External Combustion. In a broad and general view of power generation there are three origins for such power. (1) Muscular force of men or animals; (2) Forces resulting from chemical combinations, giving heat or electrical energy; (3) Gravity, or the force by which the earth draws toward its center the masses which are capable of such motion. It will be obvious that the limitations on muscular force imposed by the vital energies both in extent and continuity preclude anything like large installations dependent on it: and the expense of generating electric energy from battery reactions is equally conclusive. Gravity and heat energy from fuel are the two practical sources of power.

To utilize gravity as a source of motive power, masses must be lifted away from the center of the earth and then be allowed to descend towards it, doing work in such descent. The only usable masses which are lifted by natural forces and without expenditure of costly power are those of the water and air which are raised by the heat of the sun to the upper levels of the atmosphere. The descent of the colder air in adjustment of barometric pressures causes the winds which are used in air motors such as wind-mills; but the small power and lack of continuity of their motion restricts them to special classes of service, such as pumping or intermittent electric generation where storage devices can be made available. With water-power utilization, there must be high land or mountains available, upon which the water may fall and be accumulated in water-courses, natural or artificial, and be led in quantities to the motor to be driven. Where such elevations are not available, and the head of water is only that of slow moving streams, the motor becomes of inconvenient size and cost. Hence the liberation of heat energy from fuel is the most important and generally significant of all the systems of manufacturing mechanical energy.

This liberation of stored energy from a fuel in the form of heat, and its transformation by natural law into mechanical energy may be done in two general ways, or by the use of two media. In either system, the fuel must be burned or oxidized. This means a chemical union of its heat-producing elements with the oxygen of the air in proper proportions, so that the heat may be released or made available in that chemical process. The research of Joule showed that such release of heat gives 778 foot-pounds of work for each unit of heat available in the fuel if the transformation into mechanical energy is complete. This released heat may be used in the one system to heat the air which supports its combustion within the space enclosed behind a piston in a cylinder and by means of the pressure resulting in that air from its effort to expand by heat, the effort is exerted upon the piston. Such motors as operate with heated air upon this system are called internal combustion motors. The fuel may come into the cylinder in the form of gas or liquid, and the engine will be called a gas engine, or a kerosene, or gasoline, or alcohol engine. Power plants working upon this system form a class by themselves and have been elsewhere treated by the author.* The second system causes the heat of the combustion to impart pressure to a medium contained within an enclosed vessel to which the heat-source is external. The pressure resulting in the vessel from the expansion of the medium by heat is led to a cylinder within which is a piston, or is exerted by the living force due to its velocity against vanes, doing work as in the first case. This is called the external combustion system, and steam or the vapor of water is the most convenient and accessible medium. It costs nothing but the interest on the investment to bring the water in, or the water-tax to the municipality to compensate the latter for its expenditure for the same object. Other media such as ammonia compounds, ethers, alcohols and petroleum derivatives have been proposed in order to avail of their greater volatility, and to escape the objections to water from its high pressures with low temperature and its readiness to condense in the motor cylinder and in the piping. This again is offset by the high specific heat of water and the fact that for this reason a given cylinder volume brings in more heat than with a less dense medium; or a smaller cylinder with steam does the same work as a larger one supplied with a more volatile medium.† This treatise concerns itself, therefore, only with the external combustion system and the steam engine and boiler as means to carry it out in practice.

3. Measurements of the Work Unit of Output. Indicated Horsepower. Brake Horsepower. When the steam engine replaced the horse motor it was convenient to standardize the new power in terms of the old. James Watt established by experiment with powerful British or Norman draft horses that their capacity was a work of 33,000 foot-pounds per minute. In other units than the foot and the pound its equivalents are:

* The Gas Engine, a treatise on the Internal Combustion Engine using gas, gasoline, kerosene, alcohol, or other hydro-carbon as source of energy.

† See Heat and Heat Engines, by F. R. Hutton, Chapters on Vapors as Heat Carriers and Vapor Engines. John Wiley & Sons.

Horsepower.	English. Foot-pounds per Minute.	French. Kilogrammeter per Minute.	Austrian. Foot-pounds per Minute.
English and American	33,000	4,572.9	25,774
French	32,470.4	4,500	24,561
Austrian	33,034.2	4,549.5	25,800

The English horsepower is 1.0163 force de cheval.
The French force de cheval is 0.95363 English horsepower.

If the power is to be transmitted in the form of electrical energy over wires the unit is the kilowatt, which is energy at the rate of 1000 joules per second, the joule being the energy expended by the international ampere against an international ohm. The watt is equivalent to $\frac{1}{746}$ of a horsepower, or the kilowatt per hour is equivalent to 1.34 horsepower per hour.

Brake horsepower, or effective horsepower, is the net work in foot-pounds or other units delivered from the engine or motor after all losses from its own friction have been deducted. It is measured by a brake or power absorbing or measuring device and is, therefore, also called dynamometer horsepower or net horsepower.

Indicated horsepower is the capacity of the engine measured at the cylinder by an instrument devised by James Watt which he called the indicator. This is an apparatus in which a piston of known area receives the pressure of the steam at the same time as the engine piston. Its motion is opposed by a calibrated spring which is compressed or extended proportionately to the pressure. If the end of the rod attached to the piston and resisting spring is fitted with a pencil or tracing point, it can be made to indicate at every point in the traverse of the engine piston the pressure which was acting on it: a mean of these pressure ordinates upon a traced diagram will give the mean pressure, which, multiplied into the area of the engine piston, and by the feet traversed in the minute of time will give the horsepower supplied to the cylinder in that time. This mean pressure can also be mathematically computed from formulæ of sufficient accuracy.

The Nominal Horsepower is an old term now properly disused, which was based on an untenable assumption that all engines of a given diameter and stroke (or cylinder volume) were of the same horsepower, whatever the mean pressure on the piston or the speed of its traverse

4. Elements and Analysis of the Steam Power Plant. The simplest possible case of a power plant reduced to its lowest terms must include a grate to burn coal and a chimney to make draft and carry off smoke and burnt gases. In contact with the heat of this fire and its hot gases there must be a metallic vessel to hold water to be boiled and made into steam. The vessel must be strong enough to resist the pressure. There must be a pump to force water into this boiler against the pressure, and a pipe with a valve in it to carry steam over to the engine. There must be the engine, at whose revolving shaft the continuous effort appears in available form to do work. Referring to the diagram sketch of Fig. 1 and the analysis of Fig. 2, the coal from the pile, No. 2 and 3, is thrown upon the furnace grate, No. 6. Air enters as No. 4 to burn the coal, and the gases and smoke pass off as No. 9 through the chimney. The utilized heat, No. 11, reaches the boiler, No. 16. The feed water, No. 18, goes through the feed pump to the boiler and is there evaporated into steam by the heat. The steam passes through the steam pipe and past the control of the throttle-valve to the steam engine, No. 34, and does the useful work at its shaft,

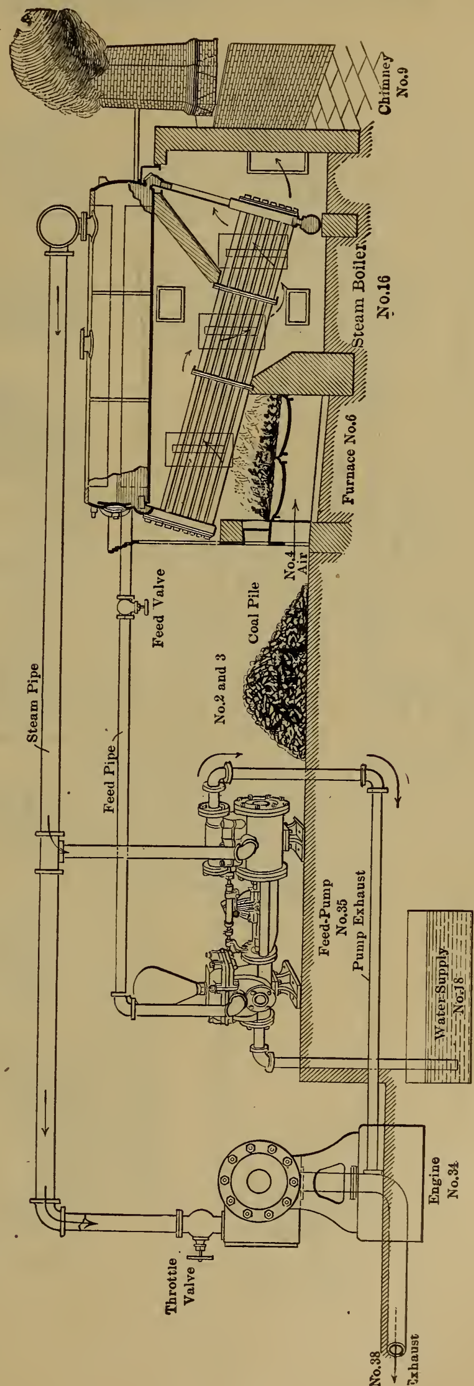
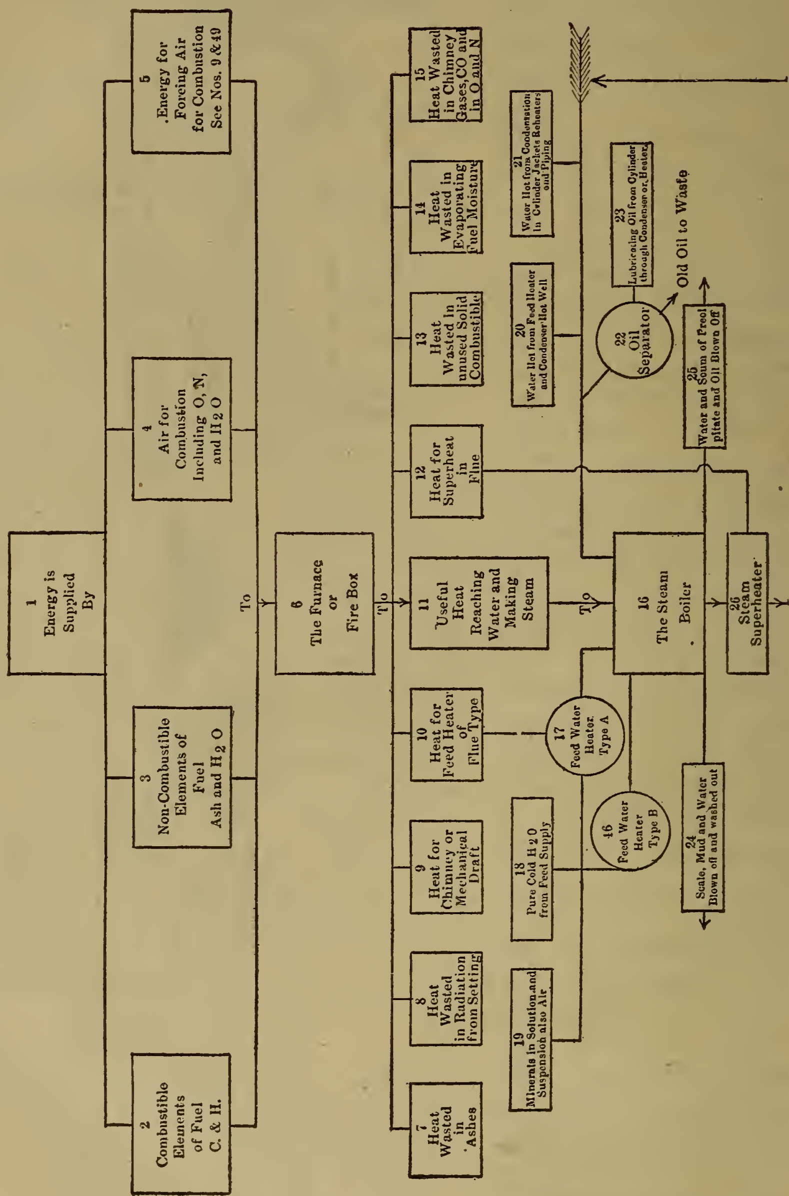
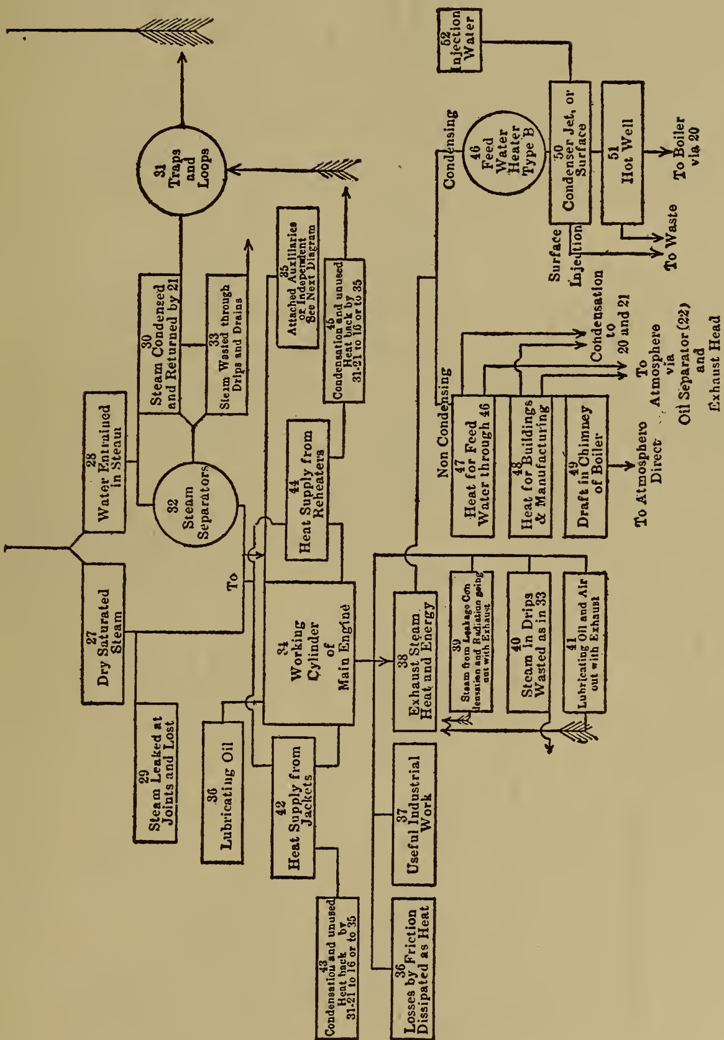


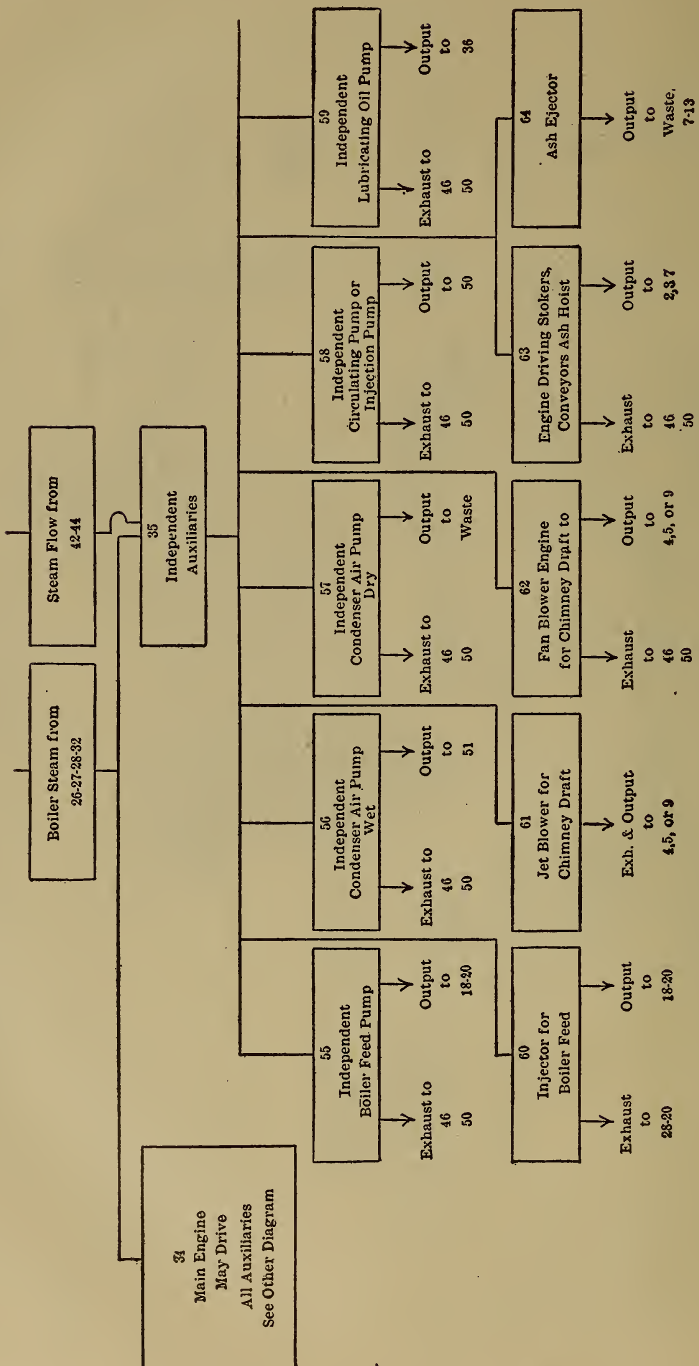
FIG. 1.


1 Energy is Supplied By
2 Combustible Elements of Fuel C. & H.
3 Non-Combustible Elements of Fuel Ash and H2 O
4 Air for Combustion Including O, N, and H2 O
5 Energy for Forcing Air for Combustion See Nos. 9 & 49
To
6 The Furnace or Fire Box
To
7 Heat Wasted in Ashes
8 Heat Wasted in Radiation from Setting
9 Heat for Chimney or Mechanical Draft
10 Heat for Feed Heater of Flue Type
11 Useful Heat Reaching Water and Making Steam
12 Heat for Superheat in Flue
13 Heat Wasted in unused Solid Combustible
14 Heat Wasted in Evaporating Fuel Moisture
15 Heat Wasted in Chimney Gases, CO and in O and N
To
16 The Steam Boiler
17 Feed Water Heater Type A
46 Feed Water Heater Type B
18 Pure Cold H2 O from Feed Supply
19 Minerals in Solution and Suspension also Air
20 Water Hot from Feed Heater and Condenser Hot Well
21 Water Hot from Condensation in Cylinder Jackets Reheaters and Piping
22 Oil Separator
23 Lubricating Oil from Cylinder through Condenser or Heater
Old Oil to Waste
24 Scale, Mud and Water Blown off and washed out
25 Water and Scum of Precipitate and Oil Blown Off
26 Steam Superheater



I. Analysis of a Power Plant as respects Energy, Materials, and Apparatus.

Fig. 2.



II. Analysis of Power Plant. Auxiliaries Independently Driven.

Fig. 3.

No. 37. The steam cooled down by doing work and reduced in pressure goes off through the exhaust pipe as No. 38. But from Fig. 2 and Fig. 3 it is apparent that a power plant of any completeness can have many more elements or units than the few listed in the fundamental form, and that some of these will be introduced to secure economy of fuel and more perfect utilization of heat; others will tend to greater mechanical or plant efficiency; and others again will be for safety. Furthermore there will be a wide range for critical selection among the elements which are arbitrary or where alternative forms may be offered. Hence the field which this treatise will aim to cover may be divided as follows:

I. Introductory and General.
II. The Boiler and its Accessories.
III. The Boiler Plant Auxiliaries.
IV. The Steam Piping and its Accessories.
V. The Engine and its Accessories.
VI. The Engine Room Auxiliaries.
VII. The Operation and Testing of the Engine.
VIII. General Considerations.
IX. Historical and other Appendices.

CHAPTER I.

THE QUANTITATIVE BASIS OF THE STEAM POWER PLANT.

5. Introductory. In every engineering problem involving the application of physical law, there are two distinct lines to be followed. The one is the functional study, concerned with the working of the machinery in practice: what it does and how. The other is the quantitative study, concerned with the necessary sizes of the organs or elements to do a given amount of work.

The study in the following chapters is to be the functional research, in answer to the question, "How does each element work?" But to give definiteness and meaning to such study it will be worth while to glance briefly over the field from above and note some few quantities which enter into the problem of design, and which form, therefore, the quantitative basis of such a power plant.

6. The Horsepower of a Piston Motor. The Cylinder Volume. The problem in industry always imposes a certain resistance to be overcome, and the effort of the motor must be powerful enough to exert the required force through the required space in the given time. If the unit of resistance is in pounds and the unit of space is in feet, the work from the motor will be the doing of a certain number of foot-pounds in a given time: or, if both members of the equation be divided by 33,000 (paragraf 3),

$$\frac{\text{Work of motor in foot-pounds per minute}}{33{,}000}$$
$$=\frac{\text{resistance in foot-pounds per minute}}{33{,}000}.$$

or, Horsepower of motor = horsepower of resistance.

The first term must be enough larger than the net resistance of the second term to overcome all friction and inertia and hurtful resistances. If the motor have the fundamental design of a piston traversing a cylinder it will be apparent that the pressure below the piston in pounds per square inch of area must be sufficient to lift the weight W in pounds: and to move this weight through the space in feet of its traverse at the speed specified (Fig. 4). If the pressure be

P pounds per square inch, and the area be A square inches, the effort on the piston to lift the weight W is PA pounds. If the traverse in feet be L, then the work of one traverse is PAL, which balances the resistance work of the weight WL. If the traverse be made N times in one minute instead of only once, the total traverse is LN feet per minute, and the work of that minute is

$$PALN = \text{Work in foot-pounds done in one minute,}$$

or the horsepower will be found by dividing both members by 33,000, so that

$$HP = \frac{PLAN}{33{,}000}.$$

If P were not the same in all traverses, or were not constant in any one traverse of the piston, no error would be made in using the mean value during the stroke or during the minute. In a crank-engine, the piston traverses L twice in one complete revolution, so that if a revolution counter is used instead of a stroke counter the quantity N must be twice the number of complete revolutions.

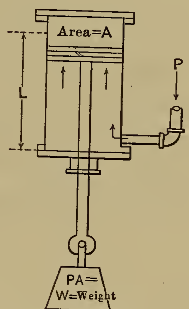


FIG. 4.

The quantity L is in feet, and the quantity A is in square inches: if LA be divided by 144 the quotient is the volume of the cylinder swept through by the piston in cubic feet. But if the factor 144 is introduced into the denominator, the same factor must be introduced into the numerator to keep the equation true: hence the pressure per square inch, being multiplied by 144, becomes the pressure per square foot, or

$$H.P = \frac{\text{Pressure in lbs. per sq. ft.} \times \text{cylinder volume in cubic ft.} \times N}{33{,}000}$$

$$= \frac{P'V'N}{33{,}000}.$$

For one stroke or traverse this becomes

$$H.P. = \frac{P'V'}{33{,}000}.$$

After the cylinder has been embodied in a metal casting $\dfrac{V'}{33{,}000}$ becomes a constant for that particular engine, which, if called K, gives the expression in either units,

$$H.P. = PNK.$$

The variables therefore with varying resistance can be only P and N. If the condition be imposed as is usual that the motor makes the same number of strokes each minute or turns at constant speed, then N cannot vary, and the only variable with varying resistance is the mean pressure P.

The volume swept through by the piston, or the piston displacement, is the measure of the power of an engine, when the pressure P is given or fixed. The relations of stroke to diameter in standard designs of cylinders are fixed by convention between a stroke equal to the diameter and a stroke twice the diameter as limits, to secure the maximum volume or power with the minimum cooling surface of contact between hot steam and metal which can radiate heat away.

7. The Work Unit and Heat Unit are Convertible. The Volume of Steam per Horsepower. The unit of heat is the quantity of heat required to raise one pound of water one degree Fahrenheit at the temperature of its maximum density near 39° F. This is called the British Thermal Unit. Some authorities prefer the value at 62° F. in raising the same weight to 63°. The researches of Joule and Rowland have shown that 778 foot-pounds are the mechanical equivalent of one British thermal unit. In the metric units this is 428 kilogrammeters per degree Centigrade. Hence if the equation for work in foot-pounds be divided by 778 the quotient will be the equivalent heat units; or

$$\text{Heat units} = \frac{PLAN}{778}$$

when the factors are known or assumed. But this assumes that the conversion of heat into work is perfect and is effected without loss; which is far from the case in actual conditions. It gives the theoretical ideal for the heat which the coal should furnish to do the work in question. The relation between the actual coal burned in the plant, and the computed quantity by this calculation measures the efficiency of the plant:

$$\text{or} \quad \frac{\text{Computed coal consumption per horsepower}}{\text{Actual coal consumption per horsepower}} = \text{Heat efficiency}.$$

If in the expression for the horsepower in the form

$$H.P. = \frac{P'V'N}{33{,}000}$$

the first term be made one horsepower and N be assumed and the pressure P per square foot is the mean during the piston movement, it would appear that

$$\frac{33{,}000}{P'N} = V',$$

or the volume of dry steam at the mean pressure P' which is required per horse power in this engine. But experience shows that the actual engine requires more than this by an amount which measures the losses from condensation and other causes in the cylinder and pipes and passages. It has been called the " missing water," and is the difference between the computed and the measured water per horsepower which the engine requires. When the computed volume has been corrected by this coefficient of condensation loss, then the weight of water per horsepower per hour can be ascertained from any reliable table of the properties of steam. (See Appendix, paragraf 566.)* The line for the observed pressure being found, there will be columns for the weight of the cubic foot at that pressure; the weight of one cubic foot multiplied by the volume V' in cubic feet per minute and this again by 60 minutes for the hour will give the pounds of water required by such a cylinder volume per hour.

8. The Water Rate of a Steam Engine. The pounds of water to be furnished to an engine cylinder per horsepower per hour form a unit which is called the " water rate " of this engine. Since this water is evaporated into steam by heat and this heat is obtained from coal which is paid for in dollars per ton, it is a means of reducing the operating cost of the plant or the cost of a horsepower to lower this rate. Many of the elements listed in the analysis of Fig. 2 are introduced for this purpose: such are the superheater, No. 26, the jackets, No. 42, the reheaters, No. 44, the condenser, No. 50, the types of complicated valve-gear and the use of the compound engine principle, as well as a number of the auxiliaries. This diminished water rate is secured by increased expenditure in first cost, a more complicated plant, and an expenditure of energy to operate it. But the net result is a gain where the coal consumption is large; or to put it otherwise, the gain is the algebraic sum of a loss made to effect the gain and the gain so secured. It becomes unsafe, therefore, to say what may be the water rate of an engine unless the factors are known which cause the rate to vary. In general terms, however, and over the broad range of practice, the probable water rate of engines in good mechanical condition, with tight pistons and valves and well lubricated, would be not far from the values in the accompanying table. The size and capacity of the feed pump of Fig. 1 will be determined by the water rate of the engine.

The reasons which lie back of some of these diminished water rates will be developed to some degree in the later chapters, or belong to an

* Consult also Hutton, "Heat and Heat Engines," p. 206. Peabody, "Steam Tables." Kent, Mechanical Engineer's Pocket Book, p. 663. Suplee, Mechanical Engineer's Pocket Book.

advanced study of the principles of the heat engine appropriate to the specialist and beyond the scope of this treatise. The record and reference here is to make it apparent why the simpler elements of the fundamental power plant enumerated in paragraf 4 became the complicated aggregate of Fig. 2.

TABLE I.

APPROXIMATE WATER RATES.

Type of Engine.	Water Rate in Pounds per Horsepower per Hour.	
	Probable Limits.	Mean Assumable.
Triple-expansion, low-speed, condensing..	18 to 12	16
Triple-expansion, high-speed, condensing.	23 to 14	17
Compound, low-speed, condensing........	20 to 12	18
Compound, high-speed, condensing.......	24 to 16	20
Compound, high-speed, non-condensing...	30 to 22	26
Large size simple, low-speed, condensing .	24 to 18	20
Large size simple, low-speed, non-condensing..................................	32 to 24	29
Medium size simple, high-speed, condensing	25 to 19	22
Medium size simple, high-speed, non-condensing.................................	35 to 26	33
Small, simple, condensing...............	40 to 30	35
Small, simple, non-condensing	60 to 50	55
Pumps without fly-wheel, non-condensing	140 to 100	125
Rotary steam engines, non-condensing....	100	100
Small steam turbines, condensing	60 to 40	50
Large steam turbines, condensing........	20 to 12	15

NOTE. It may easily happen that variations in boiler pressure, expansion, valve gear and speed will produce greater variations in water rate in any particular case than the variation in type as listed on the lines of classification above. The use of the average figure is therefore dangerous in inexperienced hands.

9. The Heating Surface of a Steam Boiler. Transfer of Heat. The heat liberated in the chemical union of oxygen with the carbon and hydrogen of the fuel in the boiler furnace is expended in heating up the fuel itself and the air which supports the combustion, and the hot gases which result from the process. This heat must be absorbed by a transfer to the metallic plates of the boiler shell and to the water and steam. Heat is transferred from a hotter to a cooler body by radiation from the solid matter of the fire and the glowing solid matter in the flame; it is transferred by contact from the invisible hot gases which have no radiating effect. There will be some transfer to the masses of brick in a brick-setting, and some transfer to the external air from the boiler itself by radiation and contact (Fig. 2, No. 8); there will be some heat potential carried off in the flues and chimney unused because unabsorbed (Fig. 2, No. 15), and some energy will be used to carry the burnt gases away (Fig. 2, Nos. 9 and 49). There are also losses Nos. 7, 13 and 14, which are hardly recoverable. But the adequacy of the absorbing metal surface of the boiler or heater (Nos. 10 and 11) must be depended on to reduce No. 15. (See paragraf 144.)

Transfer of heat is effective in proportion to the difference of temperature between the hot and the cooler body; hence the heating surface of the boiler which is the cooling surface as respects the fire, flame and gas should have water on one side of it while exposed to the heat on the other. Heating surface is therefore the area of metallic plate of the boiler shell which has water in contact with it on the one side and is exposed to radiation from fire and flame and to contact with hot gases upon the other. If a part of the heating surface becomes uncovered by lack of water, it becomes as nearly red-hot as possible, and is not only ineffective (since the specific heat of steam is less than half that of water), but by its softening by the heat and its oxidation and the sudden contraction when it cools, it becomes a source of danger.

The number of square feet of heating surface to supply a required weight of water evaporated into dry steam to supply a cylinder of a given capacity is a matter of test and experience. The accepted figure is to give one square foot of heating surface for each three pounds of water to be evaporated per hour, which would result in $11\frac{1}{3}$ to 12 square feet of such heating surface to each 30–36 pounds of water to be evaporated per hour, which is the usual allowance.

10. The Horsepower of a Boiler. A. S. M. E. Standard. The term horsepower is not properly applicable to a steam boiler, except in the sense of a capacity to supply to an engine the weight of steam per hour to enable it to develop a rated horsepower. Table I shows, however, the wide range of capacity demanded as the type of engine and the scope of the auxiliaries in the plant vary. The best that can be done is to reach a convention as to the figure for water rate to be used in computations so that all may refer to the same unit. Such a unit was agreed to in 1886 in a Committee of the American Society of Mechanical Engineers and is therefore known as the A. S. M. E. Standard. The water rate of such a conventional engine was fixed at 30 pounds of water when the evaporation took place at 70 pounds pressure and with a preheated feed-water (Fig. 2, Nos. 17 or 46) at 100° F. This is equivalent to an evaporation of 34.488 pounds of water at 212° or at atmospheric pressure with the feed-water at 212° F.

The square feet of the heating surface will be usually related to the square feet of the grate surface in certain definite relations (paragraf 12).

11. The Pounds of Coal to Evaporate a Weight of Water. Every pound of fuel has a capacity to liberate and transfer on its complete combustion a certain number of units of heat. The value of this capacity is called its calorific power. It is measured experimentally by an apparatus called a fuel calorimeter.

From steam tables (paragrafs 7 and 566) it appears that to make water into steam it must first be raised as water to the boiling point at that pressure and then changed into steam at that pressure; the units of heat for each pound at each pressure are known. At atmospheric pressure, for example, 965.7 units are required to make one pound of water at 212° into steam at 212°; if the feed water is colder than 212° so many additional heat units per pound will be required.

If the transfer of heat could be complete, and the calorific power of the fuel had been ascertained, the pounds of water per pound of coal would be

$$\text{Pounds of water from and at } 212° = \frac{\text{Calorific power of fuel in heat units}}{965.7}.$$

For a fuel of 12,000 heat units (a usual value) the water evaporation would be 12.4; for the best American coals with little ash, such as those from Georges Creek, Md., or the Pocahontas veins in W. Va., the calorific power of 14,000 gives a theoretical evaporation of 15 or 14 pounds. In the actual furnace, however, neither transformation nor transfer are perfect; hence from 9 to 7 pounds of water per pound of coal are good results. See Tables II and III below. This means $\frac{30}{9} = 3.3$, or $\frac{30}{7} = 4.3$ pounds of coal per boiler horsepower per hour or per engine horsepower if its water rate is 30. If its water rate is 12 or 16 the coal per horsepower per hour becomes $\frac{12}{9} = 1.3$ or $\frac{16}{7} = 2.3$. If the evaporation rate falls, the coal rate rises, of course, to furnish a given weight of steam. When the unit becomes 1000 horsepower and the coal consumption becomes 2300 pounds, or over one ton per hour, both the water and the coal rate become financially significant, and it pays to invest money in auxiliaries to save coal, water, and labor.

12. Grate Surface to Burn a Weight of Fuel in a Given Time. Rate of Combustion. Since the weight of coal to evaporate the required weight of water is known, it follows that if the water is to be evaporated in a given time — an hour — the required coal must be completely burned in that same time. There must be a grate area of sufficient extent to accommodate the weight of fuel needed in an hour, but the rate of combustion must be so related to that area that the full amount of heat shall be liberated from that weight. This rate of combustion is usually given as the number of pounds of coal burned per square foot of grate per hour. It is rarely or never less than 10–12 for anthracite coal and 15–18 for bituminous, and this is for chimney draft. With forced draft from pressure blowers it may rise to 60 or even to 125 pounds. When so much more heat is liberated per unit of time, the fire is intensely hot, and it becomes more difficult to absorb the heat; the evaporation rate is usually less with a high combustion rate. The following tables give some relations:

TABLE II.

RATIO OF HEATING SURFACE TO GRATE SURFACE.

Kind of Boiler.	Grate Surface.	Heating Surface.	Combustion. Rate in Pounds per Square Foot Grate.
Internally fired, stationary	1	26 to 33	10 to 20
Internally fired, stationary	1	40 to 50	20 to 40
Marine boilers, internally fired	1	40 to 50	15 to 30
Marine boilers, internally fired	1	50 to 60	20 to 50
Portable boilers, internally fired	1	30 to 70	....
Locomotive boilers, internally fired	1	60 to 70	40 to 100
Water tube boilers, externally fired	1	35 to 65	15 to 25

TABLE III.

RELATIVE EVAPORATION BY SOME AMERICAN COALS.

Source.	Kind.	Calorific Power.	Pounds of Coal per Square Foot Grate.	Heating Surface to Grate Surface.	Pounds of Water per Pound of Coal at 212°.
W. Va. Pocahontas	Semi-bitum's	14,600	15	45	9.5
Youghiogheny, Pa	Bituminous	13,500	17	48	8.7
Hocking Valley, O	do	11,700	18	45	7.6
Big Muddy, Ill	do	11,700	20	50	7.6
Mt. Olive Ill	do	9,900	20	45	6.4
Lackawanna, Pa	Broken anthracite.	12,700	15	35	8.5
Do	No. 1 buckwheat.	10,700	13	32	7.5
Do	Rice, anthracite.	9,200	12	30	7.0

13. The Cost of a Horsepower. There will be two costs to be considered under this term; the one is the cost of the plant per horsepower to install, and the other is the cost per horsepower to operate when installation is complete. Obviously figures of cost must be subject to wide variations according to the standard before the mind of the computer. A cheap engine of small or medium power costs about $14 per horsepower; a more expensive engine with better valve gear and more economical performance costs about $25 per horsepower. The boiler unset to supply the cheap engine will cost about $9 per horsepower capacity; a larger boiler in its setting costs from $14 to $15. Exclusive of the buildings to house them there is a range between $23 and $40 per horsepower. With buildings included the range is from $45 to $75 per horsepower for installation.

The charge or buying and selling value for a horsepower per annum is open to the same wide limits. A full and complete computation on the basis of mill conditions in New England gave a production cost without profit added, varying from $24 to $33. A low cost when profit is charged is $35 per annum. When a large investment interest must be provided for, the price rises where coal and labor are costly to $75 per annum per horsepower. In many of the cities the large electric producers sell current at 2 to 2½ cents per kilowatt hour. If there are 10 working hours of the day for 300 days, or 3000 working hours, the 3000 kilowatt hours will cost $75 per annum, or at the rate of $56 per horsepower. Private plants claim to be making their own current at $0.85 per kilowatt hour, charging themselves no profit for dividends: this makes the yearly cost per horsepower as low as $42. These figures must be subject to frequent revision as conditions change.

14. Summary and Conclusions. It has been the purpose in the foregoing synthetic and quantitative study to give a glance over the entire field to the ultimate purpose of the plant before turning to the analytical study of details. The object of the plant is to make money or to be a successful investment. There are certain elements or units which are inevitable, or without which there would be no plant (paragraf 4 and Fig. 1); the analysis shows an apparent complication or multiplicity of detail (Fig. 2), in the study of which the reader and student might lose sight of the fundamental unity, unless this inclusive outlook had been first given him. From this point on the study is analytical, of each unit by itself. The student is expected to put the element in its place in the general scheme as being of necessity or merely desirable according to the key which these chapters should have put in his hands.

PART II.

CHAPTER II.

THE BOILER. FORMS, MATERIAL AND MANUFACTURE.

20. Introductory. The fuel introduced and burned under the boiler is the source of energy in a steam plant and therefore the furnace and its attachments should logically be the first objects of study. The capacity of the furnace to burn fuel and the capacity of the fireman or stoker to introduce it are the limits set, beyond which the plant cannot furnish power under those conditions however big the boiler itself may be. On the other hand the boiler is typically so much more massive than the furnace details and the latter are structurally adjusted to the boiler for this reason, that these causes make it convenient to take up the boiler first and its grates and furnace afterward. The boiler will be the same furthermore whatever the fuel, whether coal or gas or liquid hydrocarbon is used, while the grate or furnace will be varied.

21. The Function of the Boiler or Steam Generator. The steam boiler or pressure generator in a power plant has three functions. First, it must receive, store and control the medium which is to be used to transform the heat energy of the fuel into pressure or into mechanical energy by its expansion. Second, it is to receive and absorb the heat energy of the fire with the greatest completeness and economy or with the least unproductive waste of heat. Third, it must serve as an accumulator or equalizer between the slowly varying or continuous heat supply from the fire and the rapidly varying or intermittent demand of the engine or consumer of steam. In the steam boiler plant, the medium is water; but in the general case of an external-combustion engine (paragraf 2), it may be any convenient cheap innoxious vapor which expands readily by heat so as to produce pressure. The vessel must be closed to retain the pressure when generated and must be tight against inconvenient leakage and loss. It must not be unduly attacked by corrosion from the medium.

The generator must not be injured by the heat of the fire, and expansions and contractions due to variations of heat in the water or the fire.

The surfaces exposed to heat should be metallically clean or free from coating or incrustation outside or scale of precipitate from the water inside. The shape should be such as to receive and absorb and transmit exterior heat to the water within very rapidly, since the coal and flame do not last long, and the hot gases should be moving quickly through fire and furnace to make room for fresh oxygen as needed. The metal surfaces surrounding the water should be as thin as consistent with safety, because the thicker the plate the greater the chance for the outer layer to get further from the lower temperature of the inner layer, which is close to the water. What is desired is that the outer layer shall be as nearly as possible at the temperature of the cooler water which it encloses. Then the transfer of heat from hot gas or radiating fire and flame is most efficient, and the gas is most effectively cooled before it escapes the boiler setting and passes into the chimney. This principle lies at the base of the efficiency of small diameter boiler units.

Radiant heat from the solids of the fire and from the volume filled by a luminous flame is the most effective to heat the water, not alone as a matter of laboratory observation with luminous points, but because the transfer is not affected by interferences, and every cubic inch of radiating mass acts as if the other cubic inches did not exist. In transfer by contact from non-luminous gases on the other hand only those layers of gas which come in thin films into contact with the metal to be cooled by it do any work. Any cubic inches of hot gas which do not touch colder metal go out unreduced in temperature and ineffective for heating, since hot gas does not radiate its heat at all. This is why the flaming or semi-bituminous coals are the best steam makers (paragraf 12). This explains why retarders in boiler tubes are of any advantage (paragraf 58); it explains the effectiveness of nests of water-tubes (paragraf 80), and the necessity and economy of small fire-tubes where the draft is powerful and the velocity of the gases is high (paragrafs 56, 70)

22. The Storage of Energy in a Steam Boiler. The type and form of the steam boiler are more determined by the opinions of its designer respecting the third function of the boiler than by any one of its duties. The demand for steam for the engine cylinder is confessedly irregular, varying from the maximum capacity of the furnace to supply energy down to zero at intervals and fluctuating all the time on either side of a mean demand. The supply of heat from combustion of solid fuel is not an instantaneous process, although with gas and oil fuel it is very nearly so. But the transfer of heat to water and its volatilization into steam gas is again a process never normally instantaneous. The volume of steam gas actually on storage in the boiler at any time is not large as compared to the cylinder volume which draws from it, and a few strokes of the engine would so exhaust it if it were a permanent gas only, that the pressure would fluctuate inconveniently and fall below an admissible limit. But it must be remembered that the heat energy from the fuel is stored in the heated water and not in the steam gas. This is by reason of its high specific heat — it takes more units of heat to heat a pound of water than any known convenient and cheap natural substance — and by reason of the physical relation of the boiling point of water to its pressure.

For the steps which succeed each other in steam making are as follows. First, when the cool water is pumped into a vessel partly full of water and with a fire under it, the water is gradually warmed and expands in the process as water until the boiling point due to that pressure is reached. This boiling point is of record for every pressure in steam tables (paragraf 566), increasing as the pressure increases. Second, a pause in which no further expansion or increase of sensible temperature occurs, during which the additional heat necessary to change water at that pressure and temperature into steam at that pressure and temperature is absorbed by each pound of water. Third, when this necessary heat has been taken in, the water changes into steam gas in bubbles, through the mass and at the level surface. This is boiling or ebullition and is most active where the water is hottest by reason of most effective transfer of heat, or where the pressure is lowest, if there are any reasons why it is not uniform throughout. Such steam gas frees itself from the water by reason of the much greater weight of a volume of water than an equal volume of gas, and the layers of gas get on top of the water, pressing it down. If the boiler is closed, this volume of steam gas, being larger than the space it formerly occupied as an equal weight of water, produces an increase of pressure in the water-surface, causing, fourthly, a cessation of the bubbling or manufacture of gas until additional heat is supplied from the fire to meet the increased boiling point due to the increased pressure. Hence, fifthly, the pressure and temperature continually increase by stages until some gas is drawn off to the engine or through a relief or safety valve or the capacity of the fire to supply further heat energy is surpassed. When the boiler is not completely closed, but the throttle valve to the engine is continuously open, there is no pause for steps two and four, but the process is continuous if the fire is adequate. If the fire is too fierce, or too much energy per unit of time is entering the water, the pressure rises in spite of the demand; if the fire is weaker in supply of energy than the demand for it from the engine, the pressure falls.

The existence of the second step, which the fourth only repeats, explains why the mass of water is the storage element in the boiler. If the mass of water is large and has stored in it a great number of heat units, the water will keep on forming steam (albeit at continually but slowly reduced pressure), after the fire has been withdrawn. It will bridge over the time required to slice the fire and remove clinkers and put on fresh fuel. It enables the fire to be kept by the process of banking over night, so that a mass of highly heated water is ready for early demands, and yet the pressure may not have risen inconveniently in the interval. Where the wide fluctuation of the steam pressure is regarded as objectionable, and where the fuel supply at the same time is irregular or intermittent, large water volume must be provided. Such large masses of water, on the other hand, make slow steam-makers of such boilers, and when by rupture of the containing shell the stored energy is all released at once, there is so much force to be expended in a very short time that the most appalling destruction is likely to occur by the throwing about of large masses of metal and the disengagement of great volumes of the steam gas from such water.

In a middle class will be those boilers in which a smaller mass or weight of water is enclosed under pressure, and the surface exposed to heat and to transfer such heat is relatively large. Such are boilers in which the water space is full of tubes carrying hot gases, and dividing the water into smaller volumes by their displacement. Such boilers are rapid steam-makers, or quick-steamers. Pressure accumulates rapidly, and disappears rapidly on demand for steam. They reach a dangerous pressure quickly when not watched, with its attendant danger of rupture if it is not relieved. But the pressure falls quickly when relieved and when they rupture there is not so much water to make so much steam gas and the disaster and destruction are not so great.

In a third class are what are called flash or semiflash boilers, in which there is practically no water on storage, or very little. The heating surface — usually in coils of steel pipe — is so great that the interval required for the passage through the steps or stages of evaporation is inappreciable or too small to be measured, and water pumped into the generator appears to flash into steam-gas. Such boilers have no steady steam pressure at all, but are subject to pulsations of pressure which synchronize with the strokes of the feed pump. There can be no explosion from accumulated energy for there is none in a true flash boiler. As the excessive fluctuations are inconvenient, the semiflash type is more in use, in which there is some water present in storage, but less than in the preceding type. As soon as all water has become steam the absorption of heat becomes much slower, and the inconvenience then is mainly from the injury to the heating surface from the overheating to which it is subjected.

It is plain that to meet the storage function of the boiler, the prime requisite is sufficient strength to resist internal pressure, tending to produce rupture or excessive deformation. The problems of theory will, therefore, be those of physics as respects transfer of heat, and those of mechanics respecting stresses and resistance in the shapes and materials.

23. The Problems of Physics Respecting Transfer of Heat. The conditions to be filled for effective transfer are:

1. The difference between the temperatures of the absorbent body and that parting with its heat must be a maximum at all the stages of the process. That is, the water when it is as warm as it is going to be should then be nearest to the hottest fire or gases. If the hottest water meet only the coolest gases, the transfer as a whole process is less effective. This means generally that the currents of gas and of water should

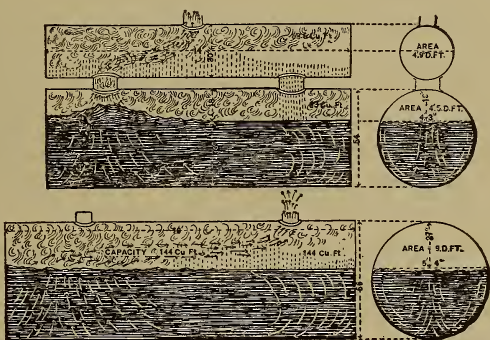


Fig. 5.

flow in opposite directions on the opposite sides of the plate, the coolest water coming in at the end furthest from the fire. This also makes it desirable that the water should "circulate" within the shell, the heavier, cooler water tending to go to the lowest part and displace upward the warmer water and that which by reason of steam gas bubbles in it is lighter than the solid water. Fig. 5 illustrates how these currents will flow in a large mass of water with steam above the water and fire below at the left hand.

2. The surface of metal between the hot element and the water which is to absorb the heat must be in the best condition to transfer this heat. That is, it must be as thin as is safe and convenient, and it must be metallically clean or free from any non-conductors of heat, especially those which would keep the water from effectively

touching the plate. The outside surface next the fire is liable to a coating of tar, or soot, or flue dust, or a mixture of all three. The inside is liable to a coating of mineral scale deposited from the water as it is boiled; to a film of oil or grease if these are not effectively separated from the feed water; and to a film of steam gas which fails to be carried up to the surface by the convection or circulation. Anything which keeps the water from the steel will permit the latter to overheat to its detriment, and impedes the transfer. Vertical surfaces or those inclined at small angles from the vertical, are least subject to such deposits, and vertical circulation is most direct and active. On the other hand, small vertical tubes of any length closed at the bottom are liable to a sluggish or indeterminate circulation. If a large steam bubble forms, it may be unable to get out except by forcing out all the water above it in the tube. By inclining the tube, the steam rises to the upper segment and follows this to the top; a descending current of water follows the lower elements of the tube. The circulation must carry away or wash off the steam gas from the heating surface, because the specific heat of the steam is less than that of the water, and the heating surface is less effectively kept down in temperature. The Field tube (Fig. 6) used in fire-engine and tug-boat boiler practice in America and in English and continental practice secures a determinate circulation by having solid cooler water descend through the inner tube while the bubbles rise against the walls of the hotter exterior tube. If circulation stops at any time, solid matter may gather at the bottom; and if it forms hard enough not to be washed out, the tube burns at the end.

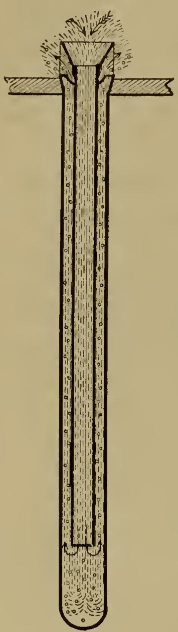


Fig. 6.

3. The disengagement of steam gas from the upper water surface must be free. If the area is small, the effort of the steam to free itself from the water is impeded when the boiler is driven. Hence unnecessary effort is required, and the bubble leaves the water explosively and entrains spattered water mechanically with it. This is called foaming or priming and not only is the steam wet and therefore wastefully used, but the water may cause breakage of cylinder castings, or bed-plate, or bend or shear or break forgings in the mechanism (paragraf 377).

4. The surfaces receiving radiated heat should be normal to such radiating volumes to receive the full effect of transfer by this method.

When the fire is horizontal or the current of flaming gas, the surface over such fire or flame is the most effective. Hence the metal should be concave downward for best effect. But this shape is ill-adapted to resist pressure from within, and has not been used for external shells since the days of very low pressures (Fig. 7). In horizontal tubes or flues the upper part meets this requirement, and the lower part is apt to be ineffective by ash deposit. When the water is inside and the fire and flame are outside and below it, the principal transfer is by contact in any event (Fig. 8).

5. The gas must remain long enough in contact with the metal for the process of progressive transfer to take place. Strong draft requires more heating surface than with gentle draft, since not only will the gases be hotter by reason of the higher combustion rate, but the velocity may compel them to reach the chimney before all possible transfer has been made.

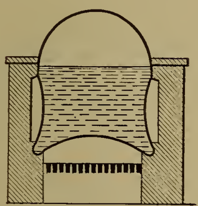


Fig. 7.

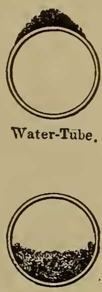


Fig. 8.

6. The gases in contact must be divided into thin films and so touch the plate, giving every cubic inch of such flowing gas a chance to give up the heat it carries. This is very effectively done with small tubes in fire-tube practice, and in water-tubular designs by having the axes of the tubes at right angles to the flow of the gas, so that the tubes act as baffles, and subdivide the currents.

7. The flaming currents of gas must not be so lowered in temperature by contact with cooler metal as to arrest the chemical processes of combustion in such flames. If the transfer under these conditions is so effective that the gases are cooled below the temperature of ignition or of combination with oxygen, the flame becomes a smoke, combustible goes off unused, and the effect of the flame is lost. This will be referred to again under smoke prevention (paragraf 129).

Of course the furnace is assumed to meet the requirements of complete combustion from the chemical point of view, supplying so much atmospheric air as will furnish the weight of oxygen required to form CO_2 gas and H_2O from the weights of carbon and hydrogen supplied per minute or per hour. This opens up the well-known computations of chemistry, when the atomic weights of carbon, hydrogen and oxygen are given, to discuss which completely would be aside from the present purpose.*

* See The Gas Engine, pp. 22, 23, for greater extension of treatment.

24. Combustion. The typical computations for combustion include: carbon, C, burning to carbonic dioxide, CO_2; atomic weight of C = 12; atomic weight of oxygen 16

$$C + O_2 = CO_2; \; 12 + 32 = 44.$$

Hence the oxygen is $\frac{32}{12}$ of the weight of carbon supplied; if the weight of carbon is one pound, the weight of oxygen is 2.66 pounds. If there is too little oxygen supplied, the carbon may burn to carbon monoxide, CO; then $C + O = CO$, or $12 + 16 = 28$, and the oxygen weight in pounds is $\frac{16}{12}$ of the pound weight of carbon.

If hydrogen (H, atomic weight = 1) burns to H_2O or steam vapor $H_2 + O = H_2O$; $2 + 16 = 18$; or one pound of hydrogen demands $\frac{16}{2} = 8$ pounds of oxygen.

With compounds made up of carbon and hydrogen, the process is identical. Assume the gas ethylene, C_2H_4; then

$$\begin{array}{rl} C_2 + O_4 = 2\,CO_2, & \text{or } 24 + 64 = 88 \\ H_4 + O_2 = 2\,H_2O, & \text{or } \underline{4 + 32 = 36} \\ \text{Totals,} & \phantom{\text{or }} 28 + 96 = 124. \end{array}$$

The gas weighing 28 requires $\frac{96}{28}$ of its weight of oxygen, or one pound requires 3.43 pounds of oxygen.

To furnish air for these weights of oxygen, one pound of air at the temperature of melting ice has 0.236 pounds of oxygen in it. Hence $\frac{1000}{236} = 4.25$ times as much air must be furnished as there is oxygen required. Hence for carbon $2.66 \times 4.25 = 11.3$ pounds of air are required; and for hydrogen $8 \times 4.25 = 34$ pounds of air. For the compound, $3.43 \times 4.25 = 14.58$ pounds of air.

If one pound of air occupies 12.39 cubic feet at this temperature:

$$\begin{array}{rl} C \text{ requires } 11.3 \times 12.39 & = 140 \text{ cubic feet of air.} \\ H \text{ requires } 34 \times 12.39 & = 421 \text{ cubic feet of air.} \\ C_2H_4 \text{ requires } 14.6 \times 12.39 & = 180 \text{ cubic feet of air.} \end{array}$$

25. The Problems of Mechanics Respecting Form and Internal Stress. The simplest function of the boiler as a storage reservoir for pressure is that of resisting such internal fluid pressure: — that of the water with the vapor tension upon its surface acting downward, and that of the vapor tension acting upward and laterally. Such tension in any high pressure boiler is practically equal in all directions, as the weight of the water per square inch becomes inappreciable in comparison with the other force. Such internal pressure tends first to produce deformation if the containing envelope is capable of changing shape under equal pressure in every direction until a permanent shape is reached. After that without change of form the equilibrium of effort of the pressure and of resistance in the material forming the envelope will be reached at a point called the bursting pressure, beyond which the resistance of the material is overcome and the shell ruptures at some point where it happens to be accidentally weakest. The question is therefore twofold: what is the permanent form of a closed vessel under internal pressure; what fixes the rupturing pressure of such permanent form?

The permanent form for such a containing envelope of pressure is the sphere. The internal pressure at every elemental area of the inside surface is decomposed equally and symmetrically in every direction, in the plane of the tangents to such surface at that point. The resistance of the shell from its tensile strength is in the line of these tangents. The elastic deformations of the material under these stresses cause the sphere to grow larger in diameter in every direction equally, but do not change the shape. The same conditions continue till the sphere ruptures. How will it rupture? If at some point on the great circle of the sphere a weak spot occurs, a split or crack develops where the continuity of the metal is broken. Instantly the strain at the two ends of such crack or split becomes greater than it was just before, supposedly then near or at the limit of such resistance, because the fractured area is holding no longer and the pressure has a moment or lever arm to prolong the split around the circle. The sphere then parts into two hemispheres, unless the pressure is so released at the break that there is not force enough to complete the tear in a ductile and homogeneous shell. If Fig. 9 represent a section of such a sphere in the plane of the page, the part above the horizontal AB tends to leave the part below it under stress, the direction of the line AB relative to the horizon being determined by the location of the weakest point at one of its ends.

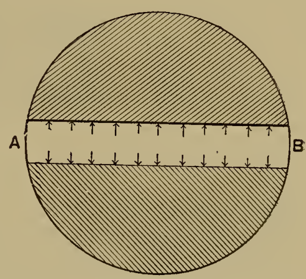


Fig. 9.

The sphere was early used as a form for the boiler, not only because the boiler is the derivative of the domestic kettle and the spherical concept was thus familiar to the first designers, but because the early boilers of soft copper sheets took this shape of themselves. It is not well-adapted to receive the heat of fire and gases, nor to keep the hot gases in prolonged contact with it (paragrafs 23–5); the only modern use has been in sectional forms where comparatively small cast-iron spheres were jointed together into banks or aggregates of such units, the ultimate shape of such aggregates becoming a parallelopidon and not the sphere. But the same permanence of shape under internal pressure attaches also to the cylinder. If the enclosing heads be made hemi-spherical, the same results are retained as with the sphere, except that quantitatively the stresses are modified. Such cylindrical boiler utilizes the heat of the gases much more adequately, and is easier to support and to set. Fig. 10 shows the type of such boiler of permanent shape, known historically as the " egg-ended " boiler, as its heads were spheroids. The flat head is much easier to make in heavier material, and hence the advantages of this shape are rarely utilized.

The sphere and the cylinder offer the same advantages of resisting uniform pressure without deformation when the pressure is from without radially inward, tending to collapse the cylinder or sphere. Here, however, it is imperative that all deformation be prevented, since the moment such a strained envelope loses the true circular section the areas exposed to pressure are no longer equal in every direction. The flattened areas have more aggregate pressure on them than the curved areas, and the flattening is progressive until the opposite sides come together or the metal tears. This trouble must be faced in large internal

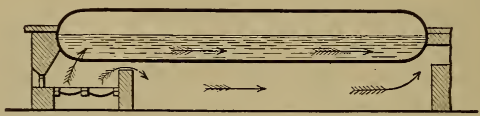


Fig. 10.

flues used as furnaces in types hereafter to be discussed and great pains must be taken to keep such cylinders as arches up to their proper shapes by stays or stiffening rings around them (paragraf 61).

Flat surfaces exposed to fluid pressure bend or deflect if fastened at the edges only, bulging outward until the stress in the plate due to its flexure resisted at the edges exceeds its tensile resistance. Such deformation must be resisted by reinforcing the low resistance of these plates to such flexure, either by stiffening bars or beams or by "stays," which are rods attached to the flexible surface at one end and fastened at the other to some point which is rigid or which tends to move under pressure in opposite sense. The tension in the stay replaces the lacking capacity of the flat surface. (See Stays, under paragraf 44.)

26. Values of the Stresses in Boiler Shells. Thickness of Boiler Plate. Assuming the cylindrical form for the boiler with flat ends it will be exposed to pressure, or its inside tends to part along a roundabout or ring seam by pressure against the heads, and it tends to part along a longitudinal seam and open out into a flat plate. If the pressure on each square inch be denoted by P, and the diameter of the cylinder be D in inches, the area exposed to pressure to blow out the head or rupture a ring seam will be the area of the head in square inches multiplied by the pressure on each square inch:

$$PA = P\pi r^2 = P\pi \frac{D^2}{4}.$$

The resistance to this pressure is offered by a ring of the boiler-metal whose area is

$$2\,\pi r t$$

when t is the thickness. If f be the tensile strength per square inch, the rupturing force just balances the holding resistance of the material when

$$P\pi\,\frac{D^2}{4} = 2\,\pi r t f.$$

This simplifies into

$$PD = 4\,tf.$$

For the resistance to rupture along a longitudinal seam it can be proved mathematically that the tendency to rupture in any plane will be the sum of the components of the normal pressure at every point which are perpendicular to that plane.* Therefore on each inch in length of such longitudinal seam the total pressure is PD. This can also be made clear by the expedient of imagining each semi-cylinder of the boiler to be nearly filled with a solid material like wood, and that the pressure P is introduced into the narrow space left between the two semi-cylinders which are held together by the enveloping ring of boiler-plate (Fig. 9). The resistance to separation of these semi-cylinders is the sum of the areas of boiler-plate at the two sides, multiplied by the tensile strength per square inch of that plate. This resistance is denoted by $2\,tf$ when the ring has a length of one inch. Hence the equilibrium of bursting pressure and resistance along a longitudinal seam is reached when

$$PD = 2\,tf.$$

It will be noticed that the boiler is twice as strong against blowing out the head or rupturing a ring seam as it is against rupturing along the longitudinal elements of the cylinder. This explains why boilers are double-riveted or are made with special joints for their longitudinal seams.

The above calculation is for solid plate, or for welds which are as strong as the solid plate. Where riveted seams are used an allowance must be made for the reduction of the value for f due to the weakening caused by removing the metal at the rivet-holes, which the rivets do not replace. This factor is called the "Efficiency." See paragraf 40.

Boilers are usually designed with a factor of safety of six in their

* See paragraf 36 at end of this chapter.

shells; or in other words, the working pressure is one-sixth that at which the shell would be expected to rupture from internal pressure.

Boiler-plate can be bought of all thicknesses, but it is usual when the calculation brings out an inconvenient figure to pass to that practical thickness which is next above. Usual thicknesses of plate are, in fractions of an inch:

$$\tfrac{3}{16},\ \tfrac{1}{4},\ \tfrac{5}{16},\ \tfrac{3}{8},\ \tfrac{7}{16},\ \tfrac{1}{2},\ \tfrac{9}{16},\ \tfrac{5}{8},\ \tfrac{3}{4},\ \tfrac{7}{8},\ 1,\ 1\tfrac{1}{8},\ 1\tfrac{1}{4}.$$

It is inconvenient to handle, curve, and rivet plate thicker than one and one-half inches. The difficulty of manufacturing thicker plates also stands in the way of their use. A less thickness can be made to serve by using a smaller diameter. An allowance of $\frac{1}{16}$ of an inch is generally made above the computed theoretical thickness for the effect of corrosion, external and internal, while still leaving the boiler of full strength until that wastage has occurred.

27. Materials for the Boiler Shell. Specifications and Tests. The material of which the shell is made must not only withstand the internal pressure stresses, but also those due to expansion and contraction by changes of temperature within wide limits. The stresses are often sudden, are unequal in different parts of the shell and therefore local in their action, and their magnitude may reach the limit of the resisting power of the shell if little or no time is given to yield to them by changes of figure of the plate. Such sudden changes come from cold feed-water suddenly pumped into a partially empty and, perhaps, overheated shell, and from an inrush of cold air into a fire-box from a suddenly opened fire-door in cold weather. Hence the boiler material must not only have tensile strength, reasonable conductivity and ease of manufacture, but it must be ductile (the term used in the sense of being not brittle), so as to stretch and yield without danger of sudden breaking.

The only materials in universal acceptance are three grades of soft or low carbon steel, made by the open-hearth process, acid or basic, and meeting specifications as in table on page 31 for the qualities known as Flange or Boiler Steel, Fire-box Steel, and Extra Soft or Rivet Steel.

It is generally considered undesirable for the engineer to limit the manufacturer respecting his processes or chemical methods of attaining the result, and to hold him for the result only. Hence while the carbon-steels referred to above will be usually lower in carbon than 0.50, it is quite likely that alloy steels containing nickel and other strengthening elements will come into use with lowered prices, provided such strong steels are not brittle under stress and heat. Other materials for boiler steels have been of significance in earlier uses, such as copper, cast and malleable iron and wrought iron (see paragraf 37) and offered some advantages at that time.

TABLE IV.

SPECIFICATIONS FOR BOILER STEELS.

Element Specified.	Flange Steel.	Fire-box Steel.	Extra Soft Rivet Steel.
Tensile strength pounds square inch	55,000 to 65,000	52,000–62,000	45,000-55,000
Yield point not less than	½ Tens. St.	½ Tens. St.	½ Tens. St.
Elongation per cent in 8 inches not less than	25	26	28
Phosphorus not to exceed	Acid 0.06 Basic 0.04	Acid 0.04 Basic 0.03	0.04
Sulphur shall not exceed	0.05	0.04	0.04
Manganese	0.30 to 0.60	0.30 to 0.50	0.30 to 0.50

The material shall be tested by testing machine, and by bending and for homogeneity. The bending test shall be made both cold and after heating to cherry red and quenching in water, the piece bent to 180 degrees flat upon itself without fracture on the outside of the bent portion. The homogeneity test shall be made by nicking or grooving the piece under test with three grooves $\frac{1}{16}$ of an inch deep and 2 inches apart. The nicked piece is then put in a vise, and broken off above the nick by light blows of a hammer, the bending being away from the groove. This is to open up and render visible any failures to weld up, or cavities or foreign matter.

Steel plate can be procured of any reasonable length, since it originates from an ingot which can have the necessary weight to give both width and length. Plate is stronger in the direction in which it has been rolled, than at right angles to this axis. Hence it is better to make up the length of the shell by several rings than to make fewer ring seams and more longitudinal by using the plate the other way. In Fig. 11 is a design made up of the two heads, the dome piece and its head and one sheet to form the cylindrical portion. Such boiler can have only the circumference imposed by the greatest distance between the housings of the rolls by which the plate was made. This is in the neighborhood of twelve feet in the largest mills.

In former years badly chosen or improperly treated steel gave trouble by cracking along the rivet holes, and this could only be prevented by troublesome and costly annealing. It is no longer liable to occur in good material and with the rivet holes drilled or reamed effectively.

28. Shaping the Shell Elements. Curving Plates. The plates for the cylindrical part of a boiler shell of large diameter will be received flat from the manufacturer, and must be bent into the cylindrical shape. This is done by rolling them cold between three driven rolls, so arranged that as the plate is moved and driven by two of them it shall be continuously pressed by the third and caused thereby to receive a continuous curvature. Such rolls are called bending-rolls, and may be arranged with their three parallel axes horizontal or vertical. The horizontal arrangement is much preferred in America by reason of its convenience (Fig. 12). The three rolls may be arranged relatively to each other in two ways. Two of the rolls may be fixed in position, both driven by power and with their axes in a horizontal plane; the third will lie above the space between the other two, or with its axis in the plane of the common tangent to the other two. This third roll will be the bending-roll, will have its axis adjustable, and will not be driven. Its position further from the lower rolls or nearer to them will determine the radius of the curvature of the plate (Fig. 13). The rolled and curved plate will gradually enclose the upper roll, so that if the bent edges are to come together the bearing or housing of this upper roll must be removable at one end to allow the completed cylinder to be removed endwise (Fig. 12).

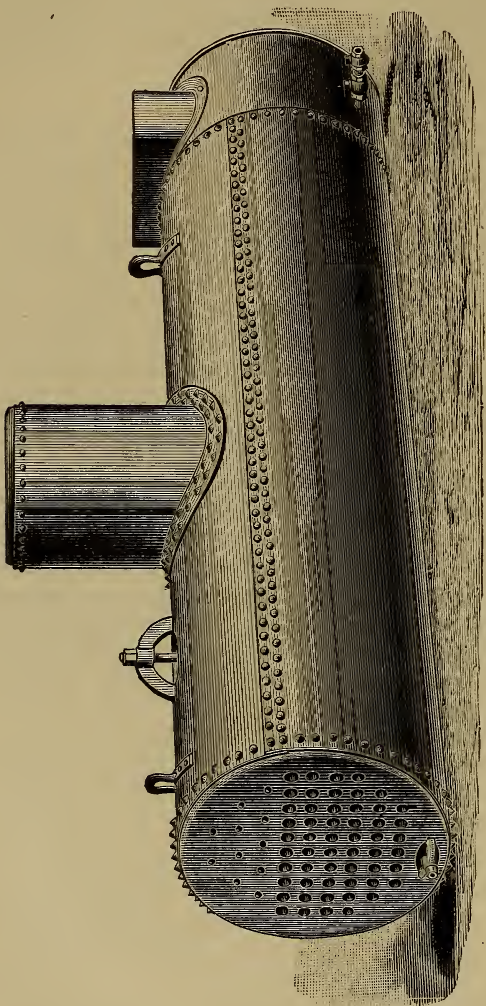


FIG. 11.

This arrangement of rolls does not cause the plate to be curved all the way to the edges which are parallel to the axis of the rolls, since a distance equal to the radius of the roll or more cannot receive the curving action of the upper roll with large diameters of cylinder (Fig. 13). A modification is to arrange the two driven rolls over each other, and to make the third approach the opening between them at an angle from below (Fig. 14). This arrangement brings the curving effect close to the edges, and has the plate positively driven against the bending-roll by the nip of the two rolls which are driven. The only difficulty arises from a change of lengths of the contact surface as the cylindrical shape is developed. The two driven rolls would develop equal lengths as they revolve without slipping upon the shorter inner surface and longer outer surface of the curved plate. If this were not overcome, the driven rolls would exert a calendering action on a plate of sensible thickness and undo the curving effect of the third roll. The difficulty is met by driving one of the two rolls from the other by a differential or " box " gear, by which the motion reaches the second roll from the first through a couple of pairs of bevel-wheels. The axes of one pair are independent of the fixed

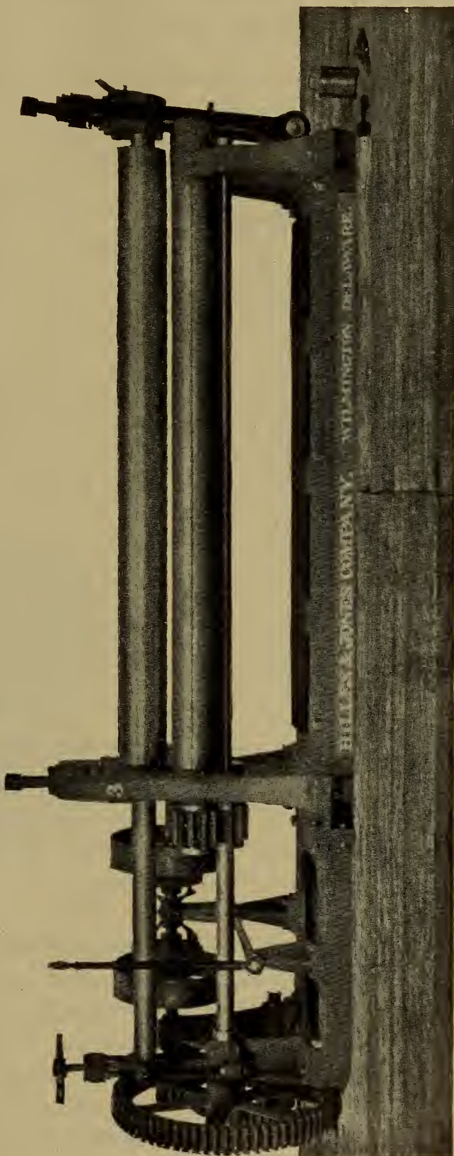


Fig. 12.

frame, so that if one roll has farther to move than the other the difference in path is able to be compensated by a motion of this movable axis which allows the gears to roll through a space while still transmitting the full driving effort necessary.

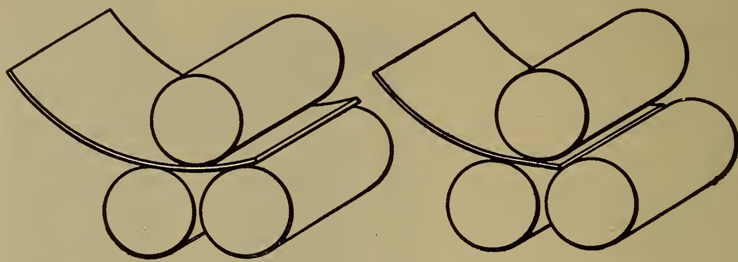


Fig. 13.

The rolling process is effected by rolling the plate back and forth through the rolls with continuous adjustment of the third or bending roll until the gauged diameter of the cylinder or segment of cylinder is reached. The rolls have to be of diameter sufficient to withstand the

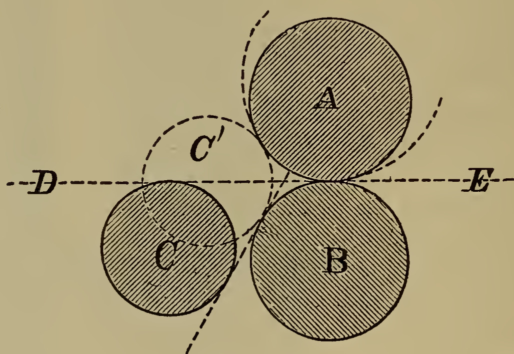


Fig. 14.

tendency to flex, and of length sufficient to handle the longest or widest plate used. The rolls limit also the thickness of plate convenient for shells. Their convenient diameter imposes a lower limit for the diameter of flues to be made by their use. Their length imposes a limit upon the length of boiler shell to be made in one piece, or in two pieces if the joint is to be longitudinal.

29. Arrangement of Rings of Plate in Shells. It is desired to have as few joints in the shell as possible, and yet the boiler must have a practical length. The least number of joints is reached in the arrangement shown for a shell boiler in Fig. 11, where the shell part is one long plate joined lengthwise. The size of such boiler is limited by the attainable size of single sheet both as to length and diameter, so that it is much more usual to arrange the length of the plate circumferentially and to get the necessary length by jointing such rings or zones by one, two, or more ring seams. In very large diameters, such as are usual in marine boilers, the rings themselves will each be made up of two or more segments, jointed by longitudinal joints. (Fig. 100.) In the ordinary shell of stationary practice the ring or zone is in one piece, joined at the edges, and the usual diameter of such shells is fixed by the length of plate usually to be had. If there are three such rings or belts as in the usual iron boiler and in many steel boilers, these rings may be jointed to each other at the roundabout or ring seams in one of three ways. The three rings may be true cylinders, each a little smaller than the preceding, so that they fit inside successively like the joints of a telescope, and the larger laps over the smaller one (Fig. 112); or one ring may be smaller than the other two (usually the middle one smaller than the two end ones), so that it will fit inside of both and form a lap

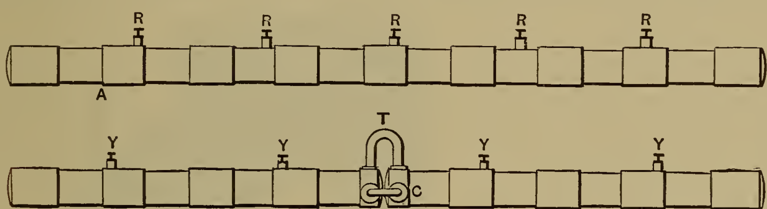


FIG. 15.

(Fig. 15, 69, 70, 73, 134, and 188). The third plan is to taper each ring slightly, so that it will fit outside at one end over the end of the next ring. This end has the same diameter as the small end of that same ring, so that the two ends of the boiler are of the same diameter. The taper of the rings is laid out so that currents of hot gases or flames shall not impinge against the ends of such lapping ring joints, but shall flow over the ridge which the lap makes. (Fig. 84.)

30. Shaping Shell Elements. Flanging Heads. The head of the boiler shell is that flat or arched surface which closes the two ends of the cylinder. It has to be jointed to the cylindrical portion. American practice is to have the cylinder fit over the outside of the head, and to bend up the edges of the head all around to form a surface parallel to the cylindrical shell by which the joint can be made. This bending up of the edges of a flat disk to form a projecting ring or flange is called "flanging." It may be done by hand or by machine. By hand the edge is heated locally, a sector at a time, and the hot metal is bent over the edge of a properly moulded anvil or former by means of heavy wooden beetles or mauls in the hands of skilled strikers or smiths. Wooden heads do not draw down the metal in bending as metal sledges would, and the blow is delivered over more surface. The objections to hand-forming are the cost of labor, the impossibility of uniform heating all round the edge, and the inaccuracy of the final cylinder. Steel heads, forge heated and hand flanged, must be annealed after forming, since steel is specially sensitive to inequalities of heating, and the finished head unannealed is all distorted and unequally strained from this action. Hand flanging must be so done that the steel never cools under treatment to its critical temperature, which is found at about a blue heat. It is brittle and liable to crack under the blows of even the wooden mauls.

Machine flanging is much to be preferred where practicable. It is usually a process of hydraulic forging with proper dies. When done at one process two cast-iron formers are used, one male and one female. The disk is heated uniformly all over, and when at proper temperature

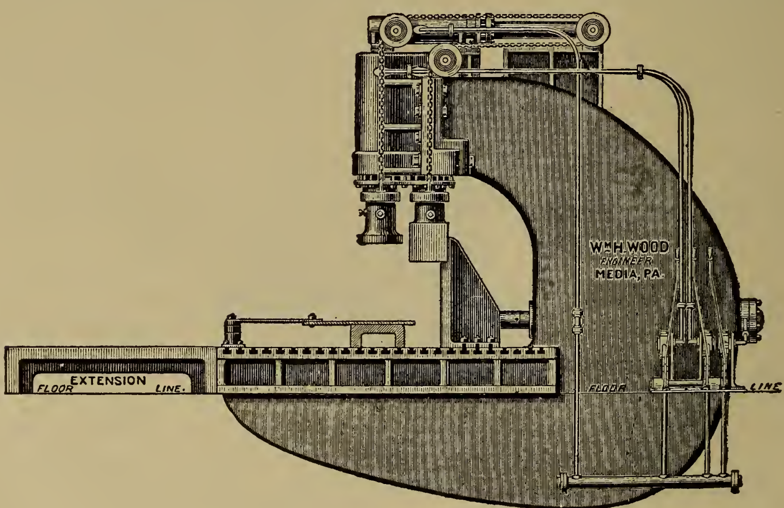


FIG. 16.

is laid upon the top of the hollow female die. The male die descends concentric with the other by hydraulic pressure, and forces the plate to bend up uniformly all around and take the shape of the standard male former. The head is thus shaped at one heat to the required shape and diameter without distorting strains. In other forms of flanging press (Fig. 16) the plate is held at a proper temperature between the faces of the hydraulic vise, while pressure comes radially upon the edge of the disk from hydraulic cylinders which carry shaping heads and bend down the edge gradually until the disk fits the former, which is the face of the hydraulic vise.

Earlier European boilers will show the head jointed to the shell by a ring of angle-iron section, or by a ring of plate forged into that shape. The use of more ductile and superior metal for heads has made flanging more usual.

The flange is usually placed inside the boiler. This keeps it protected from the rapid oxidation or burning to which projecting flanges would be exposed if hot gases or flame impinged on them and they were only cooled by conduction from the water at some inches' distance.

Flanging stretches the metal right at the bend, but compresses the metal beyond the bend which forms the flanged surface. The sharper the angle of the bend, the more severe these concentrated strains. Hence to bend flanges with a radius not less than four inches has been specified, to diminish this source of trouble. Flanging is also necessary in jointing rectangular fire boxes and for the attachment of large flues to boiler heads.

31. Joints in Boiler Shells. Welding. The rings which form the cylindrical shell of the boiler are curved from flat plates, and must be jointed at the edges and at their ends. The requisites of such a joint are: (1) strength to resist the strain from internal pressure; (2) tightness against leakage of water or steam, with a construction which shall not be too costly; (3) ability to withstand heat; (4) ability to undergo changes of shape from expansions and contractions without injury to the metal.

The two edges of the plate which are to be joined are arranged so as to lap over each other to be secured together, and this attaching can be done by welding or by some form of the rivet joint. Bolting with a thread and nut will not meet the second requirement of tightness against leakage unless the joint surfaces are planed and finished and the bolt holes reamed and the bolts turned. This is prohibitory from its cost; and even if this were not a barrier, the friction of the nut so reduces the clamping power of the screw bolt that it would make a much weaker joint than is secured by the other plans.

Welding of boiler plates to make the joint with itself or other parts of the shell offers many advantages. The welding property of wrought iron and ductile steel enables them to unite at clean surfaces when pressed together with sufficient force in a state of sufficient plasticity from heat. The presence of oxide of iron or dirt or cinder between the contact surfaces will prevent a satisfactory weld, or if there is no adequate pressure to unite the surfaces together.

When welding is satisfactory it may be expected to be as strong as the rest of the metal — which has, in the case of wrought iron, been fabricated into plate by availing of the welding property through the entire course of manufacture.

Welding of plate is done by lapping the two edges over for two or three inches, heating the lap to a welding heat on both sides by a flame or jet of gas free from sulphur or other oxidizing tendencies, and then bringing the lapped surfaces together either by the force of percussive hammer or sledge blows or by steady pressure of cams or roller presses. Some fluxing material like borax which will make a fluid glass with oxide of iron may be used as a protection for the contact surfaces, so as to prevent oxidation from exposure to air, with the expectation that it will be expelled from the joint by the welding pressure, and carry with it everything which would interfere with good welding.

Welding of boiler joints offers these advantages:

(1) It makes the joint as strong as the rest of the plate, or nearly so.

(2) The plate is no thicker at the joints than elsewhere. This avoidance of a lap keeps the tensile strain from internal pressure always in the axis of the plate and without a tendency to flex at the lap or joint (paragrafs 32 and 196).

(3) Double or extra thickness is avoided at laps or joints. The plate gets unnecessarily hot at multiple thicknesses, and oxidation is more rapid there.

(4) No rivets are required, which makes the boiler lighter and less liable to leak.

(5) A good welded seam is water-tight and requires no calking.

The objections to the welded seam in boilers are:

(1) It cannot be inspected for its satisfactory quality unless it is so bad as to allow water to leak through it under pressure. But it may be water-tight and yet be far from having full strength. While a test by hammer taps to observe the resonance of the metal at the joint will reveal much to the practiced ear, it lacks the convincing force of an inspection of each single rivet in a riveted seam.

(2) Welded joints in large shells can only be gotten from a few firms with facilities and experience for such work. This has some effect upon the cost of such joints. But when a satisfactory welded seam can be obtained it makes an ideal joint.

In cylinders with closed ends the last seam must be riveted even if the others are welded. The exception is where the head is flanged outward, or is convex inward so as to bring the closing joint outside the shell (paragraf 60).

32. Riveted Joints for Boiler Shells. When two plates are to be joined by rivets, they are lapped over each other, and through a hole which matches in the two plates a rivet is introduced red or white hot. This rivet has a head formed at one end in its manufacture, but the shank is straight. When in place through the holes, pressure is brought upon both ends of the rivet, whereby the projecting shank is upset and forced back upon itself, thereby enlarging its diameter in the hole until it fills it completely, and when the metal can no longer be displaced laterally in the holes, the metal of the rivet still projecting beyond the plate, spreads sidewise over and beyond the hole and forms the second head of the rivet. The rivet when completed has two heads connected by the shank which is still red hot when the head is finished, and which in its contraction on

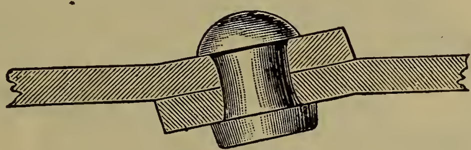


Fig. 17.

cooling draws the two plates together with a force measured by the modulus of elasticity of the rivet metal and by the cross section of the shank. It is a force much in excess of that which any bolt and nut can exert.

The riveted joint meets the requirements of a boiler joint in that it is

(1) Strong.

(2) Water-tight.

(3) Cheap.

The difficulties which it introduces are

(1) The hole for the rivet cuts out just so much metal from the solid plate, and therefore the joint is not as strong as the plate where there are no holes.

(2) In simple lap joints the strain on either side of the joint is not resisted in the axis of the plate on the other. High pressure tends to flex the lap joint till the two plates come into line (Fig. 17), and this flexure causes the deterioration called "grooving" (paragraf 196).

(3) The boiler shell is thicker at joints than elsewhere.

There are certain further disadvantages attending a badly made rivet joint which will be noted hereafter. The design of special riveted joints is to diminish these difficulties.

33. Construction of a Riveted Joint. Punching and Drilling. The holes in the plate to receive the rivets may be made by punching, by drilling, or by punching out a small hole and enlarging it by reaming. Formerly, and with iron plate, punching was universal. More recently, and with steel, the latter methods are used.

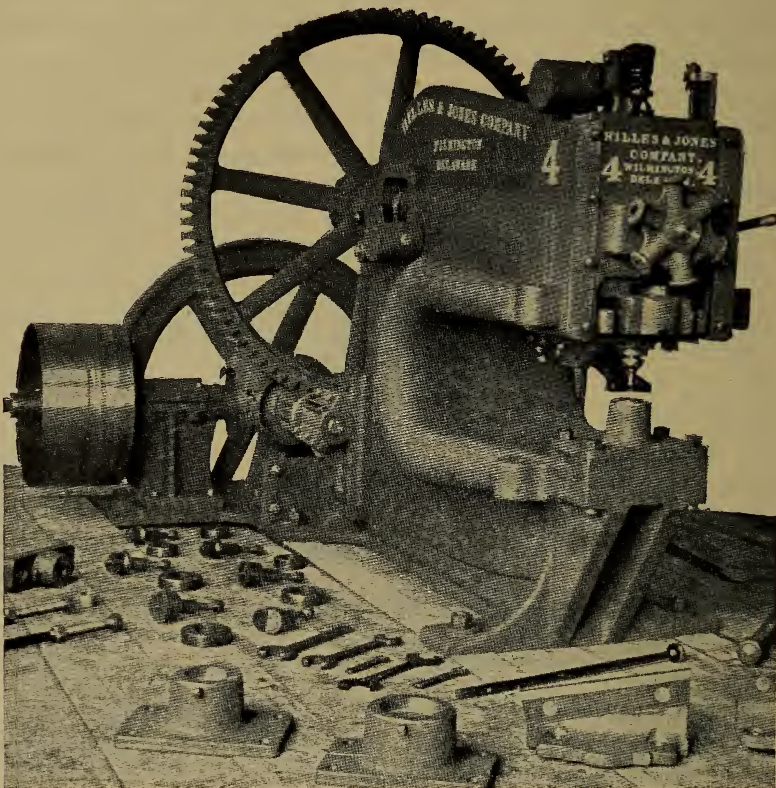


Fig. 18.

The punching of the hole is done in a punching press (Fig. 18) in which a hard and tough steel cylinder comes down upon the plate supported upon an abutment or female die having a hole in it slightly larger than the punch (Fig. 19). The punch shears its way through the supported plate and extrudes a blank of the punched plate cut by its stroke. While at first the punch cuts the plate, after a fraction of an inch of penetration it tears its way through the rest of the thickness without true shearing action; and in a plate of laminated structure such as wrought iron has, it is largely the reaction of the abutment or die which limits the lateral spread of the tearing effect. The extruded blank is conical, since the die is larger than the punch in order to free the latter and pass the blank.* The punching presses may be crank presses as shown, or the punch may be driven by hydraulic pressure. Flanged plates are usually punched in horizontal punching presses. A spiral shape has been given to the impact face of the punch so as to make the cut a gradual and progressive one around the circumference of the hole, and to help secure a true shearing action (Fig. 20).

Drilling of plate is done by the ordinary machine shop drill of two cutting planes meeting at an edge. Twist drills are most convenient, although the flat drill is still to be met. The drill will be run by the ordinary drill press.

The gang punch or multiple punch has a number of punches mounted in a fixed relation in a holder, so that one stroke of the holder punches two, three or more holes at once and at a standard distance apart.

Gang or multiple drills have a number of revolving spindles driven from a common source, each carrying its own drill and drilling a number

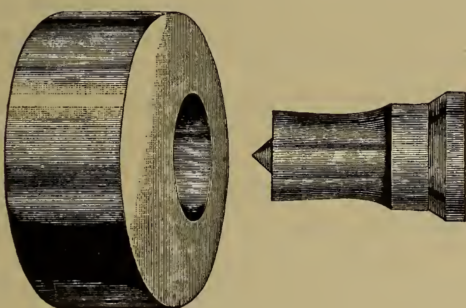


Fig. 19.

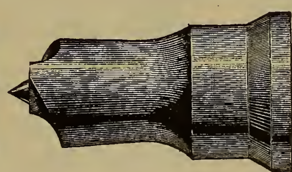


Fig. 20.

of holes at once and at a fixed distance apart. These gang drills usually drill alternate holes in a seam to permit a convenient distance between the spindles.

34. Punching and Drilling Compared. The objections to punching the holes for the rivets are:

(1) The injury to the plate. The impact-pressure of the punch upon carbon steel produces an effect upon the metal around the hole similar to that of hardening by heating and rapid cooling. The metal has its modulus of elasticity raised, so that it stretches less before breaking or cracking, which is the same as becoming brittle and liable to fail in service under strain suddenly applied. Experiments would

* Usually larger by $\frac{1}{10}$ the diameter of the punch.

appear to show that the carbon of the steel enters into combination with the iron under the shock, and, to restore the metal to the normal ductility after punching, the plate must be annealed. Otherwise the deteriorated metal must be removed by the reaming or enlarging of the hole until good metal is reached at a distance from the punched place beyond the effect of the blow of the punch.

(2) The spacing of the holes is likely to be inaccurate in punching with a single punch, and when punched independently the holes in the plates which lap will not match, or will be "half blind" (Fig. 21). This difficulty arises when massive plates are presented by hand to the punch and the work is done too rapidly. The holes are laid out or are marked on the plate, and the punch mechanism thrown into gear when the mark is under the axis of the punch. Even when the punch has a "tit" (Fig. 19), to serve to guide it to the axis of the hole it may seem to take too long to adjust the plate, and the stroke may be made before the setting is perfect. Gang punches avoid this trouble so far as each set is concerned, but best results are had from the use of feeding tables on which the plate rests, and which are fed forward by racks or similar feed devices, so that the plate moves each time through the same fixed distance, thus securing uniform spacing of the holes upon a line. Errors may creep in even here from a divergence laterally of the lines of holes which are accurately spaced lengthwise in two plates. Inaccurate spacing which causes the holes to come half blind to each other must be corrected either by reaming out the holes till they do match, or by stretching them by the drift-pin to be referred to hereafter (paragraf 43). If they do not match at all, the two holes are blind.

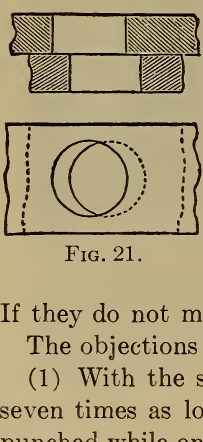


Fig. 21.

The objections to drilling the holes are:

(1) With the single drill the process is slow. It takes from five to seven times as long to drill as to punch, or five or seven holes can be punched while one is being drilled.

(2) This makes drilling costly unless gang methods are used.

(3) The point of a drill is not a true point, but an edge where two cutting planes meet. Hence the drill in starting has a tendency to work sidewise away from the true axis of its hole and follow one or the other of the corners of the edge plane. If this tendency is disregarded, the holes do not come true except by accident. If time is taken to keep the drill starting true, the work is slow.

(4) A drilled hole in thin plate usually has a burr or projecting ridge raised around the edge of the bottom of the hole, where the feeding

pressure on the spindle and drill forces the latter through the thin film of metal which remains in the hole after the point of the drill has come through, and cutting and resistance is at the edges only. This burr would prevent the joint of plates being water-tight unless it was carefully removed by filing.

(5) The drilled hole is cylindrical; the punched hole is conical. It is an advantage to have the hole conical if the two small bases of the cones can come together (as at B in Fig. 31). The sloping sides give greater holding power to the head, and give a form of rivet better calculated to prevent the head from snapping off in service.

The points in favor of punching are its rapidity and cheapness. The points in favor of drilling are its harmlessness to the plate and the probable greater accuracy as to the matching of the holes.

The plan of punching small and enlarging to size by reaming out the holes offers the advantage of rapidity and cheapness, and leaves no deteriorated metal. One-tenth of an inch of metal cut away from the edge of the hole will remove the hardened material, and such reaming is much more rapid than drilling out the solid metal. The reamer may be tapering if conical holes are preferred. A great deal of work is done by this method, as combining the commercial advantages of one and the advantages as to quality offered by the other.

Drilling, however, must be exacted for thick plates and large holes, and is best at all times. In the very highest standard of practice it is further exacted that the holes shall be drilled after the plates have been curved and assembled, so that the holes shall be drilled truly radial in both plates and with the sheets in place. This prevents troublesome burring, and prevents mismatching of holes. Special machines have been erected for this grade of work.

35. Hand- and Machine-riveting. The pressure necessary to upset the shank of the rivet into the rivet-hole so as to fill it and to form the second head can be exerted either by hand-hammers in the hands of skilled riveters, or a die or swage may be put over the end of the shank and struck by heavy sledges so as to upset the shank and develop the form of the die on the projecting end; or the pressure can be brought upon the rivet by a machine called a riveter or riveting machine.

In hand-riveting, the rivet is pushed up from within wherever possible, and when in place a massive swage is held up against the inner head of the rivet by a helper with all the force possible, by leverage, while rapid blows are delivered upon the end of the hot shank by the riveters without. Hand-riveting is necessary for the closing seam of a shell in order that resistance to the heading of the rivets may be offered from within, and by the riveters' helper with his swage. The design of the shell must be such as to allow the " holder-up " of the swage to get out of the boiler when the seam is completed. But machine-riveting gives so much better results in filling the holes by upsetting, and in forcing the plates to contact before the heat comes to press upon them and draw them together, that machine-riveting is used wherever practicable. Swage-riveting with sledges is better than light hammer-work with long or thick rivets, but is also less effective than the work of good machines. The compression of the machines gives an added resistance to the joint by the frictional resistance which the pressure opposes to a sliding of the two plates upon each other. This resistance adds to the shearing resistance of the rivets by preventing the shearing edges of the plate from commencing on the rivet until the friction is overcome.

The usual types of riveting-machine are three: steam or air riveters, hydraulic riveters, and lever machines. In all types there will be a movable head actuated by power to compress, upset, and head the

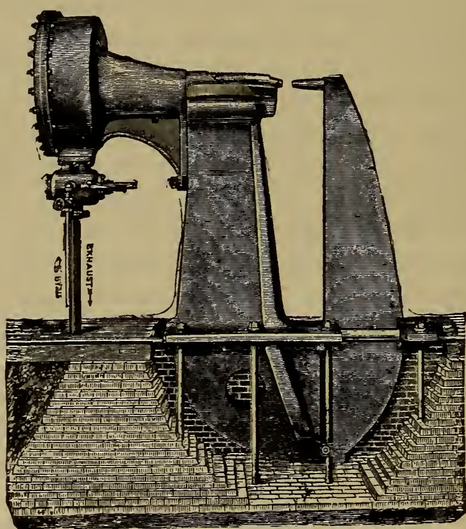


Fig. 22.

rivet against a fixed abutment or " stake " which replaces the upheld " swage " in the hand of the helper in hand-riveting. This stake requires to be a stiff and powerful organ of the riveting machine; and since the longer its length the more metal must be in it for strength and stiffness, it will be apparent that the stake limits either the diameter of flue which must pass over it, or the length of zone to the end of which the stake will reach; or it may limit both. The stake is fitted at its upper end with a die which fits the manufactured head of the rivet (or else will reshape it), and the rivet is pushed through the hole from the stake side. The movable head is then allowed to exert its force endwise upon the rivet, upsets and heads it, and is then retracted.

The steam-riveter shown in Fig. 22 has a piston of large area, which receives a relatively light pressure of steam or air upon each square inch of area, so that the necessary aggregate force is secured. The exhaust-steam after the working stroke comes round also to the front side, and is exhausted first from the working side so as to leave a pressure to retract the piston before the exhaust occurs from the front side. The

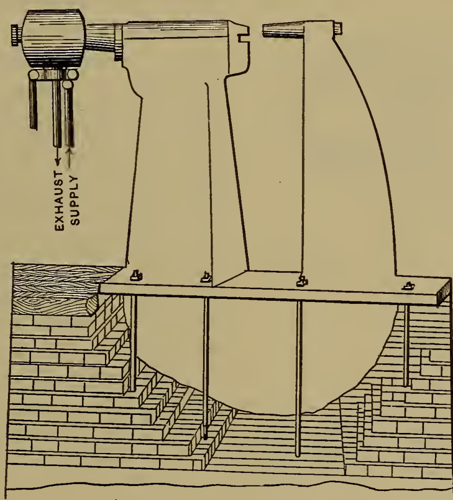


Fig. 23.

hydraulic riveter uses a plunger of small area, exposed to a water-pressure of considerable amount, perhaps 250 to 350 pounds or more per square inch, so that a much less area under greater pressure does the same work as the large area under less pressure (Fig. 23). The lever or press riveters have an elbow-joint linkage which hangs flexed when the movable head is at rest, but can be straightened out by means of a cam or a third link, and in its straightening it compresses the rivet in its place against the stake with the great force of the elbow-joint combination (Fig. 24). Some portable riveters are constructed on this principle with a fluid acting upon a piston to cause the elbow-joint links to straighten. Such are much used in bridge-shops and for girders.

The hydraulic riveters are the most compact, but the high pressures used in them give trouble at the packings. They move more slowly than the steam riveters in coming against the rivet end, and their effect is more that of pressure and less that of a blow. This latter is hard to prevent with an expanding fluid like steam, over which the valve exerts no control after it has been passed. Either of the fluid machines has the advantage over the lever machine that the pressure can be gradually increased to its maximum as the rivet yields and cools, and furthermore the pressure can remain upon the rivet an appreciable time. They have the further advantage over the lever type that the stroke or travel of the movable head is not fixed in length, but is fixed only by the refusal of the rivet to yield further to pressure. This is convenient when rivets of different length are in question for differing thicknesses or number of laps of plate. This has been met for the lever riveter by having the abutment-joint of the linkage mounted upon a bearing adjustable by a wedge for different lengths of rivet, or upon a yielding bearing which is held to its seat by springs, or by heavy hydraulic pressure maintained by an accumulator. If the resistance offered by the stake to an upset of the rivet is too great as the linkage comes straight, the back end of the linkage yields and prevents such excess. It does not serve, however, if the rivet is shorter than the normal. Then the lever riveter does not get its full pressure upon the metal, while the hydraulic and steam riveters are not subject to this difficulty, but follow the rivet to refusal.

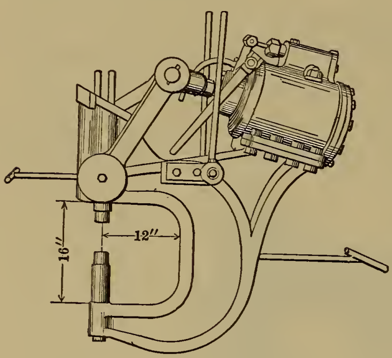


Fig. 24.

The very intensity of the pressure in upsetting rivets by machine has sometimes caused the metal of the rivet to squeeze sidewise into the joint between the plates, wedging them apart and leaving a thin film between them. This is fatal to tightness of the seam. It is best to have a double ram construction, whereby an outer annular ram forces the two plates together as by a vise pressure before the inner or heading ram proper comes forward against the rivet. This closes the joint tight before the rivet begins to press upon it, and gives much the stronger and tighter joint of those made by machine. The use of such a riveting machine is specified by some designers.

The riveting machines require adequate overhead hoisting appliances so that massive rings and shells can be rapidly and easily handled, and the joints and rivets presented truly in line and normal to the motion of the heading die. This justifies a travelling crane in a busy shop.

36. Mechanics of the Stress in a Thin Cylinder. The computation for the magnitude of the stress in the longitudinal elements of a cylinder or at the longitudinal seams of a boiler can be presented as follows in connection with Figs. 9 and 25:

Let an element of the arc of the semicircle be denoted by dx in Fig. 25 measured along the tangent, and suppose its length perpendicular to the paper to be one inch. Then the area of that element will be $1 \times dx$ and the normal pressure on it Pdx. If this normal be decomposed into horizontal and vertical components, only that perpendicular to AB tends to rupture the joints at A and B. The horizontal components tend to produce rupture along EF. Hence the component V which is $Pdx \cos a$ produces the same effect as the force P acting over the area $1 \times bc$, since $bc = dx \cos a$. Therefore the total upward force is the sum of the projections of all elements of the semicylindrical arc, or is equal to PD. Or, again, $Pdx \cos a$ may be integrated between $a = +90$ and $a = -90$, which becomes $P(R + R) = PD$. The sketch in Fig. 9 shows the reasoning when a solid mass like wood transmits the pressure to the arc just as the fluid does, and shows the rupturing force to be proportional to the diameter. This was first proposed by Forney.

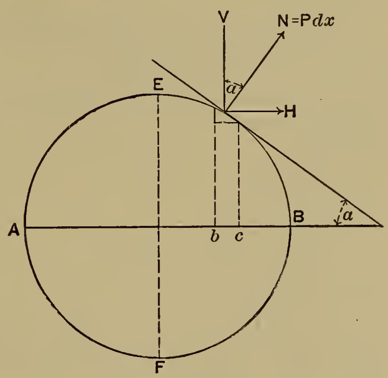


Fig. 25.

37. Copper, Cast Malleable and Wrought Iron for Boiler Shells.

Copper is:

1. Highly conductive of heat.
2. Easily molded with light machinery.
3. Resists corrosion from water and gases.
4. Scale does not adhere to it as to steel.

On the other hand it:

5. Has a low tensile strength, and must be thick.
6. Is costly to buy as compared with steel.
7. Loses its tensile strength as it gets hot.
8. Sets up galvanic action in acid waters with other metals.
9. Is liable to mechanical injury from fire tools.

Brass is the same as copper.

Copper has been used in some locomotive fire boxes and for ferrules in tubular boilers.

Cast iron is:

1. Cheap.
2. Easily molded by casting into shape.
3. Rivets are not required, as units will be made in one piece.
4. Less liable to corrosive effect of gases.

On the other hand it:

5. Has low tensile strength; hence has to be thick.

6. Is not ductile, but is brittle, and breaks without yielding.
7. Is liable to blowholes or defects below the surface.
8. Is liable to lines of weakness at corners due to unequal cooling.
9. Repairs after breakage are troublesome.

No. 7 is the prohibitory defect, when the metal is liable to sudden changes of temperature. Cast iron is prohibited in Great Britain where it must withstand pressure and is reluctantly insured in the United States. Its thickness made it popular where fittings were to be screwed in, as the thread would be long enough to be tight.

Malleable iron is used in fittings and parts of special boilers. Its objections are:

1. If malleableizing process is not complete the material is variable and unreliable exposed to heat and stretching.
2. Its coefficient of expansion by heat is not the same as steel or wrought iron.

Wrought iron is hard to get under modern manufacture except in tubes. Old classes were:

1. Tank iron.
2. Shell iron.
3. Charcoal No. 1.
4. Charcoal-hammered No. 1.
5. Flange iron.
6. Fire-box iron.

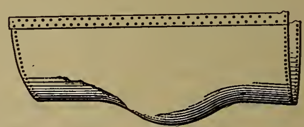


Fig. 26.

It was made by welding, piling, reheating and welding the pile until the cinder was mechanically expelled and a plate built up of required quality. Such material was liable to blisters or lamination due to defects in welding and from pockets or threads of cinder. These made bags or blisters in the shell under localized heating, which were elements of weakness and danger (Fig. 26).

CHAPTER III.

BOILER RIVETING, STAYING, AND STRUCTURAL DETAILS.

40. The Mechanics of the Riveted Joint. Efficiency. The theoretical design of a riveted joint would involve giving to the thickness of the plate (t) and to the diameter and cross-sectional area of the rivets (d and a) such values that when the rivets are spaced at a distance apart from each other called the pitch (p) measured from center to center, the required strength was secured with the unit stresses in rivet and plate within safe limits, and equal to each other in each element.

The superior limit of the strength of the joint is the unperforated plate, or the strength of the metal between holes. The efficiency of a riveted joint is the ratio which the highest allowable stress in such completed joint bears to the unperforated plate. The ideal is to have the efficiency in detail of the tension on the plate, the shear of the rivet and its resistance to compression in bearing upon the side of the whole equal to each other.

If a force in tension F be transmitted through a single riveted lap joint such as Fig. 32, the net area of the plate is $(p - d)\,t$. If the tensile resistance be denoted by f, then for the plate $F_t = tf\,(p - d)$ and the efficiency

$$E_t = \frac{(p-d)tf}{fpt} = \frac{p-d}{p}.$$

For the bearing area of the rivet in compression

$$F = dtC$$

when the compression resistance is C and the efficiency

$$E_c = \frac{dtC}{ptf} = \frac{dC}{pf}.$$

For the rivet in shear, the stress is

$$F = \tfrac{1}{4}\pi d^2 S,$$

and the efficiency

$$E_s = \frac{\tfrac{1}{4}\pi d^2 S}{ptf}.$$

These can be solved in any case by assuming a value for the tension and compression stresses, say 55,000, and for the shearing resistance, say 45,000, and dimensions for the pitch, rivet diameter and plate thickness and solving. The lowest value for E is the efficiency of the joint as a whole, and in the ideal case are all equal to each other. If the double riveted lap joint is used there is additional rivet area for bearing and shear introduced and in triple joints still more with increased efficiency of the joint and the equations can easily be written for each case. The butt joint by eliminating the cross bending of the two plates at the joint adds strength and efficiency, particularly when double or triple riveted. Usual values for the efficiencies are:

		Per Cent.
Single riveted lap..........................	Fig. 32	55
Double riveted lap..........................	Fig. 33	70
Triple riveted lap..........................	Fig. 34	75
Double riveted butt..........................	Fig. 35	80
Triple riveted butt..........................	Fig. 36	87

Figs. 17 and 30 illustrate the flexure which double butt-joint designs are intended to remove. They diminish also the danger from grooving the plate at the joint, to be discussed hereafter under corrosion of the plate.

The tendency of large diameter rivets is to lower the area of the unperforated plate: small pitches have the same effect. The boiler joint requires to be water-and

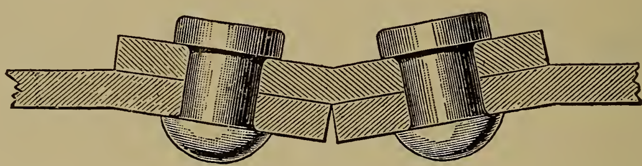


Fig. 30.

steam-tight as well as of best allowable strength. The following tables give data upon lap and butt joints from the practice of the Hartford Steam Boiler Inspection and Insurance Company and shows proportions which they recommend.

TABLE V. DOUBLE AND TRIPLE RIVETED LAP JOINTS.

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets.	Distance Between Rows of Rivets.	Edge of Plate to Center of Rivets.	Pitch of Girth Seam Rivets.	Efficiency of Joint.
			Double Riveted.			
						Per cent.
$\frac{1}{4}$	$\frac{11}{16}$	$2\frac{7}{8}$	$1\frac{15}{16}$	$1\frac{1}{8}$	$2\frac{1}{16}$	74
$\frac{5}{16}$	$\frac{3}{4}$	$2\frac{7}{8}$	$1\frac{15}{16}$	$1\frac{7}{32}$	$2\frac{1}{8}$	72
$\frac{3}{8}$	$\frac{7}{8}$	$3\frac{1}{4}$	$2\frac{3}{16}$	$1\frac{13}{32}$	$2\frac{3}{8}$	70
$\frac{7}{16}$	$\frac{15}{16}$	$3\frac{1}{4}$	$2\frac{3}{16}$	$1\frac{1}{2}$	$2\frac{7}{16}$	70
$\frac{1}{2}$	1	$3\frac{3}{8}$	$2\frac{1}{4}$	$1\frac{19}{32}$	$2\frac{1}{2}$	68
			Triple Riveted.			
$\frac{1}{4}$	$\frac{5}{8}$	3	2	$1\frac{1}{32}$	$2\frac{1}{16}$	77
$\frac{5}{16}$	$\frac{11}{16}$	$3\frac{1}{8}$	$2\frac{1}{16}$	$1\frac{1}{8}$	$2\frac{1}{8}$	76
$\frac{3}{8}$	$\frac{3}{4}$	$3\frac{1}{4}$	$2\frac{3}{16}$	$1\frac{7}{32}$	$2\frac{1}{8}$	75
$\frac{7}{16}$	$\frac{7}{8}$	$3\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{13}{32}$	$2\frac{3}{8}$	75
$\frac{1}{2}$	$\frac{15}{16}$	$3\frac{15}{16}$	$2\frac{5}{8}$	$1\frac{1}{2}$	$2\frac{1}{2}$	75

TABLE VI. DOUBLE AND TRIPLE RIVETED BUTT JOINTS.

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets.	Width of Outside Butt Strap.	Width of Inside Butt Strap.	Thickness of Butt Straps.	Distance between Rows of Rivets.	Edge of Butt Strap to Center of Rivets.	Pitch of Girth Seam Rivets.	Efficiency of Joint.
				Double Riveted.					Per Cent.
$\frac{5}{16}$	$\frac{11}{16}$	$2\frac{1}{4} \times 4\frac{1}{2}$	$4\frac{1}{2}$	9	$\frac{1}{4}$	$2\frac{1}{4}$	$1\frac{1}{8}$	$2\frac{1}{8}$	83
$\frac{3}{8}$	$\frac{3}{4}$	$2\frac{3}{8} \times 4\frac{3}{4}$	$4\frac{7}{8}$	$9\frac{7}{8}$	$\frac{5}{16}$	$2\frac{7}{16}$	$1\frac{1}{4}$	$2\frac{1}{8}$	83
$\frac{7}{16}$	$\frac{13}{16}$	$2\frac{15}{32} \times 4\frac{15}{16}$	$5\frac{1}{4}$	$10\frac{1}{2}$	$\frac{3}{8}$	$2\frac{5}{8}$	$1\frac{15}{16}$	$2\frac{1}{4}$	82
$\frac{1}{2}$	$\frac{7}{8}$	$2\frac{9}{16} \times 5\frac{1}{8}$	$5\frac{5}{8}$	$11\frac{1}{4}$	$\frac{7}{16}$	$2\frac{13}{16}$	$1\frac{13}{32}$	$2\frac{1}{4}$	80
				Triple Riveted.					
$\frac{5}{16}$	$\frac{11}{16}$	$3\frac{1}{8} \times 6\frac{1}{4}$	$9\frac{1}{4}$	14	$\frac{1}{4}$	$2\frac{1}{8}$	$1\frac{1}{4}$	$2\frac{1}{16}$	88
$\frac{3}{8}$	$\frac{3}{4}$	$3\frac{1}{4} \times 6\frac{1}{2}$	$9\frac{1}{4}$	$14\frac{1}{4}$	$\frac{5}{16}$	$2\frac{3}{16}$	$1\frac{7}{32}$	$2\frac{1}{8}$	87.5
$\frac{7}{16}$	$\frac{7}{8}$	$3\frac{3}{8} \times 6\frac{3}{4}$	$10\frac{1}{8}$	$15\frac{5}{8}$	$\frac{3}{8}$	$2\frac{1}{4}$	$1\frac{13}{32}$	$2\frac{7}{16}$	86
$\frac{1}{2}$	$\frac{15}{16}$	$3\frac{3}{4} \times 7\frac{1}{2}$	11	$16\frac{3}{4}$	$\frac{7}{16}$	$2\frac{3}{8}$	$1\frac{17}{32}$	$2\frac{1}{2}$	86.6

A considerable addition to the initial strength of a riveted joint, particularly when machine riveted, is given by the pressure of the plates upon each other and the friction between them. When the rivet head is quickly formed and the metal upset in the hole while still essentially red hot the shank in contracting draws the plates together with great force. This force must be overcome by a slipping of the plates over each other before the rivets begin to shear or the plate to tear. Experiment has shown that the shipping load for rivet on a double-riveted plate with $\frac{3}{4}$-inch rivets was 7 tons and with 1-inch rivets from 8 to 10 tons and over twice that required

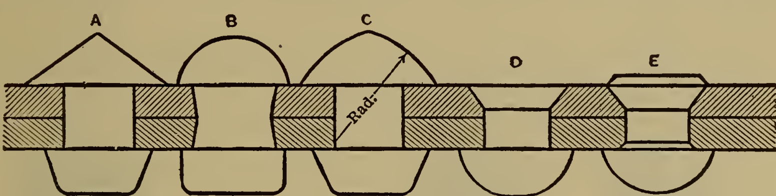


FIG. 31.

when the same joint was hand riveted. With the factor of safety of 6 referred to above, these slipping loads should not be reached in service. After slip has once occurred, however, and in old joints which have been loosened by flexure or corrosion of either plate or rivet, this extra strength can no longer be counted on. To give effective contact area for the plates and to leave plenty of sound metal outside of the holes both for strength and tightness, the first row of holes is usually put at least one and one-half times the hole diameter from the edge of the plate.

41. Arrangement of Rivets in a Joint. Special Joints. Fig. 31 shows the conventional types of rivet in section. *A* has the conical head made by hand, the bottom showing the usual pan tail which the

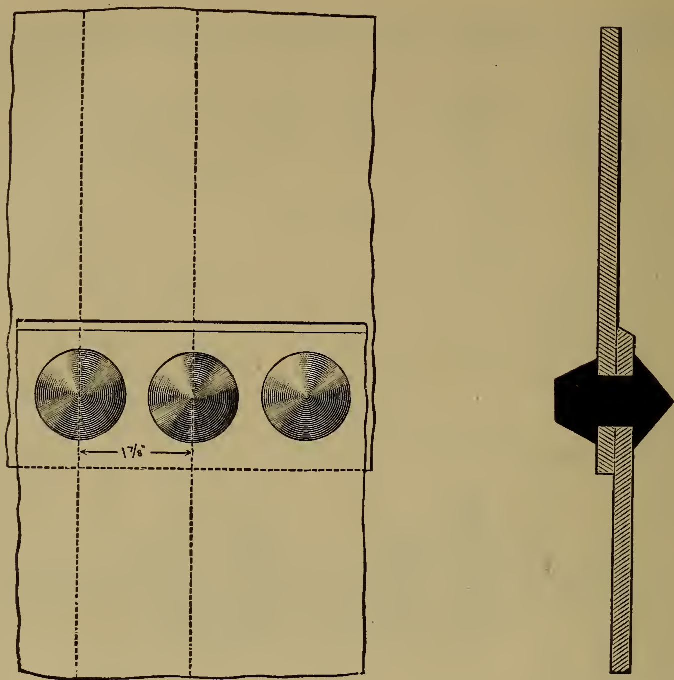


FIG. 32.

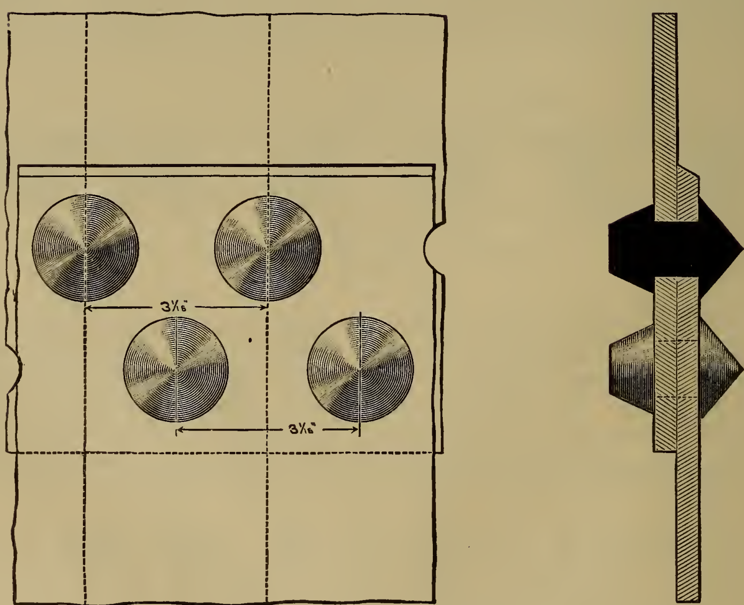


FIG. 33.

rivet has as manufactured. *B* is the cup or button head resulting from the use of a swage. *C* or *A* will represent forms given by the dies of riveting machines. *D* is the countersunk head usual in ship work or where a smooth skin surface is to be sought.

Rivets of iron boilers should be of iron, and for steel boilers they should

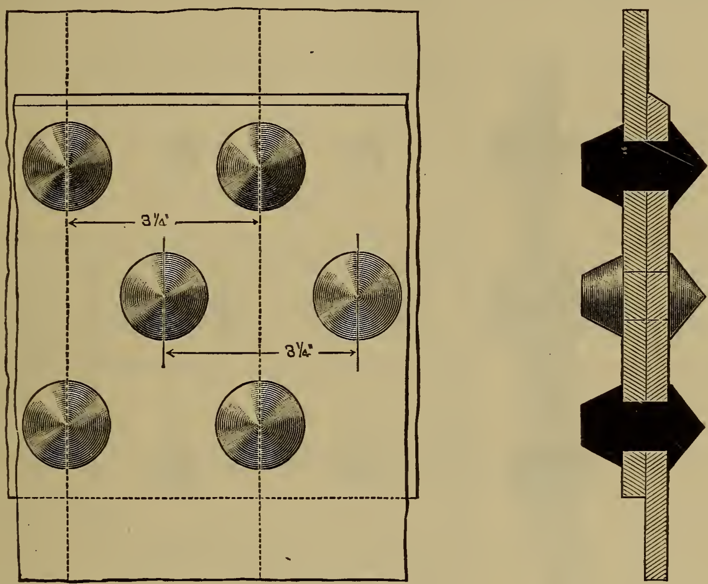


Fig. 34.

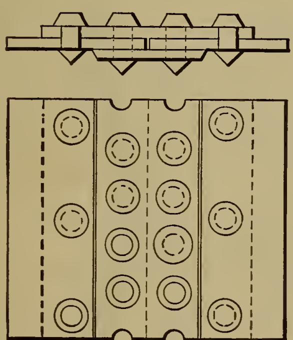


Fig. 35.

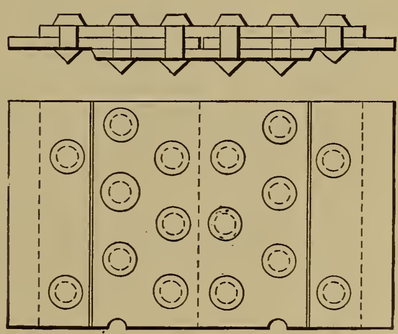


Fig. 36.

be of a mild steel able to stand the proper forge tests. Their tensile strength is usually taken the same as the shearing strength (or $\frac{9}{10}$ of it), and may be put at 55,000 pounds to the square inch for steel. The forge tests are:

(1) Bend double close when hot.

(2) Bend to a U over a bar of its own diameter cold, and show no cracking in either case.

(3) The head should hammer hot to form a disk $2\frac{1}{2}$ times the diameter of the shank.

(4) The shank should hammer cold to a flat $\frac{1}{8}$ of an inch thick, and

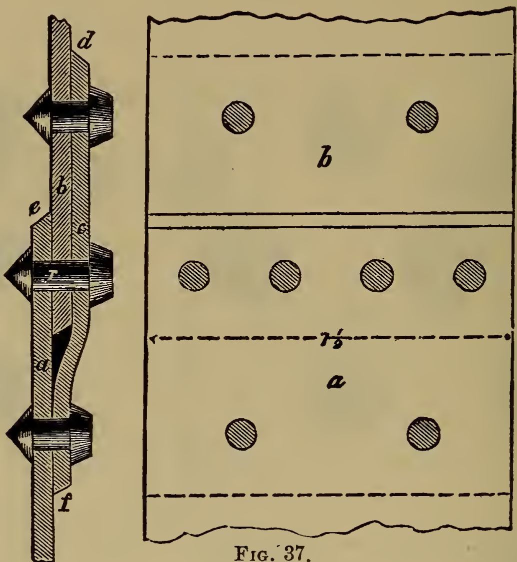


Fig. 37.

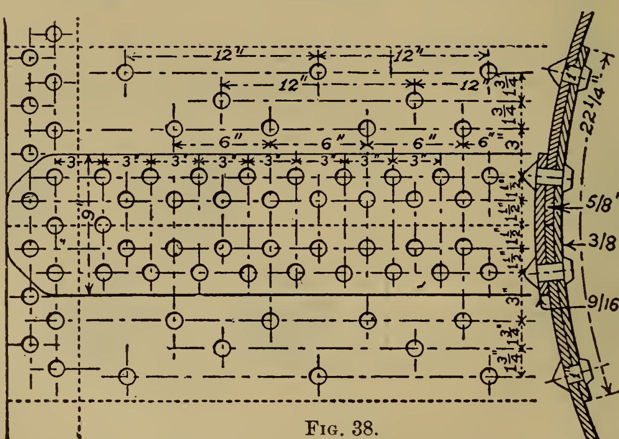


Fig. 38.

then withstand punching with a solid punch, making a hole of the size of the original shank; both of these without cracking or splaying at the edges.

For wrought-iron rivets the tensile strength may be called 50,000 pounds per square inch, and the shearing strength 40,000 pounds; the metal should withstand the same forge tests.

The arrangement of the rivets in a boiler joint will be either ordinary or special. The ordinary riveted joints are presented in Figs. 32, 33, 34, 35, and 36.

In chain riveting the two rows are one behind the other in line (Fig. 35): in staggered riveting the intermediate row is placed opposite the spaces in the other two (Figs. 34 and 36). The triple-riveted joint (Fig. 34) gives a longer lap and more plate area and more rivet-shearing area, as well as a stiffer joint. The butt joint with double cover (Fig. 35) doubles the number of rivets required as compared with the same class of lap joints, but the strain is in line and without tendency to flex the plates, and the rivets are in double shear. It is not so with the butt and single cover (Fig. 30). The double-cover butt (Figs. 35 and 36) is liable to have the outer cover overheated when exposed to fire. The special joints are departures from the four conventional

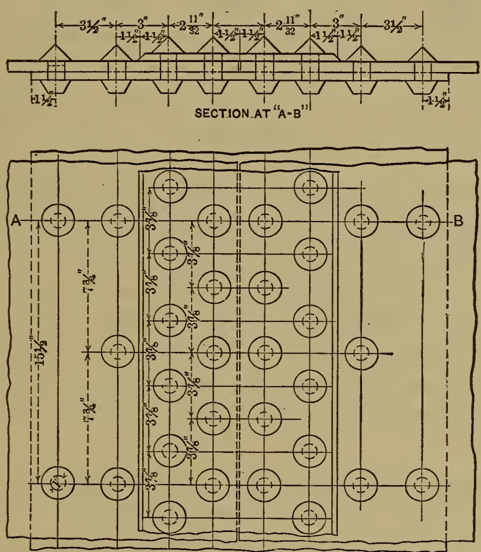


Fig. 39.

types, seeking to secure the advantages of the double butt with greater or less expense. Figs. 37, 38, and 39 will serve as types of such joints. The strength of single lap joints being from 55 to 60 per cent of the original plate, and of double riveted laps 70 per cent, such special double butt joints as Fig. 38 will show a strength of 85 per cent of the solid plate. The joint of Fig. 39 with $\frac{15}{16}$ rivets in one-inch holes in $\frac{3}{8}$ plate would be called a quadruple riveted butt joint and has an efficiency of 93.5 per cent.

42. Failure of the Riveted Joint. While the riveted joint fails in one of two generic ways, either by shear of the rivets or by failure of the plate, the latter may occur in several ways. The rivet may (*c*) buckle or (*e*) shear the plate in tearing its way out, or the plate may crack and tear (*b*) either between rivets or (*d*) between the rivet and the edge of the plate. The excess of rivet area to secure tightness for the seam usually makes the failure occur in the plate (Fig. 40). The danger from (*c*) occurs when hard steel rivets are used in soft iron plate; (*e*) may happen when the line of rivet holes is too near the edge of the

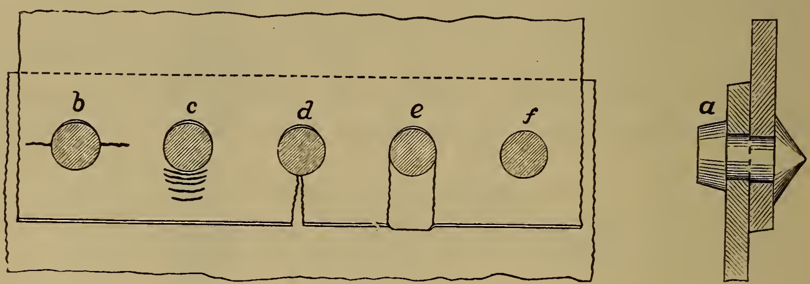


Fig. 40.

plate and the rivets are hard and dense. It is the least usual. The failure (*b*) is most common. The line between rivet holes along the pitch is the shortest line or line of least resistance, and any maltreatment of the plate in making the joint has tended to make it weaker. Such maltreatment may come from punching without reaming or annealing, whereby the steel is more brittle and less tough than it should be (paragrafs 27 and 34); or if the use of the driftpin has been permitted, the metal has been initially strained locally thereby beyond its elastic limit.

43. The Driftpin is a tapering pin of hard and tough steel which is used to force and draw into coincidence two holes in a seam which have not come opposite to each other. (Fig. 21.) The taper pin is inserted in the half-blind holes and driven downward, so that it wedges the projecting edges of the holes over and draws the metal around the holes out of shape until the distorted holes agree. This will buckle the metal in front of the hole if the error in alignment is at right angles to the pitch, and cause failure (*c*), as well as strain the metal along the line of the pitch and start the crack which ends in failure (*b*). If the error in alignment is along the line of the pitch, the driftpin tends to start failure (*d*) and injures the plate between rivets, which renders it liable to failure (*b*) also. The driftpin is fatal to good metal in steel boilers, and its use should be forbidden by the specifications. If holes must be expected to be inaccurately spaced, the coincidence should be brought about by use of a cutting reamer, whereby no injury to the material is incurred. Drilling the holes in place makes both drift and reamer unnecessary.

The failure of the joint by gradual action of overpressure by methods (*c*) and (*e*) is apt to show itself by leakage before it is imminently dangerous. Inspection may also reveal failures (*b*) and (*d*) if they are not the result of some sudden strain. It is an element of safety in the riveted joint that it should give warning of its probable failure by the leakages which accompany the first stages of such failure. Old seams may fail from corrosion or grooving by other methods than these, determined by the character of the deterioration which has weakened them (paragrafs 191–6). But except where the solid plate is weakened by corrosion or grooving, such wear and tear is most apt to hasten a failure at one of the four weak points above discussed.

44. Stays and Staying. It has been seen (paragraf 25) that the sphere and the cylinder are the only forms which have no tendency to change shape under internal pressure. Or, in other words, that the circular is the limit form towards which all sections tend, and which they will assume if the elasticity of the material will permit such deformation of section to occur without breaking. The modern tendency in design is toward the use of many units of small diameter so that the concave or convex heads or ends of such tubular units under pressure shall be stiff enough without any bracing whatever. In the older forms of large external diameter, and particularly where the furnace element was placed inside the external shell, flat or nearly flat surfaces were numerous and their tendency to deform under pressure must be resisted by positive means other than the tensile or transverse strength of the material. Rods, bolts, bars, or braces used to prevent such deformations of flat or arched or non-circular surfaces are called by the general name of "*stays*."

The simplest case is where two parallel surfaces, flat, or parallel with one concave and the other convex to the pressure and not far apart from each other, are to be tied together to resist the pressure between them which tends to force them apart. This occurs at the sides of the fire box in locomotive boilers, in some marine and upright boilers, and at the crown sheets of some designs of locomotive boilers. The most ready solution is to tie the two surfaces together by round bolts or rods whose area of cross section shall be sufficient to resist the pressure upon the area they support, and for which the distance between centers shall be so small that no deflection or bulging of the plates can occur between them. With thinner plate this center distance used to be four inches in locomotive practice; recent practice raises this distance to six inches or over. These stay bolts are either headed over hot on the outside of the two plates, like an ordinary hand-made rivet, or more usually the holes in the two plates are threaded and the stay bolt is screwed into both plates, and is slightly upset on the ends when in place, to prevent working out and leaking, and also to reënforce the strength of the threads. Thicker plates give sufficient length of thread for strength. Hollow stay bolts are also used on the water legs of locomotive boilers, both because they will manifest the beginnings of failure by leakage of steam through the crack of the initial fracture, and because the air which goes through the hollow keeps them cool and helps supply oxygen for the fire. The simple heading of the stay bolt like a rivet was troublesome on account of the tendency of the shank to bend and also because an unequal contraction of the group of stay bolts made some too tight and others loose. The holes for the screwed stay bolt are tapped by a long tap so that both plates have their threads parts of the same screw, and the two plates are under equal tension if all bolts are of the same length. This method is most satisfactory if the stay bolts are short and of equal length. As they heat by contact with the steam or hot water they lengthen and slack their hold, and the longer they are the more they yield. Hence, while this same method can be used to stay the two flat cylindrical heads of a cylindrical shell boiler to each other and prevent their bulging outward, it is usual only for boilers of comparatively short length, such as are used in marine practice, or unless the pressure is to be so high that no other plan seems advisable. Such "through stays" will be of round rods of sufficient size, threaded at the two ends, which have been upset so that the bottom of the thread on the enlarged ends shall have a diameter equal to that of the body of the rod. The hold of the rod in the plate by its thread is reënforced by a nut on the outside which caps over the end of the rod, and a flexible copper washer between the plate and the nut helps to make the joint water-tight. A jam nut on the inside with a washer helps to keep all snug, and prevents working loose by expansion and contraction. Since through stay rods of this type cannot usually be put close together, but must be spaced far enough apart to allow a man to pass between them for inspection and for work, their centers will be sixteen inches apart at least; so that they must each withstand the pressure in such case exerted over an area of 256 square inches, and it is usual to stiffen the head by means of angle- or channel-irons or similar structural shapes, whereby the holding power of the stays shall be distributed over the more flexible head. This can also be done by large washers on the outside of the head. Figs. 41 and 42 will show the detail of such through stays.

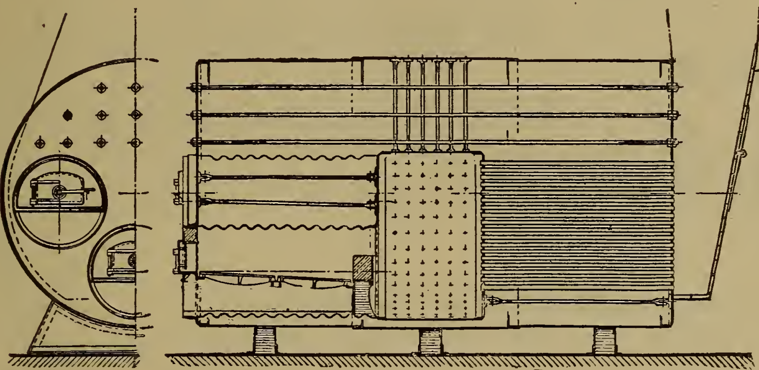


FIG. 41.

To avoid the threaded hole in the heads and the projecting end of the stay, the stay rod has been fastened to stay bars on the heads by a pin-joint. Fig. 43 shows a form of this method where the stay bars are

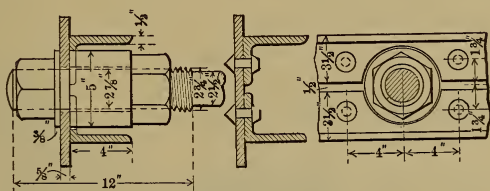


FIG. 42.

relatively heavy forgings of two to two and a half inches square, with lugs bump-welded on the inside. The stay rod ends in a fork which spans the lugs, and a bolt or pin connection ties the bars together upon the two heads. Somewhat lighter than this is the similar arrangement of Figs. 44, 45, and 46, where angle- or tee-irons are riveted to the head, and the stays pinned to them by pins in single or double shear; but in these arrangements the obliquity of the stay rod indicates a prevalent arrangement for medium pressures. The inner end of the stay rod in this case is fastened to the cylindrical shell at a convenient distance back from the head by rivets, and thus the bulging tendency is withstood by the tensile strength of the shell lengthwise, in which direction it is abundantly strong. Instead, again, of structural iron bars, single or independent sockets may be used, as in Figs. 47 and 48, whereby the action

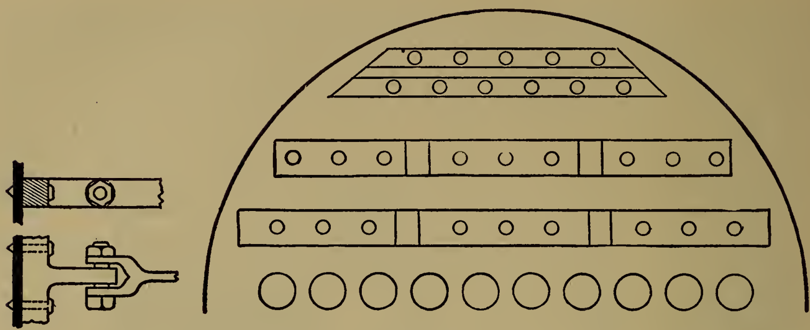


FIG. 43.

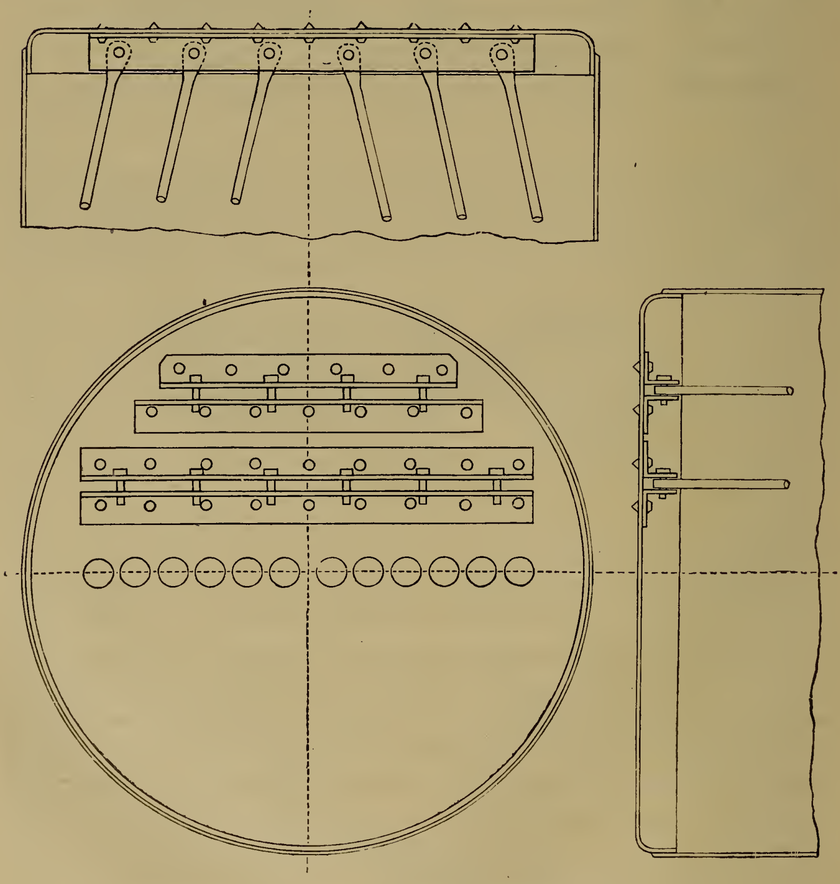


FIG. 44.

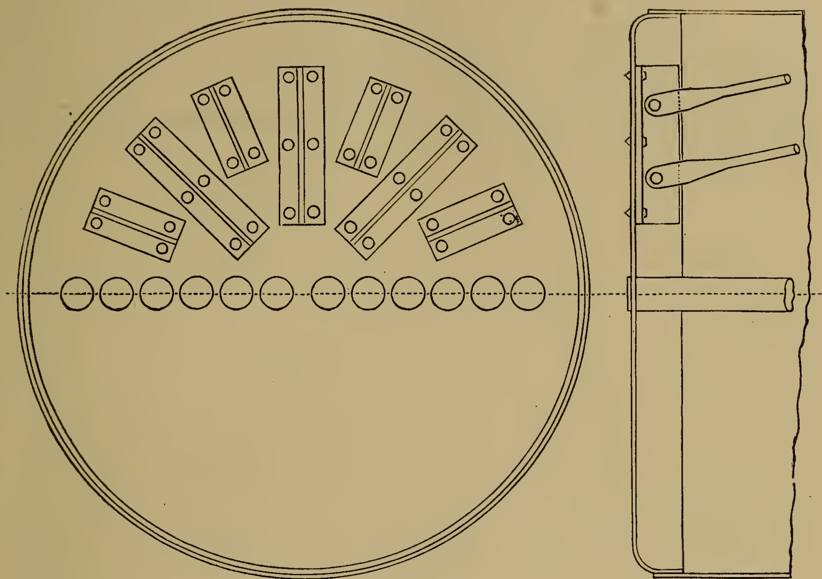


FIG. 45.

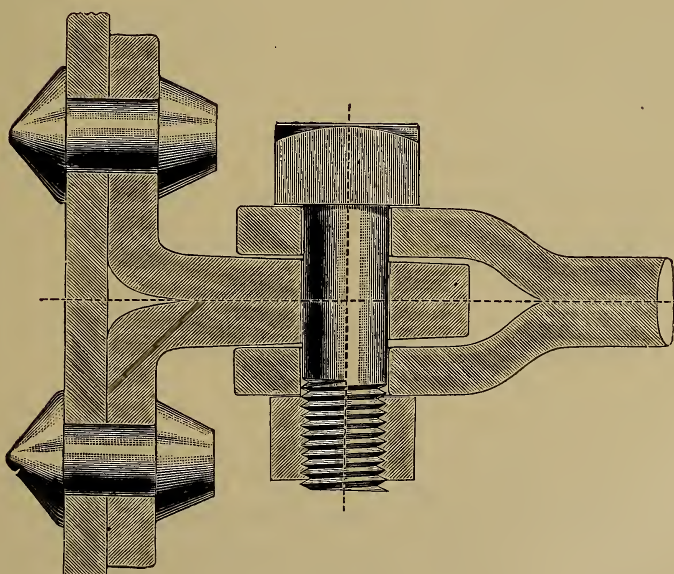


FIG. 46.

of the stays is distributed even more generally over the surface to be stayed. Fig. 46 will serve for detail of these also. It is apparent that the diagonal stay must be stronger than the straight one to withstand the same strain (Fig. 50). The cheapest and most uncertain of the diagonal stays is the plain rod, flattened at both ends as Fig. 47 is at

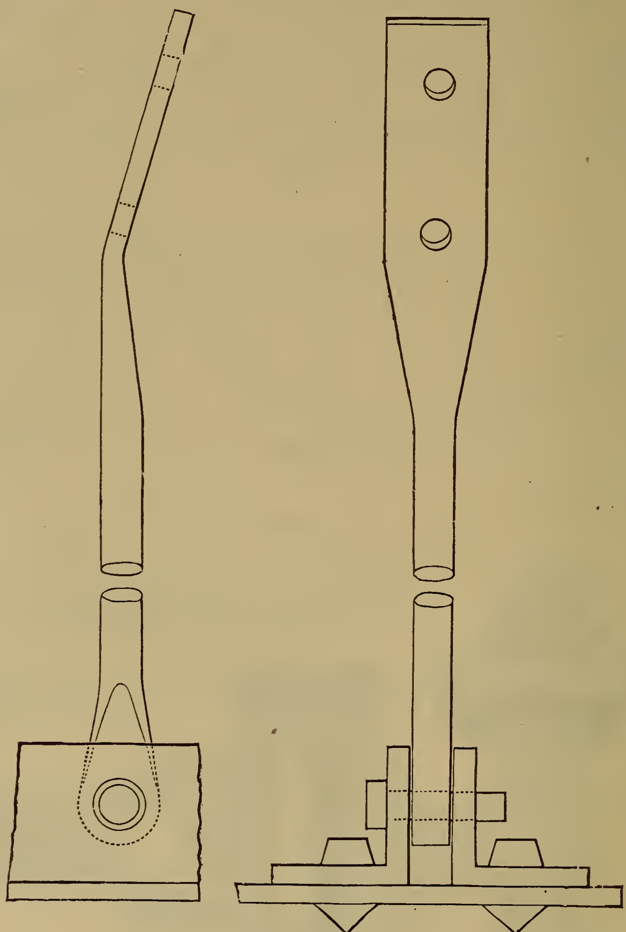


Fig. 47.

one end, and riveted by such flattened ends to head and to shell. Modern practice limits the use of this type to comparatively light pressures, and prefers the form of molded steel forged up without welding from steel plate to the welded types. Fig. 49 shows some of these forms. Such stays are strong as they approach parallelism to the shell for they must withstand a stress which is related to the straight or normal pull on the area which they support, as the length of BC in Fig. 50 is greater than AB.

Gusset-stays are a form approved for heavy pressures. Triangular or trapezoidal pieces of boiler plate are riveted to angle irons on the head and cylindrical shell and bind them into a rigid structure or the plate of the stay is turned up at right angles to form a flange which

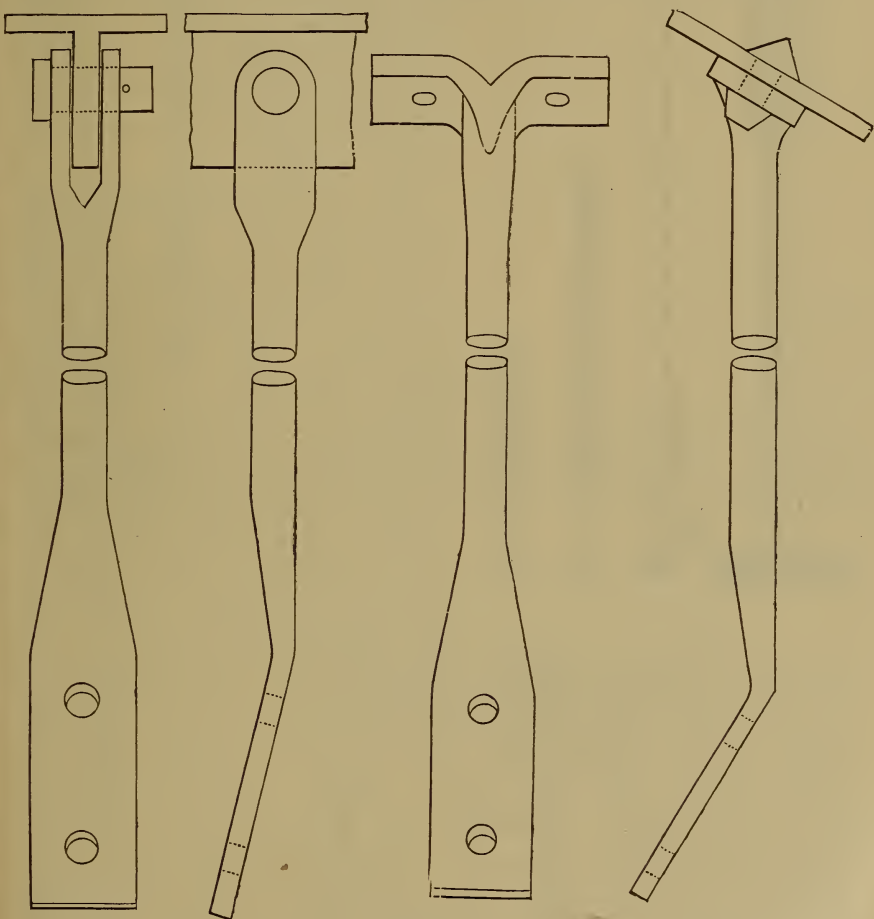


FIG. 48.

is then riveted to head and to shell. The stays do not come close to the corner of head and shell, so that in cutting away the heel of the right-angled triangle the fourth side may become parallel to the hypothenuse of the original triangle. The stays are usually placed radially upon the head. (See Figs. 98, 105, and 115.)

Where a flat surface has no surface parallel to it to which it can be directly stayed, and the length is too short for wise use of diagonal bracing, the surface must be made stiff by bars acting by their stiffness like girders to prevent deformation or collapse. This is met in the flat crown sheets of locomotives and in combustion chambers of marine boilers. The problem is complicated by the intense heat upon such surfaces, which precludes the use of solid bars, which would keep water from the metal.

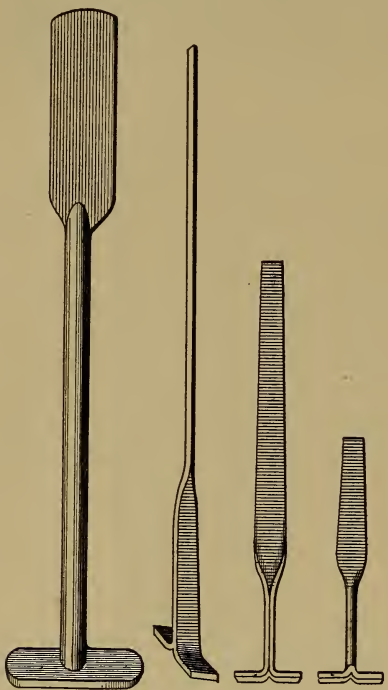


Fig. 49.

The crown-bar method is shown in Figs. 112 and 114, in which the bars appear in pairs, running across the flat sheet from side to side of the furnace. The sheet is stayed to these bars by $\frac{7}{8}$-inch bolts which pass up through the plate and between the two bars of each pair. The joint between head and plate is made by a copper washer, and the washer under the nut serves to bind the bars together. A taper washer or distance piece keeps the bars from the plate, so as to cause water to touch as much plate as possible, and keep the plate flat when the bolts are tightened. The deflection of these bars is prevented by sling stays when they are long, or their own resistance to bending is depended on if they can be short. Figs. 51, 52, and 114 show other methods, used either where the crown sheet is arched to approach parallelism with the outer shell, or where the outer shell is made flat to become parallel with the flat crown sheet. The stay bolts have taper surfaces under their heads, which draw into tapering reamed holes in the sheet by the pressure of the steam, and copper washers under the head help to secure tightness. Fig. 52 is called the Belpaire fire box. (See also Fig. 115.)

Staying should not be too rigid, and it is very objectionable to have a flexible and a rigidly stayed surface attached to each other. The motion of the flexible part either from heat expansion or by pressure produces a great strain or a concentration of the deformation at the margin where these tendencies to move and to resist motion meet. Tests have shown that in locomotive fire boxes where the difference in temperature of the outside and inside plates may be considerable

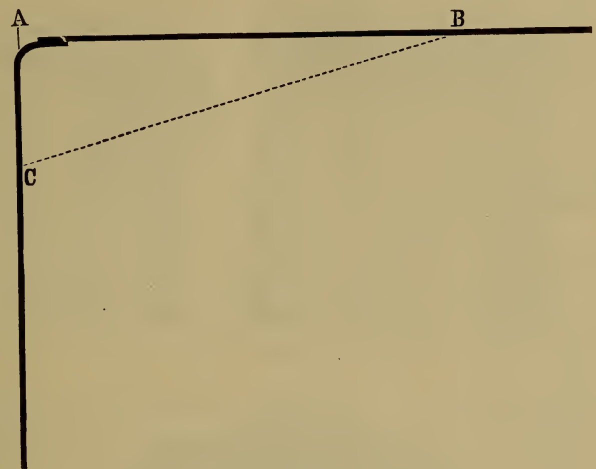


Fig. 50.

the expansion of the latter brings so serious a cross stress or flexure upon the rigidly held bolts as to exceed their elastic limit. Ultimately therefore by repeated stress these bolts fail close to the thread, and the

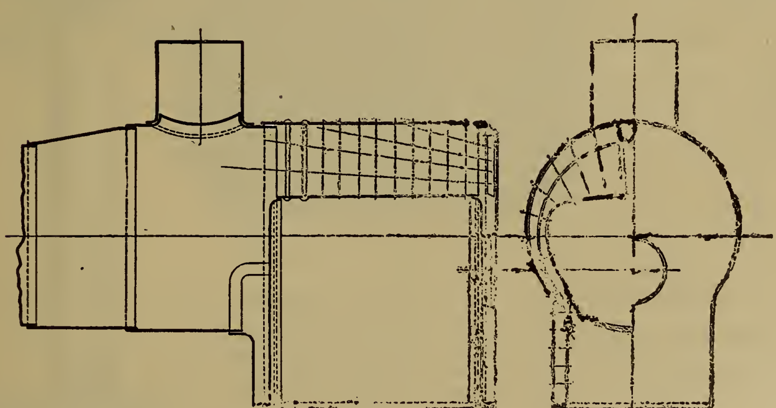


Fig. 51.

plate bulges and leaks, or the plate cracks. To eliminate these difficulties and allow lateral motion while resisting deformation in the direction of greatest stress, forms of flexible socket connection have been devised of which one is shown in Fig. 53. The end of the bolt is spherical headed, fitting in a cup. This cup screws into the outer plate from without, and a cover cap makes all tight against leakage. The angle of the bolt can vary through more than the necessary range

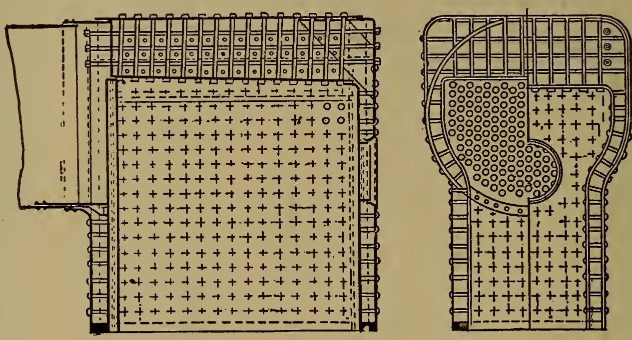


FIG. 52.

without cramping the bolt. The other end screws into the inner plate and is headed over, and a simple flexure of a beam fastened at one end takes place instead of the complicated flexure and shearing which must appear when both ends are rigid. When motion is restrained at corners as at the joint of shell and head, or of a rectangular fire box the phenomena of grooving are apt to occur where the flexure of the plate is greatest.

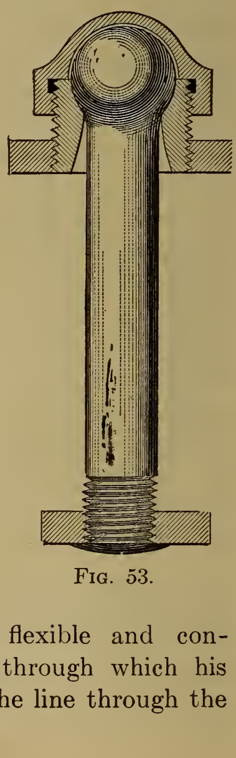


FIG. 53.

45. Manholes. In the construction of riveted shells a provision must be made to allow the helper to get out who has "held up" for the final riveting of the last joint. Access must also be had to the inside of the boiler for inspection when in service and for repairs. The function of this hole is thus to let a man in and out, and is for this reason called the manhole. It should be as small as possible to effect its purpose, because the metal of the shell removed to make it is equivalent to just so much strength removed from the boiler. Measurements show that the average man is fourteen inches on the axis of the longest dimension through the articulations of the hip-joints with the pelvic bone. The shoulder dimension, though naturally larger, is flexible and contractile, and any man can pass through a hole through which his hips will pass. The dimension at right angles to the line through the hip-joints is normally less than the other, and is a flexible one when it is not less. Hence the manhole receives an elliptical shape with its long axis 14, 15, or 16 inches long, and its short axis 9, 10, or 11 inches, or four or five inches less than the other. This elliptical shape has furthermore a very practical advantage, in that the lid which is to cover the hole and must have a size larger than the hole, so as to lap over the edges, can be made to fit upon the inside of the hole and can yet be itself passed through the hole from without. The lap over the edges must be less than one-half the difference between the long and short axes of the elliptical hole. If the hole must be circular, the lid has to be external.

The lid is held to its seat over the manhole when internal partly and mainly by the pressure upon its inner side; but to make the joint steam-tight, and to hold it from displacement at other times, the lid has one or two studs symmetrical upon its long axis, which pass up

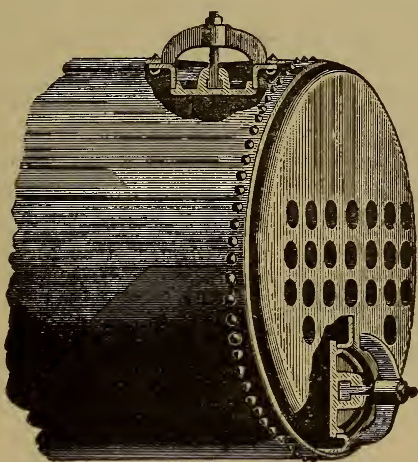


Fig. 54.

through a proper hole or holes in a bridge or "dog" of cast or wrought iron which spans the manhole opening, so that when the nut is screwed down upon the stud and bears on the outer surface of the dog, the lid is drawn to its place and held firmly by the nip of the dog upon the edges of the hole (Fig. 57). The joint between the lid and the plate is made tight by a gasket of rubber or asbestos board or similar material whose compressibility shall compensate any inaccuracy of contact surfaces. It is rare that finish of surfaces can be secured or maintained which will make a true metal-and-metal joint without gasket under the conditions prevailing around a manhole. The hole cut in the plate of the boiler leaves the strength less than when the metal was solid, at the zone whose width is the span of the hole. All the circumferential strains in the ring of plate are transferred around it till at the edge of the hole they are balanced by no counteracting force except that supplied by the reluctance of the material to split into filaments by yielding sidewise. The tendency can be illustrated by a band of elastic material like rubber with a hole punched in it. Under strain lengthwise the hole becomes deformed, and most so at the ends or at the

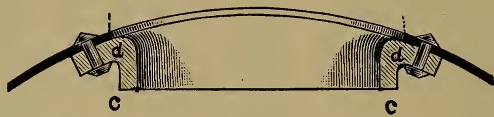


Fig. 55.

points farthest from the solid material at the sides of the hole. Hence it is desirable not only to place the hole, if in the shell, with its short axis lengthwise, but also to reënforce the weakened plate around the edge of the hole; and this practice has given rise to manhole mouthpieces or nozzles. The simplest form is a forged ring of wrought iron (more desirable than a similar ring of cast iron) riveted around the edge of the hole in the plate. The lower surface will be plane to form the flat seating for the cover, while the upper surface conforms to the shape of the boiler. The rivets are in countersunk holes on the face of

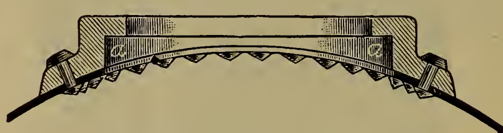


Fig. 56.

the seating, or else the ring is broad enough to allow the line of rivetheads to come beyond the lap of the cover. Such ring resists the tendency to flex which will occur when the manhole is upon a cylindrical surface and metal has been cut away which would maintain the shape when the pressure came upon the continuous ring of plate. Fig. 55 shows the ring made of a flanged plate riveted within the shell, and Fig. 56 the exterior nozzle arrangement. The interior seating offers some advantages from the resistance to flexure which it gives. Fig. 57 shows a full detail of the manhole with seating and lid of cast iron, and Fig. 58 the more approved steel seating and lid. For greater security in large holes the dog and bolt may be doubled, one on each side of the short diameter.

The location of the manhole will be either upon one of the heads, or upon the head of the dome of the boiler, or upon the shell, or upon that attachment called the mud drum. On the cylindrical or spheroidal

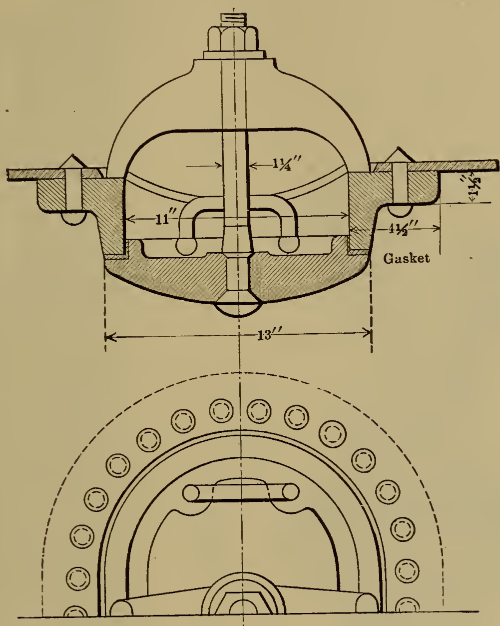


FIG. 57.

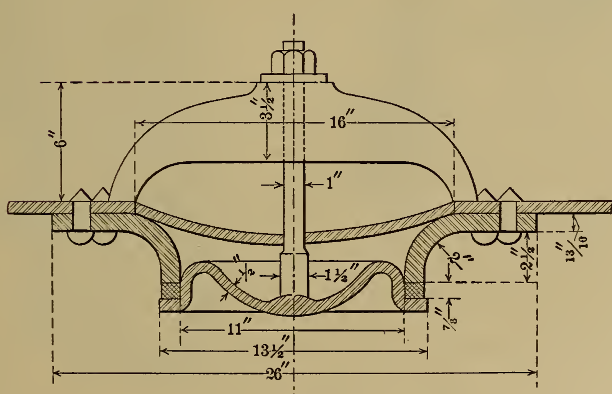


FIG. 58.

surfaces of shell or dome or drum, seatings or nozzles are a necessity, to secure planes for the covers to seat themselves upon; on flat surfaces they are desirable for strength and stiffness. The construction of the boiler may require more than one manhole, a condition frequent in marine practice.

46. Hand-holes, as their name indicates, are smaller openings in the shell to give access to the hand for an inspection by touch, or for convenient cleansing or minor repair. The construction is the same as for the manholes, the reënforce or seating being of boiler plate or a flat ring of wrought iron. Their location and number will be determined by the design of the boiler in order to serve their purpose and leave no corner which inspection cannot reach. Fig, 59 shows a typical hand-hole structure.

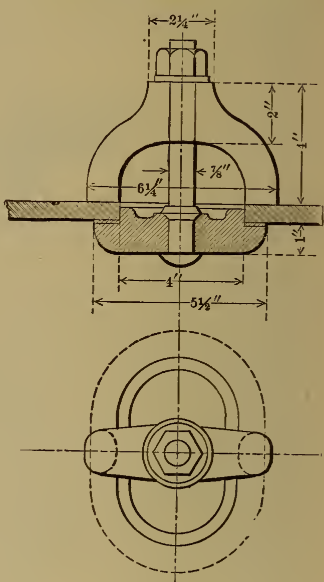


FIG. 59.

47. Edge Planing and Calking. The shearing of the steel plates to size has left an edge or selvage of metal which is brittle and unreliable from the effect of the shearing plates. This deteriorated metal should be planed away by a cutting tool. Fig. 60 shows such an edge planer for plate, the sheet being held by the clamping screws as in a vise, while the tool traverses along the edge. It is convenient to give the edge a bevel in planing, which is not only of service for appearance' sake, but gives an edge at the lap or joint to be used in calking the seam.

It is too costly to finish the flat surfaces of the plates where they lap so as to make a joint which would be steam- or water-tight under the compression of the contact areas by the rivets. There is always some scale or roughness from the hot rolling process and the later bending and flanging. Hence the seams must be calked, from within or from without. It is usually easier to do it from without and the process is that of upsetting the lower edge of the bevelled sheet into the joint by means of a round-nosed chisel held against the edge and struck with a hammer. Fig. 61 shows the method of calking with a round-nosed tool, which is much to be preferred to the sharp-nosed chisel, although the latter is easier to use. The sharp corner of the sharp tool may indent the lower plate at the joint, and thus start the first crack whose ultimate consequence will be the weakening of the plate at that point, which the illustration suggests.

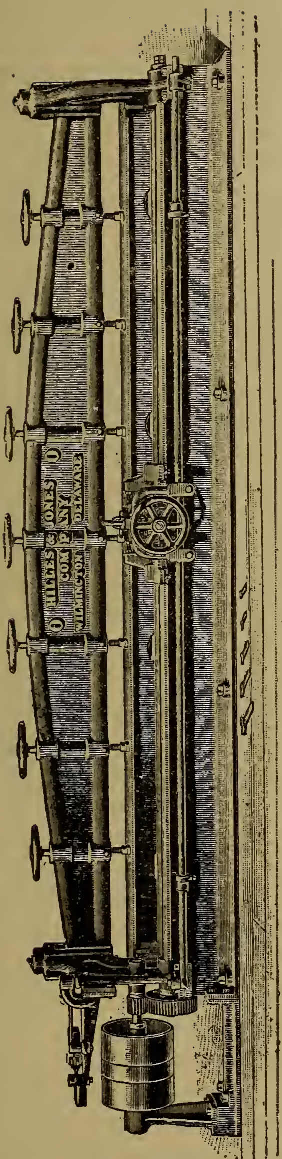

HILLES & JONES
COMPANY
WILMINGTON DELAWARE

Fig. 60.

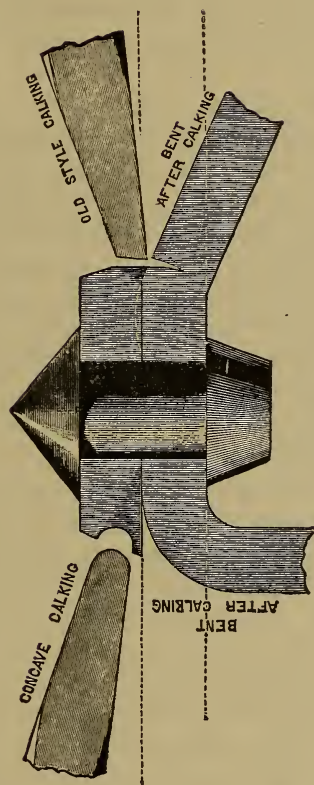

OLD STYLE CALKING
BENT
AFTER CALKING
CONCAVE CALKING
BENT
AFTER CALKING

Fig. 61.

48. Disengagement Area. Water Space and Steam Space. Following the analogy of the tea-kettle as the prototype of the modern boiler in cylindrical form, how full of water is it desirable to keep it when at work making steam? The more water it holds the more heat stored and the less the fluctuation of pressure range. If too full of water, the free surface of contact of boiling water and steam gas grows less and less, and the volume of stored steam is less, with attendant tendency to fluctuation of pressure. This surface will be a maximum in the cylinder when the water stands at the level of the horizontal diameter. To put it lower than this makes it difficult to prevent overheating of the shell, because the hot gases are likely to come in contact with elements of the cylinder which are not water cooled. Hence the water will normally be at a higher level than this. How much can the free surface be safely diminished by carrying the water higher than this?

If too much steam must escape from too small an area, the rising steam gas keeps the surface of the water bubbling and frothing. The steam lifts the top layers of water, so as to give a fictitious indication of level of water, and entrains with it a proportion of water in drops or mist or even in some mass in its too rapid flow from the surface of the water to the outlet pipe. Such foaming or frothing occurs when the boiler is hard pushed and there is oil or floating scum on an alkaline water. When the foam becomes solid water, and such water passes over into the steam pipe, the boiler is said to "prime." The presence of a scum of dirt or of grease increases the tendency to foam and prime, because the steam gas forces its way out by bursting through this scum, but to carry the water level too high is not only to bring the disengagement surface nearer to the pipe outlet for steam, but it is also to diminish its area. A boiler forced to evaporate faster and disengage more steam than at its normal rate will also be likely to foam and prime.

The accepted standard for early practice was to make the volume filled with water (called the water space) to be two-thirds of the volume of the boiler, while the space in which steam is confined above the water (called the steam space) should be the remaining one-third. This was reached by making the area of the head to be divided by the water line into segments whose area was as 2 is to 1.

This was an empirically correct ratio, based on observation, but lacked a rational basis, because the real store of steam is not in the steam space, but in the heated water. A more satisfactory basis is the disengagement area basis, and experiment has shown that when the flow of steam from the water into the steam space is at a rate such that it would fill the steam space three times a minute, the disengagement was slow enough to give no trouble from priming. This experiment was made on a marine boiler, and trouble was found from entrained water when the evaporation had to be so rapid that the steam space was filled five times per minute; at four times per minute trouble was occasional but not continuous. Stated otherwise, a linear velocity of flow of steam faster than 2 feet per second through the water surface will entrain water with the steam. The larger the cylinder volume to be filled per stroke, or the greater the number of strokes per minute, or the greater the volume of steam required per minute, the larger the aggregate steam space required if pressure is not to be allowed to fluctuate when the disengagement rate is normal and slow enough to prevent priming.

49. Domes and Steam Drums. The difficulties in the engine cylinder from entrained water will be hereafter referred to (paragraf 259), and are so serious and important that special pains must be taken to prevent priming or foaming as a continuous process. This is done by so reducing the rate of flow of steam at its intended entry into the steam pipe that time and opportunity shall be given for water entrained by the outrush of steam from the disengagement area to settle back within the boiler by gravity before such water acquires the greater velocity in the pipe itself. An enlargement in the cross section of the pipe will permit such lowered exit velocity and this is secured by the dome or steam drum. The dome which appears on most massive or shell boilers is an upright cylinder of boiler plate of some considerable diameter, up to two-thirds that of the shell and so attached to it as to form part of the steam space or an addition to it. In horizontal boilers it will have its axis at right angles to that of the boiler, and will be attached to it by flanging the sides of the dome outward, and curving the sides so as to fit the curvature of the top of the shell (Fig. 65). The shell is cut away under the dome in the type form to the full diameter of the dome. The steam outlet will be a pipe passing from the top of the dome or near it.

It will be apparent that the dome accomplishes three purposes:

(1) It removes the inlet to the steam pipe farther from the disengagement area than it could be if it were on the shell directly; water is less likely to spatter or be projected into the steam outlet.

(2) The linear velocity of steam is low in the large cross section of the dome. Entrained water is less likely to be carried at low linear velocities than high, and it has time to separate out from the steam by its greater specific gravity.

(3) The steam flows to the dome from a larger proportion of the disengagement area than it would to a small neck or nozzle on the shell. Under a neck or nozzle the water is heaped up (Fig. 5), and the greatest disengagement occurs at that point, a condition favorable to priming.

The dome is usually put at that point on the length of a boiler at which experience shows the disengagement to be most active, so as to avail of this action, and to prevent injury caused by a disregard of the tendency there.

The objections to the dome are the weakening of the shell from the cutting away of the metal under the dome, whereby not only is the strength affected, but the cylindrical shell tends to flatten under the pressure, and this results in leakage at the dome seam at the top. The shell is an unstiffened curved stay where the hole is cut, and the double thickness of the lap of the dome joint does not replace the strength of the unbroken cylindrical surface. Hence the dome joint is further

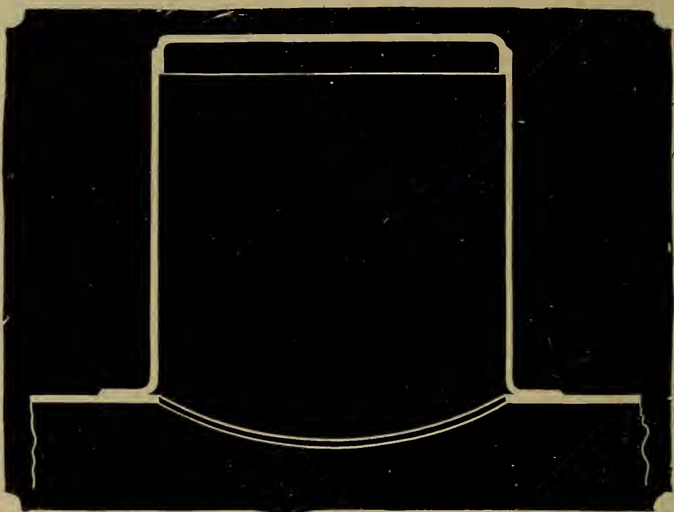


Fig. 65.

stiffened, either by turning up the plate into a vertical flange, or by a stiffening ring, as was described in manhole seatings (paragraf 45). Fig. 66 illustrates these methods.

This objection to the weakening of the shell by the hole for the dome has induced designers to seek to secure the functions of the dome without such cutting of the shell.

(1) The shell has been perforated with either many small holes or one larger one under the dome, but not cut away entirely (Figs. 67 and 68). The area of the holes should aggregate several times greater than the area of the pipe. The objection to this is that the tendency to straighten is not removed, because the pressure in the dome balances the pressure below the perforated surface, and there is no tendency to keep the

cylindrical shape. The plate acts like a curved stay only. Moreover, the dome cannot be used as a means of entry to the boiler, unless the hole in the shell is the full size of a manhole, and the top of the dome is a convenient place to enter and to place the manhole.

(2) To attach the dome by a neck (Figs. 69 and 72). The flanges of the neck return some strength and stiffness to the shell. It is more convenient, if this is to be done, to make the dome a horizontal drum (Fig. 73).

(3) To use a horizontal drum or pipe of large diameter overhead, to which the boiler will be connected by a neck if but one boiler is used. This plan is specially convenient where several boilers are side by side

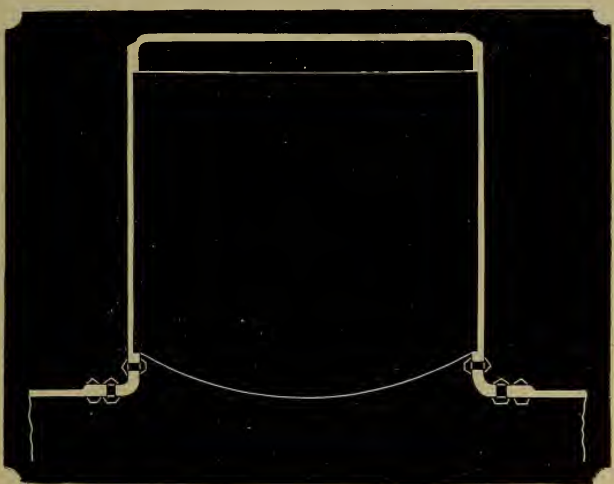


Fig. 66.

or in a "battery," as it is called. All can deliver into a common drum, and from this a drainage connection may remove any entrained water which may be carried through neck or nozzle by high velocity of steam currents. Figs. 69 and 73 will illustrate this arrangement when the drum is transverse, and is really a large pipe merely jointed to each boiler by piping which allows of expansion without cross strain. (See also Figs. 137, 148 and 189.)

(4) The use of a dry pipe with perforations. Figs. 71, 98, 106, 122 and 123 will show this arrangement. The steam leaves the disengagement surface to pass into the steam pipe through a number of small holes in the interior pipe which is a prolongation of the steam pipe inside the boiler. The gentle current into each opening prevents entrainment of water, because the aggregate area of openings is in

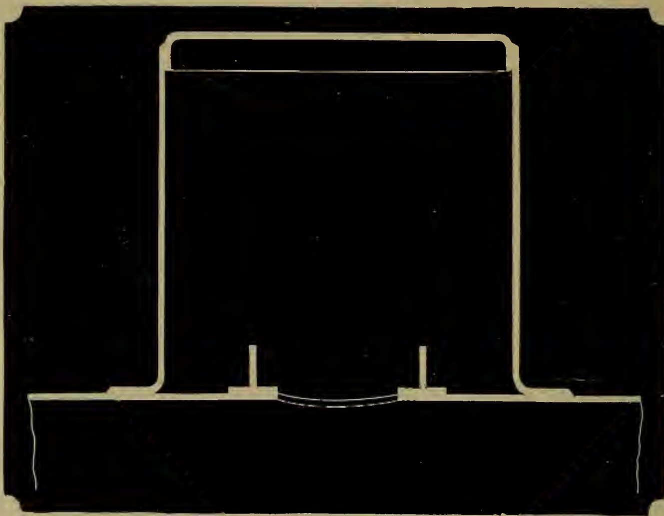


Fig. 67.

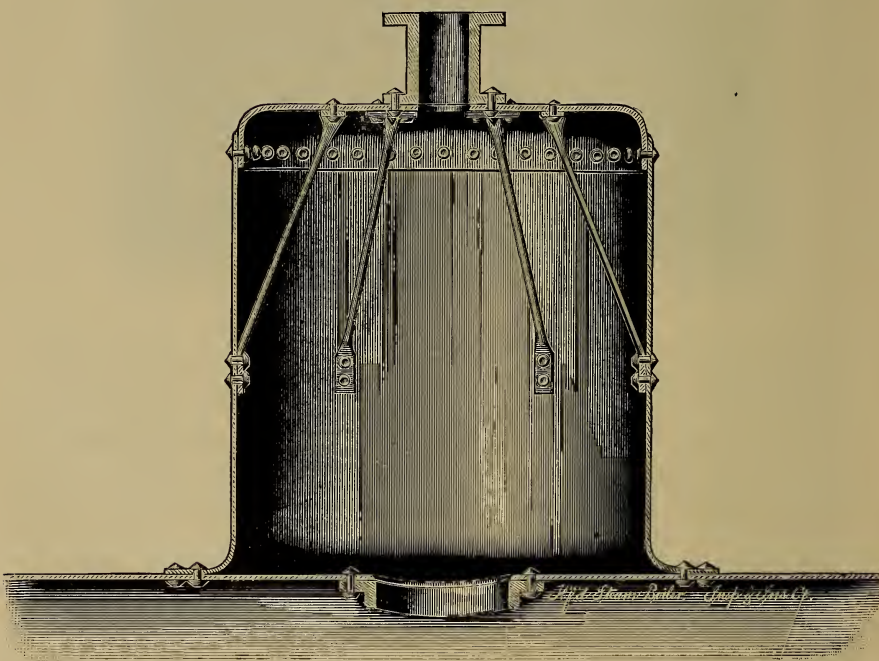


Fig. 68.

excess of the area of the pipe. The objection is the stoppage of the inlet holes in muddy waters. This is an arrangement used in marine practice and in some locomotives and is favored more and more in high-pressure land practice. The weight of the dome is an objection on board ship, and an elevation of the center of gravity of the boiler, and on some locomotives the dome has been objected to because, in addition to its other drawbacks, it stands in the way of the view of the engine-man. Where the locomotive boiler has a dome the throttle box will be near its top, and the pipe to the cylinders runs down through the steam-

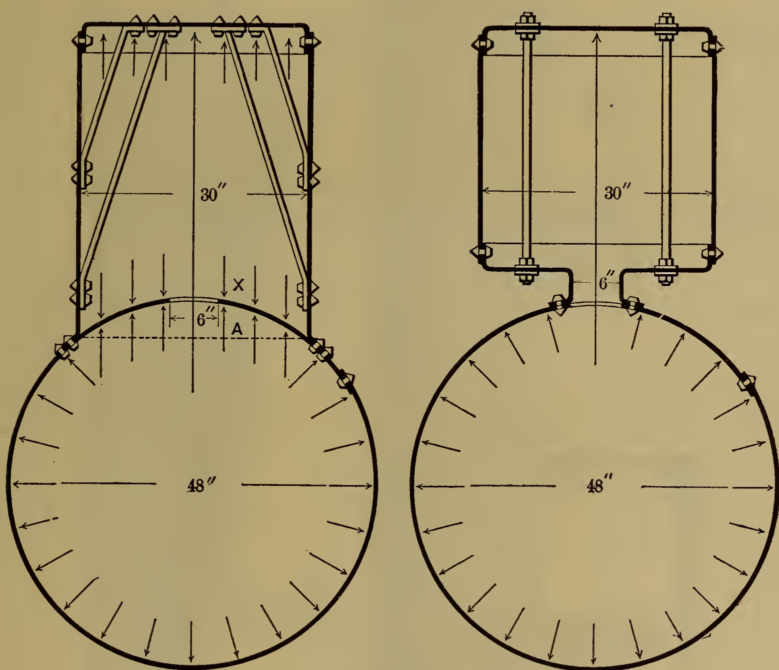


Fig. 69.

space. With a perforated dry pipe the throttle box will be at that end of it at which it comes out through the front head of the boiler. The arrangement of Fig. 72, while known as a " mesh separator," is in effect the same as a perforated dry pipe, but is removable for cleaning or repair.

(5) A form of separate dome has been used for marine boilers on smooth waters in which the dome is an annular cylinder and has the smokestack-flue pass up through it. This has prevailed in river-boat practice, and was particularly convenient in wooden hulls, because the

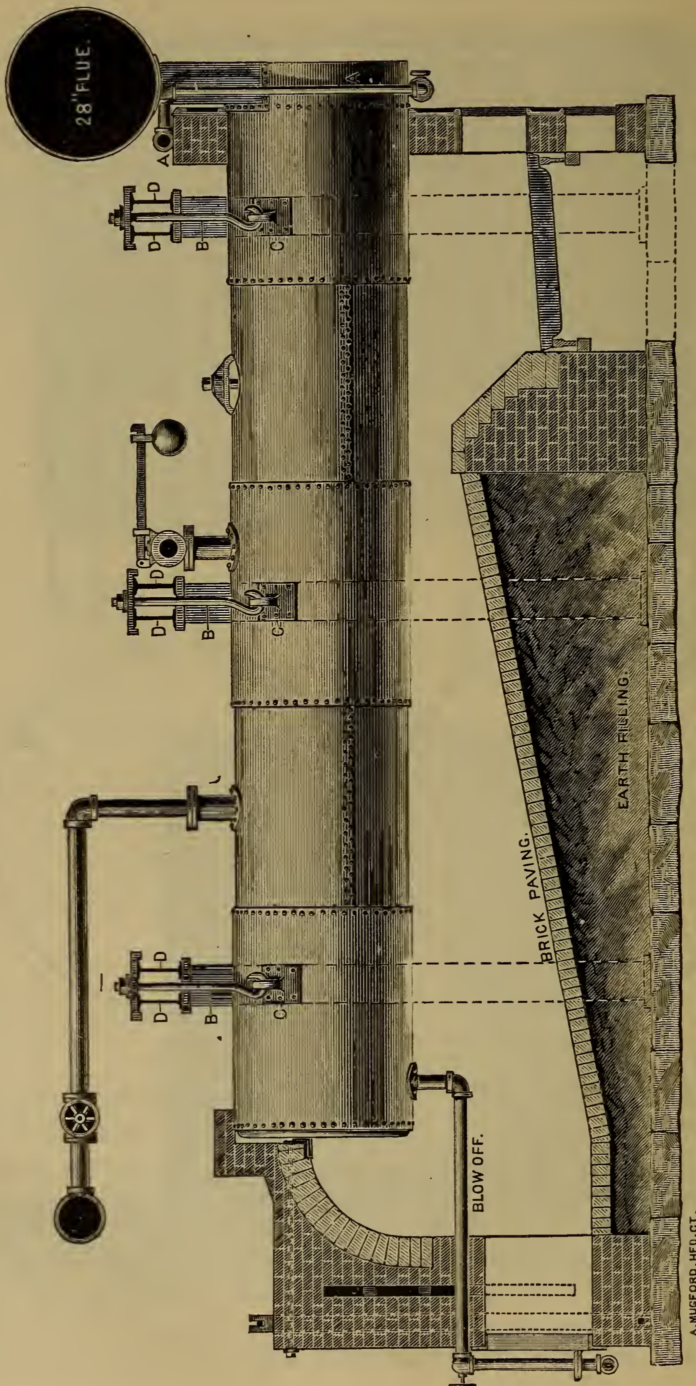

28" FLUE.
A
A
B
C
D
BLOW OFF.
BRICK PAVING.
EARTH FILLING.
A. MUGFORD. HFD. CT.

FIG. 70.

"steam chimney," as such dome was called, could get no hotter on its outside than the heat of the steam. The dome was high, and the effect of the hot chimney gases within it was to dry or even to superheat the steam in the annular space (Fig. 101).

Dome heads are rarely made of cast iron and only for moderate and

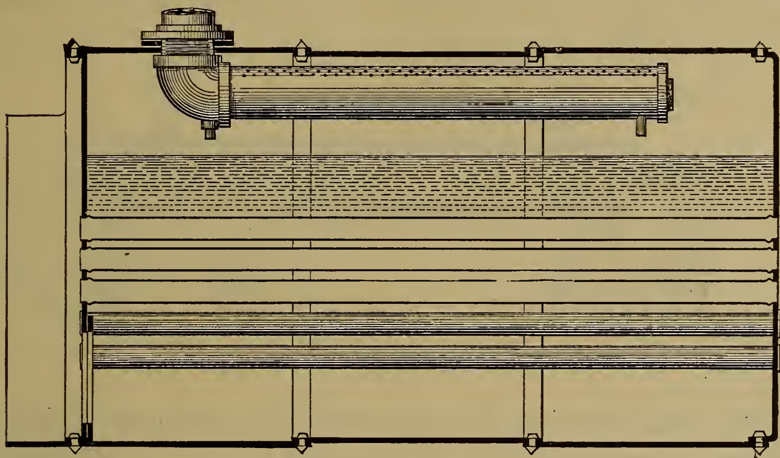


Fig. 71.

low pressures. The greater thickness of metal required with cast iron is convenient for attaching manhole fixtures, valves, and pipe outlets, and the dome head is not exposed directly to heat nor to sudden changes of temperature. The unreliability of cast iron (paragraf 37), is still against it even here. Flanged wrought iron or steel is better and

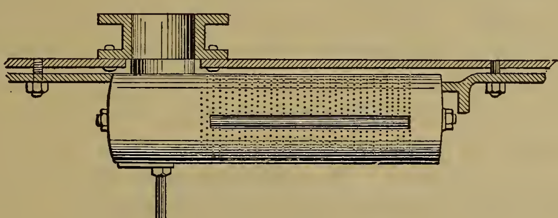


Fig. 72.

will be generally used, and will be universal for high pressures and large diameters, especially where staying must be done (Fig. 68).

50. Mud-Drums. It will be noted in detail hereafter that many boiler-feed waters are impure and contain or deposit mineral matter which remains in the boiler after the water is evaporated (paragraf 185). It is convenient to gather such solid material under the general name of "mud" in a special part or element of the boiler, and keep it if possible from settling upon parts of the heating-surface where its presence would do more harm. Hence the mud-drum will be an inverted dome or a drum at the bottom or coolest part of the boiler, and connected with it by a neck or nozzle in line with those currents of circulation within the boiler which will direct descending solid matter

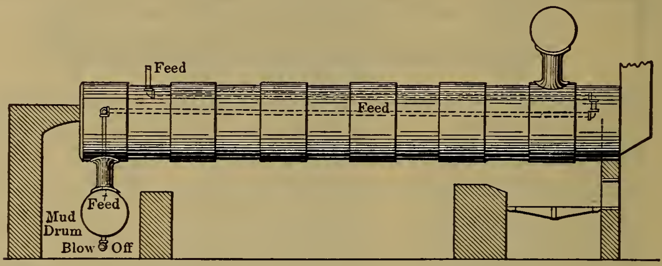


Fig. 73.

into the drum, with the view that, when once within the drum, the absence of circulation therein would prevent any mud or like material from coming out again (Fig. 73). The mud-drum is therefore withdrawn from contact with hot gases by encasing it in brick, or by having it where the gases only meet it when cooled by contact with other parts of the boiler. Fig. 74 shows the mud-drum *B* with an axis parallel to that of the boiler. It is more often transverse to the boiler. It usually has a manhole-opening when large, or when much trouble is expected from hard scale from the water. In small sizes a hand-hole will be enough. From it the blow-off pipe is led off so that mud can be blown out by opening the valve with pressure within the boiler. The feed-pipe delivering fresh water to the boiler sometimes enters the mud-drum, but this is not the best place. (Fig. 73.)

The mud-drum in pure waters often reduces to a very small appendage or disappears entirely. The difficulty with it occurs when care has not been taken to guard against its expanding at a different rate from that of the boiler itself, because cooler, while rigidly attached to the latter and not free to move. This brings strain at the connecting neck or necks, followed ultimately by leakage and by corrosion at those points.

51. Concluding Comment. Classification of Types. In the foregoing paragraphs of this chapter, the purpose has been to discuss the structural detail of all boilers belonging to the steam-kettle or shell type in which steam is made by fire under a pressure-resisting vessel

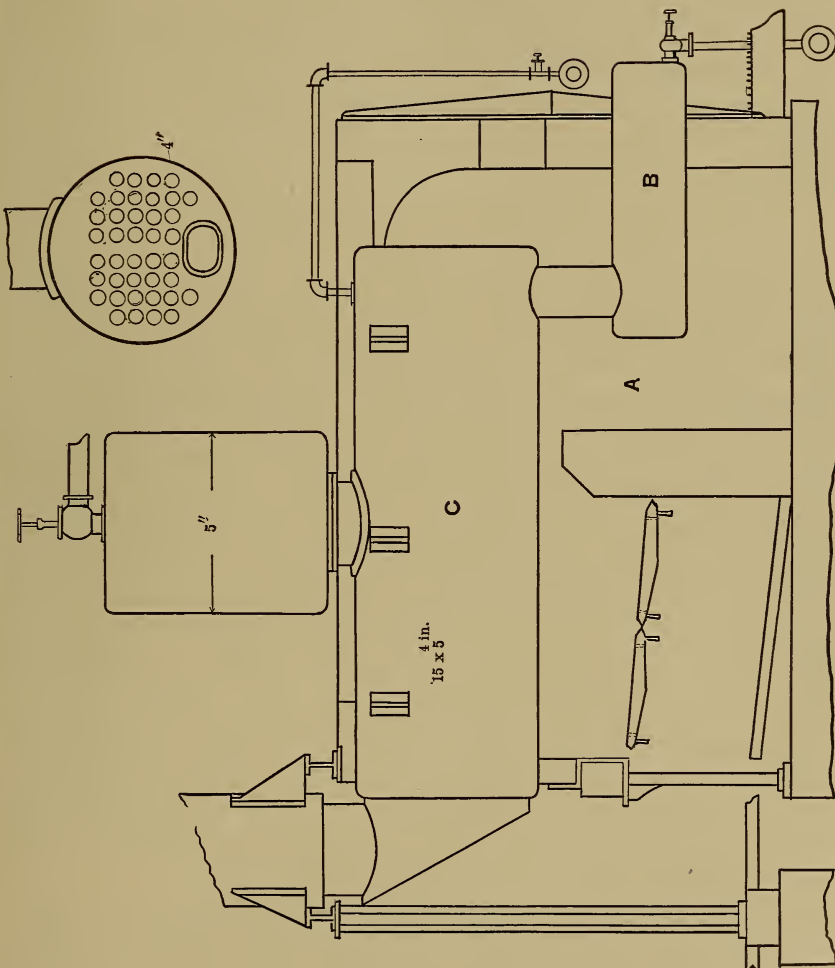


Fig. 74.

of considerable volume. While the illustrations include also other details as of the furnace, support, and operating accessories, these are apart from the primary purpose. The next step must be a differentiating of types and study of the derivatives of form from the simple fundamental conception heretofore before the mind.

The demand for increased power has continually called for the liberation of more heat in the furnace per unit of time, and the more effective transfer of that heat to the water and steam. With more rapid combustion rate (paragraf 12) the fire temperature increased, and therefore the extent and amount of heating surface to absorb it, and prevent the hot gases from leaving the boiler at too high a temperature. The one object sought has been to bring every particle of water to be heated and evaporated into close and effective contact with every particle of gas which was hotter than such water; and in general the method to do this must be to break up water and hot gas into small volumes and thin films or elements of small thickness between their two nearest sides to secure such rapid and complete transfer of heat. This fundamental purpose has been sought along the two possible lines. The first is to subdivide the volume of the hot gases and cause these to pass in small streams through the water whose mass was thus subdivided. This may be called the "fire-tube" idea, embodied in tubular and flue boilers, and reaching its limit of principle when the fire itself is placed within the mass of water to be heated. The second line is the keeping of the fire undivided, but so subdividing the water volume by the use of many smaller cylinders or tubes as to secure the intimacy of contact with fire and hot gases. This may be called the "water-tube" idea, in that the water is within the pressure unit as in the kettle, and the fire is outside. In the other or fire-tube idea the water is outside and the fire is within. In fire-tube designs the tubes or cylinders are exposed to collapse from pressure directed radially towards their center: in water tubes the pressure tends to burst them radially outward. The sectional boilers are all in the water-tube class, as are the coil and flash types.

It would have been convenient if the distinction of terms "externally-fired" and "internally-fired" could have been used as essentially synonymous with the water-tube principle and the fire-tube principle respectively. But while all water-tube types are externally fired in both senses, some fire-tube types also have the fire external to the water, and are therefore properly called externally fired. In the following the fire-tube principle will be first treated, external and internal, because historically these were first. The water-tube or sectional principle is the important modern development and is specially well adapted for use with the higher modern pressure both ashore and afloat. The classification of type will therefore be:

- Fire-tube
  - Externally-fired
    - Tubular
    - Flue
  - Internally fired
    - Cornish and Lancashire
    - Locomotive
    - Marine
    - Upright and fire engine
  - Combinations
- Water-tube
  - Plain cylinder
  - French Elephant or Union
  - Sectional
  - Coil or pipe
  - Flash and semiflash
  - Combinations

It is obvious that the two principles of subdividing the heating and cooling masses may be combined in one design, adding water tubes to a boiler whose basal classification is in a fire-tube class. These, however, can be covered in both groups under the general designation of combinations.

CHAPTER IV.

FIRE-TUBE BOILERS EXTERNALLY FIRED.

55. Types of Boiler in the Fire-tube Class. In the externally fired fire-tube class are those in which a cylindrical shell is used to contain the water and resist the pressure, and the grate and fire are under this shell, usually at one end. (Figs. 75 and 76.)

The depth of the grate may be enough to burn the desired weight of coal per hour, provided it is not so deep as to exceed the limit of easy use of the firing tools. The fire on the grate imparts its heat by radiation from its solids to the part of the shell over it, and the flame passing backward radiates its heat as long as the flaming process continues. After the flame ceases the hot gases lick the bottom of the shell until the back of the boiler is reached; here they pass by subdivision either into a small number of large diameter flues, or into a large number of small diameter tubes. The latter is the type of its class and is called the tubular or multitubular boiler. The other is the flue boiler. These tubes or flues pass through the water-space of the boiler (paragraf 48), and by breaking up the volume of gas into thinner bulk cause the cooling effect of the water or heating effect of the gas to be more effective. The cooled gases leave the shell at the front head and join into one stream again in the smoke-box.

56. The Tubular or Multitubular Boiler. Fig. 76 shows a cross-sectional elevation of a multitubular boiler. From Figs. 75 and 76 it is apparent that for a boiler of D feet in diameter and L feet long and having n tubes of d inches external diameter the water contact for absorbing heat or the heating surface in square feet will be

$$\text{Heating surface} = \frac{\pi DL}{2} + \frac{\pi dLn}{12}.$$

As n is a large number, such a boiler has a large heating-surface, and is, therefore, effective in steam-making and in cooling the gases. It is a question whether the lower elements of the tubes are as effective as the upper ones, since the former become quickly coated with dust (Fig. 8), — a non-conductor of heat, — and if any liquation or separation by specific gravity takes place in the tube, the hot gas is at the top and the cooler at the bottom. The bottom elements on the water side are

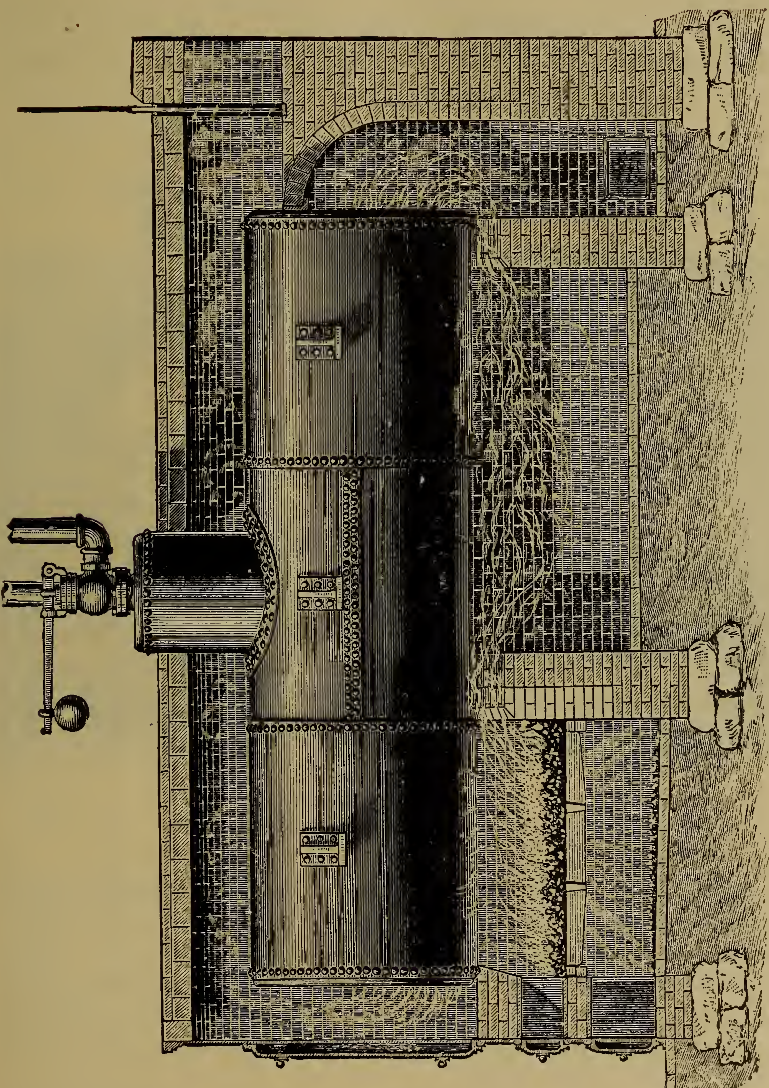


FIG. 75.

not good steam-makers either. Some designers compute only the upper half cylinder of the tubes as heating-surface, or introduce a factor 2 in the denominator of the last term. (See Fig. 8.)

An empirical rule of proportion for the chimney area in terms of the grate surface which has seemed to work well has been to make the chimney area one-eighth of the grate area. As the gases are hotter

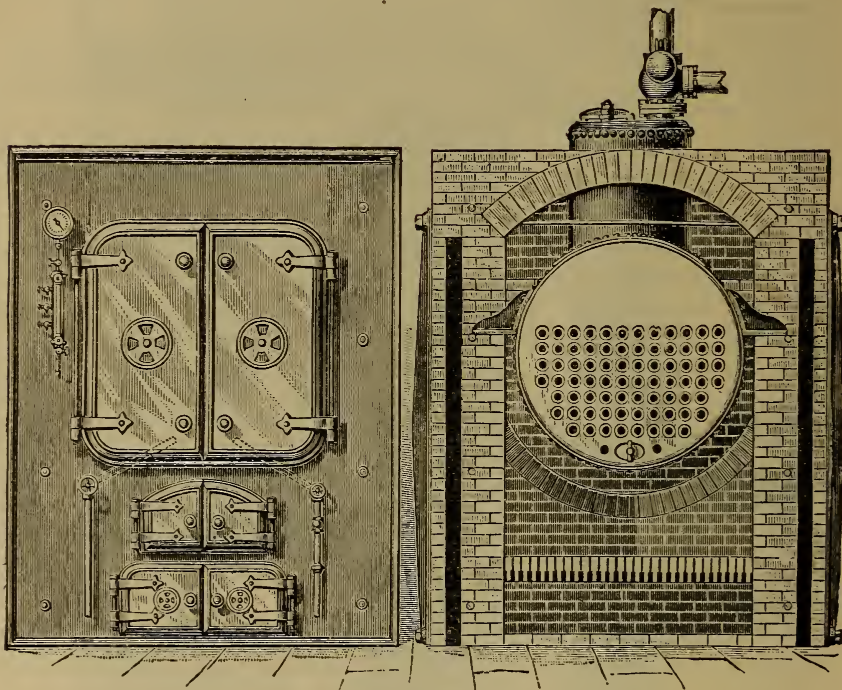


Fig. 76.

before entering the tubes or flues, it is best to make the aggregate cross-section of the open or internal diameter of such tubes to be one-sixth or one-seventh of the grate area for flaming or gaseous fuels. With anthracite it can be reduced to one-tenth so far as easy passage of the gases is a factor at not too high velocity. This tube area was formerly called the calorimeter of the boiler — a most unfortunate term at any time, and now no longer in use.

Another empirical rule of practice is to make the tube diameter bear to its length the ratio of one-inch diameter to four feet of length in bituminous or flaming-coal practice, and to five feet of length with anthracite. The table in the next paragraph gives some further data.

In common practice the top row of tubes is from four to six inches below the normal water line; five inches is usual.

57. The Tubes for Fire-tubular Boilers. The tubes for boilers are of a special grade of stock, and are designated by their external diameter, and not by their opening or internal diameter as in the case of pipe. The plate to be used in making the tubes should be either of steel or of charcoal iron. The joint is to be lap-welded, and most carefully and completely done and tested. Drawn tubes of steel have been used in a few costly boilers, so as to avoid the possible defect of the weld. Copper and brass tubes are used in a few special boilers, such as small motor-vehicle and fire-engine practice where great conductivity for heat is demanded, but always in small diameters only.

The hole for the tube in the heads or tube-sheets is made just large enough to take the tube. In thin plate, and for small tubes, these holes may be punched by a spiral punch (paragraf 32, Fig. 20), but a drilled hole is better, beveled or chamfered to give additional hold for the tube. Or it may be specified that the hole be rounded on each side with a radius of $\frac{1}{16}$ of an inch so as not to cut into the tube. Fig. 77 shows the usual type of "pin" or "tit" drill used for larger holes, a small hole being first punched or drilled, and then the remaining metal being cut away like a washer, without reducing all the material removed to chips. The cutter may be made adjustable. Special reamers then finish the hole.

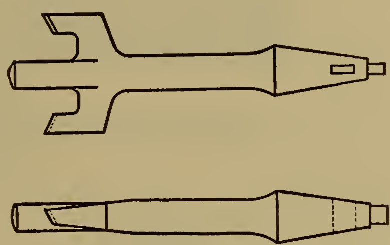


Fig. 77.

The tube is then secured in the hole by the process called "expanding." A tube-expander (Figs. 78 and 79) may be of the roller type or of the sectional type.

In the roller form three rollers are borne upon segments which guide them, and upon which radial pressure can be brought by the central conical or tapering pin. When the rollers inside the tube are opposite the tube-sheet or just behind it, upon the inside of the tube, the rollers are forced outward and revolved inside the tube. The rolling pressure causes the metal of the tube to flow outward until the resistance of the hole in the tube-sheet is encountered. Then the tube and the hole are pressed to fit each other (Fig. 81), with a force usually sufficient to be steam-tight and having a very considerable strength. To prevent the tube and its sheet from sliding under changes of length due to temperature, the end of the tube is slightly turned over or upset — called beading — so as to grip the outer face of the tube-sheet. This both

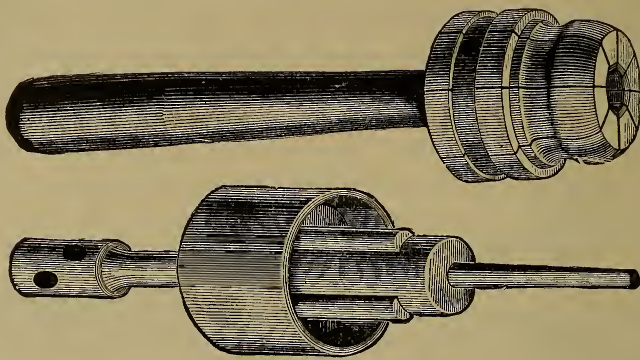


FIG. 78.

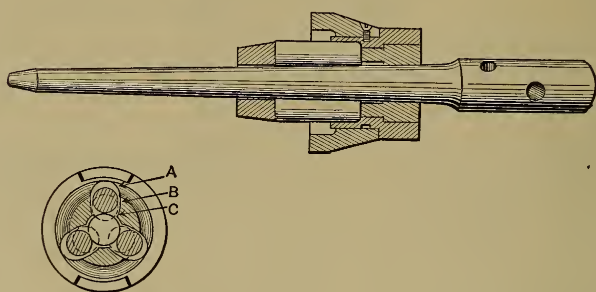


FIG. 79.

adds to strength and serves to prevent leakage. The beading is usually done with a special form of swaging-chisel (Fig. 80), often called a "thumb-swage" from the shape of its pressure end, which has a longer prong which enters the tube, while the shorter prong projects over the annular tube end, and gives an aspect suggesting the combination of index-finger and thumb on the human hand.

FIG. 80.

There are other forms of tube-expander, acting by wedging or direct pressure, but the roller form takes less power and gives better results. The fit of the expanded tube is a pressure and frictional one. Hence any tendency to push or pull the tubes through the tube-sheet (from heat or from yielding to pressure) causes the expanded tubes to leak in their holes. With high pressures a certain proportion of the tubes —

one in five, often — are made of extra-heavy stock, so as to allow for a greater strength due to the more efficient heading or beading of the thicker metal. Or, again, with thicker stock in the stay-tubes, the outside which projects beyond the sheets can have a thread cut on

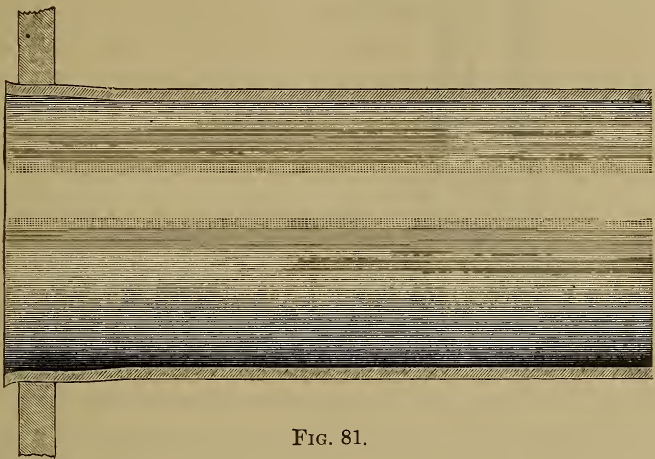


Fig. 81.

it, upon which a stay or lock-nut will be screwed home, and thus convert the tube into a through-stay. Such stay-tubes will be located where the tendency of the heads to flex under the pressure needs particularly to be guarded against.

Expanded tubes can be re-expanded so as to be made tight if leakage should be developed by service; but this cannot be done very often, since the metal must undergo a pressure in expanding which transcends the elastic limit of the material (otherwise it would spring back when the expander was withdrawn), and as the result the tube is apt to crack when the process is overdone. The leakage occurs when the tubes become overheated from any cause which prevents the water from effectively cooling them. In such overheating they expand, and in the effort to lengthen they force themselves through the holes in the tube-sheets opening the joint, which does not re-form itself on subsequent contraction. Upright boilers of certain type are specially prone to this.

The cutting off of the tubes to length is done after the tube is in place, either with a special gauged cutter revolved by hand or power, or in simpler practice by putting a gauge-plate of hard steel against the tube-sheet over the tube and cutting down to it by hand. The thickness of the gauge gives just the amount for heading and beading.

After tubes have leaked and cracked and re-expanding is impossible, or when a tube has failed from any cause in its length, it can be removed by cutting out. The same tool can be used for cutting tubes inside the heads as is used for cutting off outside (Fig. 82). The defective

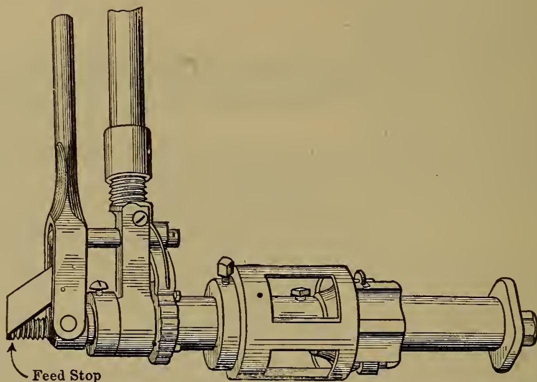


Fig. 82.

tube can then be replaced as in the first instance with a new tube, or new ends can be welded on such as are defective at the ends only, using the good stock of the old tube. The commercial length of boiler-tubes, in 12, 16, or 20-foot lengths, usually fixes the length of the boiler itself. See Table VII.

58. Ribbed Tubes. Retarders. A special form of boiler-tube has been used to some extent, which is fitted with ribs lengthwise, or is thickened at several points of its inner circumference (Fig. 83). The object of this is to arrest more completely the available heat in the hot gases flowing through the tube, so that from this extra metal the heat may be abstracted by conduction. A somewhat similar function is discharged by what have been called retarders in the tubes. A cross-shaped bar the length of the tube is laid within the ordinary cylindrical tube, and absorbs heat from the gases, which it transfers to the tube by radiating such absorbed heat, as well as acting to retard the too-rapid flow of gas at such a rate as would prevent complete transfer of heat to the tube. Such retarder may be twisted into a helix to hold the gas longer in the tube.

Fig. 83.

An objection to the ribbed tube is the difficulty in cleansing it when the gases deposit a sticky residue on the cooler surface of the metal. The retarders are cleansed by being taken out.

59. Arrangement of Fire-Tubes. While the purpose for which fire-tubes are used is best served by a large number of them, yet this may be carried so far as to sacrifice some other good features. In the first place the nest of tubes should not come too near the external shell. Space should be left to permit a descending circulation of water outside the tubes (Fig. 5), and to permit inspection and cleansing of the plate from above or below, or both. Three inches in small boilers and four to six inches in larger ones is the recommended space.

It is a good plan to leave out a vertical row of tubes in the center of the nest; or, if this number cannot be sacrificed, to make the clear horizontal space between tubes two inches in the center and one inch elsewhere. When the fire is forced, steam generation will be most active in the part nearest the fire, and the current of gas will rise in the center, lifting the water here higher than at the sides, and stimulating the descent at both sides. Through this opening also cleansing of the central elements and rows of tubes can be more easily done by a tool which will reach down.

If there is a manhole for cleansing in the front or back head below the nest of tubes to give access to the lower elements of the shell (Fig. 54), this will compel the omission of some tubes at the bottom of the nest. Such space at the rear of the boiler becomes a sort of mud-drum for the accumulation of solid matter, and it should be easily cleansed. These together lead to arranging the tubes in two nests, symmetrical respecting a vertical plane through the axis.

The tubes will be arranged in horizontal rows. It is best to place them also in vertical rows or under each other in a horizontal boiler, rather than to "stagger" them by placing the tubes in alternate rows vertically under the spaces between tubes on the row above. The latter gets more tubes in; but circulation is not so easy, and cleansing of rows below the upper two becomes impossible if mechanical scraping is necessary. Tubes may be staggered in upright boilers, as scale does not settle on vertical surfaces; but cleansing of the lower tube-sheet becomes then impossible toward the center. In vertical rows with space between reasonable cleansing by scraping can be done from above. Table VII gives facts and standards of arrangement.

TABLE VII. FIRE-TUBE PROPORTIONS AND ARRANGEMENT.

Diameter of Shell in Inches.	Tube Diam. Inches.	Usual Number.	Vertical C to C Inches.	Horizontal C to C Inches.	Length Over All. Feet.	Center of Upper Row of Tubes Above Center of Boiler.
48 to 60	3	46 to 76	4	$4\frac{1}{4}$ to $4\frac{1}{2}$	10 to 12	$6\frac{1}{2}$ to $8\frac{1}{2}$
48 to 66	$3\frac{1}{2}$	34 to 70	$4\frac{1}{2}$	5 to $5\frac{1}{2}$	12 to 16	$6\frac{1}{4}$ to $9\frac{1}{4}$
54 to 72	4	36 to 74	5	6	16 to 20	7 to $10\frac{1}{4}$

The tubes of a tubular boiler not only take off area from the head exposed to pressure, but they also serve to stay the two opposing heads together. It is only necessary therefore to stay the segment of the flat head above the top row (see paragraf 44). The head is always flat for convenience of the expanding process, and the radius of the curve of the flange which is formed on it to fasten it to the shell is often specified to be not less than a minimum — usually $1\frac{1}{2}$ inches. A larger radius even up to four inches will cause less strain to the extreme fiber in flanging.

60. The Flue-Boiler. When the fuel burned on the grate is of such a bituminous or gaseous type that its flame exceeds in length the over-all length of the shell, the tubular boiler becomes uneconomical because the fine subdivision of the still glowing flame makes it go out in the cooler tubes. No more oxygen can reach the flaming gas to support combustion, and combustion in progress ceases because the temperature of the gas in the small tube falls quickly below the point of sustained ignition. The glowing flame becomes a smoke, and carbon goes off with the gas in an unburned state as CO or as lampblack or soot, which might have given heat to the water. A larger tube is required.

When the water contains much solid precipitable matter which will form a scale on the heating-surfaces which is hard to get off, the fine subdivision of the water around the small tubes of the preceding type offers many surfaces which are hard to reach for cleaning by mechanical abrasion. Hence a type of fire-tube boiler is suggested where there will be larger fire-passages and larger water-masses. If the tubes are larger, there must be fewer of them in a boiler of a given diameter of shell. Such tubes, when of 7 inches external diameter or over, are called flues, and the boiler using them is a flue-boiler (Fig. 84). The formula for heating-surface of paragraf 55 applies here also, as the setting is the same but as the value of n is smaller, the heating-surface for a given shell diameter and L is less. It may be convenient, therefore, to increase the length.

The greater diameter of the flues permits a joint with the head or tube-sheet to be made by flanging the holes in the latter and riveting the flue to the flange. These flanges may be internal (Fig. 85), or external (Fig. 84). Internal flanges should always be used where the ends of the flues are exposed to flame, as at the back end of the setting, since the ends of the tubes and flanges are cooled by conduction only, and not by contact with the water. External flanges are easier to rivet and are safe for the front or cooler end of the shell, and have to be used with small-diameter flues in any case. The number of such flues will be as great as possible when regard is paid to the limitations which they

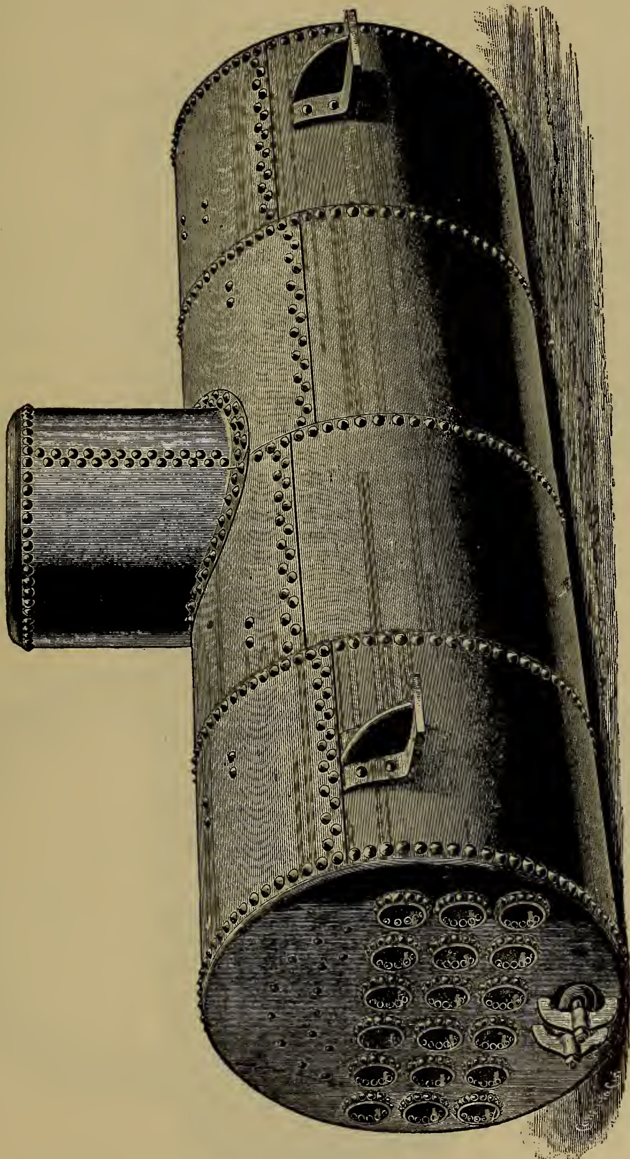


Fig. 84.

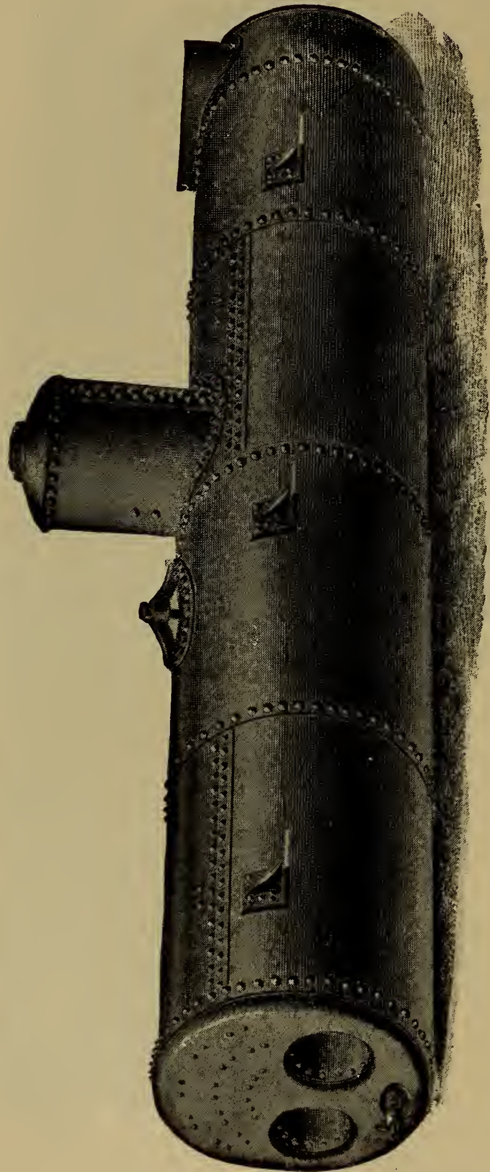


Fig. 85.

are to help to meet; two of large size being the lower limit (Fig. 85). They may be of tube up to the limit of its size of manufacture (usually 18 inches); above that of riveted boiler steel with butt or lap-joints.

61. Reinforced Flues for Long Boilers. When the boiler and the

FIG. 86.

flues become too long for the flue to be in one piece, it must be jointed either as the shell is, or by the bump-joint of Fig. 86 where one segment is locally enlarged at the end for a short distance and the other piece

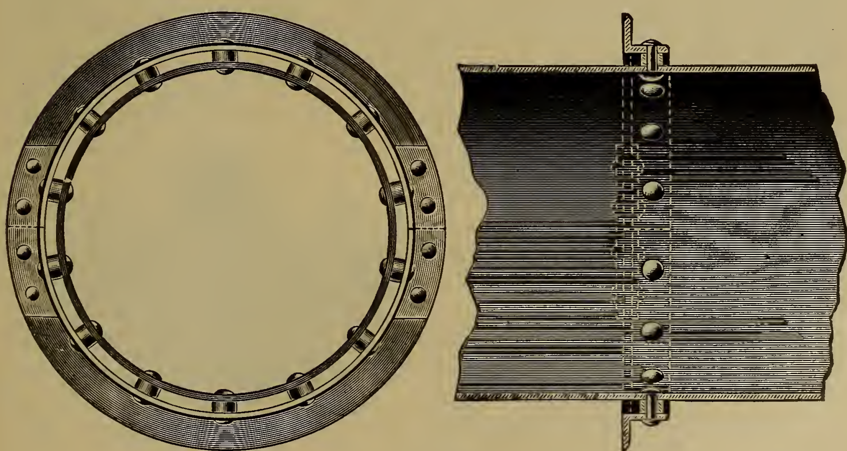


FIG. 87.

telescoped into it. With very large or long flues where they are exposed to stress and consequent deformation by their own weight, the flue must be kept to its cylindrical shape to give it strength against collapse, and this is best attained by means of rings of stiff section placed around the outside of the tube and riveted to it, or by the corrugated flue to be hereafter discussed (paragraf 68). Such stiffening-rings may be of steel of angle or tee-iron section (Figs. 87, 88), with distance pieces to give

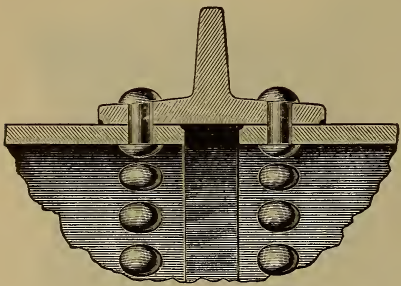


Fig. 88.

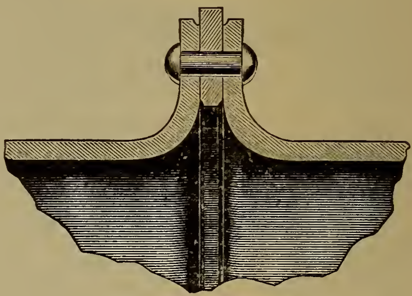


Fig. 89.

free access of water to the flue-metal; or the end of each flue segment may be flanged outwardly to make a joint with a flat ring placed edgewise between them (Fig. 89). They are corrugated ringwise with the same object. These stiffening-rings are either plain rings with light pressures, or else the ring receives a greater transverse resistance by being made of angle or tee-iron (Figs. 87 and 88). Or, again, the formation of the flue in several segments may be utilized as opportunity to form up the joints into rings of stiffening effect, as in the Bowling and Adamson flue-joints of Figs. 89 and 90.

The latter gives a longitudinal flexibility to the flue which diminishes the strain on rivets in joints and at the heads when the flue gets overheated or is suddenly contracted while the shell temperature remains constant. Illustrations of reinforced flues will appear in the next chapter, where the larger diameters are mainly in evidence.

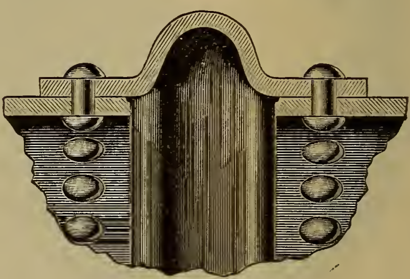


Fig. 90.

62. Conditions Suggesting Choice of Tubular or Flue Types. The distinction between the tube and flue is purely arbitrary. Under six inches is a tube, over seven inches is a flue. The tubular form is the type form for anthracite fuel and with good feed-water clean and free from mineral matter. It may almost be called the New England or Northern Atlantic Sea-Coast Standard, except where the sectional types come in. It is the form before the mind of most every one not skilled in separating the types from one another when he speaks of "a boiler." Its advantages are:

(1) Great heating-surface in a small space.

(2) This gives it ability to make steam rapidly, and to supply a great deal of steam from a given area of ground occupied, as compared with certain other forms.

(3) This evaporative capacity is cheaply bought. The tubular boiler is not an expensive one, but is the cheapest of the efficient types, costing under ordinary commercial conditions from $8 to $11 per horsepower. It can be made anywhere and by any boiler shop, with the usual equipment of tools.

(4) The water is subdivided by the tubes into small masses, securing immediate transfer of the heat of the metal of the tubes to all parts of the volume of water. Hence the boiler responds promptly to an attempt to force it.

As objections to the tubular boiler may be advanced:

(5) The water-space is so filled with tubes that access to the lower parts for cleansing is difficult, and to some places is impossible. This objection is a fatal one if water is to be fed to the boiler which has great amounts of salts in it which are precipitated on boiling.

(6) The fine division of the gas-currents in the small tubes will so lower their temperature that they are extinguished. If the flame would naturally be longer than the length from the furnace to the entry to the tubes, the extinction of the incandescent particles on lowering the temperature makes the gases smoky, and carbon is wasted as soot. The fine subdivision in tubes is also fatal to further union of carbon with oxygen, and combustion, if not completed before the gases enter the tubes, will either be incomplete, or else will take place beyond the boiler at some possibly inconvenient place — such as the top of the stack or at its throat.

The objections to the tubular boiler are the advantages of the flue type. With flaming coals and where the water is not good, it avoids some of the obstacles to the use of the multitubular. As the water becomes worse, the number of flues should diminish so as to give complete access for cleansing. As the fuel becomes gaseous and its flame lengthens, the flue should become larger in diameter.

The tubular or small flue will be used wherever possible.

U. S. FORMULÆ FOR FLUES.

63. The U. S. Federal laws fix the formulæ which are to be used in proportioning flues against collapse. They may be grouped as follows:

Case A. For lap-welded flues greater than 7″ in diameter and less than 16″, and 18′ long or less.

$$P = \frac{t}{R} \times 44.$$

For each 3′ of length add $\frac{1}{100}$ of an inch to t. One wrought-iron stiffening-ring to be used in each 5′, whose $t' = t$, and whose width is greater than $2\frac{1}{2}''$.

Case B. Flue greater than 16″ diameter and less than 40″:

$$P = \frac{FT}{C}.$$

Case C. Flues greater than 40″:

$$P = \frac{89{,}600\ T^2}{LD} \qquad \text{[Rankine uses 80,600]}$$

Case D. Corrugated flues, $\frac{5}{16}''$ thick or thicker, and the corrugations $1\frac{1}{2}''$ deep, 6″ pitch:

$$P = \frac{14{,}000 \times T}{D}.$$

In these formulæ

$$F = \frac{1760}{D};$$

T = thickness in inches;
P = allowable pressure in lbs. per square inch;
C = constant 0.31;
D = diameter in inches;
L = length in feet (not over 8′).

These were supplemented in January 1894 by adding:

THICKNESS OF MATERIAL REQUIRED FOR TUBES AND FLUES NOT OTHERWISE PROVIDED FOR.

"9. Tubes and flues not exceeding 6 inches in diameter, and made of any required length; and lap-welded flues required to carry a working steam-pressure not to exceed 60 lbs. per square inch and having a diameter not exceeding 16 inches, and a length not exceeding 18 feet; and lap-welded flues required to carry a steam-pressure exceeding 60 lbs. per square inch, and not exceeding 120 lbs. per square inch, and having a diameter not exceeding 16 inches and a length not exceeding 18 feet, and made in sections not exceeding 5 feet in length, and fitted properly one into the other, and substantially riveted; and all such tubes and flues shall have a thickness of material according to their respective diameters, as prescribed in the following table·

Outside Diameter.	Thickness.	Outside Diameter.	Thickness.	Outside Diameter.	Thickness.	Outside Diameter.	Thickness.
Inches.	Inch.	Inches.	Inch.	Inches.	Inch.	Inches.	Inch.
1	.072	$2\frac{3}{4}$	.109	5	.148	12	.229
$1\frac{1}{4}$	.072	3	.109	6	.165	13	.238
$1\frac{1}{2}$	.083	$3\frac{1}{4}$	.120	7	.165	14	.248
$1\frac{3}{4}$	.095	$3\frac{1}{2}$	.120	8	.165	15	.259
2	.095	$3\frac{3}{4}$	.120	9	.180	16	.270
$2\frac{1}{4}$	.095	4	.134	10	.203		
$2\frac{1}{2}$	.109	$4\frac{1}{2}$	.134	11	.220		

"10. Lap-welded flues not exceeding 6 inches in diameter may be made of any required length without being made in sections. And all such lap-welded flues and riveted flues not exceeding 6 inches in diameter may be allowed a working steam-pressure not to exceed 225 pounds per square inch, if deemed safe by the inspectors.

"11. Lap-welded flues exceeding 6 inches in diameter and not exceeding 16 inches in diameter, and not exceeding 18 feet in length, and required to carry a steam-pressure not exceeding 60 pounds per square inch, shall not be required to be made in sections.

"12. Lap-welded and riveted flues exceeding 6 inches in diameter and not exceeding 16 inches in diameter, and not exceeding 18 feet in length, and required to carry a steam-pressure exceeding 60 pounds per square inch and not exceeding 120 pounds per square inch, may be allowed, if made in sections not exceeding 5 feet in length, and properly fitted one into the other, and substantially riveted.

"13. Riveted and lap-welded flues exceeding 6 inches in diameter and not exceeding 40 inches in diameter, required to carry a working steam-pressure per square inch exceeding the maximum steam-pressure prescribed for any such flue in the table of section 8 of this rule, shall be constructed under the provisions of section 15 of this rule, and limited to the working steam-pressure therein provided for furnace-flues; but in no case shall the material in any such riveted or lap-welded flue be of less thickness for any given diameter than the least thickness prescribed, in the aforementioned table, for flues of such diameter."

CHAPTER V.

FIRE-TUBE BOILERS INTERNALLY FIRED.

65. Definition of Internal Firing. Advantages and Disadvantages of the Principle. A fire-tubular boiler becomes internally fired when the furnace with grate and ash-pit are taken within the fire-tube system which is surrounded by the water to be evaporated. When this is done by enlarging a flue or part of it, or combining a number of tubes into one large flue for part of their length, the water surrounds the furnace completely. The fire-tube and shell may be cut through on the bottom so that the water to be evaporated is not on the bottom of the furnace, but this is open to the air. As before, the metal surface in contact with water below the fire-flue is not efficient as a rule, and little or no circulation of such water occurs unless mechanically compelled.

Flues of this furnace type will necessarily be of large diameter; they will therefore have to be supported or stiffened by rings at short intervals, or by corrugation against collapse from external pressure, or by cross-tubes, or both (paragrafs 61, 63). The pressure within them is that of the atmosphere or less, and surrounding them is the full steam-pressure. If the flues are made of other than arched or cylindrical elements, the problem of keeping them in shape becomes more complicated.

The advantages of the principle of surrounding the fire as well as the hot gases by water and the flue-metal are:

(1) Economy. No heat is lost by radiation from brickwork external to the boiler and heated by the heat of the fire. The water to be evaporated intercepts all radiation.

(2) The part of the boiler exposed so as to radiate heat to external air is no hotter than the water and steam within it. Loss by radiation is lessened here because of the lower temperature of the radiating body. This makes fire-rooms more comfortable, especially on board ship or in contracted quarters, and is of great importance in railway practice, where the boiler must be exposed to cool outdoor air.

(3) The metal surfaces surrounding the fire are most efficient evaporating surfaces. This makes such boilers compact with a given evaporative capacity, so that great evaporation is secured in a small space.

This is of moment in locomotive and marine practice. Such boilers as are to be portable reap advantage from this.

(4) The furnace being internal, the boiler requires either no setting or one of the simplest description. In wooden hulls the internal fire was a matter of great advantage in the matter of safety from fire, and the absence of a brick setting removes the difficulty from weight. The absence of setting makes such boilers portable, and fits them to be used where this is convenient.

(5) No cool air infiltrates through cracks or porous places in the brickwork to dilute the gases and lower their temperature. Such infiltration may make a difference of ten per cent in efficiency in favor of internally-fired boilers which are self-contained.

Applicable to many types is the advantage:

(6) They make steam and reach working pressure quickly, since the relation of heating surface is usually large, as compared with other forms, to the weight of water contained. This does not apply to large marine types holding large masses of water.

The objections to the internally-fired type are:

(7) The internal fire-box exposed to a pressure tending to collapse it inward makes a costly type of boiler. This is offset in a comparison of types by the saving from the absence of setting.

(8) The efficiency of the heating-surface keeps down the temperature of gases, and thus prevents their complete combustion and causes smoky products of combustion. This is a real difficulty with coals containing much volatile matter, and vitiates economy of such boilers with such coals. Locomotives and marine boilers are usually the worst offenders in smoky cities. The difficulty is increased when high rates of combustion are used. Means must be used to keep the gases hot enough to burn.

(9) Rapid steaming capacity secured by large heating-surface, coupled with a small volume of water in the boiler at one time, makes a type in which pressure will rise rapidly from the safe working pressure to a pressure so much higher as to endanger the resistance of the shell to rupture. This makes such boilers dangerous in proportion to their liability to this trouble.

Applicable to some types are the objections:

(10) Many types introduce places in their structure which are hard to clean and inspect.

(11) Circulation is not always perfect or satisfactory, and one part may have water in it which is much cooler than the average or normal temperature. This gives rise to unequal contractions and tends to develop leaks. Or the steam may not be carried away from the heating-

surface by the circulation, but may remain and keep water from touching and cooling the heating-surface, so that it becomes overheated. These do not attach to the same types, nor is either difficulty common to all types. The special features of any type will appear in their proper places.

66. The Cornish and Lancashire Boiler. The Cornish boiler is a single-flue boiler, with the fire-box or furnace at one end of it (Fig. 95).

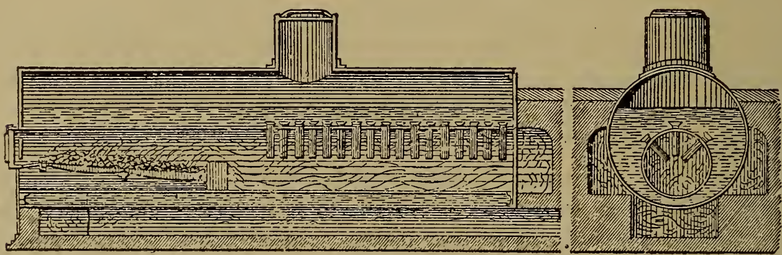


Fig. 95.

The flue is therefore of large diameter (probably five tenths of that of the shell), and has to be stiffened against deformation and consequent collapse by the methods suggested in paragraf 61. The furnace is formed by inserting grate-bars supported on bearers across the flue, and its back is made by a brick bridge-wall. The gases pass backward through the flue to a back connection, whence they come forward

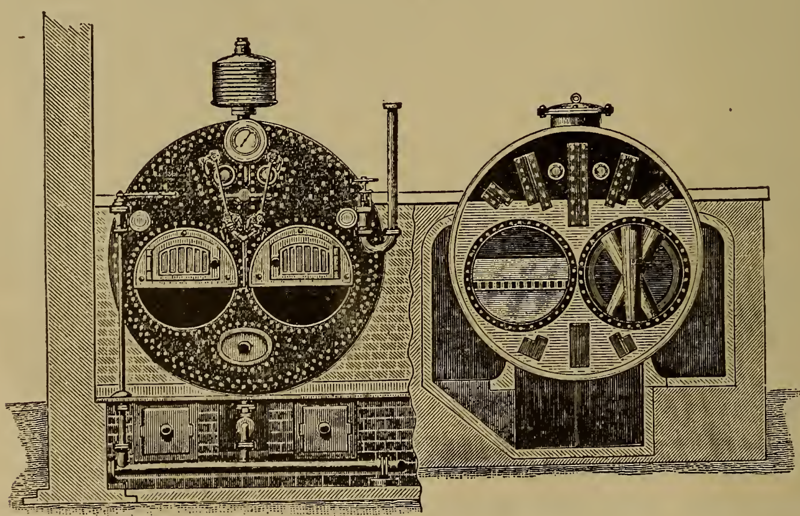


Fig. 96.

either along the sides or under the bottom if the chimney-duct is at the front; but if the chimney is at the back, the gases come to the front in side flues and return under the bottom. Such a boiler requires to be set in brick (Fig. 98).

The objection to the Cornish boiler is the large and weak flue. This early caused the development of the Lancashire boiler, which is sometimes called the double Cornish boiler. Two flues with internal fires replace the single flue of the Cornish. Each will be of smaller diameter and hence stronger, and the existence of two fires permits cleaning of fires and coaling to be done alternately in each, with advantage to the steadiness of pressure. Fig. 96 shows a Lancashire boiler fitted with the Galloway water-tubes (paragraf 67).

A modification of this type in which two furnace-flues join into one flue behind their bridge-walls has been called in England the "breeches" boiler. The American type of this has been seen in a form of locomotive boiler which the single flue serves as a combustion-chamber. The alternate-firing principle helps to keep up a high temperature in the combustion-chamber when one furnace is freshly fired with gaseous coal, and the distilled products are ignited before getting into the fine subdivision caused by tubes (Fig. 97).

67. The Galloway Tube in Lancashire Boiler. The Cornish and Lancashire boilers are not usual in America, except in the modified form caused by introducing the Galloway tube (Figs. 96 and 98). This is a conical water-tube intended to cross the flue of either of the foregoing types, and serve both to stiffen it and to add a very efficient heating-surface of water-tube directly in the hottest current of the furnace-gases. The conical shape is given to the tube to favor circulation at uniform rate, but more especially to make it possible to pass the flange of the smaller end of the tube through the hole made in the flue to pass the larger end, but not its flange. By this expedient one of the inner tubes which fails can be cut out and replaced by working from without the flue and without disturbing other tubes nearer the ends of the flue. The tubes may be alternately vertical and horizontal, or they may all diverge from the vertical. The flanges of the tubes serve to rivet them to the flue, one inside and the other outside (Fig. 98), and where the tubes brace the flue no stiffening-rings will be required. The flue usually has provision for flexibility in case of unequal expansion of shell and flue (paragraf 61).

68. The Scotch or Cylindrical Marine Boiler.—The cylindrical furnace arrangement of the internally-fired flue-boilers leads naturally to that form of boiler which is so generally used in the merchant marine. The large cylindrical shell will envelop or contain two or three internal flue-

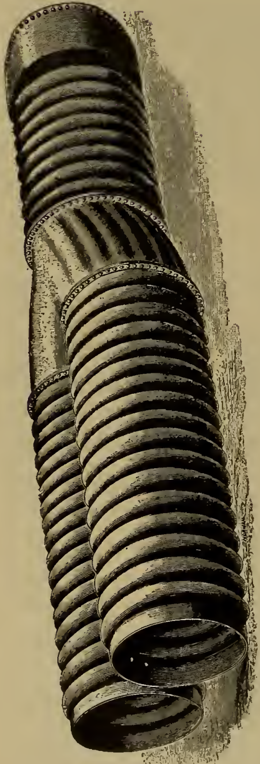


Fig. 97.

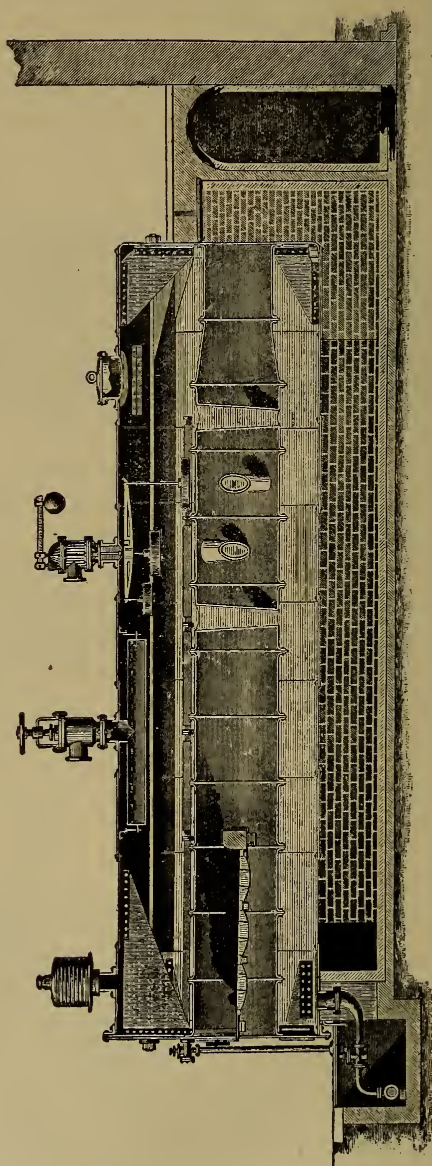


Fig. 98.

furnaces, arranged as shown in Fig. 99 (see also Fig. 100). These furnace-flues being short are usually corrugated in modern practice to give them stiffness against collapse without stiffening-rings. The flue is first welded lengthwise, and then corrugated $1\frac{1}{2}$ inches deep and with 6 inches between corrugations (Fig. 104). The end is flanged

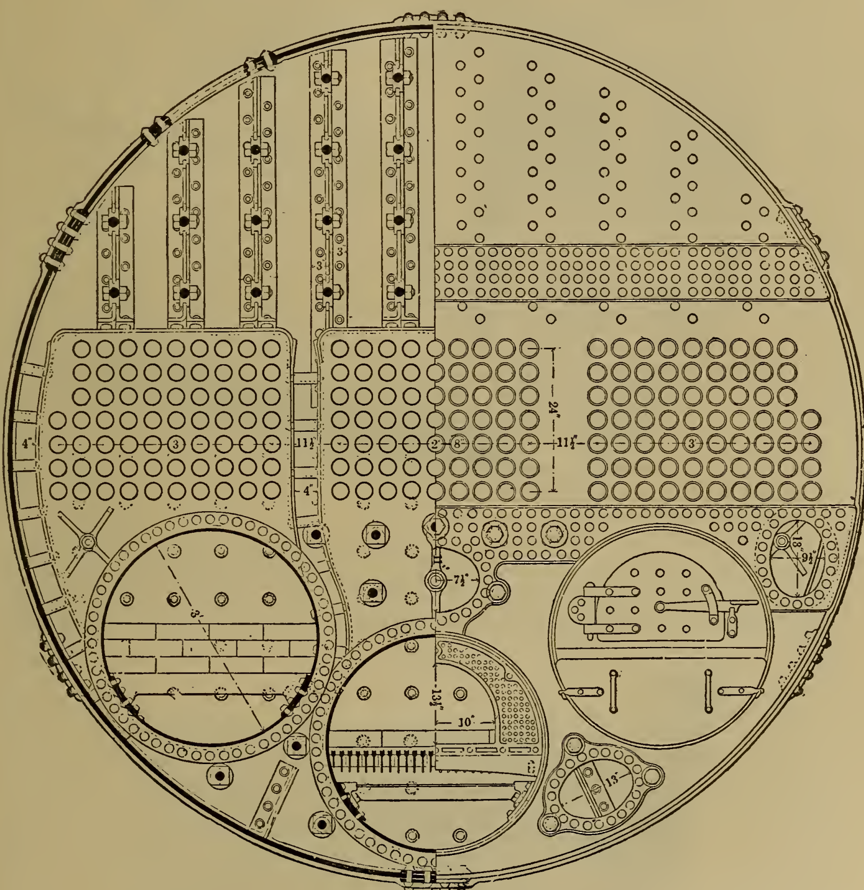


FIG. 99.

or straight to attach it to the front or rear sheets. Corrugation increases enormously the resistance to deformation by pressure, and its only drawback is the difficulty in keeping it cleansed outside and in.

At the rear of the furnace-flue the gases rise in a "back connection" to the plane of the tubes, still surrounded by water, whereby the heat of the gases is more completely withdrawn (Fig. 106); and from these tubes the gases and smoke pass into smoke-boxes and thus to the chimney-stack. Sometimes such boilers are made double-ended,

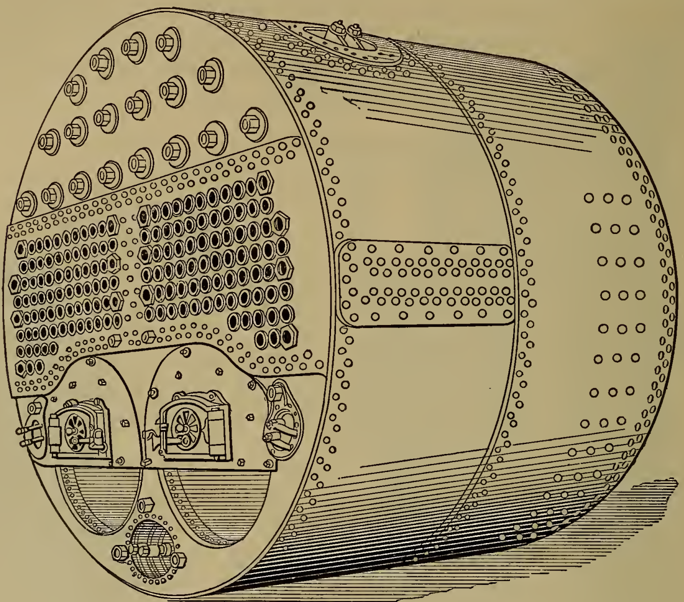


Fig. 100.

either like Fig. 103, or with the back connection partly in common. The flat surfaces of the back connection require careful staying as well as the large areas of the heads. The shells are butt-jointed and double

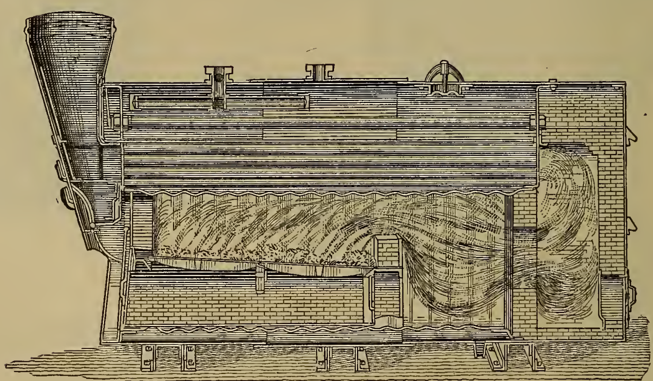


Fig. 101.

or manifold-riveted, by reason of the strength required with large diameters (paragraf 35). Such boilers usually have through-stays and stay-tubes as well.

The Scotch boiler needs no setting and is self-contained. The objection to it is the tendency of the water below the furnace-flues to cool down and remain without circulation, thus preventing the shell from getting uniformly warm. This is prevented in part by causing this lower water to circulate mechanically by means of connecting the suction of the feed-pump to the lower part of the boiler, while its delivery or forcing connection is toward the surface of the water in it. Other devices are also used for the same purpose, perhaps the best known being the hydrokineter proposed by a Mr. James Weir of Glasgow (Fig. 102). Here a steam jet from another boiler with steam up is used to induce the motion of the dead water under the furnaces by an injector action (paragraf 155).

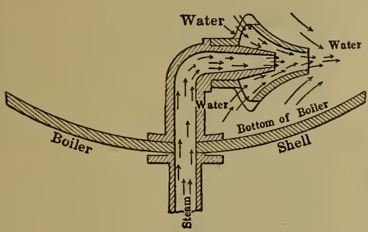
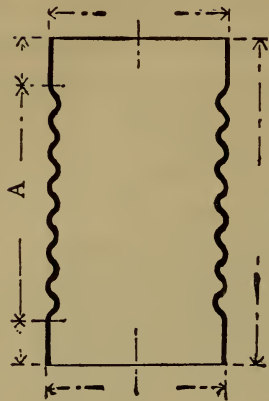


FIG. 102.

The typical Scotch marine boiler is intended to be laid athwartship and to get a length of course for the gases by a return arrangement of tubes. This makes a large diameter necessary. For high pressure, and where the boiler can be laid lengthwise, the form of Fig. 105 gives the necessary length for the gases to give up their heat, and keeps the diameter down.

69. The Rectangular Marine Boiler. With the lower pressures used in the simple condensing engine rectangular fire-boxes or furnaces have been much used, and often the shell has been made with flat or arched surfaces so as to fit the lines of the vessel to a degree. The gases may be led from the furnace by flue or tubes to the back connection and then returned by flues or tubes to the front (Fig. 106). Sometimes the gases were returned on a lower level (drop-return-tubular or flue-boilers), and many combinations have been made of the tube and flue principle. The type of Figs. 101 and 105 is gradually replacing these weaker and less reliable forms as pressures increase. Flat-sided marine boilers are still to be seen for simple condensing engines in lake and river practice (Fig. 107), but even here the furnaces are usually cylindrical and either corrugated or stiffened with rings (Fig. 101). Oil or grease settling from the water upon these furnace-crowns and causing them to soften and come down from overheating is a very frequent source of annoyance and danger in boilers of this class, and is aggravated by unwise handling of the engines.

The conditions attending the use of the marine boiler in sea-going vessels call for a type of highest efficiency and best economy with



TYPE A.
FIG. 104.

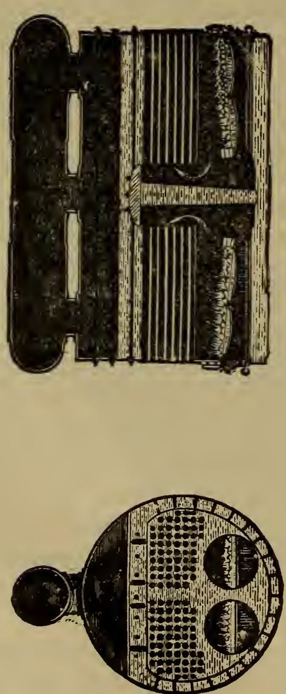


FIG. 103.

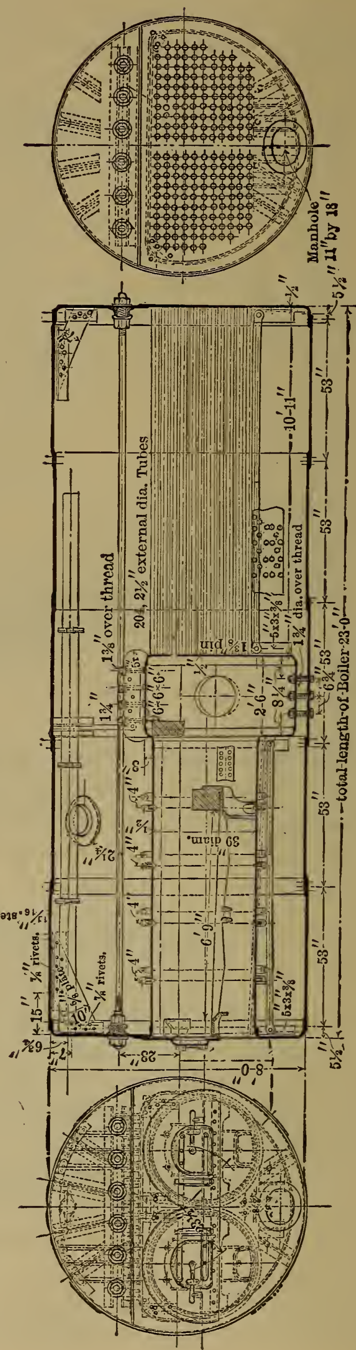


FIG. 105.

least bulk and weight. The vessel must carry its own coal, and have to spare for any delay in reaching its next coaling station. Hence boilers of this class stand very high as types. Domes for such boilers are inconvenient, and for large diameters will either be dispensed with as not required where a large steam-space is furnished by the large

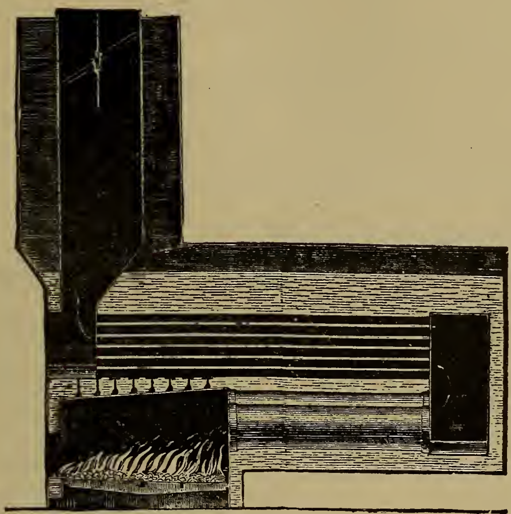


Fig. 106.

diameter, or a dry-pipe will be used. For smooth-water boats the steam-chimney is still much used (paragraf 49 and Fig. 106).

70. The Locomotive Boiler. The boiler which is to supply steam to the cylinders of the locomotive engine must meet the most exacting requirements.

(1) It must be self-contained, requiring no setting.

(2) It must evaporate a very great weight of water in a very short time to meet speed and grade resistances with heavy trains without such drop of pressure as will delay the train or time schedule.

(3) It must be exposed to low outdoor temperatures and snow and rain.

(4) It must have minimum bulk or volume.

(5) It must be efficient, since the engine and tender as a unit must transport their own water and fuel. The internally-fired type with many small tubes meets these requirements best. A high rate of combustion on the grate must be secured (paragraf 12) to enable a single grate to furnish the heat required per minute or per hour, with consequent high temperature of fire; the tubes must withdraw this

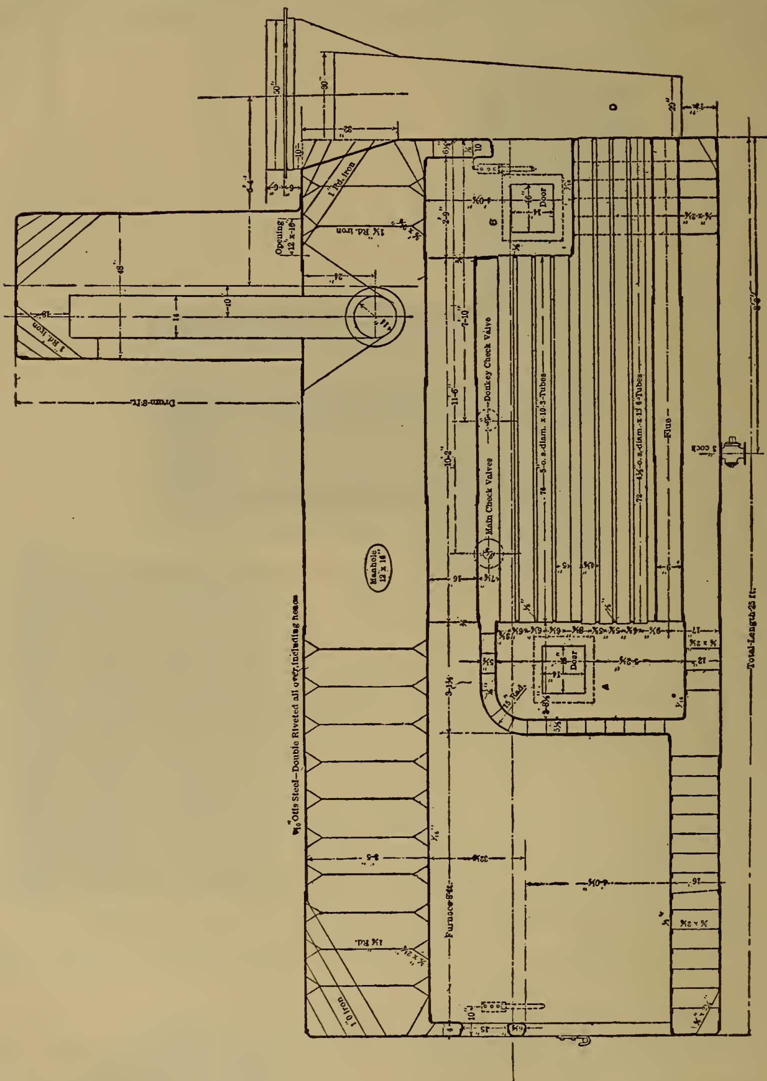

Opening 12"x16"
Drum 8 ft.
Manhole 12"x16"
Main Check Valves
Donkey Check Valve
Flue
Door
Door
Furnace 8 ft.
Total Length 25 ft.

FIG. 107.

heat very rapidly when the gases are passing rapidly, else the smoke-box and stack temperatures will be so high as to be wasteful; and combustion must not be incomplete.

The designer is limited also by the permissible diameter of the exterior shell; by the limit of width for the grates; by the possible length over all imposed by the curves of the line, and by the limit of human endurance in the fireman in the amount of fuel which can be charged into the furnace on a run of a certain number of hours or miles in length. Furthermore, unless the grate area is increased, the increased intensity of draft to get the combustion will force mechanically so much fuel out unburned through the stack as to overburden the firing process and produce coal waste. There can be little water storage in the locomotive boiler to store reserves of heat to any considerable extent, and the pressure will fall on up grades and rise on descents in spite of the best skill and experience in both fireman and engineman. If possible the feeding of cold water on up grades should be avoided.

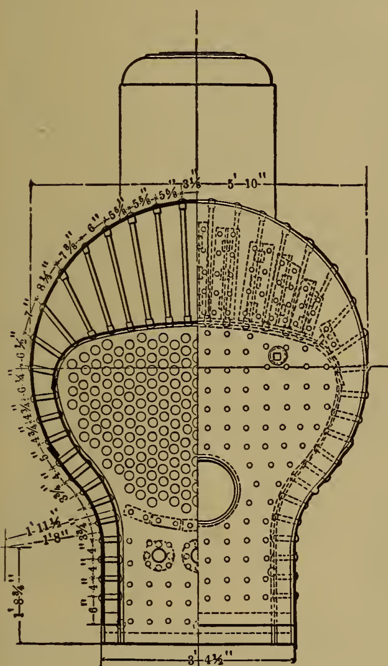


FIG. 110.

The boiler being horizontal perforce, the general lines of the preceding figure 105 will be followed, except that the fire-box or furnace will be more usually rectangular in plan, and the shell will be open on the bottom below the grate to let ashes fall into an external ash-pit. The rectangular section was early imposed by the supposed compulsion to get the fire-box end between the frames and the large driving-wheels. This gave rise to an end-view like Fig. 110, which is half elevation and half section through the fire-box. The crown-sheet is a flat arch stayed to the shell; the sides which form the "water-legs" are stayed by stay-bolts from coming in. To get a large steam-space, the boiler is carried up as high as possible over the furnace-crown, and either is prolonged on that line to the front, or is drawn in by a tapering ring beyond the fire-box to a smaller width and height. This tapering course is called the waist of the boiler, and the enlarged fire-end makes a "wagon-top" boiler from the suggestion of the canvas cover over hoops in early prairie wagons.

If it be preferred to avoid the different lengths of the radial crown-sheet stays of Fig. 110 and their different expansion under heat and stress, the flat crown-sheet and straight-side fire-box will be designed (Fig. 111), and either limited in width to the space over the frames between the wheels, or for small diameter driving-wheels carried over the latter also (Figs. 112, 113). If finally the wheels can be moved enough forward and nearer the center of gravity to get the fire-box behind the wheels altogether, the lateral limitations are removed, and the design of Fig. 114 becomes possible, limited only by the clearance width between two engines on parallel tracks. Fig. 115 shows the back-head in the Belpaire design strongly and rigidly stayed by gusset-stays.

In the illustration the fire-box is reached through the back by one door. If the fuel is to be anthracite in very small sizes, even the wide grates shown will compel too rapid a combustion rate, and a specially wide furnace with two doors of access will result (Fig. 116). The ash-pit only comes down between frames, and the cab is moved in front of the fire-box altogether. These extra wide fire-boxes were first brought out by a Mr. Wootton of the P. and R. Railway, and when made very long also (Fig. 117) have to allow an axle to come up pretty close to the top of the ash-pit.

The extra wide fire-box end shown in Fig. 114 will serve as a favorite type of locomotive boilers intended for stationary use, where the type is that best adapted for the service, but the limitations of the line are not imposed.

With the softer or bituminous coals, the rapid rush of flame into the tubes is not favorable to economical combustion. To give chance for flame combustion to be complete and at high enough temperature, fire-brick is introduced within the fire-box (Figs. 111, 115) or behind it (Figs. 117 and 118) to provide for such time and temperature as is required.

The illustrations also show the various types of crown-sheet bracing, either the crown-bar system of Fig. 112 for flat crowns, or the curved-bar and sling-stay system of Fig. 114 as substitutes for the radial bolt of Figs. 110, 113, and 117 of the arched-top types. They show also the straight and sloping back-head, and the method of constructing the fire-door opening through the latter in which the two plates are flanged inward to overlap and then riveted together. Or the space between plates at the opening may be closed by a solid forged ring like the "mud-ring" which closes the bottom of the water-legs around the

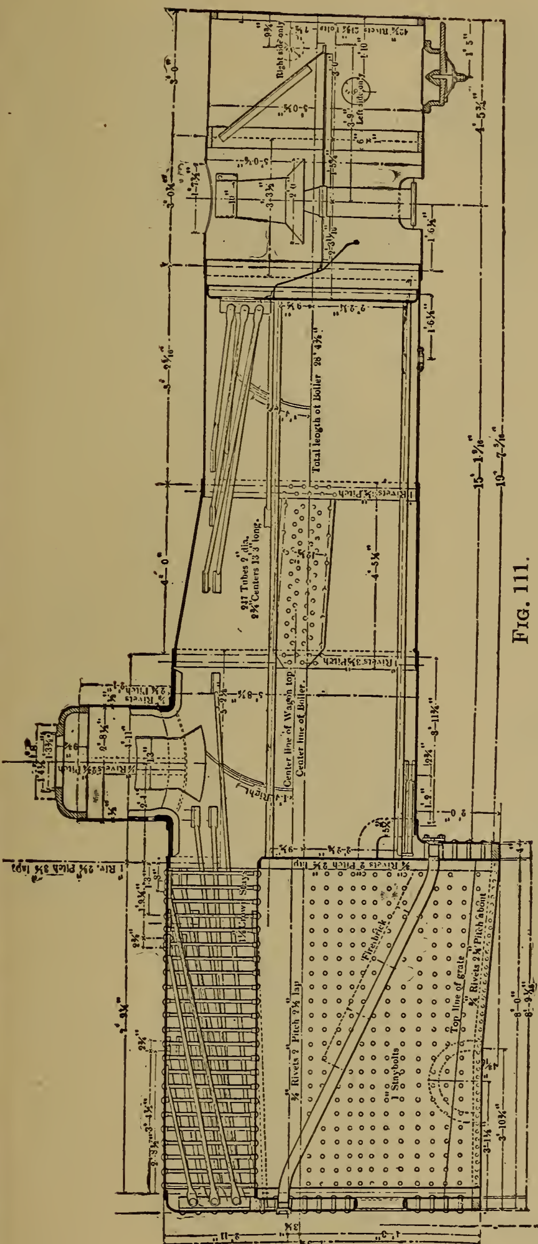


FIG. 111.

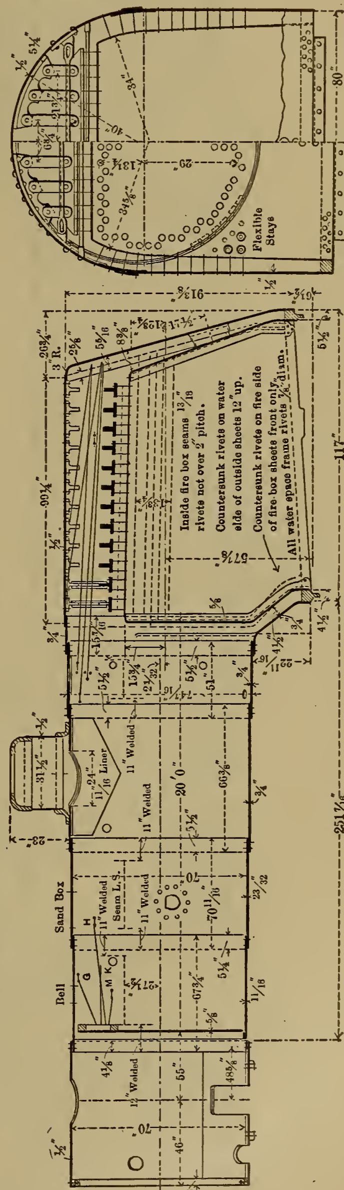


FIG. 112.

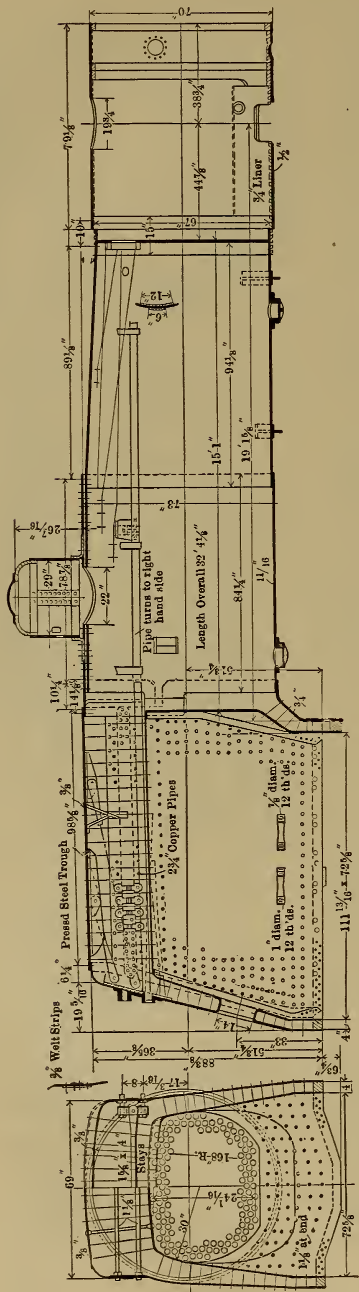


Fig. 113.

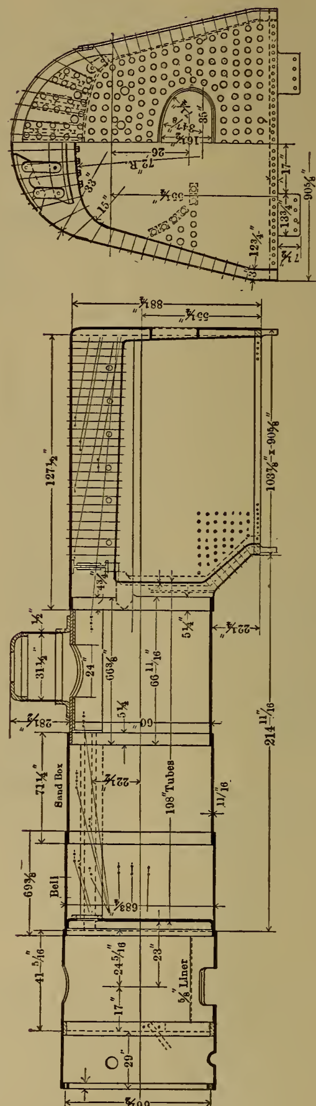


Fig. 114.

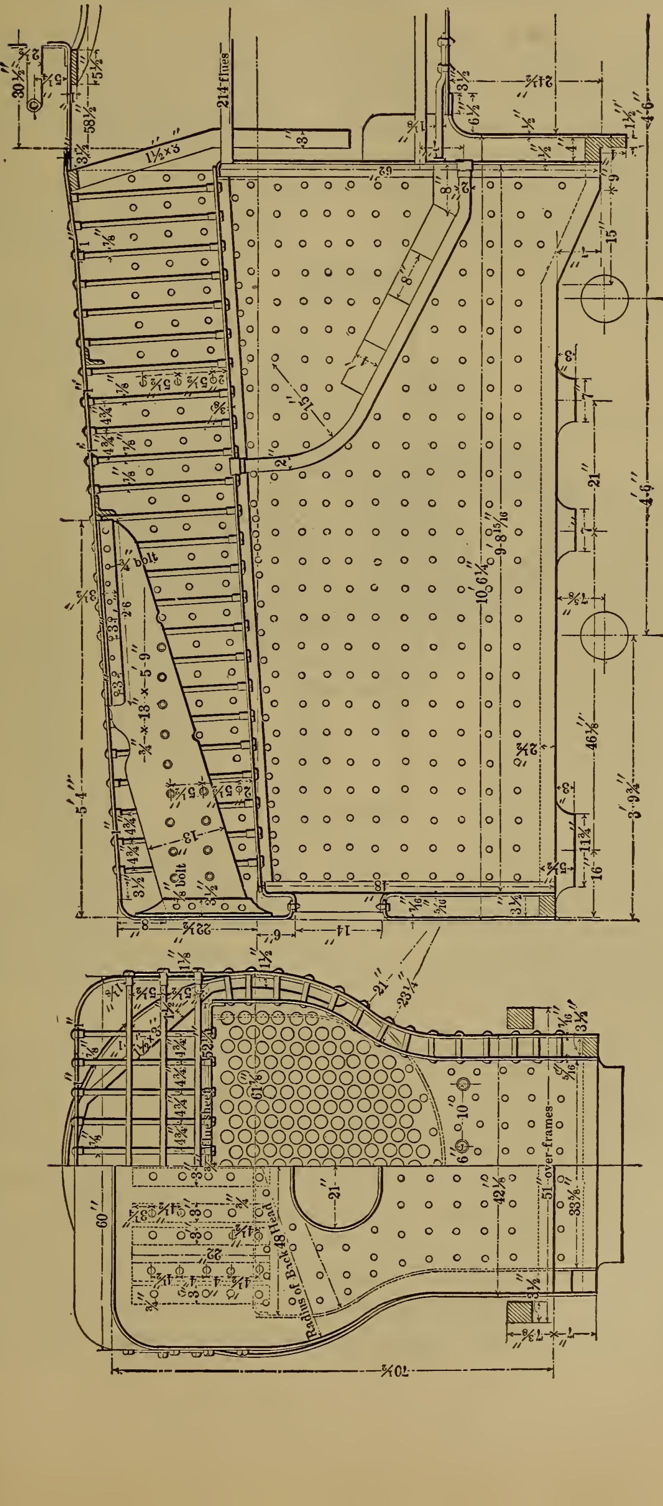


FIG. 115.

fire-box. Or, again, a ring may be made of U-section of plate of the shape of the opening and riveted to both surfaces. The mud-ring is almost universal for road-locomotive boilers; for stationary use, the inner plate may be flanged outward and downward to form the closure of the water-leg and joint with the outer sheet.

These boilers also all show the "extension front" smoke-box which enables the desirable spark-arrester or netting to be placed here below

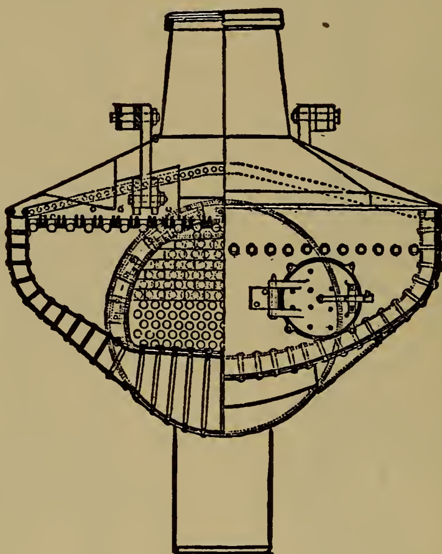


FIG. 116.

the orifice of the exhaust-nozzle (Fig. 111). The draft in the fire-box and tubes can also be damper-regulated without putting back-pressure upon the engine pistons, and the smoke-box can be used as a holder for sparks and cinders which can be conveniently removed at the end of a run.

71. The Locomotive Boiler with Corrugated Furnace. The application of the corrugated cylindrical flue to the locomotive after it had proved its advantages in marine practice has been made by Mr. George S. Strong and by Mr. Cornelius Vanderbilt, Jr. Fig. 119 shows such a boiler with three furnaces taking into a common combustion-chamber. From the front of the latter the tubes run to the smoke-box. The furnaces need no staying on either tops or sides, but the combustion-chamber with flat-top element is stiffened by crown-bars and hung by sling-stays. The ash-pits are troublesome to clean when coal is used as a fuel, and hence the single furnace is more usual. The design

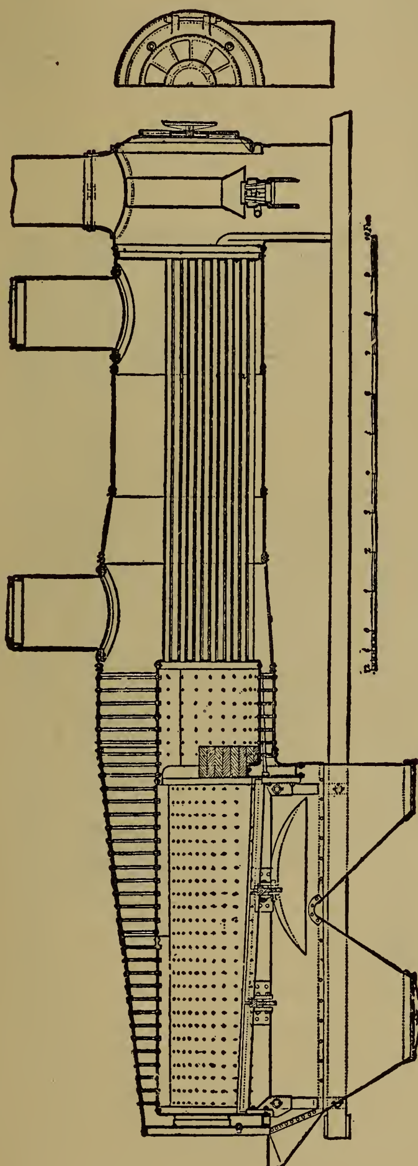


FIG. 117.

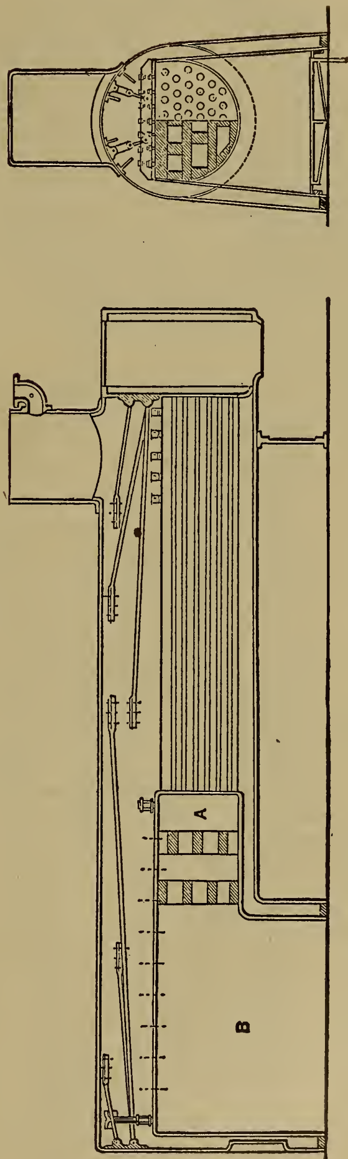


FIG. 118.

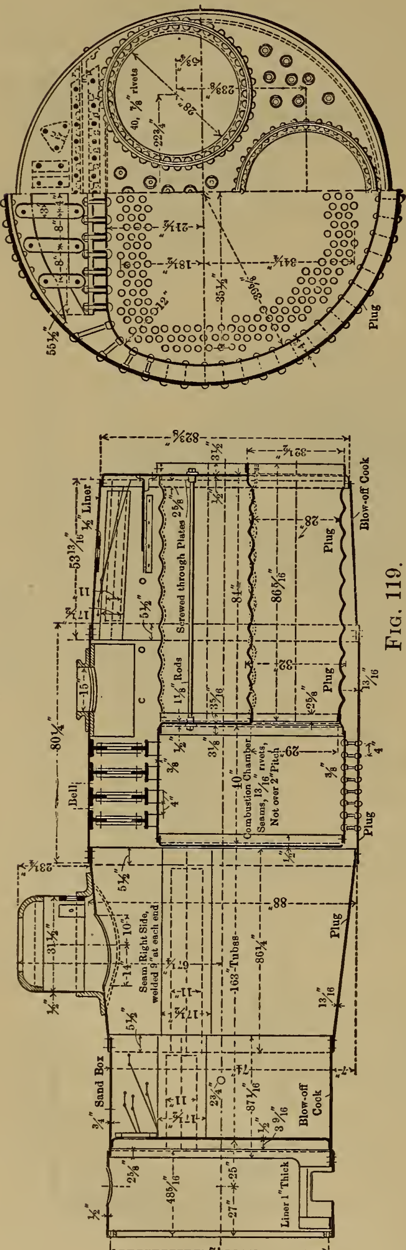
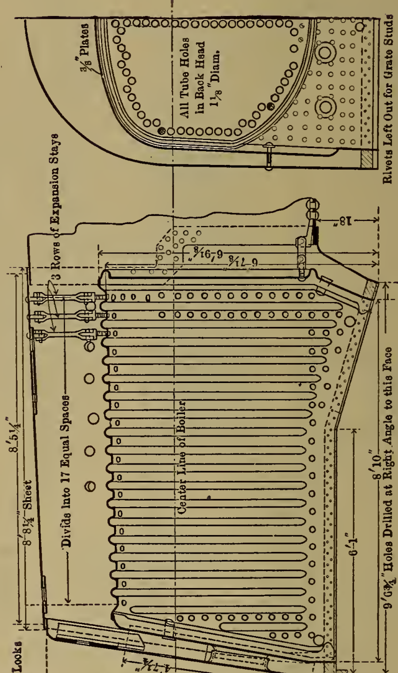


Fig. 119.



NOTE:
Holes for Gage Locks to be Left Out

Fig. 120.

shown is intended for use with oil-fuel, and this limitation is not present. Fig. 120 shows a fire-box with corrugations upon the plane elements. This is due to Mr. William H. Wood. Table VIII gives some data on the large locomotive boilers which have been illustrated, and on some others.

TABLE VIII

Fig.	Builder.	Railway.	Type.	Boiler Pressure, Pounds	Cylinders, Diameter and Stroke, Inches	Diameter Drivers Inches.	Weight on Drivers, Pounds.
	Am. Loco. Co.	N. Y. Central . . .	4–4–2	200	21×26	79	95,000
113	Am. Loco. Co.	Penna. R. R.	4–4–2	205	$20\frac{1}{2}\times 26$	80	109,000
	Am. Loco. Co.	C. R. R. of N. J.	4–4–2	210	$20\frac{1}{2}\times 26$	85	99,400
	Am. Loco. Co.	Nor. Pacific	4–6–2	200	22×26	69	134,000
	Bald. Wks . . .	C. R. R. of N. J.	2–6–2	200	18×26	63	108,000
112	Bald. Wks. . . .	Chicago & Alton	4–6–2	220	22×28	80	141,700
	Am. Loco. Co.	C. B. & Q.	2–8–0	210	22×28	57	187,000
114	Bald. Wks . . .	B. W. & Gt. Falls	2–8–2	200	$14+24\times 26$	50	128,000
119	Bald. Wks . . .	A. T. & S. F. . . .	2–8–0	210	$17+28\times 32$	57	191,400

Fig.	Weight, Total Lbs.	Heating Surface Fire-box, Sq. Ft.	Heating Surface, Total Sq. Ft.	Ratio of Heating Surface to Grate Area.	Tubes, Number.	Tubes, Length Feet and Inches.	Grate Area, Sq. Ft.	Per cent of Fire-box to Total Heating-Surface.	Fuel.
	176,000	180	3,505	70	396	16	50	5 1	Bit. Coal.
113	176,000	166	2,640	47	315	15–1	55 5	6 2	Bit. Coal.
	191,000	174	2,967	36	325	16–6	82	5 86	Fine Anthracite.
	202,000	175	3,462	74	301	18–6	47	5	Bit. Coal.
	165,000	137	1,832	33	249	13	54 5	7 47	Anthracite.
112	219,000	202	4,078	75	328	20	54	4 95	Bit. Coal.
	208,900	195	3,827	70	462	15	54	5	Bit. Coal.
114	166,900	174	2,496	44	270	16–6	56	6 97	Lignite.
119	214,600	165	4,266	. . . . . .	652	13–7	. . . . . .	3 86	Oil.

72. Derivatives of the Locomotive Boiler. The locomotive boiler received its type form from Stephenson at the successful competition in England in 1829. Its excellences in the way of rapid steaming, compactness, and economy have made it the starting-point of many modifications of type for stationary use, and it has also been adopted in torpedo-boat conditions. Fig. 121, for example, is a short, self-contained, non-radiant form without lateral water-legs, but a brick lining in both furnace and back connection. This fits it for economic use with bituminous or flaming coals.

In Fig. 122 a type of circular fire-box, combustion-chamber, and tube-nest is used without stays and their attendant cost and objections.

The whole fire-tube element can be removed for renewal without disturbing the shell or connections by cutting off the tubes and removing the front head.

These secure structural cheapness and the basal features of their type, for stationary use. The same excellences attach to Fig. 123, in which the furnace is corrugated with tubes leaving a tube-sheet at the

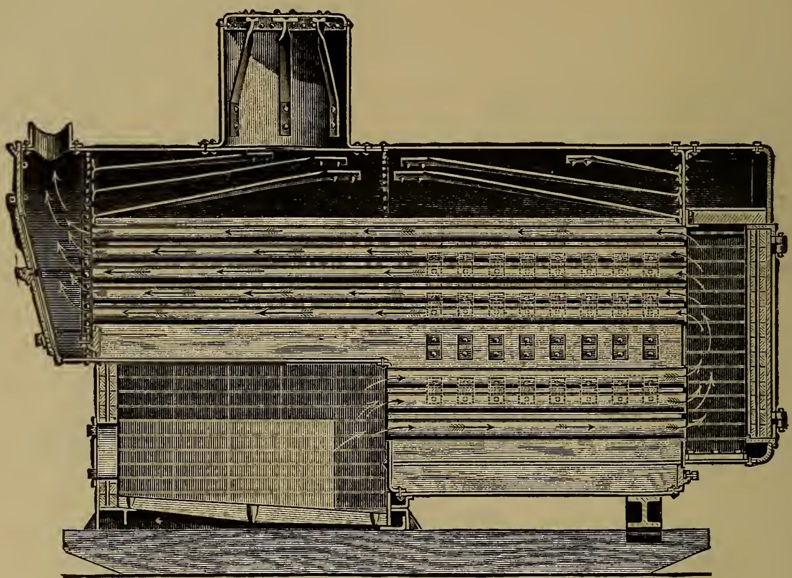


Fig. 121.

rear and leading into a surrounding casing of sheet steel lined with asbestos air-cell blocks. The water rises from the lower cylinder through necks into the water and steam-drum above, and is led off as steam from the perforated dry-pipe. The boiler carries its own setting and combines the features of external and internal firing.

73. The Upright Boiler. If the internal furnace of the Cornish type, or the marine type of Fig. 105, or the circular furnace type of Fig. 119, locomotive type, or Fig. 122 of the derived type, be any one of them turned up on end so as to make the tubes rise vertically from the crown-sheet over the fire, such a vertical boiler then becomes the upright (Fig. 125). The water surrounds the fire-box as before, and the latter is easily stayed to the cylindrical shell. The steam now, however, goes to the upper end, and the upper ends of the tubes traverse the steam-space above the water-line. Crown-stays disappear because much pressure is removed from the crown-sheet by the tube area, and the tubes stay much of the upper head. The water-legs are inaccessible except through hand-holes, and the crown-sheet filled with closely arranged tubes is hard to clean inside. There is no place for a dome or dry-pipe.

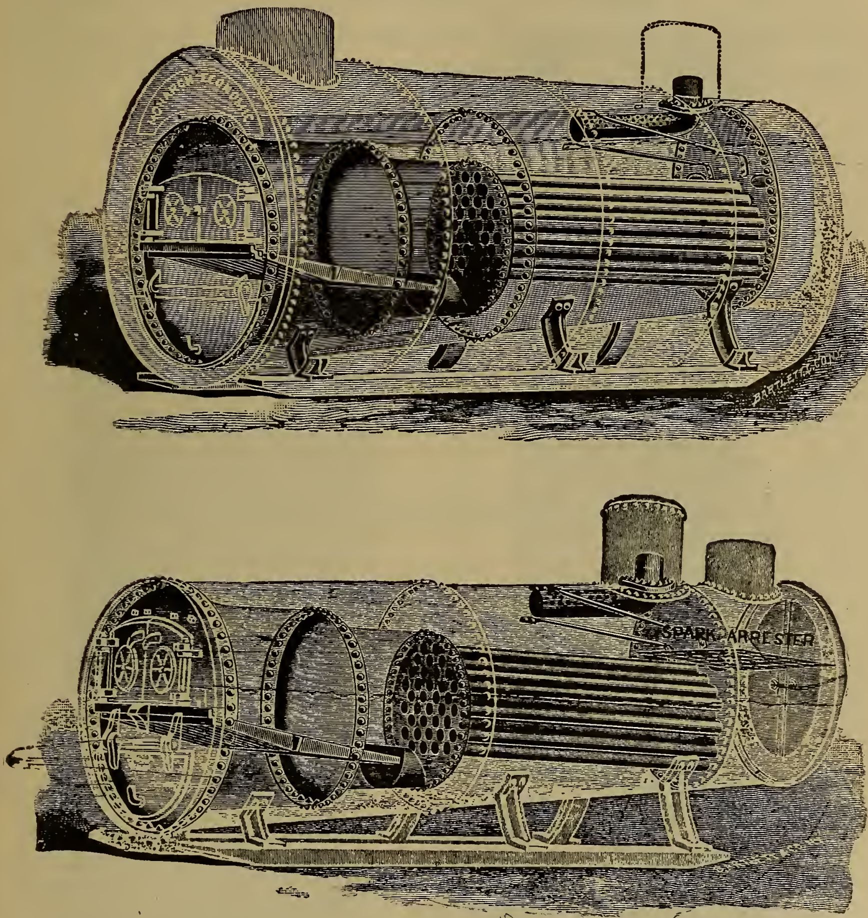


FIG. 122.

The features of the upright type are:

(1) It is light and portable.

(2) It requires no setting for the general type shown in Figs. 125, 127.

(3) It is a rapid steamer, because vertical tube-surfaces evaporate rapidly and the water is subdivided.

(4) It takes little floor-space.

(5) The upward motion of the hot gases is the natural flow of such gases.

(6) The simplicity of the stays makes it a cheap boiler.

On the other hand, it may be urged:

(7) The circulation is not determinate, and may be defective. Everything tends to ascend all over, and if water does not replace the steam made at the tube-sheet, the latter will overheat. This is reme-

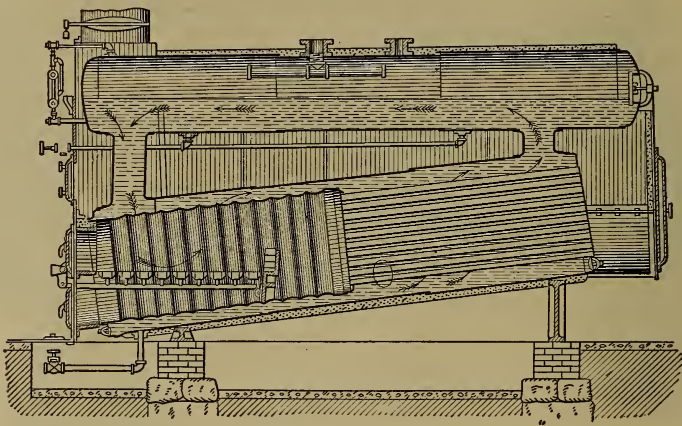


FIG. 123.

died in part by thinning out the tubes, so as to leave open spaces for water to move through, and by the use of baffle-plates or brattices to force a determinate circulation.

(8) It is troublesome to get a dome or a large steam-space. Wet steam will result if the flow is rapid.

(9) The upper ends of the tubes are not water-cooled in such a design as Fig. 125, but will grow very hot and expand so as to cause leaks at the upper tube-sheet. While such hot tubes may serve to dry the steam somewhat, the difficulty from unequal expansion is of sufficient moment to justify the design of Fig. 127, where the smoke-box is drawn down into the boiler proper to submerge or drown the ends of the tube below the water-line.

(10) The boiler cannot be entered for a personal inspection, and cleansing is not easy when it has to be done from outside. This is a very serious objection with many waters.

(11) It holds the least amount of water of any of the shell types, which makes it pass quickly from safe pressure to one which would endanger it. This danger is greater the smaller the boiler.

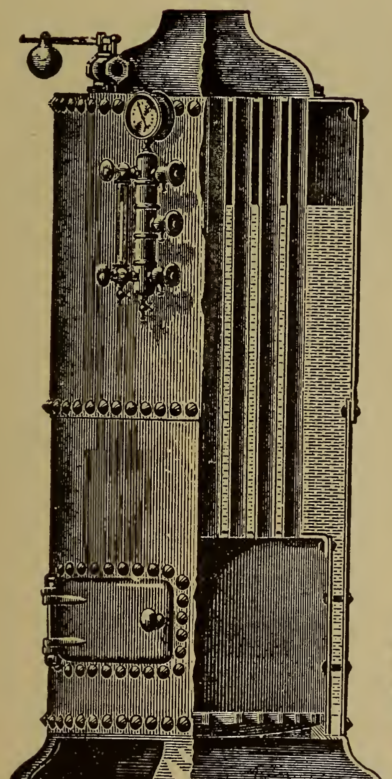


FIG. 125.

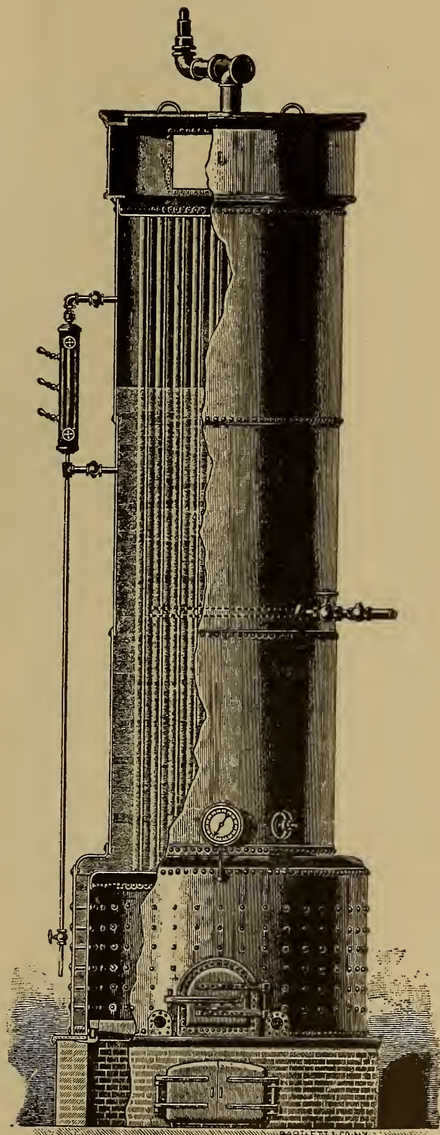


FIG. 126.

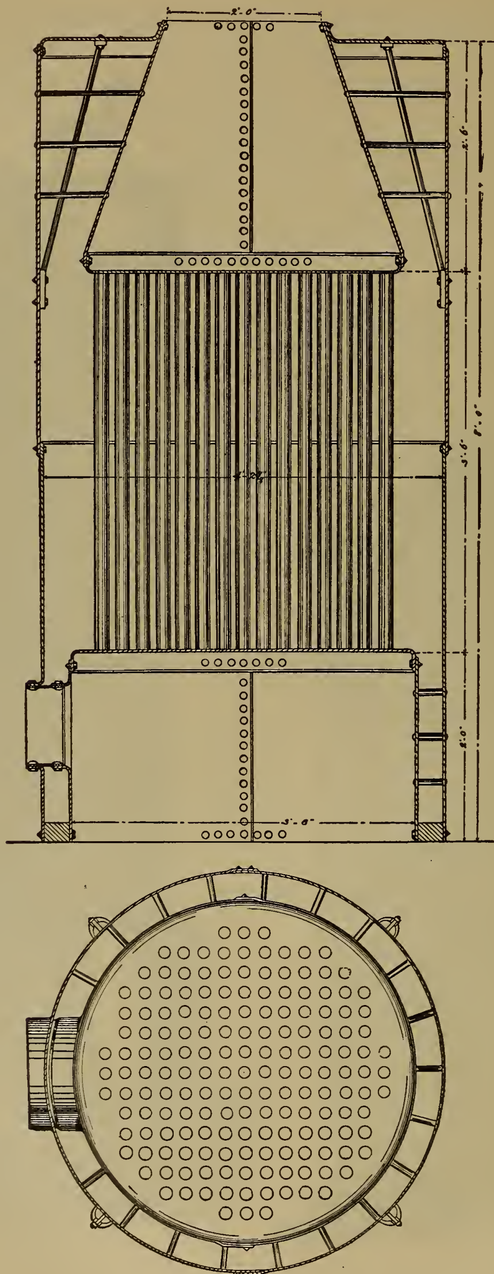


Fig. 127.

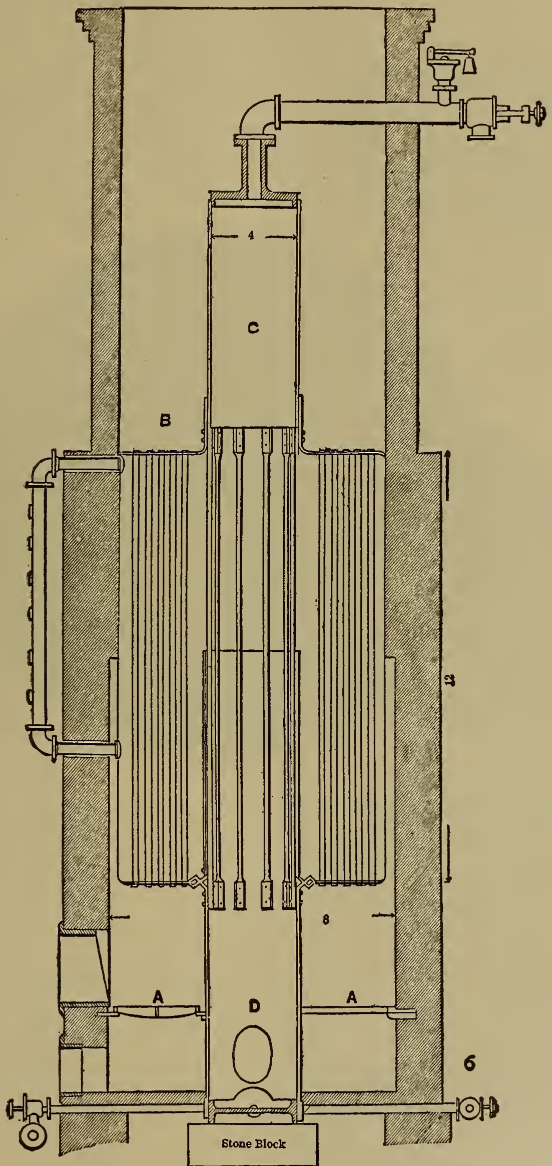


FIG. 128.

74. Modification of the Upright Boiler. To secure the convenience of the upright arrangement and at the same time avoid some of its defects, various special designs have been advanced. Fig. 126 is the Manning boiler, designed to secure a larger grate area and either a slower combustion rate or the consumption of more fuel than with the type form, and derive the advantage of reduced ground-plan for a required evaporative capacity. The upper ends of the tubes act as a superheating surface to dry and superheat the steam surrounding

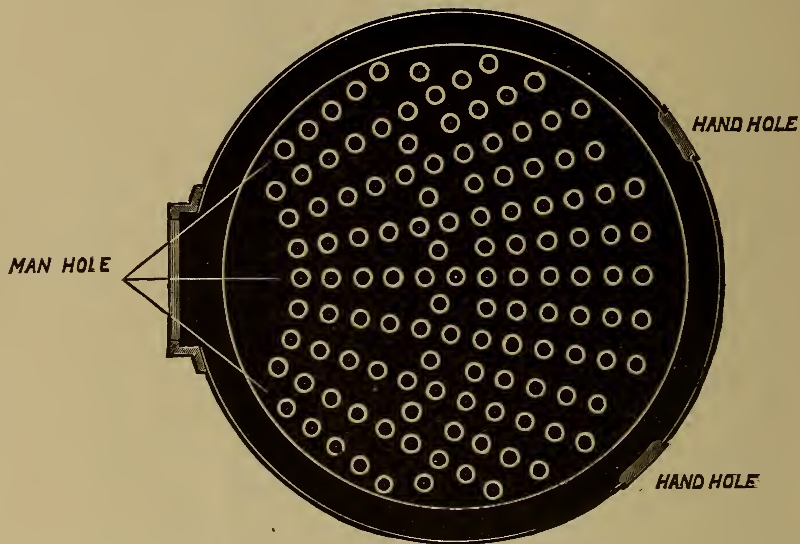


FIG. 129.

them, and the tube is long enough to compress or yield slightly under unequal expansion stress without starting the tube-sheet joints. In Fig. 127 the water-line is carried above the level of the upper tube-sheet, drowning the tubes under water and lessening the likelihood of working the expanded joints. The smoke-box cone is, however, exposed to collapsing stress from without and to high temperatures from within. It requires radial staying. Fig. 128 is the Corliss boiler, using an annular grate with several fire-doors in its brick setting, the grate surrounding a mud-drum. Above the upper tube-sheet, or in the base of the chimney as it were, is a steam dome surrounded by the hot gas and forming a superheating element. The water can be carried so near the top of the barrel that overheating of the tube-ends is secondary.

Fig. 129 shows the Reynolds boiler arrangement of tubes in such fashion that through a manhole full access is given between the rows to every part of the crown-sheet (Fig. 129).

75. The Fire-engine Boiler. The boiler for the steam fire-engine for municipal use must meet the prime necessity of light weight so that by horses or by its own motor it shall be rapidly run to the fire over paving or roads of ordinary sort. It must also get up steam to working pressure within three minutes in well-equipped cities if horse-drawn, or must have propelling steam-pressure to start with if self-propelled. The upright boiler meets the first and second requirement the best of any fire-tube boiler, and until the development of the water-tube types it was the only one in use. It was a submerged or drowned-tube type, such as Fig. 127 or Fig. 130, with copper or brass tubes as close together as they

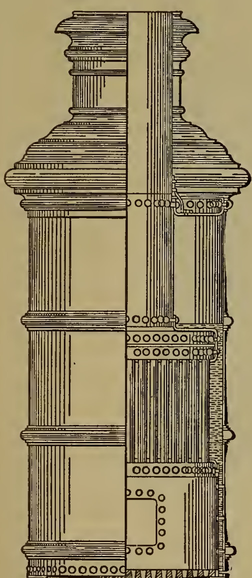


Fig. 130.

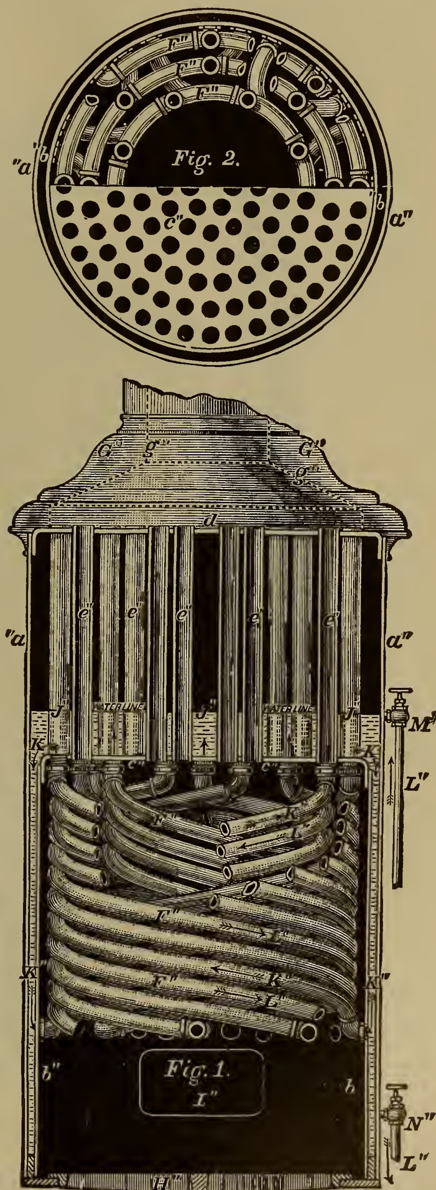


Fig. 131.

could be placed, and working with strong forced draft by steam jet. The later and other types are usually composites to avail of the advantages of the water-tube type as respects quick-steaming. Fig. 131, for example, has both fire-tubes and water-tubes, the latter drawing water downward in the water-leg and passing it upward as steam at the upper tube-sheet, while the hot gases pass through

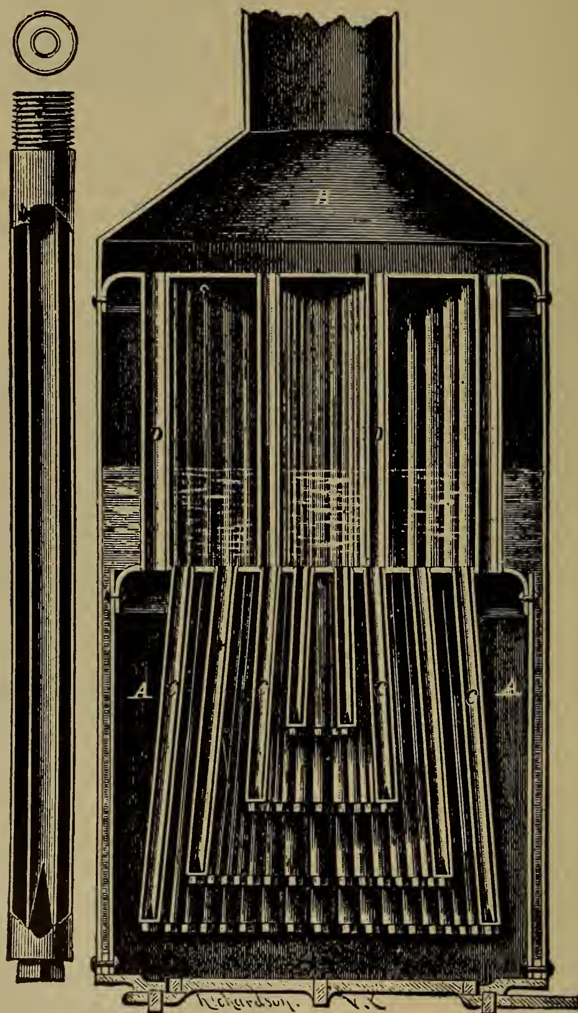


Fig. 132.

the fire-tubes and out at the top. The low water-line makes the fire-tubes mainly superheating surface, the steam being made really in the water-tubes.

76. Combinations of Type. Fig. 132 is also a boiler for fire-engine service primarily in which the fire-tube and water-tube principles are combined as in the form just referred to. The special type here is the Field tube (Fig. 6, paragraf 23), in which rapid circulation and self-cleansing are secured when the boiler is at work. It has been used also in tugs. Similar combinations have been made by putting water-tubes at the sides of an ordinary brick fire-box between the grate and the wall (Smith's); or by putting such water-tubes in the combustion-chamber behind the fire (Stead's). There are also unclassifiable propositions, such as that of mounting the horizontal shell upon trunnions, as the Corliss boiler would be (Fig. 128) if turned horizontal, and causing such cylinder to revolve slowly with little water covering a part of the tubes only (Pierce's). These belong to the specialist and the historian rather than to the practical student, and this place is made for them rather in the logic of the subject than because they are of practical import.

CHAPTER VI.

WATER-TUBE BOILERS.

80. Introductory. In paragraf 51 of Chapter III the two directions of development were pointed out. The one sought to subdivide the fire elements and surround small units of heat-carrying metal with the water in relatively undivided masses. The other development has been to subdivide the water-containing metal vessel into small units, while the fire and gas were in larger and integral masses. Instead of keeping a large kettle of water, and passing small currents of gas in tubes through the water, the tendency has been to multiply small kettles in the one fire. These kettles must all be connected together at some point — or at more than one — in order that they may deliver the steam they make into a common delivery pipe; and they must be supplied with feed-water from a common source. As the kettles must contain the water, and the fire be outside of them, such units, if tubular in shape, will be properly called "water-tubes." They must all of necessity be in an externally-fired class when the tube is small.

When these separate water-heating or steam-generating water-tube units act independently as steam-boilers and are grouped together as an aggregation to deliver their steam into a common steam-space or at a common disengagement area, the boiler as a whole becomes a sectional boiler. There have been water-tubes externally fired in the previously discussed forms (Galloway, paragraf 67; Fire-Engine, paragraf 75), and the water-tube principle was used in old navy practice with rectangular shells (paragraf 69); but here there was retained the enveloping shell within which these water-tubes performed their work. Only a part of the heat from the fire entered the water by way of the water-tubes, or these water-tubes were only part of the heating-surface. When the heating-surface is largely or exclusively made up of such water-tubes or other generating units, and the large-diameter enveloping shell disappears from around such units, the boiler becomes a sectional boiler. The forms of generator in which this water-tube principle has been embodied are:

(1) The Plain Cylindrical Boiler. (3) The Sectional Boiler.
(2) The French or Elephant Boiler. (4) The Coil or Pipe Boiler.
(5) The Flash and Semi-Flash Boiler.

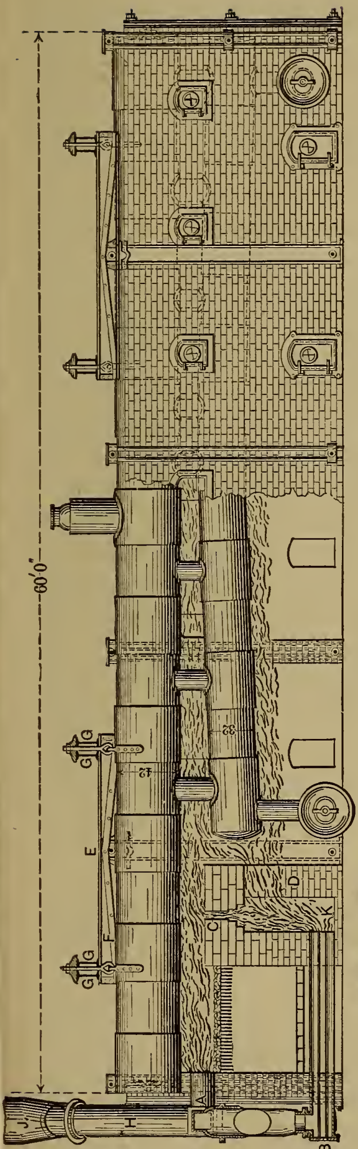


Fig. 134.

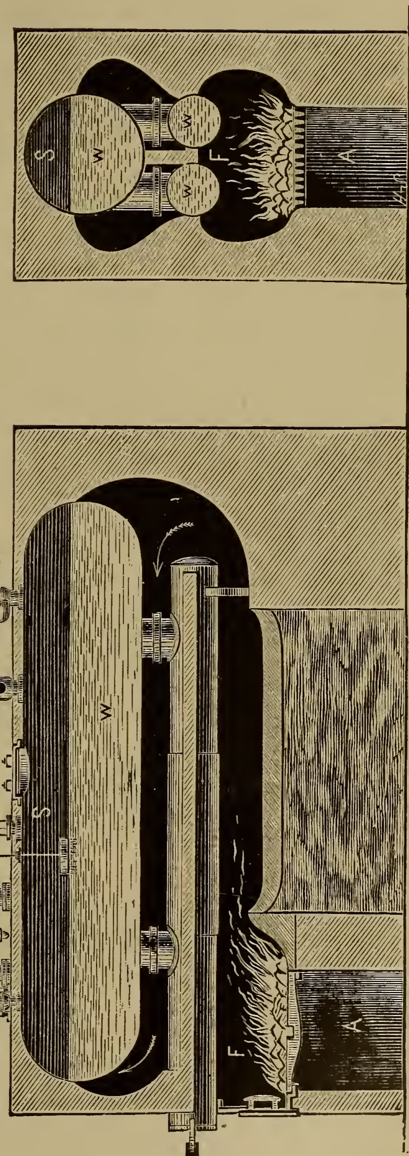


Fig. 135.

81. The Plain Cylinder Boiler. The discussions in paragrafs 23, 25, and 51 have established the cylinder with hemispherical or flat heads as the better shape than the spherical for receiving the heat of the fire and storing such energy in the form of pressure (Figs. 10, 25, 50, and 69). But when the temperature of the fire increases in an effort to burn the coal more rapidly and make more steam or at higher pressure, the only possible way to secure area of contact to absorb such heat is to lengthen the cylinder of the boiler. The same is required when the length of the flame becomes great with flaming or gaseous coals. The old solution (Figs. 7, 135) was to flex the gas or flame-current back around the sides of the boiler — the "wheel-draft" of historic settings — when the pressures were low. Such lengthening makes an expensive setting in the brick which supports the boiler and forms the external flues, and occupies ground-space which may be costly or prohibitory. If at the same time that length is demanded, it is desirable that storage capacity for heat in the water be supplied; and if it be also sought to increase the diameter of the shell, then the weight of the mass of water makes the stresses on the shell inconveniently great as respects its support. A trend of solution for the very long boiler for long-flame gas is shown in Fig. 134, where first the long cylinder is hung from equalizing levers pivoted at the center of each so as to distribute the load equally at each of the four points of support on the shell; and secondly by doubling the cylinder capacity by the water-drum connected to the upper cylinder by necks or nozzles. A second solution was to break the long cylinder into two halves (Fig. 136) at or near the middle,

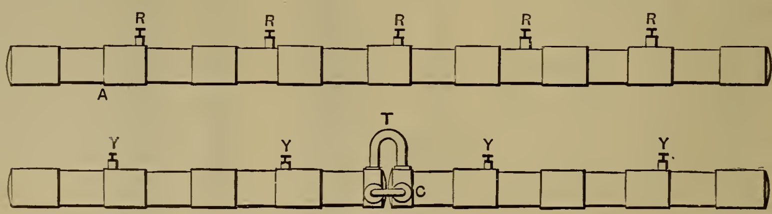


Fig. 136.

supporting each half at two points and coupling the two for steam and for water by flexible copper connections which would yield slightly as the boiler tended to flex upon its supports. Such boilers had the advantage of simplicity in construction, and access for cleaning was complete. They were bulky and not economical.

82. The Elephant, Union, and French Boiler. If now the principle of Fig. 134 of a double-deck arrangement of superposed cylinders be carried out in a shorter length of axis to get surface and storage without excessive length but by increase of diameter, the type known in the United States as the Elephant boiler results (Fig. 137). Less area in ground-plan is required, but more height; the cylinder diameter of each unit becomes less than if that same water capacity had been secured in one shell of the given length, and hence the shell is stronger against rupture (paragrafs 24 and 25) from internal pressure. If the boiler *C* in Fig. 137 was fitted with fire-tubes, as was quite a common

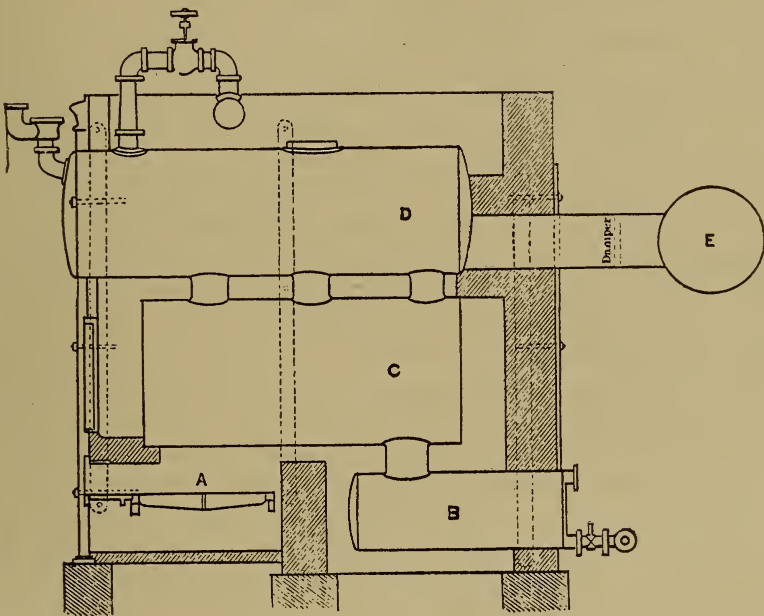


Fig. 137.

practice for non-flaming fuels, the composite was called a Union boiler; or the two or three necks uniting the elements *C* and *D* might again be replaced by an enveloping shell of plate connecting the upper and lower cylinders along a larger area of the elements of both shells. This is no longer in use.

When instead of one lower element *C* in the Elephant type, a pair or three smaller cylinders were placed below the main shell as in Fig. 135, the type becomes the French boiler, much used in industrial France, where it originated. It reaps the advantages of large water-content with small shell diameter, and increased heating-surface due to the double travel of the gas. The circulation in all forms of double boiler is indeterminate and variable after steam begins to form in the lower boiler, and such steam has to make its way through a great vertical height of water to find its disengagement area.

But by the multiplication of cylindrical units, the trend of development toward the sectional principle has been started, and its attendant advantages. What is the sectional principle, and what are its advantages and disadvantages?

83. The Sectional Principle. The principle of the sectional boiler is not the same thing as the actual construction of the generator which embodies it. There are advantages and disadvantages which are secured or missed by certain forms more than others. A sectional boiler is a steam-generator in which the plan of a single enveloping shell to contain the water and steam is abandoned and is replaced by that of a number of small generating vessels so joined together that the steam formed in all of these separate units or sections is delivered from a common disengagement surface into a common steam-space. The sectional principle may be carried in a boiler of large capacity to the extent of subdividing the disengagement area, so that the steam from several such areas shall be delivered into a common steam-drum, from which it shall be withdrawn by the steam-pipe.

The generating vessels in the forms to be discussed hereafter will be found to be tubes either straight or curved, and inclined at small angles from the horizontal or vertical plane. The only exception is the Harrison sectional boiler (Fig. 138), made up of cast-iron or cast-steel spherical or spheroidal units jointed on three or four or six faces to other similar units, forming in effect tubes or banks of spheres which deliver their water and steam at an upper disengaging area. When the water-tubular units become of small diameter, and instead of connecting at one or both ends into transverse headers such generating elements become a continuous coil of pipe, or a combination of pipe and fittings, the name sectional is less easily applicable, and the boilers become coil or water-tube generators. When the coil or pipe is used as a flash or semi-flash generating surface, the name sectional or coil is again not used, but its flash property is emphasized.

84. Advantages of the Sectional Principle. Among the advantages derivable from the use of the sectional principle may be listed:

(1) By subdividing into sections, each section has a small diameter, or one much less than that of the shell of the shell boiler. Strength to resist rupture with a given internal pressure increases as the diameter is less (paragraf 207). Hence each section is far safer against rupture than the large shell with same thickness of metal, and the danger from explosion of the boiler is much more remote.

(2) The rupture or failure of any one of the units from overpressure

or deterioration from any cause should not and usually does not cause the failure or loss of the whole structure. The failure of the unit should act as a safety feature, whereby pressure is released at one place only and before the other units are involved in any serious overstrain. Furthermore, the repair of any unit or section makes that part as good as new, and in this way the parts of the boiler may be gradually replaced and the whole structure become really new. It is, therefore, the safety of this type of boilers which has given it the great development of recent years as the pressures of steam have been increasing. The safety is not from the avoidance of all possible harm which a rupture may entail. The injury from escaping steam or hot water may be as fatal in either case. But the sudden release all at once of the enormous energy stored in the water of a boiler is much less likely to occur, and the train of disaster is avoided which would usually follow in the case of a similar failure of a large shell.

Since the great reduction in diameter which comes when the units are tubes would give unnecessary strength if the same thicknesses were used, it is more common to have thinner metal for the tubes. Hence follow:

(3) Lighter weight for a given evaporative capacity.

(4) Thinner tube-metal in the fire or gas-currents makes rapid transfer of heat to the water to be evaporated, so that the heating-surface is efficient, or a less number of feet of heating-surface becomes permissible, though not always advisable.

(5) The sectional construction makes the boiler portable and manageable so as to be put conveniently in places where access is difficult. The shell boiler must be handled as a whole, or built in place, if the doors or openings in walls are not large enough to pass it in or out as a whole. Sectional boilers can be put under finished buildings, or can be shipped beyond rail or water transportation and there assembled.

(6) Repairs and renewals are easy, cheap, and rapid, and can usually be made by available labor and skill, and entail but a short stoppage of the plant.

(7) The mass of the boiler which receives the action of flame and heat is less than in shell boilers.

(8) Sectional boilers can be driven further above their nominal capacity than shell boilers. In the horizontal tubular type such driving may be carried a little over 10 per cent; in sectional types it may be over 50 per cent excess, and even for a while as high as 100 per cent.

85. Disadvantages of the Sectional Principle. The sectional principle, however, offers certain disadvantages, also irrespective of the method followed in applying it; but the degree in which any given form suffers from them may be different.

(1) The aggregation of units must be connected together steam and water-tight under pressure. Unequal expansion (or contraction) of different parts or units must strain or loosen these joints, or flex or distort those parts whose length is changing, or wrench those to which they are attached if rigidly fitted to them. Efforts to mitigate this evil have given rise to the curved tube for the unit, instead of the straight tube, and underlie some forms of flexible connections for the units.

(2) The small unit principle precludes the idea of immediate personal access to the inside of the tube-surfaces for cleansing and inspection. This must be reached in some way or other in any form of generator which is to be properly called a safety-boiler. Hence there has resulted a prevalence of straight-tube units, to which access can be had from the end through a proper cap or hand-hole lid, and there is usually a cap at each end, in order that inspection of the inside of the tube may be made with the eye at one end of the tube and a light or torch or candle at the other. The cap feature is also a necessity if a tube is to be renewed without dismounting adjacent ones.

The objection to the cap feature is the multiplication of joints, which must be faced or ground joints so as to be tight without gasket or packing, and which are an occasion of leakage, and therefore of corrosion, when not attended to most carefully. The multiple-joint objection belongs also to some types which have no caps.

(3) The necessity for combining the evaporation of several tubes or units into one common duct or header, which is present in most of the types, makes the effective disengagement surface become only that part of the water-surface which is near the outlets of these headers. The disengagement is therefore tumultuous at such points when the boiler is driven, and water-gauges applied near such parts may show a fictitious water-level; and if the steam-outlet has to be near such part of the drum, the boiler is likely to prime.

(4) The circulation of the currents of water in any boiler is due partly to the presence of steam-bubbles, which are lighter than the hottest water, and partly to the action of the less warm water, which is heavier than the hottest water. In a shell boiler this circulation is untrammeled by any narrow passages where high velocity is called for. In the sectional types the circulation must be determinate; and if all units are to be full of solid water, the descent of cooler water must be just as fast and positive as the ascent of the steam-gas bubbles to the surface of disengagement. Where friction or scale or bad design

prevents this free descent of heavier water, and where steam formed in the units displaces water but cannot itself escape to the steam-drum, the unit becomes overheated and oxidizes and corrodes, and its overheating lengthens it unduly and produces the difficulties discussed above under (3).

Defective or impeded circulation with waters which deposit scale causes the scale to settle in the tubes or units, causing them to overheat and lengthen and produce the same trouble. Grease or oil depositing on the upper elements of tubes produces the same result.

(5) Since the water is within the tube or unit, with pressure on it, the failure of such tube or unit compels the whole structure to be put out of use for the repair. When a fire-tube fails, a plug of pine wood can be fashioned for each end and securely driven home from without. The leakage swells the wood and keeps it tight, while preventing the wood from burning further than a protecting thickness of charcoal on the outer surface. Such plugs will last for months if a shut-down is inconvenient.

Furthermore, where tubes are attached in nests or groups, the repair to a middle one can only be done by removing those tubes which are outside or around it and which may not need to be removed for any other reason. This consideration has dictated the prevalence of the straight-tube type arranged in essentially parallel rows, and with free space in the line of the tubes endwise.

(6) Tubes or units which are so shaped or fitted that they cannot be inspected inside by the human eye for their entire length, or which are so curved that cleansing by scraper is uncertain or even inconvenient, are to be objected to or condemned outright, for many conditions if not for all.

(7) The gases pass too rapidly through the necessarily limited length of the tubular units, and leave the setting at too high a temperature, without having given all their available heat to the water.

(8) The workmanship and parts of the sectional boiler make it costly, per unit of capacity, as compared with the fire-tube shell boiler. While prices vary, the sectional is apt to cost from one and one-half times to twice as much as the shell boiler, or from \$11 to \$18 per horsepower, with an average of \$14 or \$15 in large sizes.

86. Classes of Sectional Boiler. The sectional-boiler principle may be attained in many ways, but they will group themselves for examination into a small number of classes.

First, the units may be spherical or tubular. There are few examples of spherical units; the other class is more prevalent.

The tubular class may include:

(1) Straight tubes.
(2) Tubes curved at ends, straight in middle.
(3) Tubes curved for their whole length.
(4) Closed-tube types.

The straight-tube class may have the tubes inclined at about 15° from the horizontal, or inclined from the vertical, so that they are sometimes called, respectively, horizontal or vertical tubes; and the curved-tube classes pass into the coil type when the curvature becomes continuous for any one tube for more than 360°.

These several types in practice are identified by their builders' or originators' names. The predominance of one type over another is so often in any one locality a matter of business enterprise or commercial

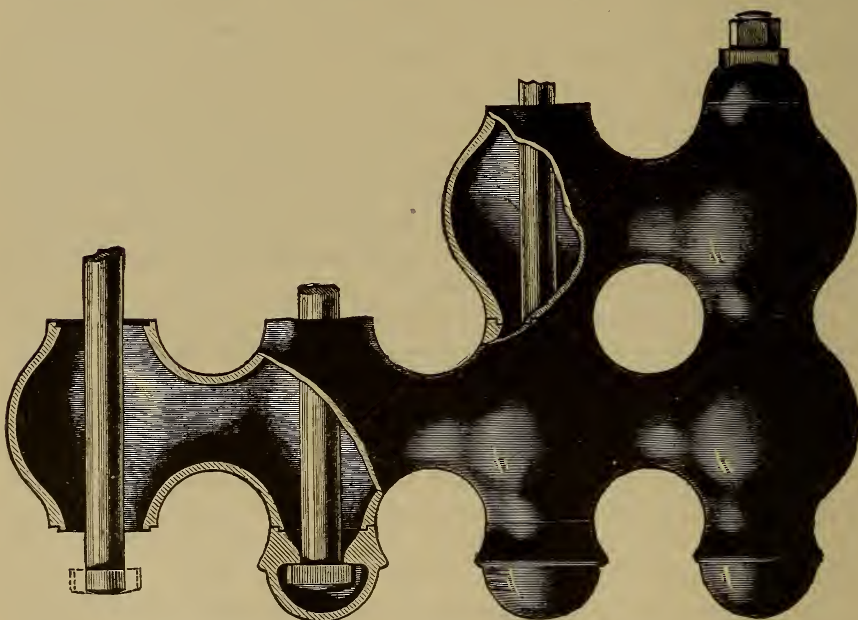


Fig. 138.

achievement, and the improvements on each type are so much conditioned upon a leading personality in each period of use, that it becomes unsatisfactory to treat of the individual types by name or at length. Certain typical forms alone are presented.

87. Spheroidal Unit Type. In Fig. 138 is shown the detail of a sectional boiler built up of spheroids. These were of cast iron with flanges in the earlier forms, and latterly of steel castings of

Bessemer metal. These each have circular openings which fit similar openings in the units above and below them, making a metal-and-metal rabbet-joint. The series of units is tied together lengthwise and crosswise by wrought-iron tie-rods, which come out through the cap which closes the last openings in any series. These tie-rods not only provide the strength to resist the tendency to separate, and furnish a flexibility for the connections, but also under excess of strain they will stretch

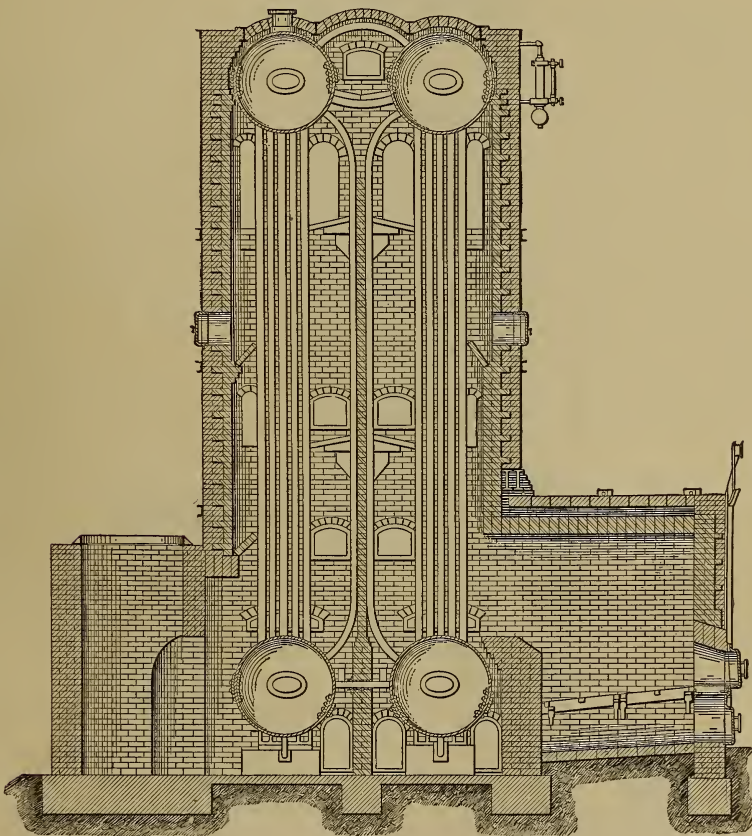


Fig. 139.

enough to cause the joints of the units to leak and relieve some of the pressure. The wrought iron also gives to the cast-iron whole some of the properties which cast iron would lack if used alone or altogether. The boiler has some surface in the steam-space exposed to hot gases, which gives a superheating area which tends to dry the steam when the boiler is working slowly. It offers some of the disadvantages discussed in paragraf 85.

88. Vertical Straight Open-tube Type. As an example of subdivided generation in straight vertical tubular units open at both ends, Fig. 139 may be cited. Four-inch tubes are expanded into molded nozzles in

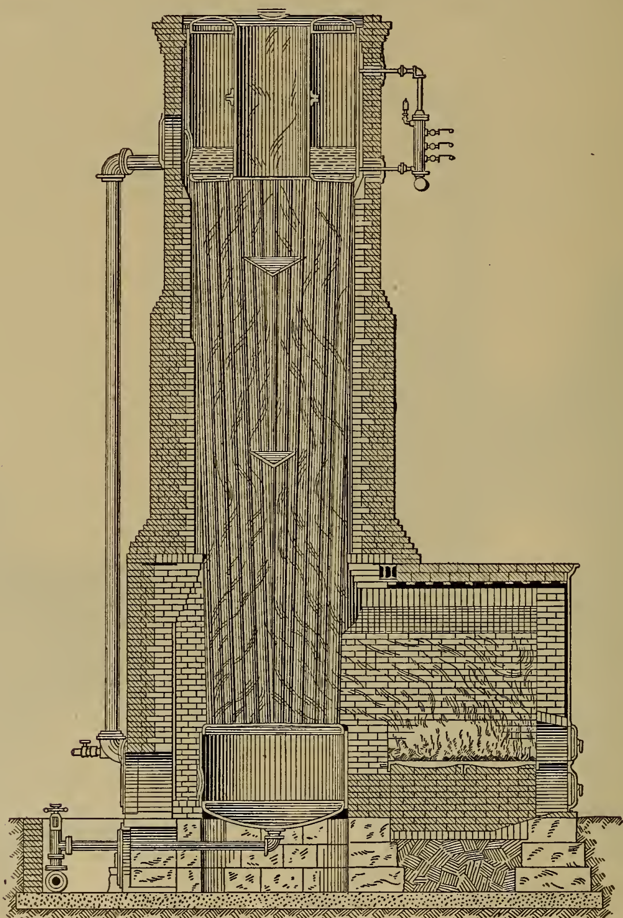


FIG. 140.

the mud-drums below and the steam and water-drums above. Both sets of horizontal drums are cross-connected. The fire in a Dutch-oven type of furnace where the radiant fuel is not cooled by the boiler-surface sends its gases up the right-hand section and down and out to the chimney on the left. Baffles force the hot gas into intimacy with the tubes. The descending current is at the cooler left-hand half and is determinate and effective. The drums do not touch the brick of the setting, enabling inspection to be satisfactory. The tubes are so spaced that a defective inner one can be removed and replaced without disturbing the others by working from within the drums. The whole boiler rests on the castings at the bottom and the vertical walls are inclosing flues. Scale is not likely to settle in such vertical tubes but in the drums below the circulating pipes.

In Fig. 140, there is no return of the gases, but they pass up through a sort of steam-chimney which is surrounded by steam. Here the circulation is made positive, definite, and powerful by a descending pipe outside the setting altogether and not exposed to action of heat. The lower drum acts as a mud-drum below the circulation pipe. As in the previous case, defective or injured tubes must be withdrawn and replaced through the gas-channel in the center. Unequal expansion is not supposed to occur, by reason of the unlikelihood of scale settling on the vertical surfaces. Inspection is done from the upper and lower drums, which are large enough to admit of entry; and the tubes being open and free above and below, they can be seen throughout their whole length, and cleaning apparatus can be introduced and effectively used. Expansion of the chimney-plates is permitted and favored by the flanged ring-joint. The disengagement from each tube is free at the top and independent of all the others.

89. Horizontal Straight Open-tube Type. The name horizontal is intended to include the types in which the generating tubular units are inclined at comparatively small angles to the horizon. The elevation of about 15° at one end both favors the circulation and compels it to be definite, and also makes room for the fire-box, in many of the types. The greater length of the water-column at the back end of the boiler gives the weight of solid cooler water by which the water and steam-bubbles are forced upward, at the front or over the fire.

In Figs. 141, 142, and 143 is presented the type which has the least sectional features. The generating tubular units are expanded into the inner plates of two water-legs of steel plate front and rear. These legs are flanged at the top and connect to the steam and water-drum above. The legs are stay-bolted. Opposite each tube at each end is a cap over a corresponding hole in the outer plate, so that any tube can be removed and replaced endwise. Expansion of any tube by heat more than its neighbors has to be taken up by bending of the tube or by compression which strains the joint with the water-leg. The leg is too stiff and strong to bend, and is held by the other tubes. Scale and oil render the horizontal tube liable to overheating, or any cause which prevents intimate contact of water and the tube-metal exposed to heat.

In Fig. 144 a side section is presented of a type of wide-spread acceptance. The generating water-tube units are in banks or nests of seven

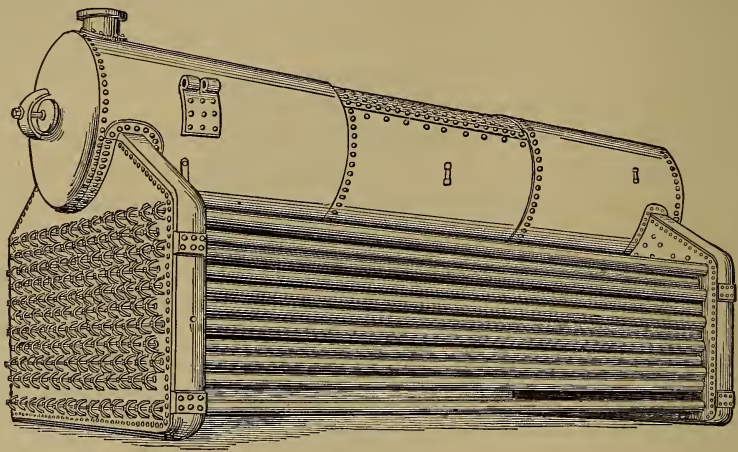


Fig. 141.

staggered vertically (Fig. 145) expanded into a steel header (Fig. 147). This header has the hole and necessary cap opposite the tube for inspection, cleansing, and renewal (Fig. 146), and all the steam made in the

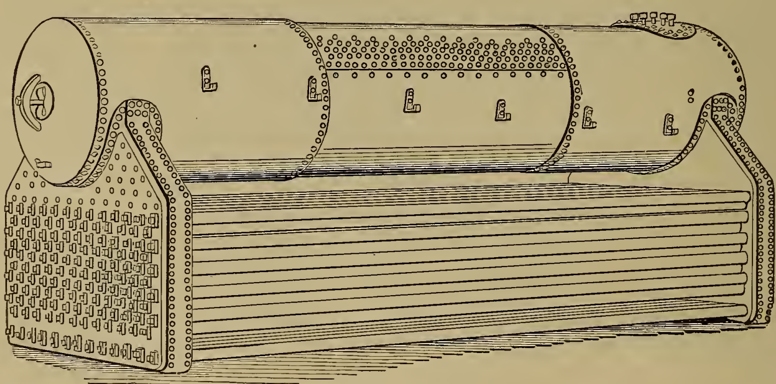


Fig. 142.

bank of tubes rises to the disengagement surface through the connection at the top of the header. This makes the circulation through that upper neck so rapid as to be tumultuous, and makes it desirable to take the steam away from the boiler at its back end. The water-level at the front will be higher than at the back when the boiler is being forced. If the number of nests or banks of tubes is small, there may be only one upper steam and water-drum. When a number are put in one furnace the drums will be two or three in order to secure disengage-

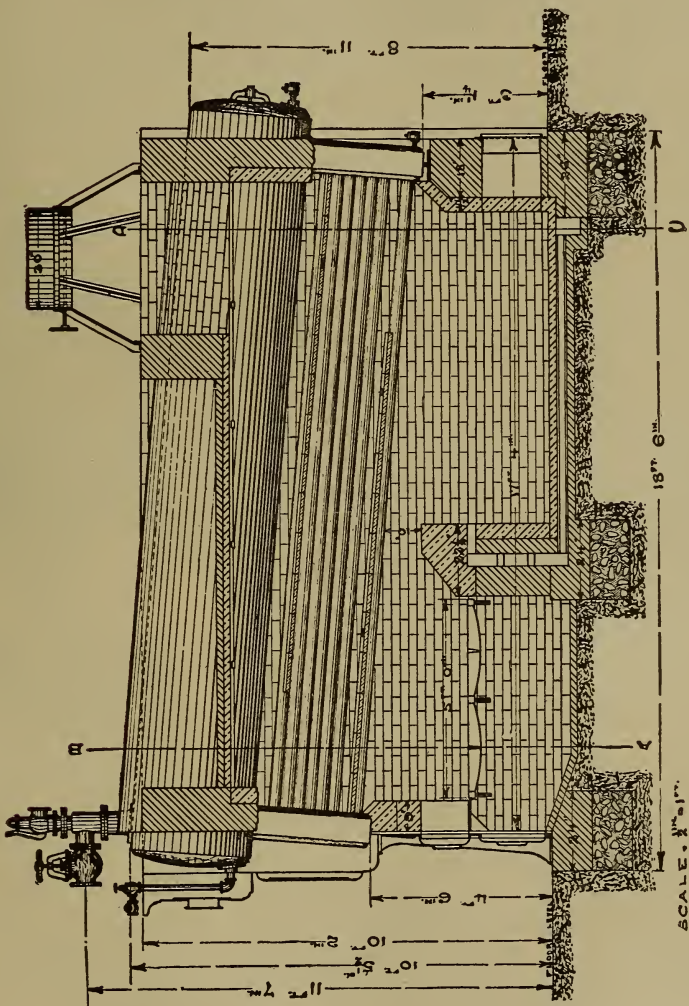


Fig. 143.

ment area without inconvenient or dangerous diameter relative to the pressure. The long descending elements at the back to each rear header are the forces which produce the circulation and keep the active generating tubes full of water. If the lower tubes in any bank, from scale or oil or inadequacy of water circulation, become hotter and longer than

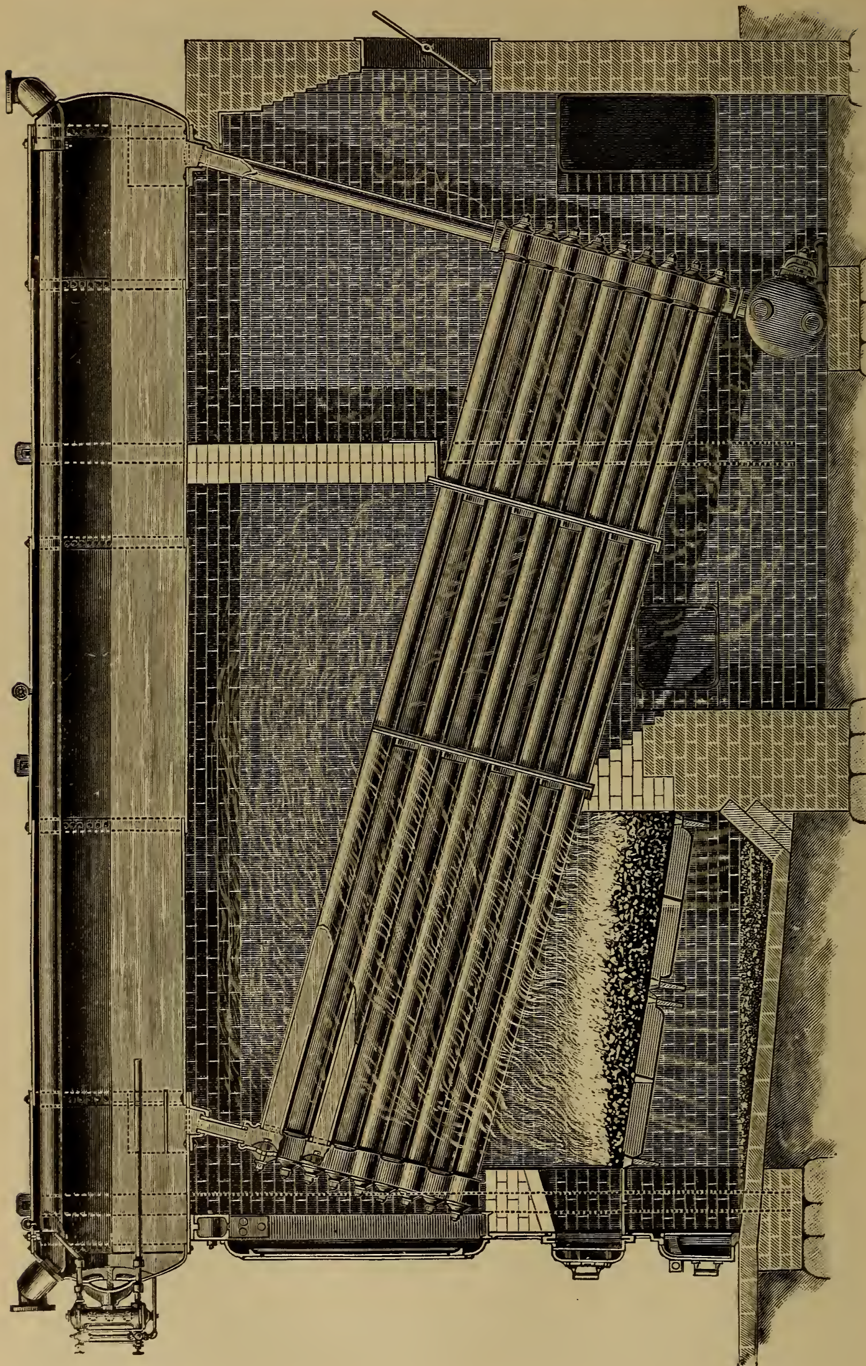


Fig. 144.

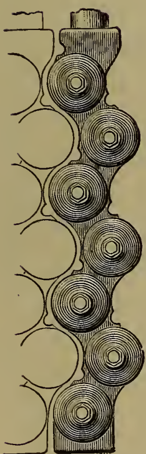


FIG. 145.

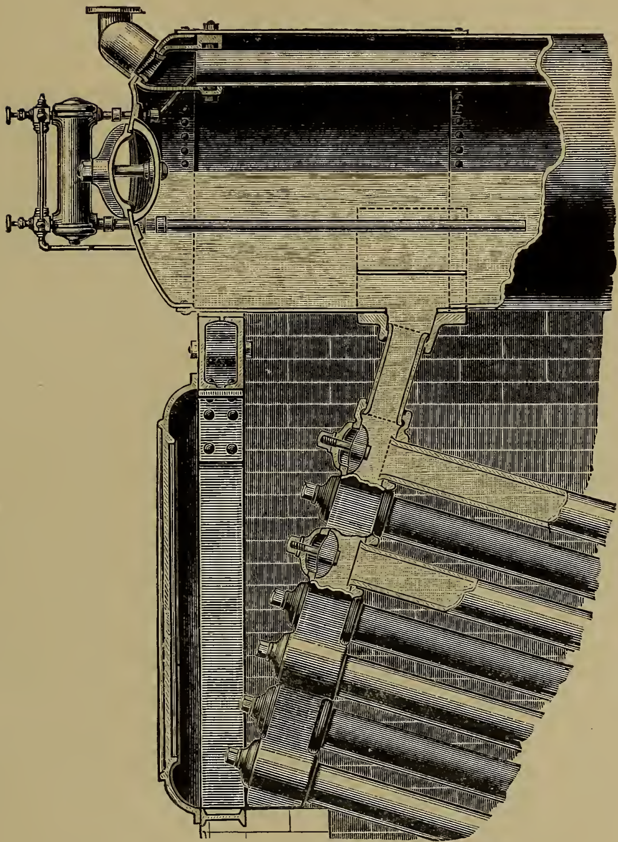


FIG. 146.

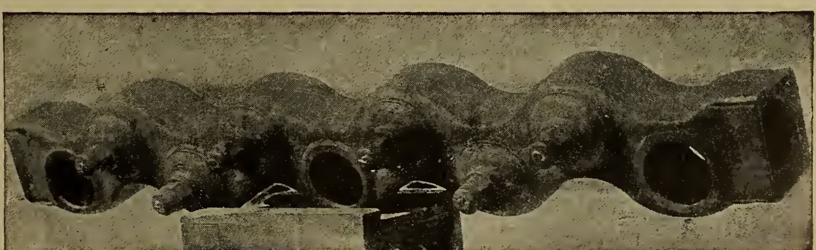


FIG. 147.

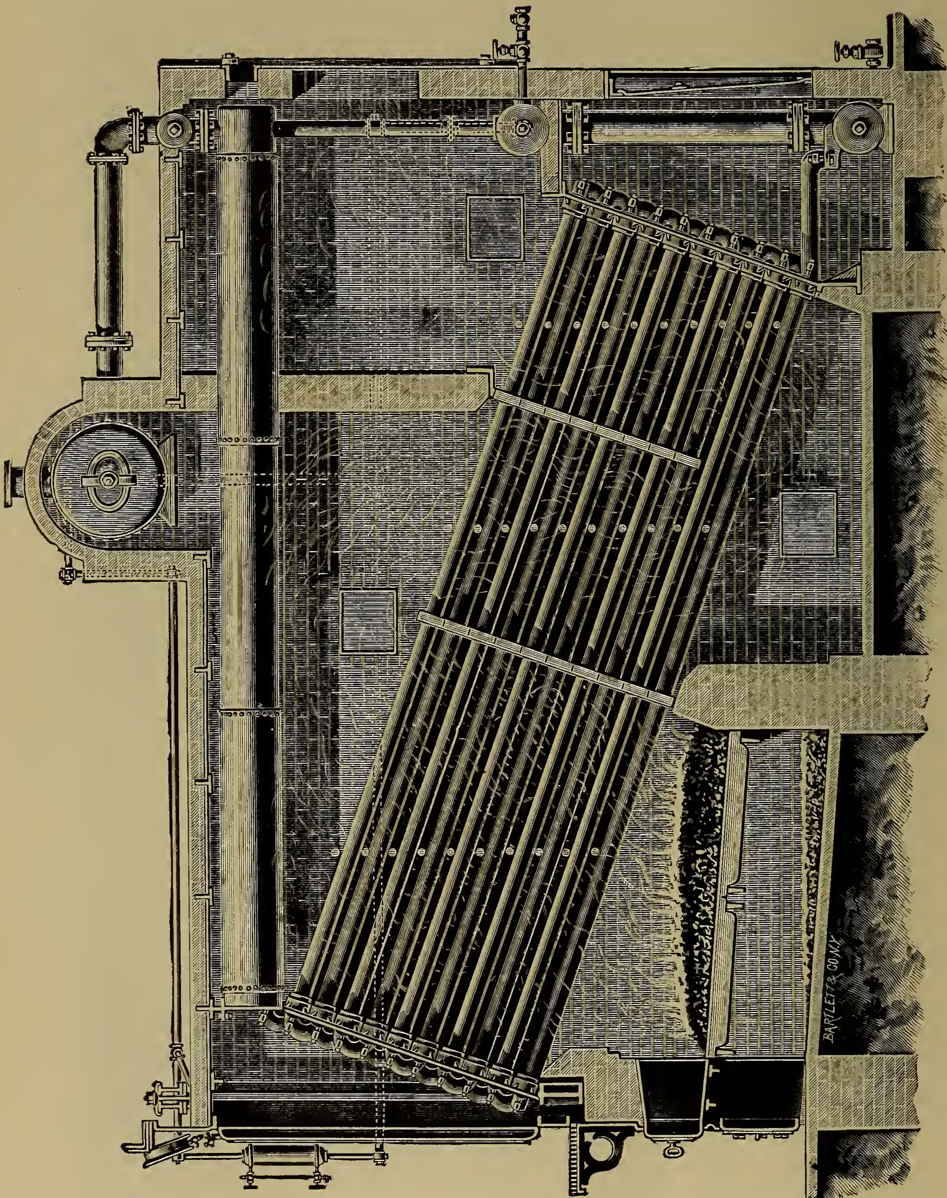


Fig. 148.

those above them, a serious cross-stress is brought upon the headers, unless the long tubes can flex. These headers were formerly made of cast iron, but such unequal expansion was likely to crack any brittle material. The gases are made to follow a serpentine course through

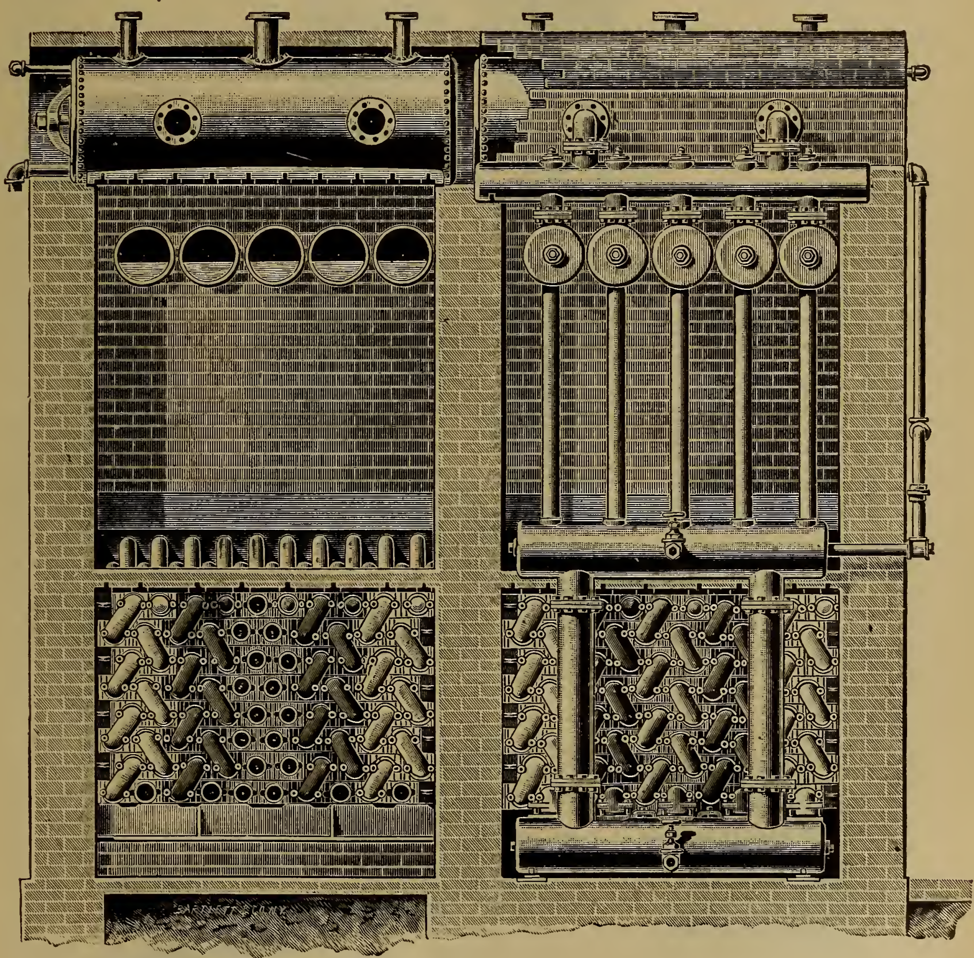


Fig. 149.

and across the generator tubes, but if the fire is at a high temperature, the gases may not be adequately cooled before they leave contact with the heating-surfaces. With long-flame fuels it may be desirable to build a Dutch-oven furnace in front of the boiler proper, so that the flame shall not be prematurely extinguished by coming too soon in contact with the front ends of the cooler tubes.

In Figs. 148 and 149 the sectional principle is carried still further, in

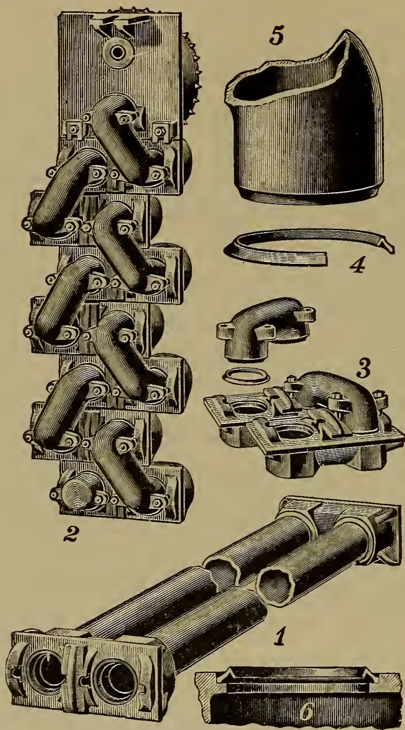


Fig. 150.

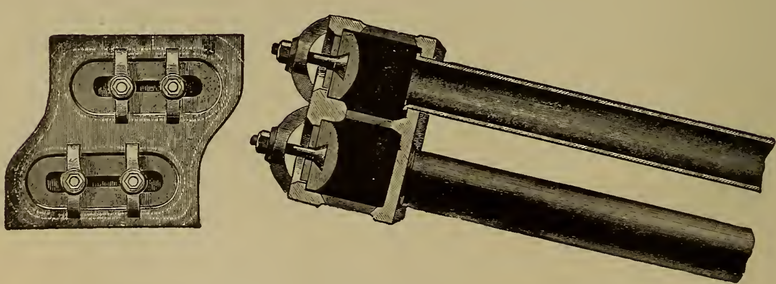


Fig. 151.

that each front header has its own steam and water-drum connected also at the back to a common water-drum, and this again to a mud-drum.

The separate steam and water-drums deliver to a common steam-drum. Furthermore, to reduce the expansion strain the headers are not rigid units, but the end of each tube is connected by a short branch with spherical joints at each face to the tube next above it, forcing a coil-like circulation until the steam escapes from the upper one in the series. Fig. 150 shows details of this construction.

In Fig. 151 are details of another construction where four tubes only are connected and not in the same plane, and with two caps only for each four tubes, and these on the inside. Fig. 146 shows the cap on the outside. If the holding-bolt breaks from overscrewing, the cap blows outward with great energy and releases hot water in a four-inch stream. Fig. 151 shows an inner plate which will be forced outward against the opening if the holding-bolt breaks, and tend to hold back the rapid flow of hot water.

In all these types it has been plain that the fire-room must be large enough in front of the boiler or behind it to permit the removal and replacement of a tube after service. If the tube is sixteen feet long, at least this clearance must be provided within walls. In marine practice such long lengths become inconvenient or impossible, and yet with forced draft the necessity of great heating-surface is imposed at the same time. This has given rise to forms of boiler using short tubes and compactly arranged (Fig. 152). Such boilers are usually set in a steel-plate envelope lined with refractory and non-conducting lining in blocks. In the form illustrated, the firing is done from the end, but it can also have the fire-door on either side, as the construction is symmetrical. Here again, however, the boilers in battery must have adequate space to remove any tube and replace it by another.

It will be obvious that all boilers of the straight-tube type open at both ends permit and favor complete inspection from either or both ends by placing the eye at one end and a suitable lamp at the other. In the water-leg or header type, this means a cap and its joint at each end of the tube, and consequent multiplicity of joints and possible attendant leakage. Such joints must be ground metal-to-metal joints, as no gasket material is appropriate for such a place. If the caps are objected to (and with reason), renewal must be by some other expedient than from the end; and the cylindrical drum of a size sufficient to be entered must replace the flat header or leg. Cleansing by impact effected by machine inside the tube is simple and sure in the straight design with capped-end openings (Figs. 263 and 264). Fig. 154 shows the Thorneycroft straight-tube design, depending mainly on the absorption of the radiant heat from the fire.

90. Vertical Curved Open-tube Type. Conceding that the cylindrical drum construction for the two open ends of the generating tubes is to be preferred to the header or manifold or water-leg construction, there is a limitation to be avoided. Either the bank or nest of tubes must include few tubes, or the entry of each tube into the drums must be specially molded in the steel to allow the tubes to enter normal to the

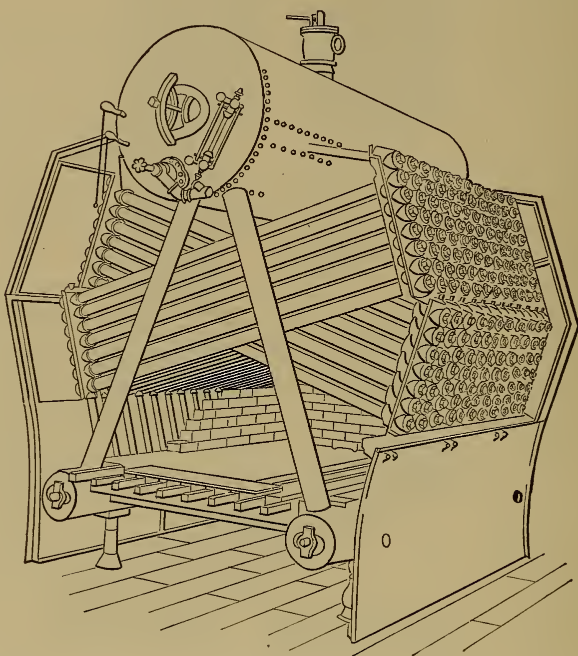


FIG. 152.

surface of the drum so that a reliable joint can be made by expanding. Or, as an alternative design, the tubes must be curved. There is only a limited surface in the drums presented normal to a common straight tube joining them together. If the tube is curved, moreover, expansion is provided for in an easy flexure of such tubes in their length without straining their joints. Repair or replacement of the inner tubes in the nests must still be looked out for, since this cannot usually be done from the ends. Fig. 153 shows a type of vertical curved-tube construction, the gases being compelled to serpentine course by the hanging fire-brick brattices or bridge-walls. The tube-sheets are made of extra thickness on the drums, and both the lateral steam and water-drums deliver into the central one. The lower acts as a mud-drum; the coldest descending current is at the back. In the Thorneycroft curved-tube type (Fig. 155) this principle is carried to its practicable limit, and by the excessive curvature all difficulties are removed which would be the consequence of different diameters of the lower and upper drums. But this amount of curvature can only be given to tubes of comparatively small diameter, leading to the coil or pipe-

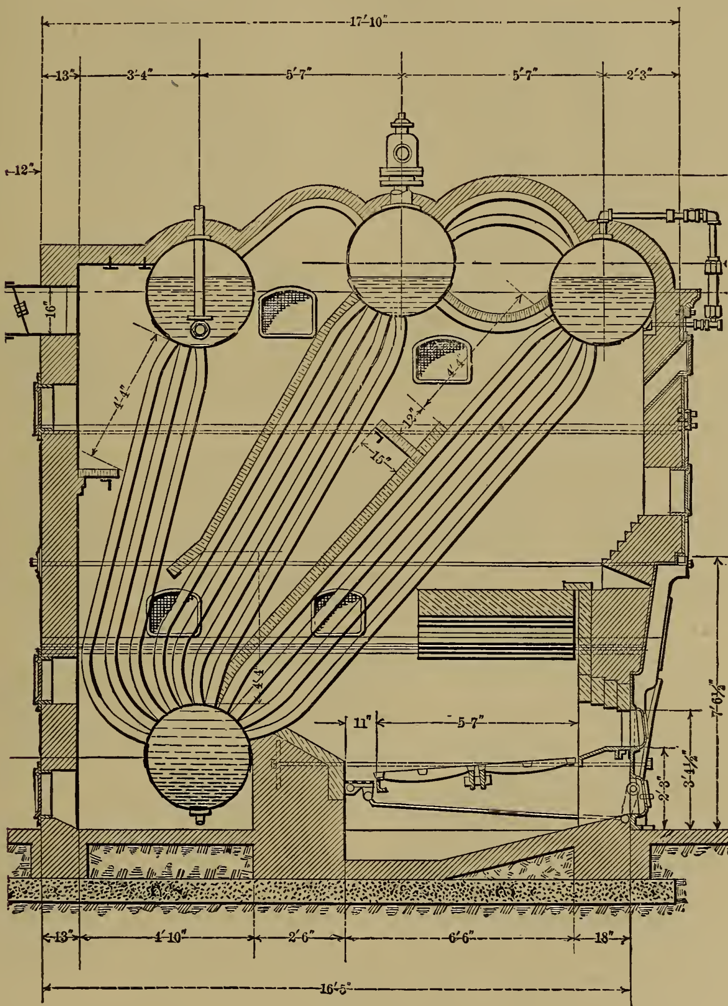


Fig. 153.

boiler. Fig. 156 shows the influence of the Thorneycroft practice in yachts and high-speed naval vessels with the curved vertical tube.

The fundamental objection to the curved tube of more than very easy curvature is the impossibility of inspecting it from end to end to see what of corrosion or other injury it has undergone in use. Machines for knocking off hard scale by impact with the tube are less easily managed. In nearly all vertical tubes, each tube disengages separately; and the large water volume at the foot of the tubes keeps them full of water.

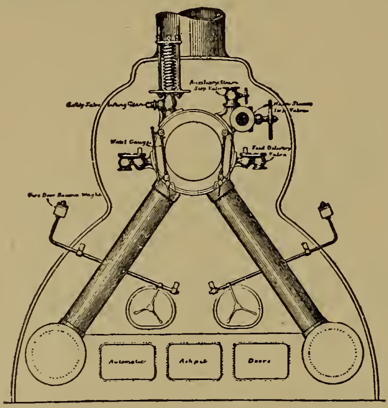


FIG. 154.

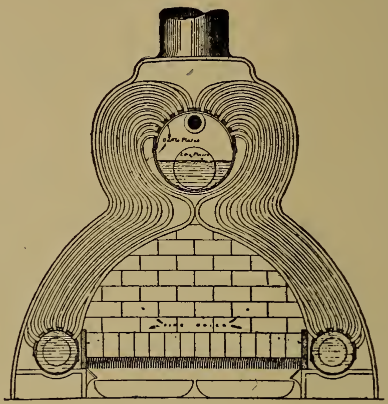


FIG. 155.

91. Horizontal Curved Open-tube Type. American practice has not developed the horizontal curved type except for yachts and similar conditions. The vertical-tube arrangement, more likely to be self-cleansing in good waters, and to a degree even with bad ones, has

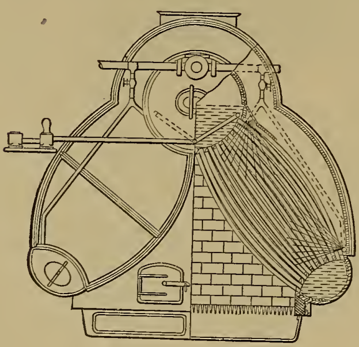


FIG. 156.

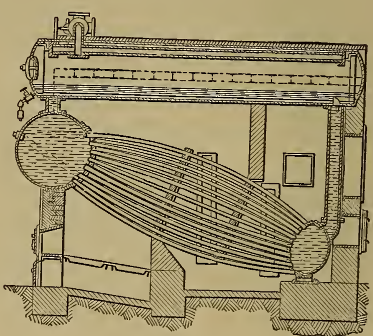


FIG. 157.

been preferred. In the design of Fig. 157, the tubes are so curved as to permit their removal endwise through a side opening of the cylindrical drum, which is covered by a proper lid which is long and narrow.

92. Closed Straight Vertical-tube Types. Field Tubes. Another direction of trial and design has been to make the generating units of tubular or cylindrical elements open at one end only and with no joint at the bottom. Such tube will therefore be closed at the bottom, and will open at its top into the water-drum in which lies the disengagement surface. The tube requires to be of sufficient diameter that the ascending current of steam-bubbles shall not interfere with the descending current of water, and this double action seems best secured when the tube-unit is inclined about 15° from the vertical. Then the bubbles formed in the tube ascend continually along the upper elements of the tube, and the lower elements (which are those turned to

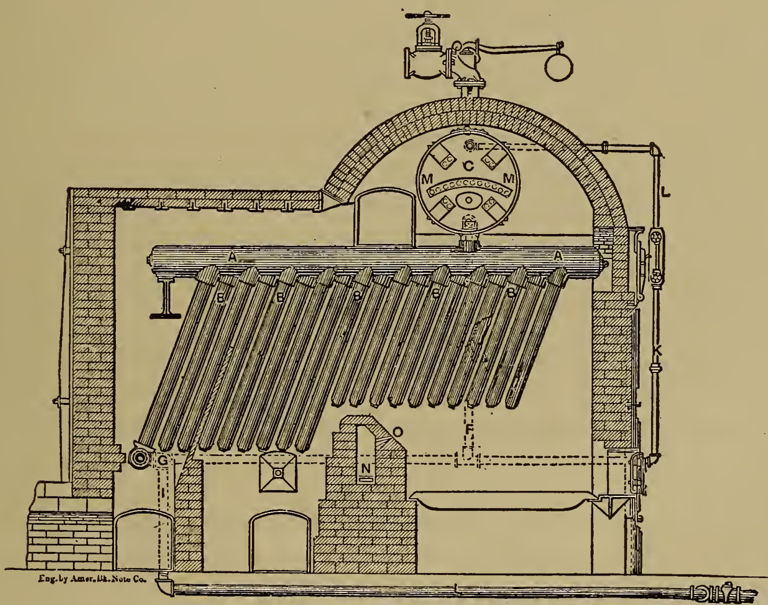


FIG. 158.

the fire and against which the hot gases impinge) are always bathed by the descending water. This was a feature of the Allen boiler (Fig. 158), and although it suffered from the difficulty of repair to middle tubes, it has been a favorite idea among German designers. If, however, the tube is of small diameter, and ebullition is too violent or the tube too nearly vertical, the steam blows the water out of the tube, and it overheats and burns.

To prevent this trouble and insure circulation in water-tubes which have to be of small diameter and essentially vertical, the double tube has been used, sometimes called the Field tube (Figs. 6 and 132).

Within the exterior and always hotter water-tube proper is placed a smaller inner tube concentric with it, kept in place by fins or lugs. This inner tube does not extend to the bottom of the outer tube, and is open at its bottom. The annular space between the inner and outer tube is to be the channel for ascending currents of steam, gas, and water. The inner tube will always be full of solid descending water. The circulation when the fire is at work should be so rapid and vigorous as to wash out and carry up any sediment or scale from the bottom of the outer tube where its presence would result in a burning of the tube. The circulation is active while the boiler is steaming, but it is not so when no steam is being withdrawn, and the circulation is that due to convection only. Under these conditions such tubes are apt to fill and solidify when least desired. It has been a favorite in fire-engine practice (paragraf 75), and has also been used in tug-boat boilers.

93. Closed Straight Horizontal-tube Types. Belonging also to the closed-tube class is a type with horizontal units projecting radially from a central vertical shell. The difficulties here are the cleansing of the inside of the units, and the indeterminate character of the circulation. It has been called the "porcupine boiler."

94. Bent or Curved-tube Types. To avoid the indeterminate circulation of the closed radial tube, it has been made an open tube by bending it back upon itself in an easy sweep, to enter the vertical water-drum at a different level. The difference of level of the two ends is to maintain a determinate circulation while steaming, the bubbles rising and escaping at the upper end while water enters the lower end to supply their place. The tubes cannot be readily cleansed nor inspected by eye except for a short distance. The limitations of size for such bent or curved tubes put them rather into a class of appendages to other types of boilers (paragraf 75 and Fig. 131), and the difficulties of repair and maintenance will always keep the design of comparative unimportance. The same results are better secured by the pipe or coil-boilers of the next chapter and with no greater disadvantages.

CHAPTER VII.

COIL AND PIPE-BOILERS. FLASH AND SEMI-FLASH BOILERS.

95. Introductory. When the demand is for steaming capacity at a rapid rate in very small bulk or weight, as in motor vehicles, launches, yachts, and small naval vessels, the small diameter of the resulting water-tube makes the previous constructions less serviceable. Rapid continuous circulation becomes imperative to keep the small tubes clean on the inside, since no inspection or access is possible. The tube should be of small caliber and the boiler hold little water at one time, so as to minimize the disaster if such an uninspected tube fails. Such small water-content is also a factor in the rapid steaming. To get length of tube or heating-surface in small bulk, the water-tube must be bent back over itself so as to form a coil; or a multitude of such partial or complete coils united into one generating whole. When the bent tube or pipe is formed into more or less helical or spiral coils of some considerable length, the boiler will be called a coil-boiler: when multiple partial or incomplete coils are connected by pipe-fittings, it will be called a tube-and-fittings boiler. The pipe or tube-and-fittings boiler will be discussed first.

96. Tube-and-Fittings Boiler. Fig. 160 will illustrate a simple type of tube-and-fittings boiler. The water enters at the bottom and is there divided into four coils of tube and return bends. These discharge steam and water at A into the annular steam and water-space surrounding the fire and the coil. The heat is radiated from the grate and fire-box to make steam also of the water in W, and the steam from the coils separates from the water as it is withdrawn from the side of the shell opposite to the delivery of the coils at A. This boiler is used for fire-engine purposes. The pipe and fittings must expand freely within the shell; circulation must be rapid enough to keep the coils clean and clear, the solid matter from the water going down into the annular separating and settling chamber.

In Figs 161 and 162 are two types of tube-and-fittings yacht or torpedo-boat boiler, designed by Almy. Here the separating chamber is separate from the generating elements and outside the casing. The fire is surrounded by the vertical risers, and each pair of vertical units unites into a common riser at the top. The boiler is inclosed in steel with a refractory and non-conducting lining. Certain fittings are special, but the manifolds, elbows, and return bends are standard except for the way in which the ends of the threads are protected from the heat. Each coil-unit, however, is shorter than in the preceding form or in the coil-boilers to follow.

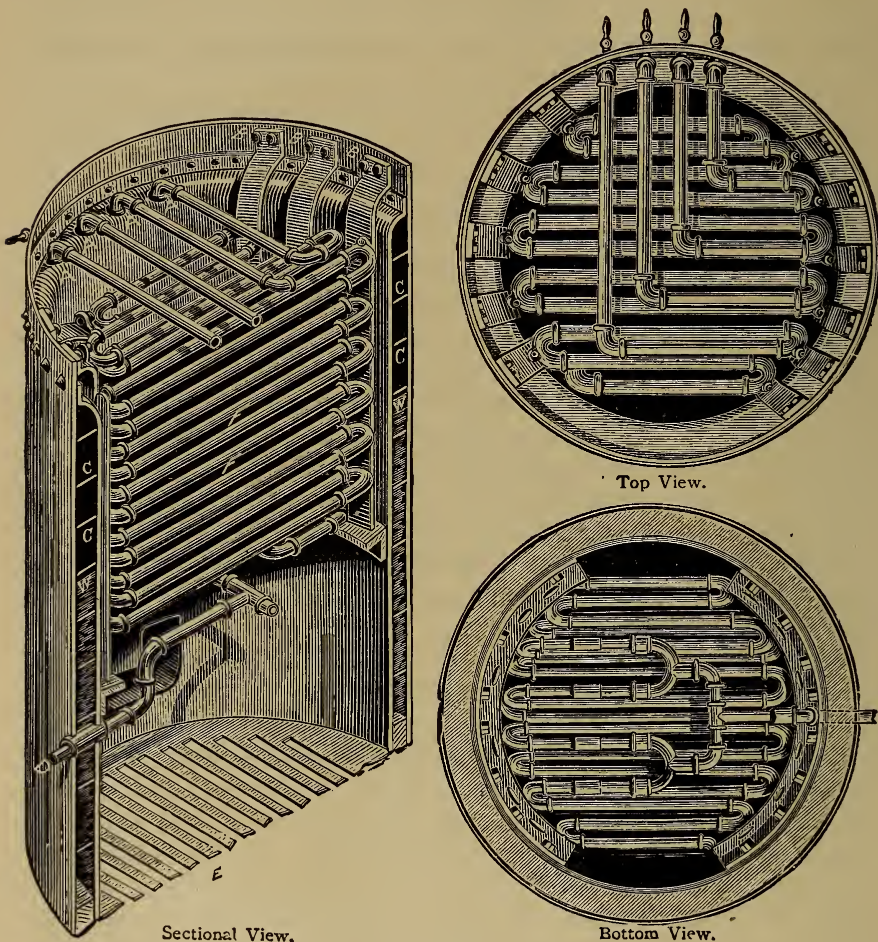


FIG. 160.

The curvature of the generating units with elbow and return bends takes up expansions; there is very little water in the boiler at one time; the subdivision is very thorough.

97. The Coil-Boiler. The coil-boiler differs from those above mentioned in that one or a limited number of continuous coils is used instead of a large number of short or separate circuits. The steam formed near the bottom end of such a coil must run through its entire length before escaping at the disengaging surface. This makes it desirable that the water in such a coil-boiler should be circulated mechanically both by reason of the advantage so far as efficiency of transfer is concerned, and to preserve the coil from burning. Such coil-boilers have given very

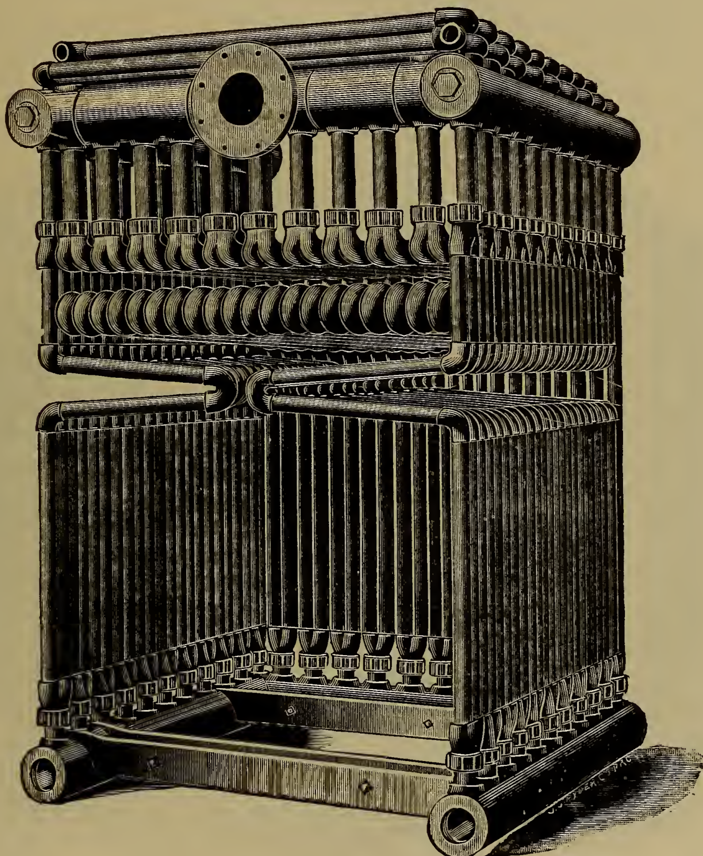


Fig. 161.

large results for their size in experimental forms. The best-known types are identified with the names of Ward, Herreshoff, and Trowbridge. Fig. 163 shows the Ward boiler, which has been approved in U. S. Navy work of its class, and Fig. 164 is the Herreshoff type.

98. Sundry Types. Conclusions. There will be types of pipe and coil-boilers which will not go naturally into the general classes above named. Such would be combinations of accepted types made for special uses, and possibly new types. The student and reader must follow his own lead in the consideration of such cases.

It seems to be held in the case of the water-tube marine boilers that they present the following features:

(1) Light weight of both metal and water; about one-half that of a Scotch boiler of equal steaming capacity.

(2) Adapted for high pressures.

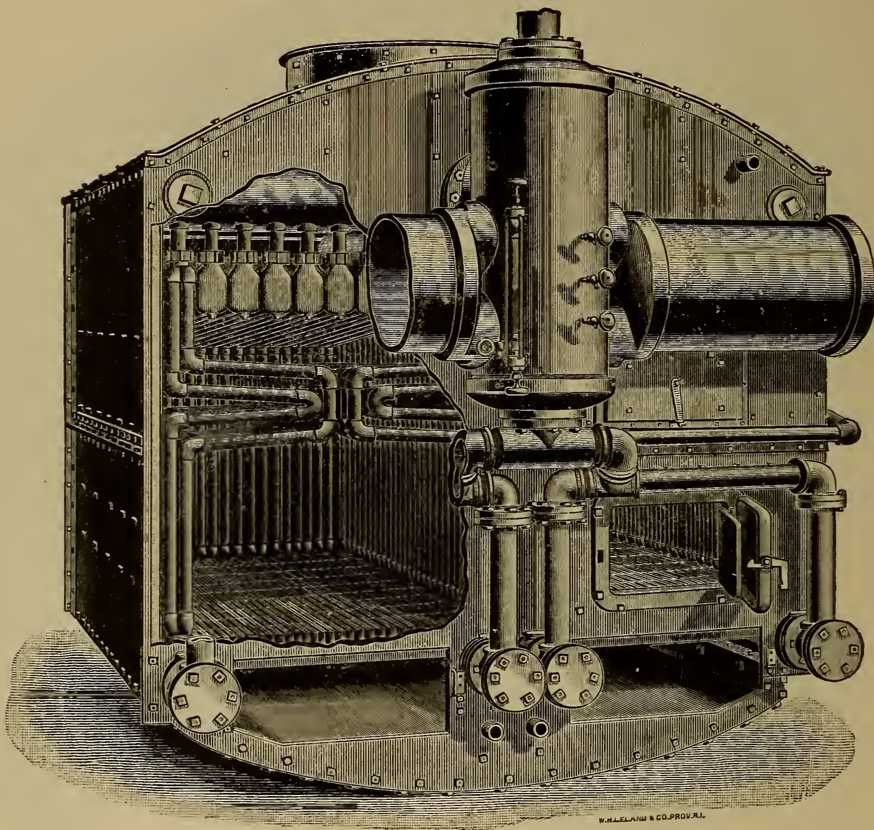


FIG. 162.

(3) Make steam rapidly, and have the pressure soon after starting fires.

(4) Are safe against a disastrous explosion, because they hold so little water.

(5) Are not injured by the intense combustion and local heat caused when forced draft is used.

(6) The parts are not difficult to renew.

As disadvantages they offer:

(7) They require more care in feeding. The water does not always remain in the lowest part of the coil and give a normal level at the water-gauges. When used in batteries the water may not remain in one particular boiler in mass, but will fly about in it or even into other boilers.

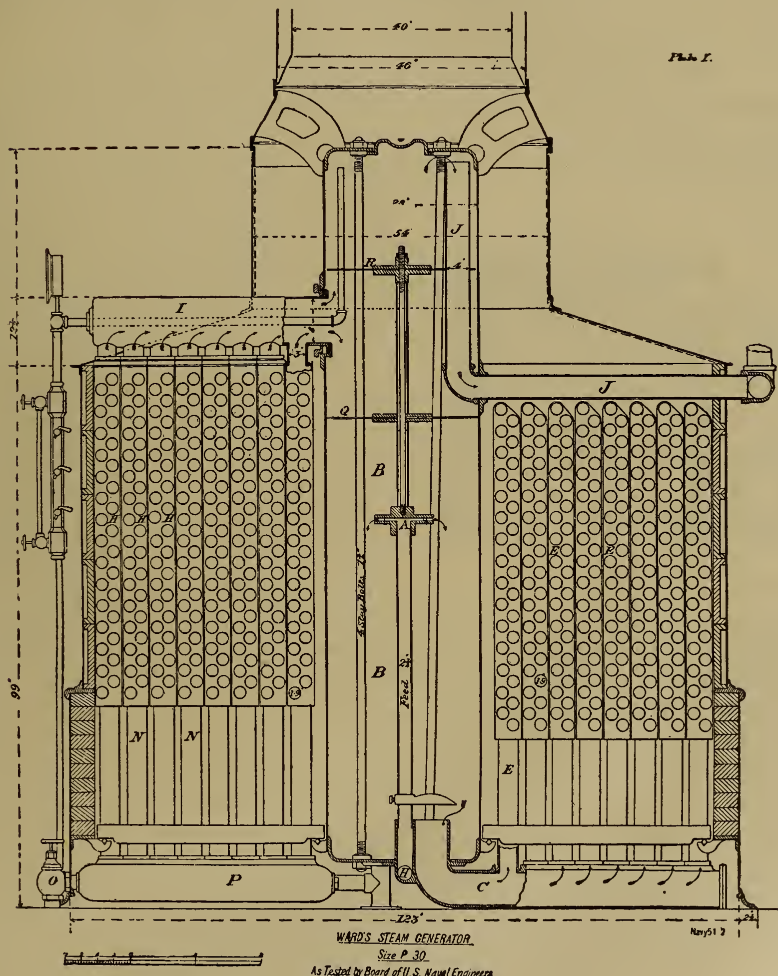


Fig. 163.

(8) Corrosion is troublesome within the coils, and access and inspection impossible. Oil also gives trouble if allowed within the coil with the water.

(9) The coil is prone to fail where the pipes or tubes are threaded into

fittings. Overheating from absence of water on the heating-surface makes the screw-threaded ends give out by oxidation and by stretching.

(10) The casing around them gets very hot and makes the fire-room or stoke-hold hot even to the point of being unbearable.

(11) A certain greater amount of air is drawn into the fire around the casing so as to dilute the hot gases and lower their temperature. This does not occur in a true internally-fired boiler.

99. Spray Boilers. It occurred to a Mr. Dunbar in the middle of the nineteenth century to make steam rapidly by throwing a thin film or spray of water upon an intensely heated plate in an inclosed vessel. Such water became steam so quickly that it was said to flash into steam. The Dunbar boiler failed, however, by the deterioration of the plate from the high heat and sudden changes of form by the impact of the cold water. It became coated also with the deposits from the water. Recently, however, a type of steam generator of the coil class has come into extensive use for small units where great steaming capacity is required with small weight or bulk, as in motor-cars and launches. The object sought is maximum necessary heating-surface with minimum weight of water in the boiler at any time, the latter both for lightness and for safety in case of rupture or failure of any part. The theory is the same as underlay the Dunbar spray boiler, which was a generator of the shell type, in which at each stroke of the engine there was to be injected into the boiler from a feed-pump the same weight of water as had just left it in the form of steam for the cylinder. The relatively great heating area exposed to fire is so hot that the transfer of heat is nearly instantaneous in some forms, and the water seems to "flash" into steam-gas. If the boiler contains no water as water for more than a fraction of a second, but only wet steam and dry steam, it is a true flash boiler; if it contains some water in the process of change, as well as wet and dry steam, it is a "semi-flash" boiler.

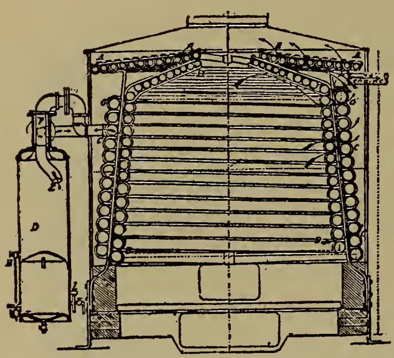


FIG. 164.

100. Flash and Semi-Flash Boilers. In high-speed engines to be supplied from a boiler of this type, the interval required for the heating of the water to the boiling-point for the pressure, the heating for the change of high-pressure water into high-pressure steam, and the further heating for the superheat so easy with this system, have made a necessary gap between the work of a given stroke of the engine and the making of steam to supply the volume emptied by such stroke. This fact has caused a drift away from the spray or true flash type, and towards the semi-flash. In the true flash type the pressure variation is excessive and inconvenient when any variation in the load on the engine occurs. There is no reserve of heat energy in the form of heated water to bridge over the interval of time required for a response to an increased demand for steam volume and weight when the latter must be met by an increase in the weight of water fed by the pump and an increase in the weight of fuel supplied per second to the flame of fire. Such gap in the action of the flash boiler is indicated by a sudden drop in the steam-pressure as revealed by the gauge on an increase, or an inconvenient rise of pressure on a decrease of the demand. In the semi-flash types, where some water is always present in the coil, it serves as an accumulator, storing heat by reason of its high specific heat for a few seconds as pressure is rising, and giving this out gradually on a fall of pressure as generated steam, and giving time for the controlling devices to diminish the injection weight and control the fire temperature on a lessening of load, or reverse these processes on an increase of demand for steam as the load increases, or the speed of revolution when the latter is variable. Plainly, however, the water can have only a limited capacity to act as such an accumulator, and when it is exhausted the pressure variation will be as in the flash class.

The leading exponents of these types are the Serpollet (or Gardner-Serpollet) on the continent of Europe, and the White generator in America. The Serpollet generator has taken several forms in its development. The earliest was a group of flat spiral coils, made up of flattened steel tubing; later, of a series of tubes first pressed so as to bring two parallel sides close together (of an inch or $\frac{1}{8}$ less) and then pressed again, so that the section of the opening between the sides was a crescent or circular arc for part of the length of each unit. The whole tube was then formed into a U, and successive elements coupled together by return bends. Later again tubes of cylindrical section were used, or flattened tubes twisted; and last of all round tubes formed into a coil of two layers, the lower with branches closely parallel, and the upper with fewer return bends and at right angles to the axes of the lower. The feed-water enters at the top of the series of coils and passes downward from the second to the very lowest one, and thence up through several (perhaps five), whence it rises outside again to the third and descends thence to the outlet. Other foreign forms of this type have been the Blaxton generator of England, using plain coils and taking the water in at the bottom; the Simpson-Bodman, using dented or Rowe tubes,

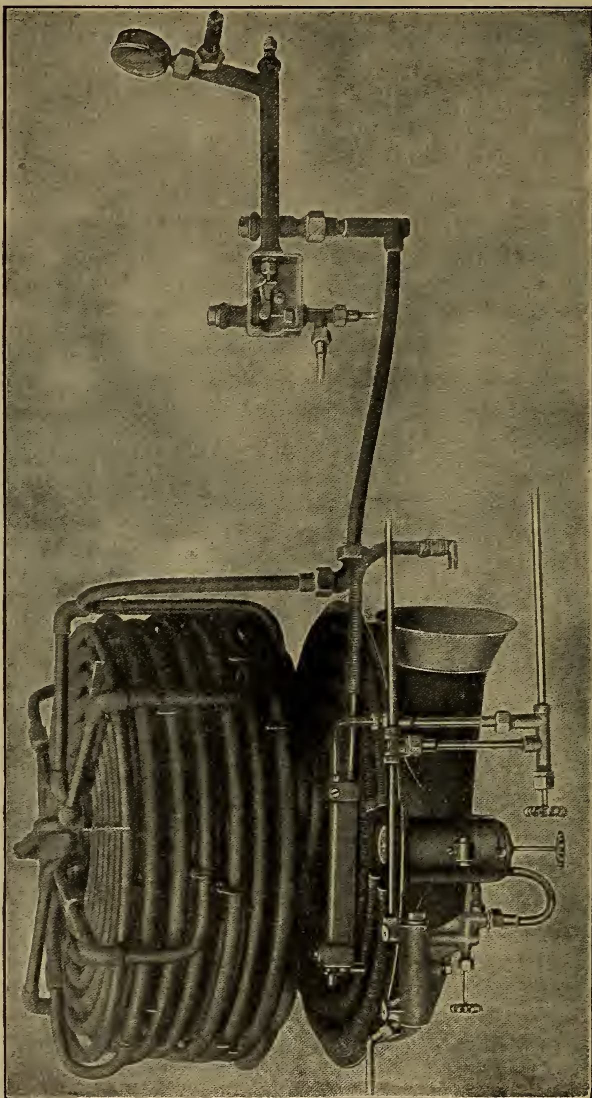


Fig. 165.

and types having the unit tubes formed into vertical helices. In the White generator the coil is of three-quarter inch steel tube, bent into the form of a series of flat spirals. For a 30 H.P. unit there are usually seven of these spirals, aggregating over 320 feet. Feed-water is pumped into the coolest coil, which in an upright boiler will be at the top, and after passing round the first spiral crosses over the top and descends to the next coil below. This method of putting together makes each coil a unit, and the inverted U tube which connects it to the next one acts as a trap or seal in each case, compelling the circulation to be definite and positive and the separation of steam from water to be always at the front or lower end of the moving column. At a point in the coil which is variable with the amount of water pumped in as feed, the liquid becomes a gas and moves forward over the more intensely heated coils close over the fire, toward the outlet at the bottom, becoming superheated in its passage, and entering the engine cylinder in this state. The last or final superheating coil being just above the fire at the bottom, while the coolest coil is at the top, where the effect of the fire is least, secures a graded intensity of the heat transfer, which releases the metal of the coils from the great stresses otherwise caused by frequent alterations of temperature, and secures a long life for the coil material, which would otherwise be impossible. The intensely rapid circulation makes deposit of scale in the coils unlikely, as such separated solids are swept forward with the steam and so outward from the generator. The superheater coil being in the fire itself, it becomes easy to use the temperature of the superheated steam as a means to actuate the metallic element of a thermostatic pyrometer, the motion of which can be utilized to open up the water supply when the pressure and temperature fall, and to shut off the water supply when the temperature and corresponding pressure rise too high. Too little water in the generator causes excessive superheat, and the fire is at once shut off, so that disaster from low water cannot occur. An excess of feed-water, on the other hand, will diminish the length of coil acting as superheating surface by increasing the length which acts to change water into steam. In a generator properly designed for its burner or grate area, the effect of this is to increase pressure at the expense of temperature, and this pressure can be used on a spring-controlled diaphragm to actuate a by-pass on the feed-pipe, sending the pump discharge around into the suction and not into the generator, and enabling the latter to void its excess. (See paragraf 170.) The advantages attaching to this system are:

1. The minimum weight and bulk of generator and contents with the maximum evaporating capacity possible for a given burner or fire.

2. Practical automatism of fuel supply and water-feed with simple and reliable apparatus.

3. Prompt response to variation in the demand for steam, without inconvenient change in the working pressure on the piston.

4. Safety from serious disaster: the stored energy at any one time is not large.

5. Scale trouble from bad water is minimized.

The disadvantages are those which it must share with all curved-tube or coil forms as respects inspection, as discussed in the previous paragraf. Fig. 165 shows the general arrangement of such a generator for motor-car service.

CHAPTER VIII.

BOILER FURNACES, CHIMNEYS, AND SETTING.

105. The Fire as the Source of Energy. It was shown in the analysis of paragraf 4 and Figs. 1 and 2 that the steam engine was a heat engine. In paragraf 20 the reasons were given for studying the boiler before the grate which supplies heat to the boiler.

The previous chapters have discussed the heating-surface or absorbing area for heat from the fire under or within the boiler, covering the forms of such generators and reservoirs of pressure and the accepted designs which work well. In this chapter attention will be directed to the liberation of that heat energy upon a grate by combustion which the fuel is ready to give up.

Every pound of fuel of constant chemical constitution has a definite capacity for heating water. Such capacity is called its calorific power. It is either computed from the heating power of its constituents, or experimentally determined in a calorimeter.* The latter method is more satisfactory. Table IX gives some data upon fuels which may be found of convenient reference.* The unit is the pounds of water raised one degree Fahrenheit by the complete combustion of one pound of the fuel. The weight of oxygen required for such combustion is found by the chemical formula of atomic combination of carbon with oxygen and hydrogen to form carbonic acid and water vapor.* (See paragraf 24.)

To effect this combustion and liberation of heat at the desired rate per hour to supply heat to the heating-surface of the boiler and to the water which it incloses, the coal must be burned at a rate to deliver that quantity. Plainly the leaner the coal, the more must be burned; also, the larger the grate the more coal it will receive at one time, and the more heat it will deliver with a given combustion rate (paragraf 12). The rate of combustion is determined by the draft, or by the supply of oxygen or air which is drawn through or over the fire to support such combustion. The draft or rate of combustion is caused by the chimney or by what may replace it as a force for bringing air to the fuel on the

* For more complete treatment of fuel calorimeters, calorific power of gases and liquid hydrocarbons and the computations for combustion, consult "The Gas Engine," by F. R. Hutton. John Wiley & Sons, New York.

TABLE IX.

SHOWING THE COMPOSITION AND CALORIFIC POWER OF VARIOUS COMBUSTIBLES. THE QUANTITY OF OXYGEN AND AIR NECESSARY FOR COMBUSTION, AND THE VOLUME OF THE PRODUCTS OF COMBUSTION OF 1 POUND OF COMBUSTIBLE.

(From Morin and Tresca.)

Name of Combustible.	Composition.				Calorific Power.	Weight of Oxygen Necessary for Combustion.	Weight of Air Necessary for Combustion.	Volume of Air Corresponding in Cubic Feet.	Volume of Products in Cubic Feet.
	C	H	Volatile Matter.	Ashes.					
Carbon	1.00	....	....	....	14,400	2.66	11.29	137.6	137.6
Anthracite coal	0.90	0.03	0.03	0.04	13,500	2.64	11.21	138.9	136.2
Bituminous coal	0.85	0.05	0.06	0.06	14,400	2.66	11.29	139.6	140.1
Lignite	0.70	0.05	0.20	0.05	11,700	2.26	9.69	120.2	116.3
Peat	0.55	0.05	0.30	0.10	9,000	1.86	7.90	97.9	102.1
Peat 0.20 water	0.39	0.04	0.50	0.07	7,200	1.49	6.32	78.3	81.5
Coke	0.85	0.05	....	0.10	12,600	2.26	9.69	120.2	116.9
Peat-charcoal	0.82	....	....	0.18	9,000	2.18	9.25	114.5	112.7
Dry wood	0.48	0.06	0.05	0.01	7,200	1.75	7.43	91.9	89.2
Wood 0.20 water	0.40	0.05	0.25	0.01	5,400	1.40	5.94	73.5	71.8
Wood-charcoal	0.80	....	0.04	0.07	10,800	1.86	7.90	97.9	96.3
Hydrogen	....	1.00	....	....	62,000	8.00	33.97	420.6	475.4
Carbonic oxide	0.43	....	0.57	....	4,320	0.57	2.42	29.9	35.6
Illuminating-gas	0.62	0.21	0.17	....	18,000	2.64	11.22	136.3	176.7
Gas from blast-furnace	0.06	0.02	0.92	....	1,620	0.23	0.99	12.2	30.3

grate. (See Analysis, Fig. 2, Nos. 2, 5, and 49.) The grate or furnace and the chimney are therefore one unit, designed to effect combustion and the liberation of heat energy from the fuel. What physical and mechanical principles underlie the grate and chimney?

106. Principles underlying the Boiler Grate. The grate must meet the following requirements:

(1) Area sufficient to take at one time the weight or quantity of fuel which at the rate of combustion then prevailing will liberate heat energy in heat units at the same rate that the engine is using mechanical energy in foot-pounds. The steam-gauge is the indicator of this process. When the fire energy is in excess, the pressure rises; when the engine demand is in excess, the pressure will fall as the energy on storage falls.

(2) Internal resistance in the material of the grate to the stresses imposed by the weight of such fuel tending to deflect the grate. Such material must not sag or break even when hot.

(3) Adequate free openings for air to pass into the fire so that the chemical reactions with oxygen can occur. As oxygen in air is a gas, it can unite only with carbon and hydrogen also as gases and heated to a sufficient temperature. When fresh cold fuel is thrown on a fire, the first stage is a distillation by heat from the surface of the lumps. If this gas meets oxygen and both are hot enough to ignite and combine chemically, a flame results and no carbon is wasted. If oxygen is lacking, the carbon gas goes off uncombined or partially consumed only, and there are unnecessary waste and smoke. If the temperature is too low for chemical union, the result is the same as if oxygen had been deficient in quantity.

(4) Resistance to deformation or warping from heat of the fire, and of ashes falling hot and burning into an ash-pit below the grate.

(5) Resistance to mechanical injury from the tools used by the firemen in cleansing the fire from clinker and ash.

(6) It must not be too costly to buy or to replace.

(7) It must not be heavy to the degree of being unmanageable in contracted places.

107. Principles underlying the Chimney. The oxygen or air to oxidize the fuel on the grate will not get there of itself. Some mechanical energy or force must be expended to bring it there in sufficient quantities at sufficient speed and under sufficient pressure to overcome the resistances (Fig. 2, Nos. 5, 9, and 49). This force may come from a fan or blower driven by expenditure of mechanical power; or the energy in heat may be used directly in the construction of the machine called a chimney. The chimney is a device to render available the weight of a column of colder air when a definite column of light heated air tries to balance it.

The most widely accepted theory of the action of the chimney was first elaborated by Péclet, and developed and quoted by Rankine and other writers. His discussion can be made most easily to be understood by the conception of the chimney as an inverted siphon, with the fire-grate at the bend at the bottom. In Fig. 166 the hatched section represents the chimney, and the dotted lines the siphon leg of cold external air. A diaphragm A–B in the bend of the siphon will have unequal pressures on its two sides if the legs are of equal length and equal cross section, because if D_a denote the density of the external air and its weight per cubic foot, and D_c denote the density of the warmer lighter chimney-air, then HD_a acts on one side, and the less HD_c on the other. To equalize the pressures an extra effort must be exerted on the lighter leg to balance the heavier, so that an extra length of column of hot air of unknown height h, and having a density D_c, must exert a pressure $p = hD_c$ to effect this balance. Or, since

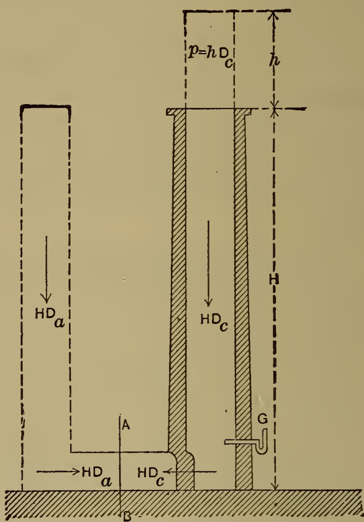


FIG. 166.

$$HD_a = HD_c + p,$$

we can write

$$p = HD_a - HD_c$$

as the pressure exerted on the diaphragm or on a film of air at the base of the siphon and which causes the flow when there is no balancing pressure at the top of the chimney. But since $p = hD_c$, the height of the column of hot gas will be

$$h = \frac{H(D_a - D_c)}{D_c},$$

and the question of the values of these two densities is a question for observation or calculation. At 32° F., D_0 for air is .0807, and by reducing D_c to 32° F., the value for h can be found in feet, or more conveniently the expression can be transformed to read in absolute temperatures instead of densities by the relation that the densities will be inversely as the temperatures, so that

$$\frac{D_c}{D_0} = \frac{T_0}{T_c}.$$

But the chimney-gas is a mixture, and not a constant or permanent gas. An accepted value for its ordinary density at 32° F. is .08424, which is derived from an averaging of many analyses and experiments which give for such gases:

Carbonic acid	10 per cent,	weighing at 32°	.12344
Nitrogen	79 "	" " "	.07860
Oxygen	11 "	" " "	.08926

Multiplying the per cent of each by its weight, we have:

Weight of CO_2	.01234
" " N	.06209
" " O	.00981
Total	.08424

If the composition of the gases differs from the above assumption as determined by analysis or otherwise, additional data are given in the following table:

	Specific Volumes.	Specific Heat in Gaseous Condition.	Density or Weight of One Cubic Foot.
Hydrogen	178.881	3.4090	0.00559
Oxygen	11.2070	0.2175	0.08928
Nitrogen	12.7561	0.2438	0.07837
Carbon dioxide	8.10324	0.2169	0.12341
Carbon monoxide	12.81	0.2450	0.07806
Water	....	0.4805	....
Air	12.3900	0.2375	0.08071
Ash	....	0.2	....

Substituting, then, for D in the formula for height the expression

$$D_c = .08424 \frac{T_0}{T_c},$$

the expression for that height becomes

$$h = H \frac{\left[.0807 \frac{T_0}{T_a} - .08424 \frac{T_0}{T_c}\right]}{.08424 \frac{T_0}{T_c}},$$

which becomes by performing the operations

$$h = H\left(.96 \frac{T_c}{T_a} - 1\right).$$

The velocity in feet per second caused by a height h in feet will be denoted by $v = \sqrt{2gh}$; the volume V per second if the cross-section be denoted by A square feet will be Av, and becomes

$$V = Av = A\sqrt{2gH\left(\frac{.96T_c - T_a}{T_a}\right)}$$

for the temperature of 32° F. If it be required to burn W pounds of coal per second, and KW cubic feet of gas at 32° F. be the result, we shall have an equation for W, since $V = KW$,

$$W = \frac{A\sqrt{2gH\left(\frac{.96T_c - T_a}{T_a}\right)}}{K}$$

as the theoretical pounds of coal which will be burned by a chimney of height H and area A when the resistances to flow of air and gas are disregarded.

108. Discussion of Péclet's Theory of Chimney-Draft. Péclet developed a later theory in which the dynamic energy for the flow of air to the furnace was a head in feet expressed in terms involving the cold gas or external air. He also developed an expression for the velocity of flow, starting from the general expression for the relation between the head in a pipe and the flow which it produces in the case of a liquid. A form for this is

$$h = \frac{v^2}{2g}\left(1 + K + K_1 + \frac{fl}{m}\right),$$

which in Péclet's form appears as

$$h = \frac{v^2}{2g}\left(1 + G + \frac{fl}{m}\right).$$

In this equation h is the head; g is the acceleration due to gravity; f is the friction against sides of pipe or duct or flue; K and K_1 or G, which combines them, are coefficients to express the resistances offered by bends, elbows, valves, and fittings in hydraulic work and by the grates, tubes, and damper in boiler-furnace work; l is the length of the pipe or gas passage; and m is the ratio of area of cross section to the perimeter, called the hydraulic mean depth. For square or round flues m will be one-fourth of the side or diameter, since $\frac{b^2}{4b} = \frac{b}{4}$ for a square flue, and $\frac{\pi r^2}{2\pi r} = \frac{r}{2} = \frac{D}{4}$ for a round one. Péclet's value for G he puts at 12 for cases where 20 to 24 pounds of coal are burned per hour, and for f his value is 0.012 for surfaces covered with soot. Hence his formula becomes

$$h = \frac{v^2}{2g}\left(13 + \frac{0.012l}{m}\right),$$

whence the expression for volume per second with a height H would appear

$$V' = Av' = A\sqrt{\frac{2gH\left(.96\,\frac{T_c - T_a}{T_a}\right)}{2gh\left(13 + \frac{0.012l}{m}\right)}}.$$

The uncertainty as to coefficients; the fact that it is not true that $v = \sqrt{2gh}$ for a flow of a gas which undergoes any notable change in pressure or temperature, and the chimney problem introduces both;

the fact that the chimney temperature T_c is not constant from top to bottom; and the necessity for the assumptions of area and temperature and velocity before a height can be worked out, have thrown designers upon the data of experience rather than upon the foregoing calculations.

The Péclet formula, however, possesses this interest. Since the velocity of the gas in the chimney increases as the square root of the height of the dynamic column, and therefore with $\sqrt{.96T_c - T_a}$ when the external-air temperature is fixed, and since the density is inversely proportional to the temperature in the chimney, the weight discharged will be proportional to

$$\frac{\sqrt{.96\ T_c - T_a}}{T_c},$$

which becomes a maximum when

$$T_c = \frac{2T_a}{.96} = \frac{25}{12}T_a = 2\tfrac{1}{12}\ T_a;$$

or the greatest weight will be discharged when the absolute temperature within the stack is $\frac{25}{12}$ of the absolute temperature of the external air. That is, if the external air be at 62° F., or an absolute temperature 522°, the temperature within the chimney for a greatest weight of gas flowing should be $522 \times \frac{25}{12}$ or 1087° absolute, or 626° F. This explains the usual preference for temperatures around 600° F. in ordinary boiler-stacks. This is about the temperature of melting lead. On the other hand, for many metallurgical purposes a higher temperature in the stack is a necessity, and a greater velocity than is usual in steam-boiler practice. When this maximum temperature prevails $h = H$; or the extra column of hot gas has a height equal to that of the original chimney, and the density of that gas is one-half of that of the external air. The formula also indicates that chimneys draw best with cold air outside and at high barometric pressures.

109. Some Accepted Chimney Formulæ and Data. Mr. William Kent in 1884 proposed a formula based on successful practice and on the idea that the effective area of a chimney was less than its gross area by a dead-space of two inches radially from each wall of a square chimney or all around a round one. This idea, if A be the gross area expressed in square feet, and E the effective area, will make:

For square chimneys

$$E = D^2 - \tfrac{8}{12}D + \tfrac{1}{9} = A - \tfrac{2}{3}\sqrt{A}.$$

For round chimneys

$$E = \frac{\pi}{4}(D^2 - \tfrac{8}{12}D) = A - 0.592\sqrt{A}.$$

This is so nearly the same for both that it can be written

$$E = A - 0.6\sqrt{A}.$$

Since the power of a chimney varies both as the square root of its height at best temperature conditions and as its effective area, it can be written that

$$\text{H.P.} = E\sqrt{h} \times C,$$

in which C is a constant to be determined from successful practice. A boiler horsepower is assumed to be equivalent to an evaporation of 30 pounds of water per hour (paragraf 10). Assuming 5 pounds of coal per horsepower per hour to take account of poor conditions, and observing the number of pounds of coal which a successful chimney will take care of, an acceptable value for C is found to be $3\frac{1}{3}$. Hence

$$\text{H.P.} = 3.33\,E\sqrt{h} = 3.33\,(A - .6\sqrt{A})\sqrt{h},$$

which can be written also

$$E = \frac{0.3\ \text{H.P.}}{\sqrt{h}},$$

when the quantities of the second member are the known data.

A series of observations by Morin and Tresca from French practice have resulted in the following table, which is a very safe guide. The grate is eight times the chimney cross-section.

Heights of chimney in feet	20	25	30	35	40	45	50	55	60	65
Pounds per hour per sq. ft. grate	7.5	8.5	9.5	10.5	11.6	12.4	13.1	13.8	14.5	15.1
Pounds per hour per sq. ft. chimney	60	68	76	84	93	99	105	111	116	121

Heights of chimney in feet	70	75	80	85	90	95	100	105	110
Pounds per hour per sq. ft. grate	15.8	16.4	16.9	17.4	18.0	18.5	19.0	19.5	20.0
Pounds per hour per sq. ft. chimney	126	131	135	139	144	148	152	156	160

A simple formula by Thurston agreeing quite closely with the above table is

$$\text{Rate of combustion} = 2\sqrt{h} - 1,$$

in which h is the height in feet.

Other designers have aimed to deduce formulæ from practice which should take account of the prevalent resistances in grates and fires with different grades of fuel, introducing the results of tests into formulæ as coefficients. But successful practice of others will remain the preferred guide. Sectional-boiler practice using water-tubes has deduced the following diagram (Fig. 167) to represent the draft in inches of water corresponding to any number of pounds of air delivered when the chimney is 100 feet high and the external air is at 60°, as well as the maximum rendition-point between 500° and 600°.

Chimneys over 150 feet in height are rarely justified; but 250 feet of

DIAGRAM OF DRAFT AND CAPACITY OF CHIMNEY.

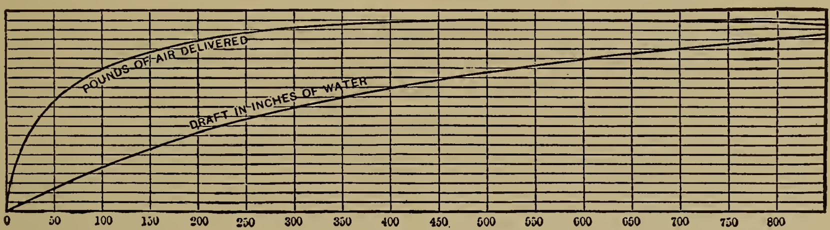


FIG. 167.

height may be compulsory in towns to carry off gaseous or noxious products without possibility of nuisance. The following table represents conservative data:

Pounds of coal consumed per hour.. Up to	100	500	1000	2000	3000	4000	5000
Height in feet........................	60	100	120	140	160	180	200

Several smaller chimneys are often used instead of one large one, where location does not compel great height, with considerable economy.

Fine anthracite coal needs a higher stack than good bituminous coal, both on account of the grate resistance and the lower temperature of the gases, and wood requires less than either of the other two.

Tallest chimneys on record are:

	Feet.
Townsend's Chemical Works, Glasgow	468
Hallsbruckner Hütte, Saxony	460
Standard Oil, Bayonne, N. J.	365
Metropolitan Street Railway Company, New York	353
Omaha & Grant Smelting Company, Denver	352
Clark Thread Company, Newark, N. J.	335
Manhattan Seventy-fifth Street Station Elevated Railroad (four)	278 and 17 feet diameter
Amoskeag Mills, Manchester, N. H.	250
Narragansett E. L. Company, Providence	238
Maryland Steel Company, Sparrows Point, Md	225
Passaic Print Works, Passaic, N. J.	200
Edison Electric Light Company, Brooklyn (two)	150

110. Dilution of the Products of Combustion. In the burning of solid fuel involving the two stages of distillation and subsequent chemical union, the latter step takes place reluctantly in an atmosphere charged with carbonic acid, which is not a supporter of combustion. Hence the chimney and the draft arrangements of the grate must be planned to give from one and one-half to twice the air supply which is theoretically required. This extra weight of air carrying inert nitrogen makes demand on temperature of the fire to heat it up, or in effect cools the fire.

In burning gas or liquid fuel made by spraying into a mist of atomized particles, no such excess of air is required, and this is one cause of the superior effectiveness of such fires. If the air can be preheated by otherwise wasted heat, considerable saving results.

111. The Grate-Bars. Stationary Grates. The problems which the boiler grate must meet (paragraf 106) have been solved by making the grate of bars, either of cast iron or of wrought iron, solid or hollow. Grate-bars may be divided into three classes: the fixed or stationary grates, shaking and dumping grates, and mechanical or traveling grates.

The stationary or fixed grates are almost always of cast-iron bars (Fig. 168). It is most usual to run these bars lengthwise or in the direction of the axis of the boiler and perpendicular to the front. It is easier to clean them when arranged this way. They will be supported by transverse bars, usually of wrought iron, let into the brickwork of the side walls (Fig. 169). There is usually one at the front and one at the back supporting the bars at their ends. It may be, however, that instead of running the bar continuously the whole depth of the furnace, it will be divided in the middle, and each short bar will rest upon a third bearer midway between the other two (Fig. 189).

It would appear that a maximum relation between the supporting function of the bar and free passage of air would be reached when each was made 50 per cent of the surface of the grate. Practically this relation cannot be reached without causing much unburned fuel to fall into the ash-pit to be wasted, or to entail the labor of picking over if it is to be saved. The difficulty is worse as the size of the fuel grows smaller. It is usual to consider the bar satisfactory for the passage of air when 25 per cent of air-space is presented by its design. The proportion of air-space to solid surface of the bar is usually determined by the expedient of laying the bar upon a piece of stiff paper, tracing its profiles of openings with a sharp pencil, cutting out the paper representing the openings, and weighing on delicate scales the relation of the weight of the air-space and solid bar in any given unit of area.

The usual deterioration and failure of grate-bars come from their warping, from fusion of the top surface, and consequent softening and loss of strength, and from breaking through by their own deterioration or from a deterioration caused by the continued heat.

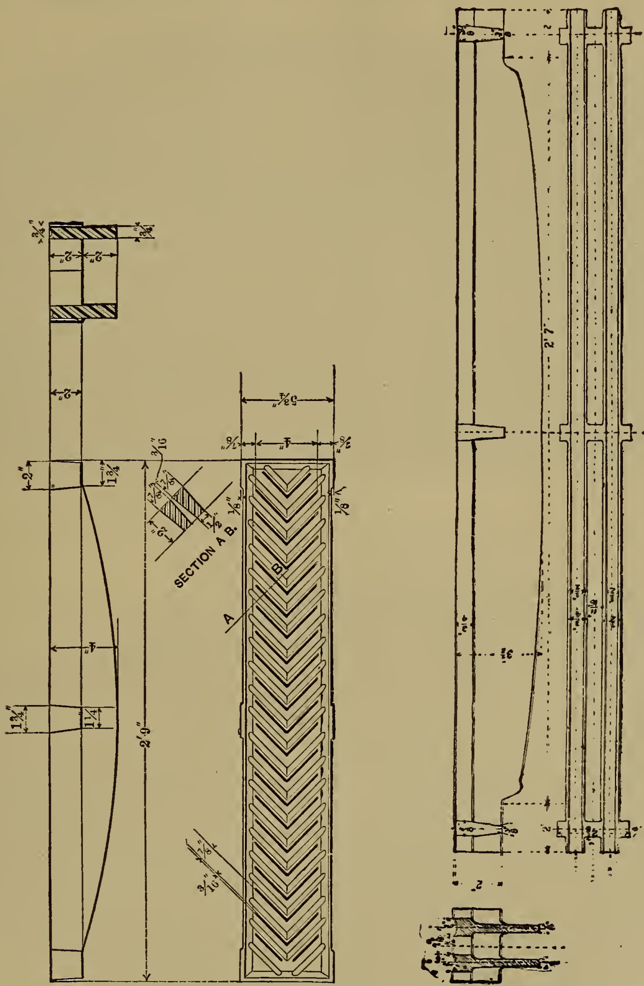


FIG. 168.

Wrought-iron bars when made solid are particularly troublesome from a tendency to warp and to bend from softening by heat. Wrought iron is less stiff than cast iron.

When for any reason the air which enters under the grates is to be preheated so as to lose its cooling effect, solid grate-bars of either cast or wrought iron give trouble by their softening. For this condition and in certain other places hollow wrought-iron grate-bars are used through which the feed-water or the water from the boiler is caused to circulate. This is to keep them from reaching the temperature of softening, and adds to the heating-surface of the boiler. The difficulties from the expansion of such water-grates make them troublesome to join to the ends, but they have formed a satisfactory solution for many problems, and are a necessity in what is called the down-draft furnace (Fig. 202), to be referred to hereafter. Some of the bars in locomotive-boilers are usually water-tubes.

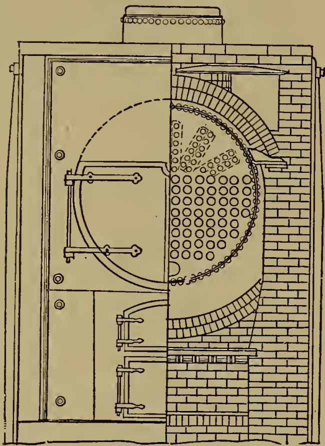


FIG. 169.

With very fine fuel, such as coal-dust, and where sawdust is used as fuel, the grate-bar has to become a perforated hollow tube or a plate (Fig. 173). Where oil or gas is used the grate-bar disappears entirely, and the gas will be passed up through the perforations made in a fire-brick or similar floor which converts the grate into a form of burner.

112. Shaking and Dumping Grate-Bars. The stationary or fixed grate-bar is cleaned by running a proper tool, called a slice-bar, over the top surface, or a poker between the bars. This is a labor of considerable difficulty and requires that the furnace-door should be open while it is going on, and the cold air thus admitted not only deadens the fire but cools the heating-surface and checks the generation of steam. What are called shaking-grates are grates whose bars are so constructed that by a lever or similar means a motion can be given to the bars, from without the setting, whereby the fire shall be agitated, the fine dust or ashes shall be shaken downwards through the openings of the bars, and the ash or clinker which has attached itself to the top surface of the bars shall be broken up and ground into pieces fine enough to drop through and leave the fire clean. This result is attained in various designs of grate-bars by different mechanical methods. In some the bars are supported by proper bearers at their ends, to which bearers such a motion is given that the alternate bars move lengthwise in opposite directions through several inches of travel when the lever of the shaking mechanism is worked (Fig. 170). In others each individual bar receives a rocking motion around the axis upon which it is supported. The rocking motion lifts the fire and lowers it, thus shaking out the accumulation of ashes and dirt.

Dumping-grates are a form of shaking-grate in which the motion which shakes the bars when carried farther opens sufficient space between the adjacent bars to allow the fire to slip off the top surface

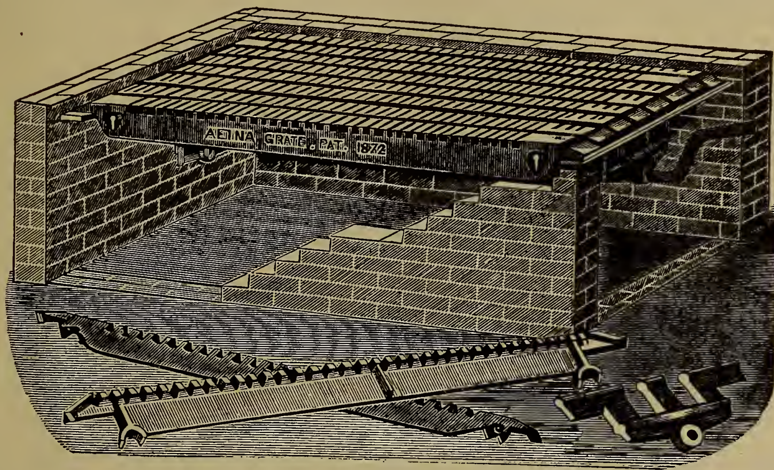


Fig. 170.

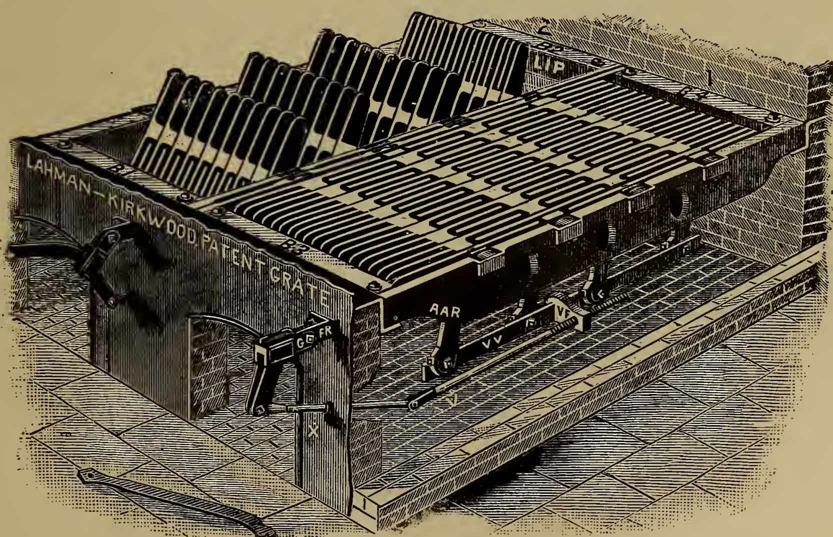


Fig. 171.

of the bar into the space thus opened and fall into the ash-pit below. The difficulty with the dumping type of grate-bar is that carelessness in its use causes a loss of an excess of fuel in cleaning (Fig. 171).

Fig. 172 shows a design in which the throwing of the bar from the level position of service to the front gives the shaking motion, and the lifting of the front of each bar or its throw to the back gives the dumping effect. In Fig. 173 the bar proper is hollow with air openings

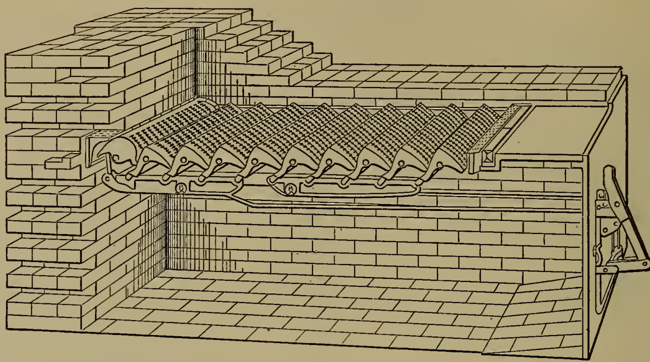


Fig. 172.

on the upper surface and is intended to operate with forced draft. The bars are rocked by a side-to-side motion across the front of the boiler.

The advantages of the shaking-grate are as follows:

(1) The fire-door is opened for coaling, but not for cleaning.

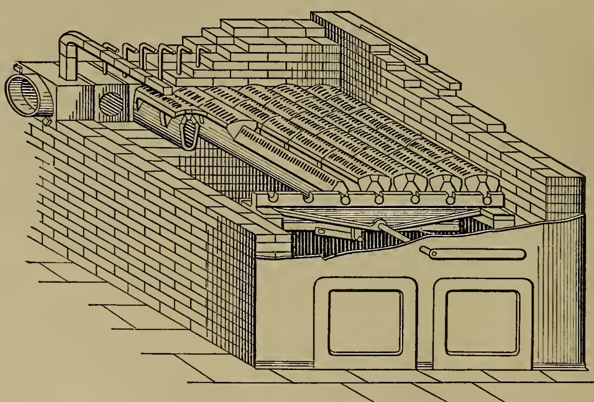


Fig. 173.

(2) The fire-box lasts longer, because not exposed to the shrinkage and deterioration caused by cold air coming in upon its heated surface.

(3) The firing is more regular, because the fires are kept in a condition of good efficiency by being always clean, and are not torn to pieces by the effort of the fireman to cleanse them. This is particularly true with

anthracite as a fuel. One man can attend to more furnaces when the labor of attending to each is so much lightened.

(4) The duty of the fireman is made less arduous and exhausting when he does not have to face the intense heat of the furnaces at the open doors for so long a time.

The objections to the shaking-grate are as follows:

(1) It does not work with all varieties of bituminous fuel. Where the coal is what is called fat, so that it fuses together on the upper surface of the fire, the shaking-grate does not cleanse the fire, but only leaves a hollow space below the real body of the fire. For coal of this class the use of the slice-bar is necessary in any case, and it might as well be used altogether.

(2) The trouble and annoyance from machinery of any sort in an ash-pit. It cannot be lubricated; it is exposed to grit and dust.

(3) The efficiency of the bar for cleansing usually throws down excess of unburned fuel into the ash-pit. The shaking-grate for stationary practice is usually considered to be a stepping-stone on the way to the use of mechanical stoking, and its advantages are usually reaped with the advantages which the latter offers.

113. Step-Grates. For the burning of fine coal, and particularly in soft varieties where a large quantity of air is a necessity, a form of grate has been long used which is called the step-grate. The bars are flat surfaces or treads arranged so that the upper one slightly overlaps the one below it, while leaving the space open which corresponds to the riser in stairway construction for the passage of air. It will be seen that this construction permits abundance of access of air with little or no possibility of coal dropping through the grate-surface. When the bars are laid across the furnace, as is usual, the slice-bar of the fireman can cleanse each bar separately by working through the vertical opening between the bars, or the method of firing may be used whereby the coal is fed first on the upper bar, and from that is gradually pushed down the steps from bar to bar until at the bottom it will be pushed off with all available combustible matter utilized, and only refuse and ash remaining.

It is very easy to make such a step-grate become a shaking or dumping-grate by arranging each bar so as to permit a motion to tip it down the steps. This can be done either by hand or by mechanical means (Fig. 174).

114. Mechanical or Traveling Grates. — The principle of successive passage of fuel from bar to bar suggested in the previous paragraf leads to a construction of grate which is known as the traveling-grate. The bars, instead of being continuous and solid, are made up of a series of short bars which are pinned together so as to form a flat chain with

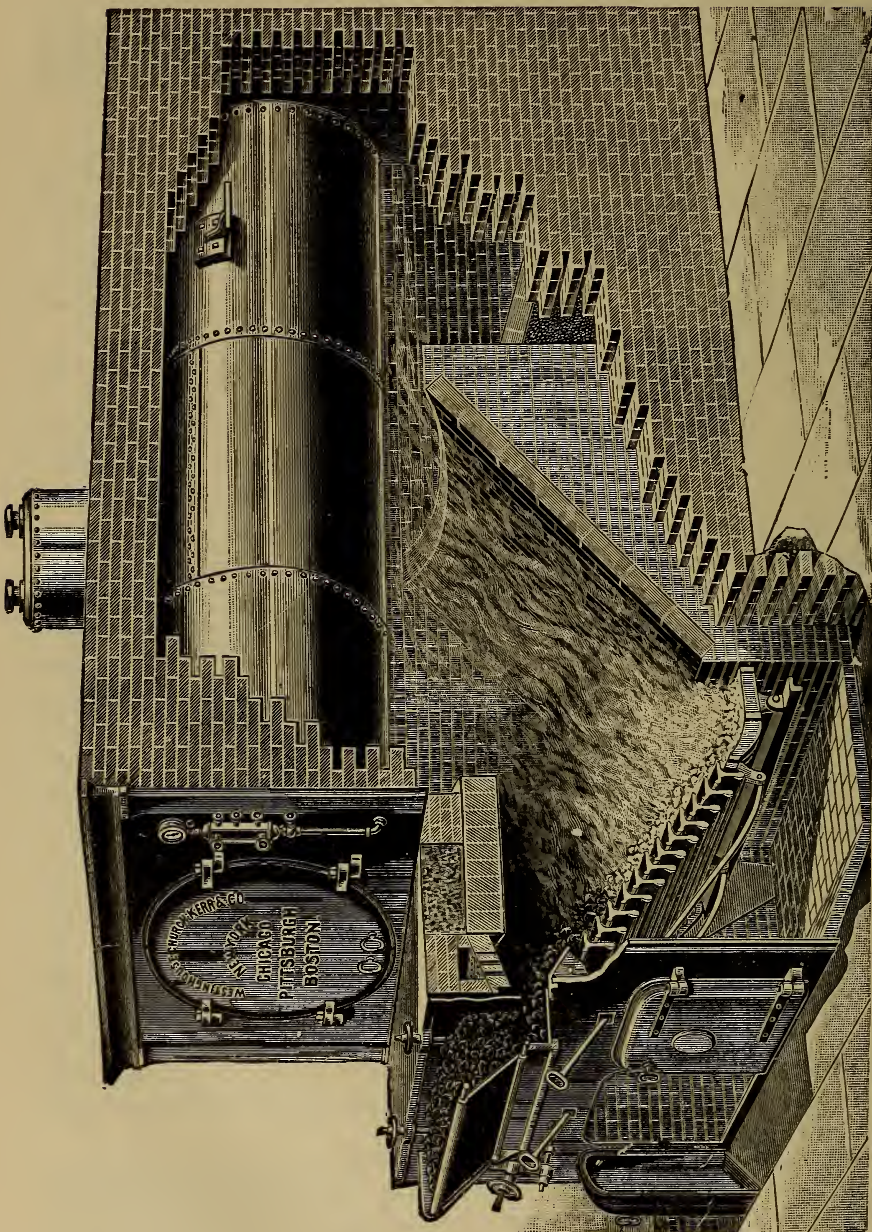

WESTINGHOUSE, CHURCH, KERR & CO.
NEW YORK
CHICAGO
PITTSBURGH
BOSTON

Fig. 174.

the links edgewise. Chains of these flat links, made endless, mounted upon proper carrying-rollers at the front of the furnace and at the rear, and having the width of the furnace area, can be driven by machinery attached to the rollers so as to draw the chain from the front of the furnace to the back, carrying on its surface the fuel to be burned. The speed of driving should be so proportioned that the fresh fuel charged at the front upon the traveling-bed of the grate should be

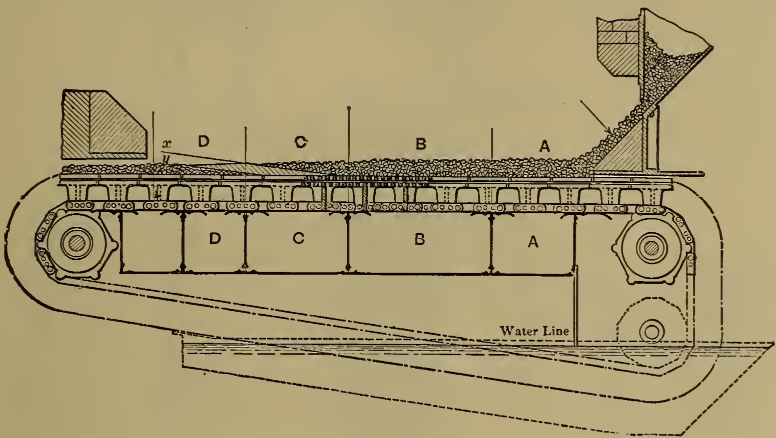


FIG. 175.

completely burned during the period of its transition to the back, so that when a given series of links reaches the rear roller and is dropped over, there is carried with it and dropped only the incombustible matter in that given amount of coal. Such a grate is practically self-cleansing and leads at once to the use of an automatic appliance for feeding the fuel to it to make it complete. Fig. 175 will show a typical traveling-grate, and Fig. 174 a type of grate in which the passage of fuel from step to step is made to be automatic by mechanical means. It will be seen that the mechanical grates of this type lend themselves and lead naturally to the principle of the automatic stokers.

115. Mechanical Stokers. A combination of a mechanical grate with a mechanical feed of the fuel constitutes a mechanical stoker. This is aimed to secure both economy of labor and complete and economical combustion. Inasmuch as the mechanical feeding is an auxiliary equipment of the power plant, the discussion of the stoker will be postponed till the discussion of such auxiliaries is taken up in Chapter XIII, as paragraf 223.

116. Inclined and Horizontal Grates. It will be noticed by examining Figs. 178 and 179 that a difference of practice prevails with respect to arranging the grate-bars horizontally, or inclining them downwards at the back in the proportion of about 3 inches in 6 feet. The practice of inclining is quite usual, in order that the under surface of the fire may come more nearly normal to the incoming air-currents, so as to invite

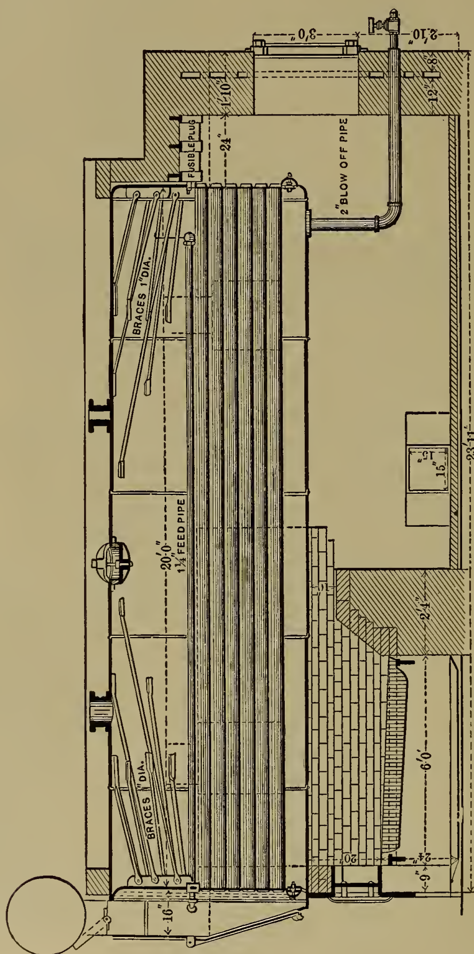


Fig. 178.

them to pass equally through all parts of the fire, rather than to take the easiest course. In sectional boilers with inclined water-tubes the inclined grate is of advantage in keeping the surface of the radiating fire more nearly parallel to the absorbing surface of the tubes. Inclined grates are also easier to clean by slicing.

The horizontal grate renders it more easy to keep the fire of even thickness at the front and back, and makes it slightly easier to withdraw the clinker and other solid matter which is to be drawn forward and out through the fire-door in arrangements of this sort. The general prevalence of the inclined bar seems to indicate that it offers advantages over the other arrangement.

The level of the grate-bars with hand-firing should be so selected as to make the cleaning and coaling convenient to the fireman. This seems to be secured by having the top of the grates from 24 to 30 inches

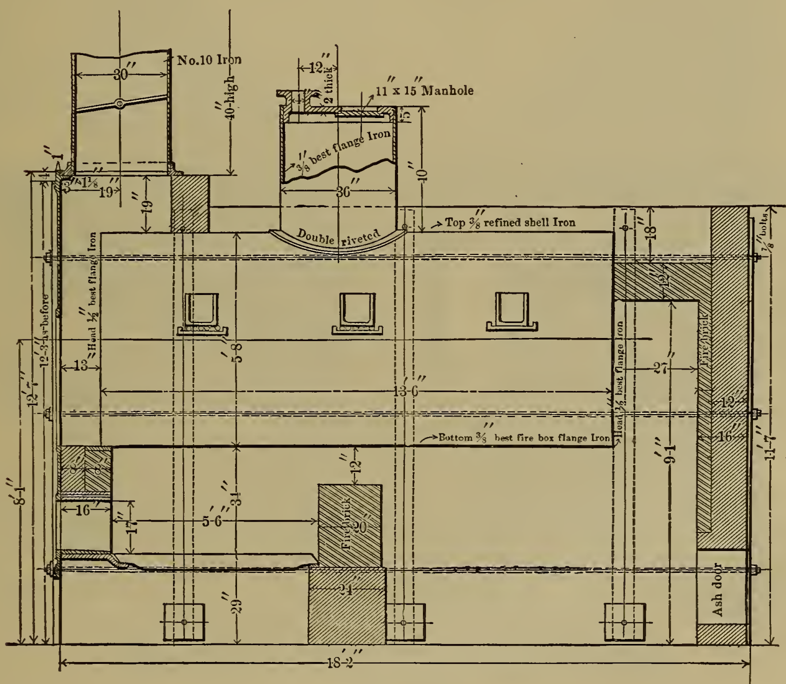


Fig. 179.

above the general floor-level. The depth of the furnace or the length of the bar with hand-firing seems to be determined by the twofold considerations of ease of cleaning and the satisfactory spreading of fuel. When the fireman stands on the floor-level he can easily deliver coal with precision at the back of the grate, which is 6 or even 7 feet deep. When he stands above the grates, as in the case of the locomotive, he can throw coal to the back of a fire-box 10 feet deep. Cleaning, however, by hand, cannot easily be done with a furnace deeper than 6 feet, and this is usually placed for the limit of the length of the grate-bar. With shaking or mechanical grates the grate could be deeper if it were otherwise desirable.

117. The Firing on the Grates. There are two systems to realize the complete double stage process of combustion with solid fuel. The one is called the side-firing system and is easily carried out when the furnace is wide enough to require two doors for complete access for firing and repairs. In this, the furnace is supposed to be in two halves lengthwise, and each half is fired alternately with respect to the other. The coal is thrown, for example, on the right-hand half through the right door at the time when the left-hand half has completed the distillation from the black fuel, and all that part is in red glow and full combustion. In side view, therefore, it looks like Fig. 180. Then when the right half is all aglow and no more smoky-looking gas is coming off, the left hand

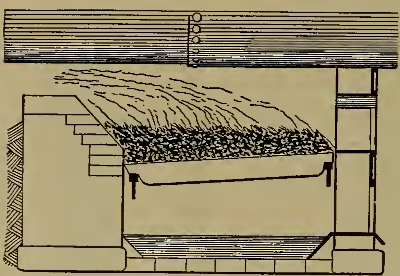


FIG. 180.

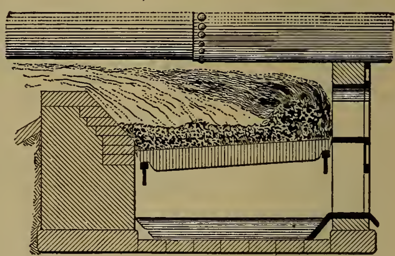


FIG. 181.

receives fresh coal, and so on. In the other or coking system, the fresh coal is always thrown on the dead area in front of the grate-bars (Fig. 182) and when warmed enough is pushed back on the front end of the bars (Fig. 181) where distillation is active, and the gases mingling with the hot flame and products of complete combustion are made hot enough to burn also. When distillation is complete the incandescent fuel is pushed back and fresh coal charged at the front. The fire should be of uniform thickness on the grate to give uniform distribution of oxygen, and there should be no holes in it to allow cool air to break through without supporting combustion. Such jets of cool air stop generation of steam and are bad for the shell.

118. The Dead Plate and Furnace Mouthpiece. The front end of the grate-bars should not project beyond the end of the heating-surface of the boiler, and therefore a distance equal to the depth of the smoke-box will lie behind the fire-door and between it and the end of the bars. This gives a width of space which serves as a dead-plate for the coking method of firing when it is desired to adopt it, but even with anthracite firing, where no coking is required, the dead-plate remains as a distance-piece, but without significance or use in firing. It usually forms the top of the opening into the ash-pit below, and is simply a plate of cast iron built into the brickwork of the setting at the sides (Fig. 182). In some cases the dead-plate has been made to drop by hinging the front end

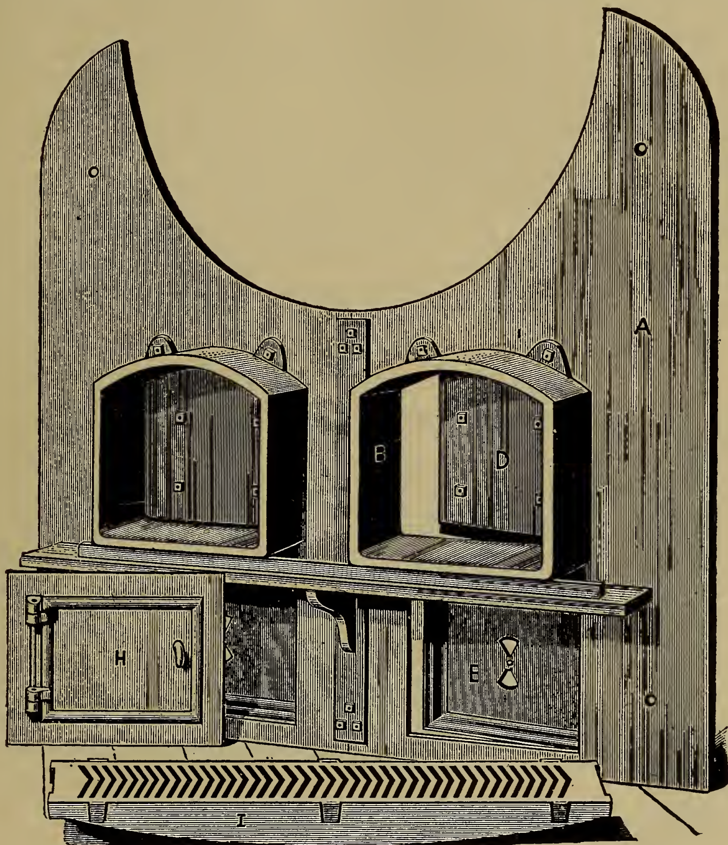


Fig. 182.

against the boiler-front and holding up the back by a latch which can be released. The object of this arrangement was to permit clinker and ashes too large to pass through the grates into the ash-pit to be dumped into the latter over the ends of the bars without coming out through the fire-door and causing unpleasant odors from any cause which such material might give off in the open fire-room.

The sides and top of the furnace-mouth opening will be made either of cast iron, like the dead-plate which forms its bottom, or of fire-brick. The latter may be either the ordinary forms of fire-brick, or specially molded shapes can be obtained whereby the mouthpiece has but a few joints in it to give trouble in service. The mouthpiece must flare towards the furnace in order that an opening smaller than the width of the grates may permit access to every part of the grate, and the injury from firing-tools and from the action of the heat of the fire makes trouble with ordinary brick construction. The arch over the door is also a flat one which it is troublesome to make and maintain if of many separate pieces. The sides are sometimes of the usual sizes of brick, and the top of cast iron.

119. The Ash-Pit. The ash-pit, as its name indicates, is to catch the refuse incombustible matter when the fires are cleansed. It is simply formed by the sides which form the furnace, and is paved on its bottom with fire-brick also. The bottom is sometimes made lower than the general floor-level (Fig. 183) in order that water may be allowed to lie in the depression thus formed. The object of the water

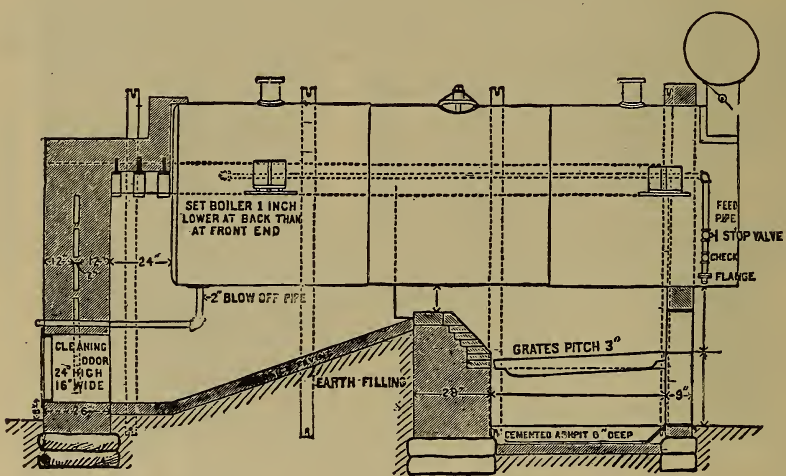


Fig. 183.

is first to quench incandescent matter which if allowed to glow in the ash-pit would heat and soften the grate-bars. It is desirable also that if sulphur-gases are given off from such ash, the process should be stopped at once. It is further urged that the steam formed from this water will tend to keep the grate-bars cool on its passage through them, and the combustion of the hydrogen, when such steam is dissociated in the fire, will add to the heat of the usual combustion. The objection to this is that the dissociation of the steam cools the fire itself exactly to the same extent that the combustion of hydrogen would raise its temperature. With short-flame fuel the hydrogen may act to lengthen the flame and increase the effect of radiation in a perceptible degree.

With long-flame fuel its effect is not observable. It is a question whether steam from the ash-pit may not act to rust metallic surfaces and to form a more active compound with the sulphur-gas given off than when that gas is dry.

When the air for combustion is to be supplied to the fire by mechanical means so as to create an artificial draft by pressure below the grates (paragrafs 126–128), the flues or ducts for such artificial draft will be carried into the ash-pit. The best places are the side walls, rather than the bottom, since it is difficult to keep ashes from dropping into the ducts when the openings are directly under the grates (Fig. 201). These openings will be controlled with proper dampers operated from outside of the setting.

In large plants where the weight of ashes to be disposed of in any day becomes very large, it is worth while to arrange the ash-pits so as to deliver their accumulations into a tunnel underneath them through which a suitable wagon may be wheeled to receive the contents of each pit as it stands under a convenient opening below it. This principle also becomes of importance when the mechanical methods of firing are used whereby the grate is made to be self-cleansing and discharges its ash and incombustible matter continuously over its end. If the wagon method is inconvenient, it may be replaced by a continuous conveyor whereby the discharge from each grate or ash-pit falls upon a continuously moving band, and is carried by it and dumped into some convenient receptacle outside.

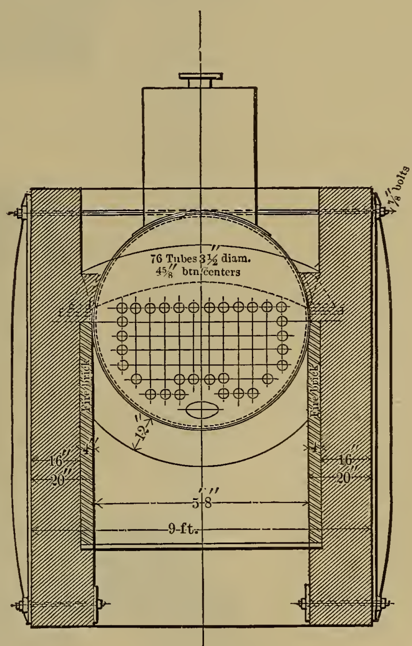


Fig. 184.

120. The Bridge-Wall. — The back of the ash-pit and of the furnace or fire-box is made by a low wall over whose top the gases and products of combustion are to pass. It separates this space in the setting from the combustion-chamber behind it. In so far as it is merely a separating wall it might be made of 8 inches in thickness, but inasmuch as with stationary grates it is liable to suffer impact from the slice-bar in cleaning the top of the grates, it is more usual to give it a thickness from the bottom to the line of the grates of 2½ or even 3 bricks lengthwise, giving a dimension of from 20 to 24 inches. It is not necessary that at the top it should be of this full width, and therefore it is quite

Fig. 185.

usual to taper it from the line of the grates backwards, either from the front or from the back, so as to give it a width of one brick or 8 inches only at the top. Examples of both methods of tapering will be found in the illustrations Figs. 178 to 188. The objection to tapering from the front or fire-box side is that so much of the fire as lies upon the sloping surface does not receive its full proportion of air, although this is corrected in part by the slanting direction which the air takes in passing from the grates to the top of the bridge. The diminished thickness at the top is of advantage in diminishing the friction of the

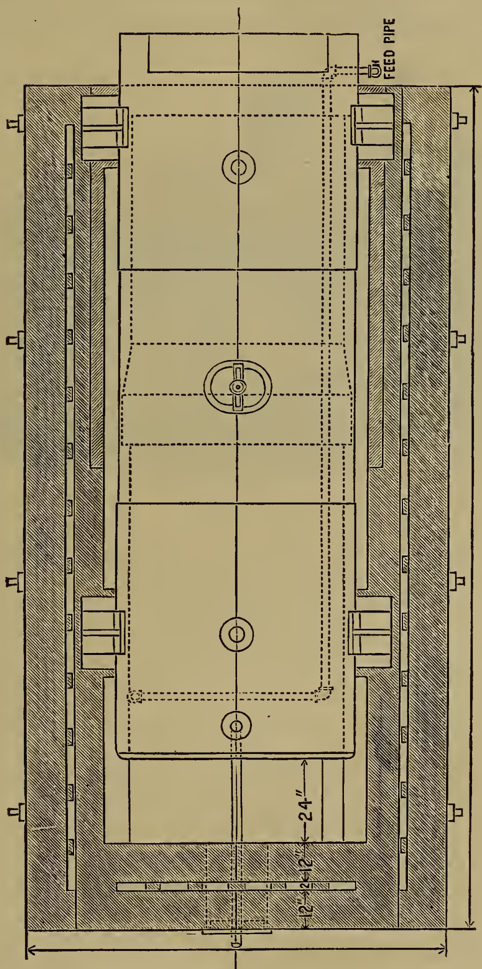


FIG. 186.

gases in passing over the bridge, and in rendering it unlikely that misdirected fuel will be caught upon it. It will be observed also that there is difference of practice as to making the top of the bridge-wall a horizontal line, or an inverted arch parallel to the circumference of the shell. The inverted arch is supposed to direct the currents of hot gas and flame close to the shell. It makes, however, a very deep corner where the height from the grate-surface is much greater than at the middle. The horizontal wall is easier to make, keeps the fire of equal intensity over its whole width, and the tendency of hot gas and flame is to keep to the upper part of its passage in any event.

The bridge-wall is represented as solid in the foregoing illustrations: it is quite common to perforate its rear at or near its top, and to make

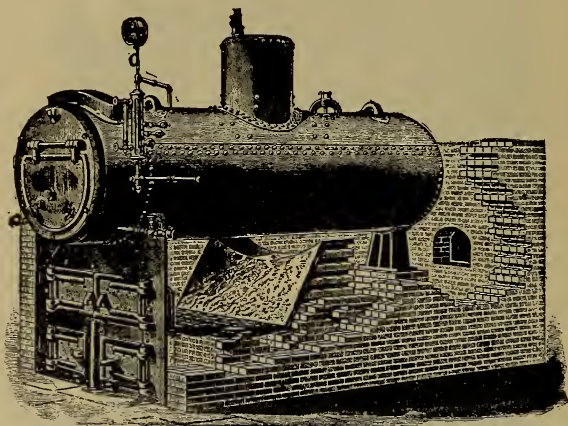


Fig. 187.

openings into a hollow within it to which air can have access from the outside. The outside of the setting will force air in through this hollow wall, where it will become heated by contact with the hot bricks, and passing through the opening will mix with the flowing products of combustion over the top, and help to complete their combustion (Fig. 188). With this same purpose the bridge-wall is often made of a hollow cast-iron box with similar perforations at its back. It will be seen also that a metallic bridge-wall may be filled with water to be evaporated, and, if proper circulation is kept up within it, it can form an efficient addition to the heating-surface. If water-grates are to be used in a brick setting, the water bridge-wall becomes, practically, a necessity.

121. The Combustion-chamber. — Behind the bridge-wall and underneath the shell of the boiler is an open space intended to permit complete combustion of the carbon which may come over the bridge-wall in the form of flame or combustible gas. For this reason it is called the combustion-chamber, even if, as is the case in anthracite practice, there is really no combustion to take place within it. It is desirable to have it with gas-fuels in order that a space may be made in which the boiler shall not be too closely forced into contact with the hot gases and extinguish them by its lowered temperature, and, furthermore, in which there shall be permitted both room enough and time enough for a proper union of oxygen with the gases. It is furthermore of advantage, if otherwise practicable, to introduce refractory bricks or similar material into this combustion-chamber which shall serve to keep up the temperature of the flame and gases above the point below which no chemical union can occur. In anthracite practice this chamber can be filled up in part or largely without disadvantage. In bituminous practice this would cause a smoky and wasteful combustion. Fig. 189 shows a type of a setting prevalent at one time in which the small size of the combustion-chamber may be credited with causing very smoky chimneys. The combustion-chamber serves also as a catch-chamber to hold some of the particles of ash and flue-dust which will be drawn out of the fire by a strong draft, but which will be precipitated by the lower velocity of the gas-currents in the large area behind the bridge-wall. This makes it necessary that there should be doors of access into the combustion-chamber, that it may be cleaned out at intervals, and such doors give also a convenient access for inspection of that part of the boiler. These doors will usually be of some size (perhaps 18 or 24 inches wide by 18, 24, or 36 inches high), and they will be made by building a flanged framework of cast iron into the brickwork which will clasp the flange, and be supported by them while the projecting plane beyond the brickwork carries the hinges (see *H* in Fig. 182). The door-openings are objectionable, because they break the continuity of the brick wall and cracks originate from them for this reason. It would be desirable not to put them at the bottom on account of this tendency to create cracks, which are less troublesome if they are towards the top. The location of the doors in the side or back wall of the combustion-chamber must be a matter of convenience and location, but the back wall is not as good a place as the sides by reason of the effect of direct impact of flame and gases.

In sectional-boiler settings the combustion-chamber is partly filled by the boiler itself, or rather it is made from a space within which are the tubes. The absence of return fire-tubes in boilers of this class compels the gases to receive a circuitous path in and out among the water-tubes, and this is secured by partitions of fire-brick like hanging bridge-walls, which force the gases to pass around them and meet complete combustion while still in contact with the tubes. It is probable that the gases will be hotter when leaving a sectional boiler than in leaving a return tubular boiler for these reasons.

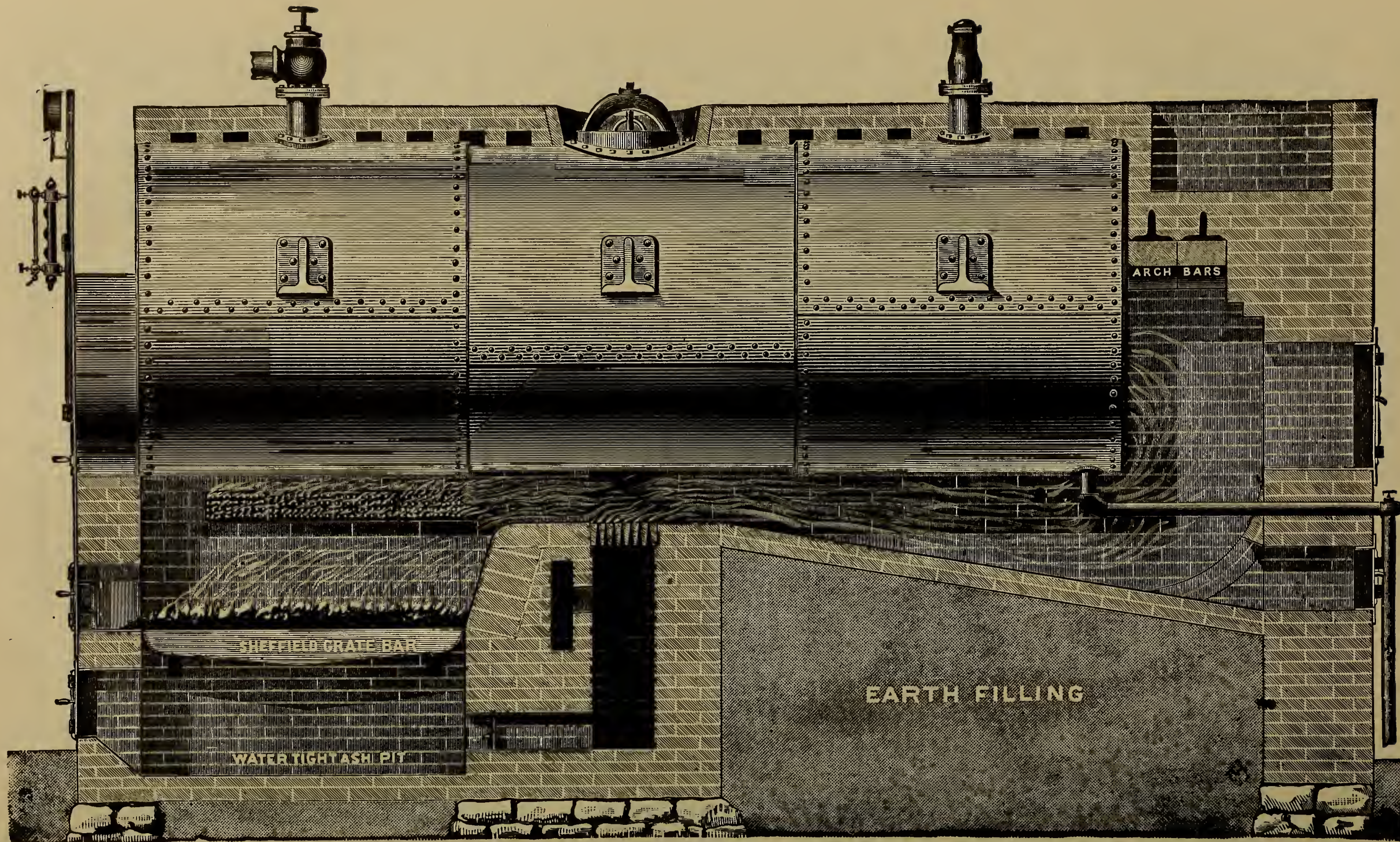

ARCH BARS
SHEFFIELD GRATE BAR
WATER TIGHT ASH PIT
EARTH FILLING

Fig. 188.

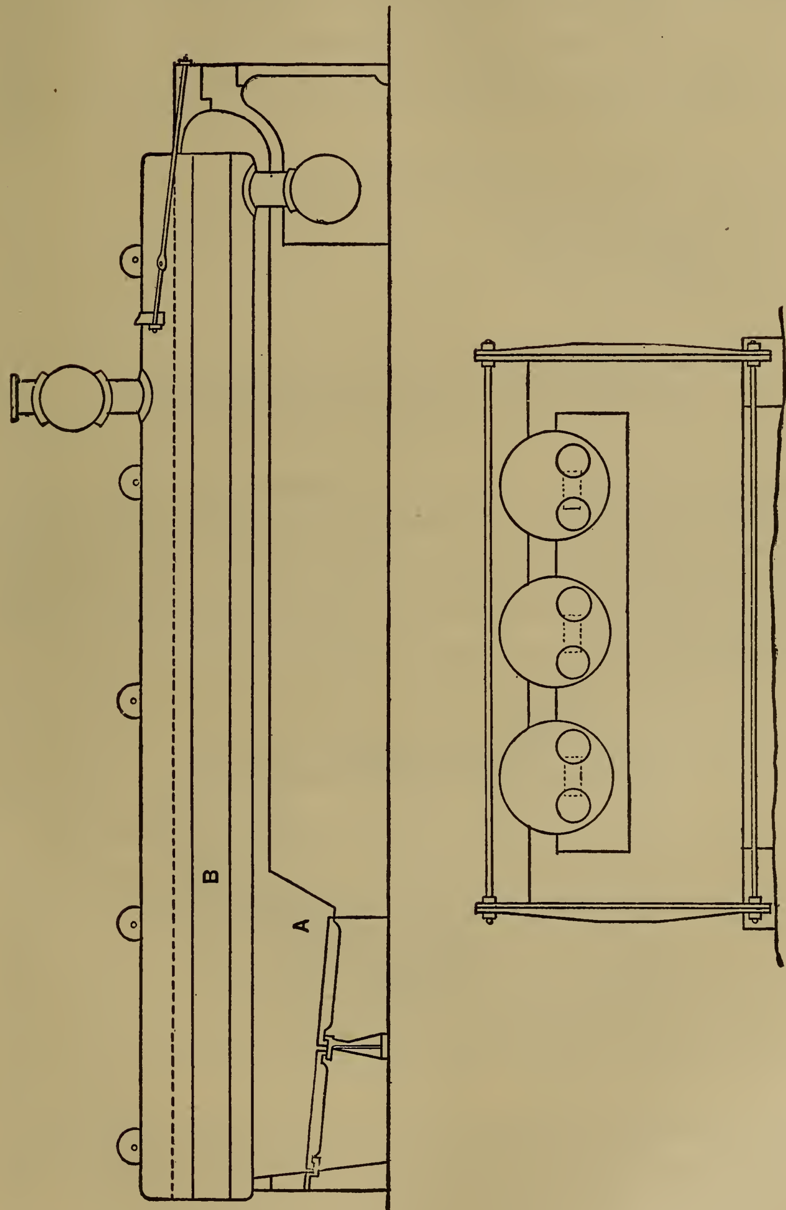


FIG. 189.

122. The Back Connection. — The hot gases passing backwards underneath the shell of the boiler are to be deflected into the tubes or flues in order to come forward through them to the front. Following the analogy of the internally-fired boilers, this space at the back end of the setting in which the tube sheet comes has been called the back

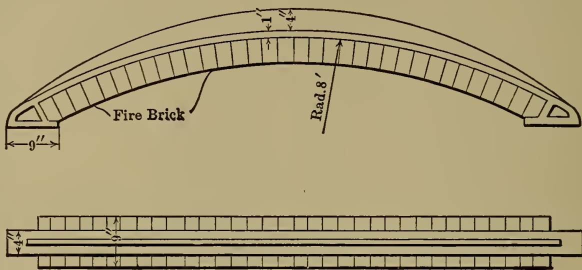


Fig. 190.

connection. It is apt to be about 2 feet deep, and must be roofed at the top at such a level that the flame and hot gases impinging against the back head shall not heat the surface of that head, which is not protected from overheating by water on the inside. It will be seen from examination of Figs. 178 to 214 that there are three methods for making this roof of the brick connection.

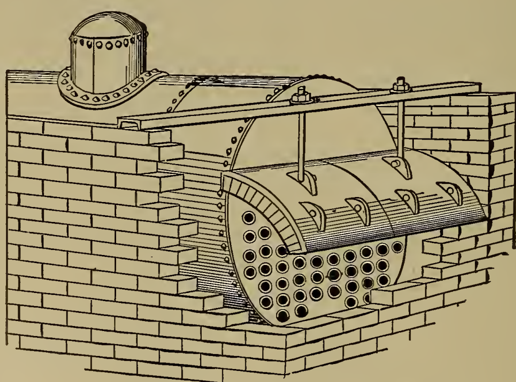


Fig. 191.

First, the roof may be made of an arch whose axis is transverse to the setting, and of which the boiler itself shall form the keystone and take the thrust of the arch (Fig. 751).

In Fig. 191 is shown the scheme of hanging this semi-arch from a bearer resting on the side walls, taking off the thrust on the boiler and giving longer life to the arch.

Second, the roof may be flat, the bricks which form it being supported upon transverse bars of cast or wrought iron which rest upon the side walls and support the bricks. Cast iron is better than wrought from its resistance to softening by heat, and the usual shape is a T iron with its cross downwards and the web of the T among and between the bricks (Figs. 178, 188).

The third plan is to arch the roof with an axis parallel to that of the boiler and with its abutments on the side walls. The fire-brick is supported as in the second plan by a cast-iron arch bar ribbed on its upper side for stiffness (Fig. 190) so that its section is T-shaped in the middle. Without the bars arch would deteriorate by heat and fall in.

If the first method is used, the back end of the boiler must be the fixed end, and expansion be from this end towards the front. The back connection must be large enough to give convenient access to the back head of the boiler for any repairs which may be called for at that point.

123. The Front Connection. The gases which pass through the flues or tubes are to be gathered together at the front head and discharged into the stack. When the front end is not made a smoke-box it will be called the front connection. The gases should have parted with a great deal of their heat in passing through the flues or tubes, so that their volume is less, and for this reason the front connection is usually about two-thirds the depth of the back connection. Sectional boilers have no front connection, but the gases pass directly from the back connection to the stack. The front connection gives access to the front head of the boiler, and the flue-doors of the boiler-front admit to it from the outside.

124. The Flue to the Chimney-Stack. When the front connection is a smoke-box in extended front settings, and in many cases of full front settings, the gases pass directly through an opening into a metallic flue which carries the products of combustion to the chimney and so to waste. If there are several boilers side by side or in a battery, short lengths of flue from each front connection or smoke-box will unite them to a larger flue increasing in size as additional quantities of gas are discharged into it, and through this common flue they pass into the chimney (Fig. 192).

When the chimney is at the back of the setting a customary arrangement has been to carry the gases to the rear in a flue formed by springing an arch over the top of the boiler from side wall to side wall. The tie-rods and buckstays withstand the thrust of this arch, and from the space thus formed the gases pass to the chimney. Figs. 75, 168, and 193 show this arrangement clearly.

It offers the following advantages:

(1) Radiation is diminished from the top and the boiler is kept warm by its own gases.

(2) If these gases are hot enough, they have a tendency to dry or even slightly to superheat the steam in the steam-space and in the dome.

The objections to this construction are:

(3) It is of small value as a superheating appliance, because shortly after starting the boiler is thoroughly covered with a coating of fine ashes or dust which is practically a non-conductor.

(4) It is difficult and usually unwarrantedly expensive to construct the opening through which the dome of the boiler must protrude, and the expansion of the boiler in the brickwork opens cracks for leakage of air into the flue.

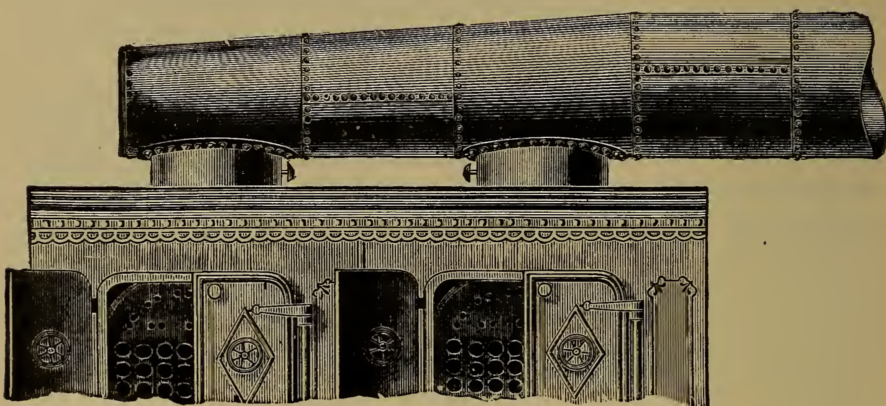


Fig. 192.

When, however, the chimney must of necessity be at the rear of the setting of such boilers, these difficulties can be avoided sufficiently well to make it a justifiable feature of settings for anthracite coal, but not for bituminous. It should be large enough to permit the access of a man for inspection.

Where it is not used, the top of the boiler will be covered with some non-conducting material laid on in sections which shall permit their removal for inspection. These non-conducting coverings catch and hold any water of leakage, and unless care is taken may occasion external corrosion.

125. The Damper and Damper-Regulator. In order to control the action of the chimney, which depends on the weight of a column of air outside of it, a valve of some sort is required in the flue from the boiler.

When closed wholly or in part it causes a friction in the discharge of the gases through it, which checks the flow of air through the fire.

It is usually made in one of two forms. The sliding or guillotine damper is a flat plate sliding in grooves across a frame in the flue (Fig. 75). The pivoted or balanced damper is a plate mounted upon an axis through its center of gravity by which it can be turned so as to

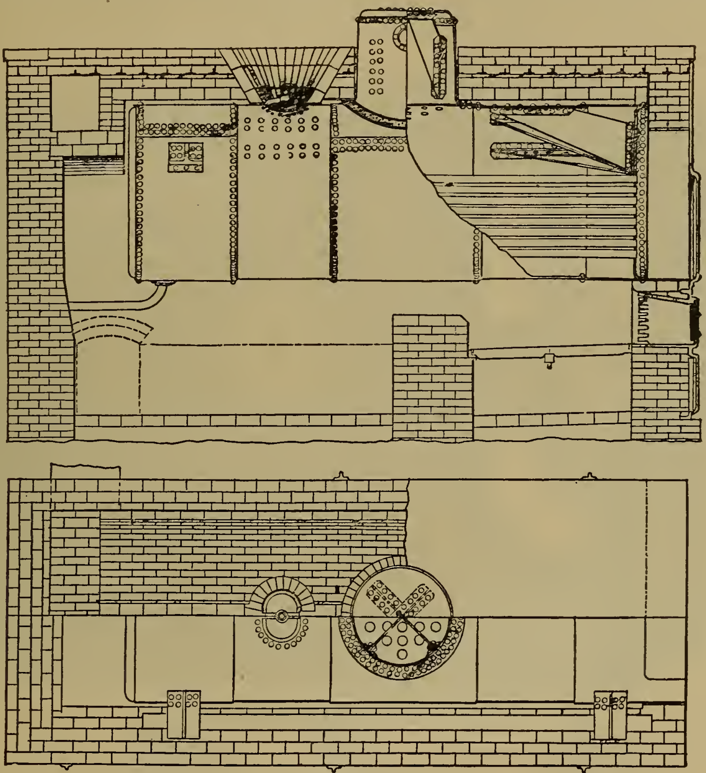


FIG. 193.

stand edgewise to the flow of gas, opposing little resistance, or flatwise to it so as to close the opening altogether. The sliding damper usually is the harder to move, and if it slides vertically has to be counterweighted in order to be balanced. The other form is in equilibrium in any position. The damper is often arranged not to close entirely even when it is nominally shut, in order that there may still be a tendency for a current to be maintained inwards through the setting, and out through the stack to prevent undesired gases from getting into the boiler-room because access to the chimney is closed.

Since the chimney is the immediate and usual method of controlling the fire, it becomes exceedingly simple to make it automatic, so that the fire shall be somewhat self-regulating. The pressure of steam can be brought against a piston, and the motion caused by that pressure can be resisted by a weight or spring. When the pressure exceeds the normal, the weight will be overcome; when it falls below the normal, the fall of the weight will move the piston the other way. The motion of the piston, which can also be made a diaphragm of flexible metal, can be

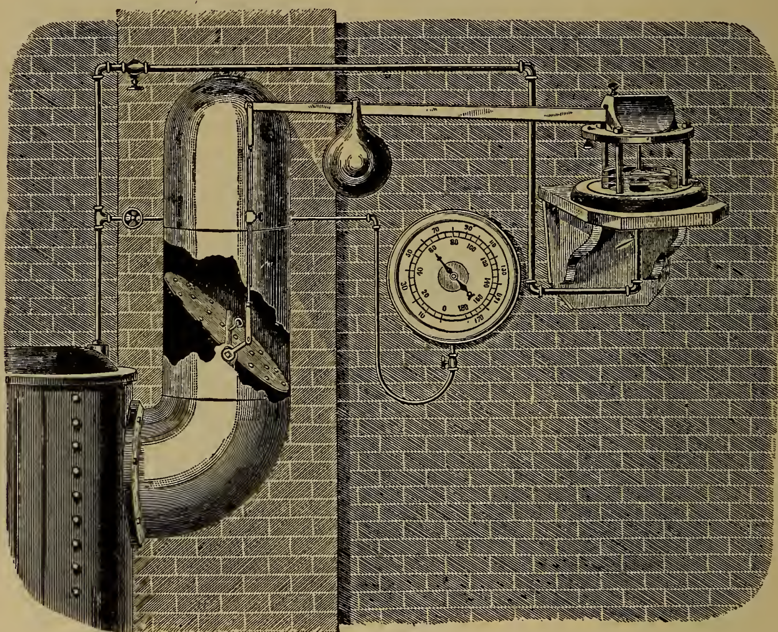


Fig. 194.

attached to the damper so as to close or open it when the pressure rises or falls. This may be done either directly, as in some of the older forms of damper-regulation (Fig. 194), or the steam-pressure may move a valve to admit the pressure which operates the damper, upon one side or another, of the mechanism which moves the latter (Fig. 195). This may be the water-pressure of the city mains, or it may be the pressure from the boiler of the steam or water in the boiler itself.

126. The Chimney. The effect of height in the chimney as causing flow of cold air into the ash-pit has been discussed in paragrafs 107–109. The weight of chimney-gas moving per second through the fire is conditioned both upon velocity and cross-section: and these vary inversely as each other for a fixed weight of gas. Too large a cross-section makes the chimney draw badly because the lower velocities cause eddies and back-draft: the chimney is unnecessarily costly to build; and the gases are unnecessarily cooled by contact with the large chimney surface. Hence the standard cross-section exists of one-eighth the grate area as representing successful relations between grate

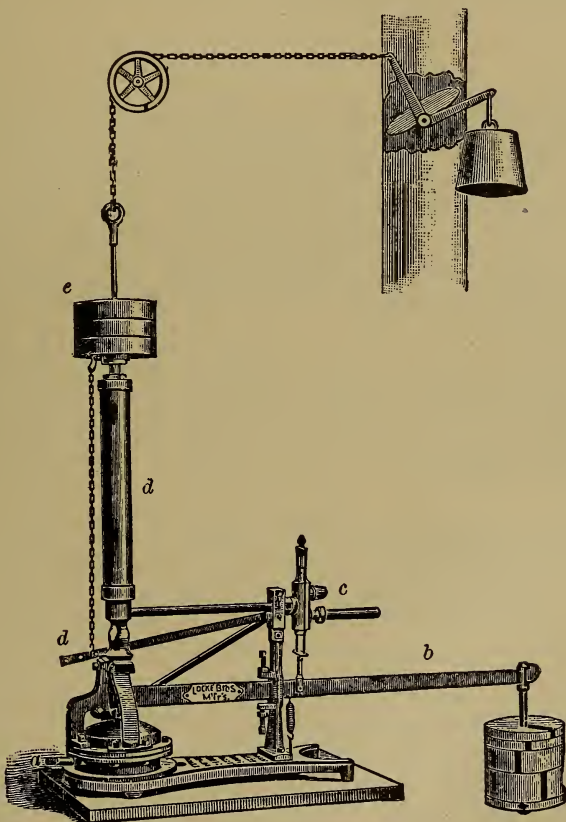


FIG. 195.

area and chimney area. This can be shown to be ample for any normal velocity, for if an area of one square foot be taken, and a temperature for maximum output $v_1 = \sqrt{2\,gH}$, and if H be taken at 64 feet of height for illustration,

$$\begin{aligned} v_1 &= 64 \text{ cubic feet per second} \\ &= 64 \times 3600 = 230{,}000 \text{ cubic feet per hour.} \end{aligned}$$

Suppose 20 pounds of coal burned per hour per square foot of grate, and 300 cubic feet of air per pound of coal; then $20 \times 300 = 6000$ cubic feet of air at 62° will be required per square foot of grate. At 626° F. in the chimney this air will have twice its volume at 62°, since

$$V_c = V_a \frac{T_c}{T_a} = 6000 \tfrac{1085}{521};$$

whence

$$V_c = 12{,}500 \text{ cubic feet},$$

which, if multiplied by the assumed relation of chimney 1 : grate 8 = 96,000, is only about $\frac{96000}{230000}$, or one-third of what the chimney of only 64 feet high will take care of per foot of area of cross-section.

The friction becomes greater if the chimney be too small, and plants are usually enlarged after some years of use. Hence, although this one-eighth value is large, it is usually best not to pass much below it in small plants. Possible excess of area is corrected by partly closing the damper in the flue to the stack.

An ingenious designer has proposed to use the dead area of the Kent formula, or the back-draft area in the above discussion, as a passage to bring preheated air down the stack so as to introduce it above the fire and avoid the consumption of fuel required to raise this air to fire-temperature.

The chimney foundation problem belongs rather to structural engineering than to a treatise upon power, and it would divert from present purposes to discuss these questions at length. Wind-pressure is not likely to reach 55 pounds per square foot of flat surface; and the chimney may be viewed as a cantilever loaded uniformly with this load. In brick structures this must never produce tension on the windward side, when compounded with the resistant weight of the bricks, which will range from 100 to 130 pounds per cubic foot; nor on the compression side must the stress exceed 8 tons to the square foot, which the brick should be able easily to withstand. That is, if h be the height in feet, d the average breadth, and b the breadth at the base, there must be equilibrium between W, the weight of the chimney in pounds, and the quantity $C\dfrac{dh^2}{b}$. In the latter, the coefficient C is a factor for wind-pressure per square foot of area. It is 56 for a square chimney, 35 for an octagonal, and 28 for a round chimney. A brick chimney so proportioned will withstand any gale likely to be experienced. It will appear, however, that a chimney from these causes and the concentration of weight on a small area is a structure particularly liable to unequal settling of its foundations. The latter, therefore, should receive most careful attention from a competent designer of foundations, and should be laid by experienced persons. Natural and undisturbed soil will carry one ton per square foot; loam, compact sand, or hard-pan can carry two tons per square foot. Where natural foundations cannot be had, piling and other artificial methods are to be resorted to.

With respect to their structure, chimneys may be grouped into

(1) Brick.

(2) Steel or iron shell, brick-lined.

(3) Skeleton iron and brick.

Brick chimneys are round, square, octagon, or star-shaped. Circular section seems best, as lighter, stronger, and more shapely. English rule is, base equals one-tenth of height; the batter or taper in American practice is from one-sixteenth to one-quarter inch to the foot on each side. One in thirty-six is the English standard.

The upper 25 feet is one brick thick (8″ or 9″); thickness increases by one half brick per 25 feet. If the diameter exceeds 60 inches, begin at top with one and a half bricks.

An inner lining or core, detached from the wall proper and running either nearly to top or over 50 or 60 feet up, prevents expansion from cracking the walls. It need not be fire-brick all the way up, or even further than one-half. The core is made of tangent-laid brick, with an occasional header to guide the core by the wall.

Another practice is to make a 100-foot chimney in three sections: first, 20 feet high, 16 inches thick; second, 30 feet high, 12 inches thick; third, 50 feet high, 8 inches thick. Core in three sections of 12, 8, and 4 inches thick, respectively.

The top of a chimney is exposed to weather and frost and snow, melting and freezing. There should be a cast-iron cap, or a stone, to protect the top edge of the brick. Large molded terra-cotta or fire-clay blocks are also used, clamped and doweled together.

Cylindrical steel chimneys of riveted plate steel, secured by a flare in the lower 10 to 25 feet to a cast-iron base-plate, which again is anchored by heavy foundation-bolts to a masonry foundation, require no guy or stay ropes and are 35 to 50 per cent cheaper than a brick stack. They take less room, are strong and safe, and no air leaks in to cool the gas. They are brick-lined part way or all the way up. They must be kept painted.

Stacks when not anchored to foundations by bolts, and all light and unlined stacks, require to be stayed by guys of wire rope. They are attached opposite the center of effort of the winds, at two-thirds of the height; are usually four in number, the first being led in the direction of the most violent wind, and each guy of a cross-section in square inches one-thousandth of the exposed area in feet.

Skeleton chimneys have been put up by iron-works, but have no advantage over steel cylinders, and for many reasons are not as good. Brick is built in between uprights of rolled iron, which are banded by flat rings on the outside.

Access should be permitted to the chimney at its base through a proper door either in the flue or in the foundation of the chimney, and it is best that a ladder on the outside of the chimney should give access to its top. In a square chimney this ladder can be made by bars let into two walls at a corner. Figs. 196 a–c show chimney constructions and the proportions which have been found satisfactory, according to which the thickness may be reduced as the chimney attains height.

127. Artificial or Mechanical or Forced Draft. It has been already pointed out (paragraf 107) that a movement of the air for combustion might be mechanically produced by a proper appliance for this purpose (Fig. 2, Nos. 5, 9, and 49).

A calculation of efficiencies shows that for heights of chimneys such as are ordinarily used the mechanical methods of securing draft are the more efficient, so that it becomes a question of consideration whether the necessary air for combustion shall be furnished by a costly chimney or group of them, or by a continuously running machine of some different type. Artificial draft can be secured by two general methods. The first type is that made familiar in locomotive practice, in which a rapid motion is given to the air to draw it out of the smoke-box so that the reduction of pressure within the latter shall cause a flow through the grates, fire, and tubes to equalize this rarefaction. This is called the induced-draft system, and as applied when fans are used, as in steamship practice, is illustrated in Fig. 197.

The other plan is to cause a pressure of air in the ash-pit below the grate-bars so that the air will flow up through the fire, the setting, and flues by the excess of pressure which prevails in the ash-pit. This is called the forced-draft system, and is becoming more usual in high-speed marine practice. The movement of the air can be produced either by means of a steam-jet inducing a current of air to flow, or fans or blowers, either of the centrifugal or positive type, may be used. If the first or aspirating principle is used, the products of combustion must pass over the aspirating appliance. These gases are hot and possibly corrosive. The heat makes lubrication difficult, and almost excludes the use of apparatus where lubrication must be provided unless all bearing-surfaces can be without the flues which carry the gas. Protection against corrosion can be secured if proper trouble is taken, but where

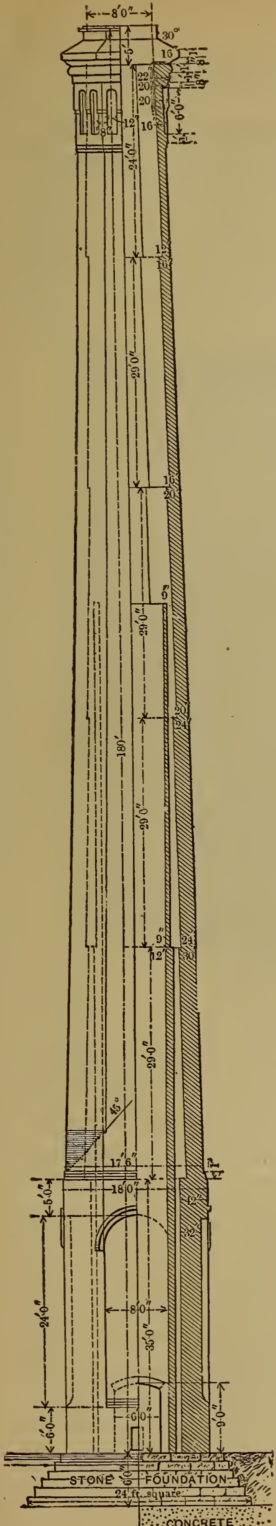


FIG. 196c.

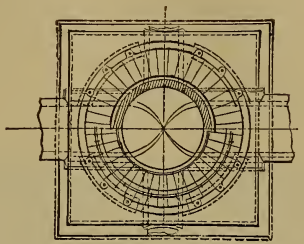


FIG. 196d.

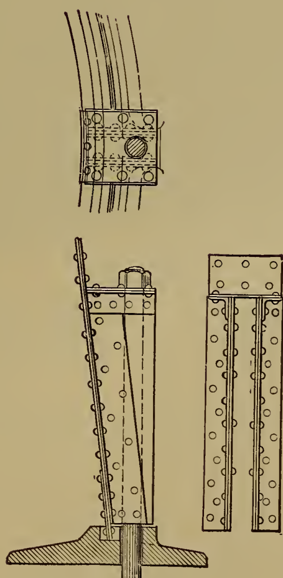


FIG. 196b.

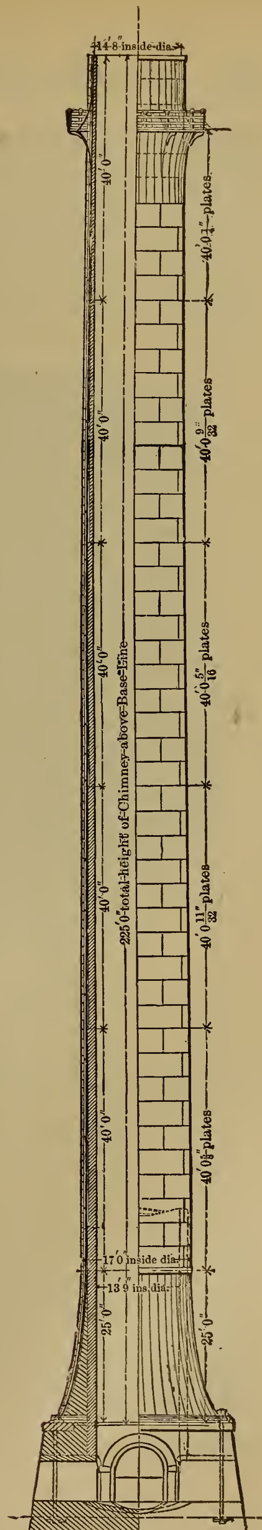


FIG. 196a.

this is not guarded against the apparatus deteriorates rapidly. The forcing system has the fresh cool air pass through the forcing appliance, and has furthermore the advantage of maintaining a higher tension within the setting than prevails outside of it, so that there is little or no tendency for cool air to leak through cracks or porous brick work into the gas-currents. This is a difficulty present where the draft is done by aspiration. On the other hand, the pressure system makes a hot and

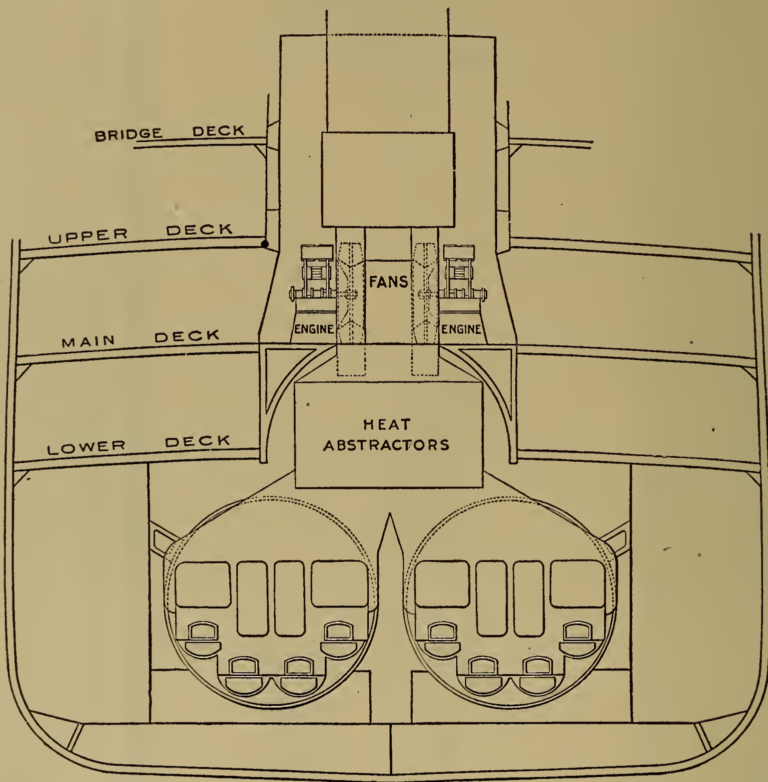


Fig. 197.

gassy fire-room if there are places where gas can escape through cracks, doors, or elsewhere from within the setting into the room. Fig. 198 shows Mr. John C. Kafer's closed ash-pit system, similar to that on the U. S. S. *Swatara* and *Kearsarge*. Since combustion is more efficient the denser the air used to effect it, the pressure system offers an advantage from this point of view, as compared with natural draft or the aspiration system.

128. Advantages of Artificial Draft. It is to be said in favor of natural or chimney draft, that when the chimney is once built and paid for, the draft-machine costs nothing to run except the heat which is used for this purpose, and it undergoes little or no deterioration with use. Furthermore, in cities the necessities imposed upon the power plant to carry the products of combustion high enough up to create no nuisance

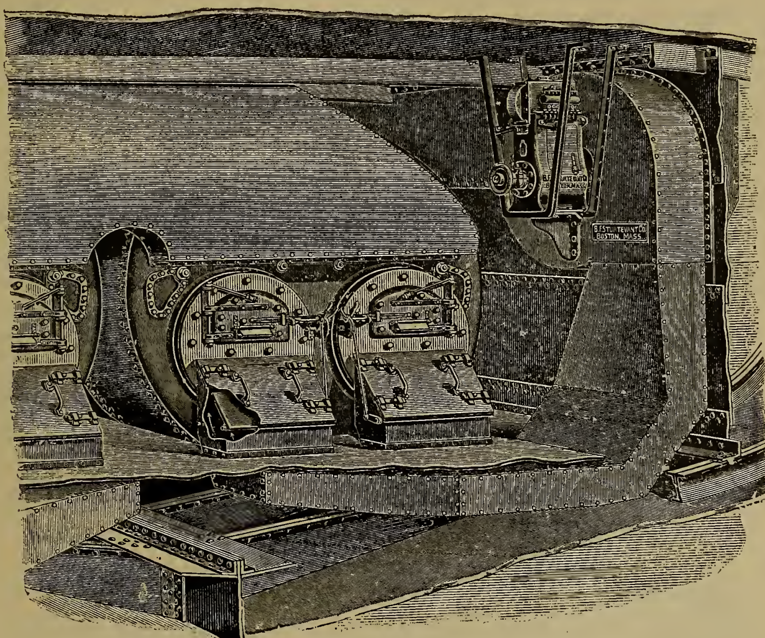


FIG. 198.

in its neighborhood compel a height and cost of chimney which make the consideration of artificial draft unnecessary, since the high chimney must be there in any case. Again, where the plant is so large that the cost of the draft-machine becomes considerable, or, what is the same thing, the cost of the expensive chimney becomes distributed over a large number of horse-power units, the advantages of artificial draft are not so apparent.

Artificial draft, on the other hand, offers the following advantages:

(1) The rapidity of combustion in the fire-box is not limited by atmospheric conditions. With a demand for high steam-pressure and great capacity in a limited space the forced draft is a necessity, as in war-ship practice.

(2) It is possible to increase the evaporative capacity of a given plant

without other change than the velocity of the draft-machine. This increase may be either permanent or to meet sudden demands for steam, such as occur in street-railway practice at busy hours. With natural draft the chimney must be designed to meet the maximum requirement, and will be partly shut off at other times.

(3) It is possible to burn inferior, cheaper, and smaller sizes of fuel with artificial draft, because a high pressure can be maintained which will force the necessary air through a compact body of fuel.

(4) The draft arrangements are more portable than chimneys can be.

(5) The plant is more flexible for changes in quality or size of fuel, and the desirable thickness of fuel-bed on the grates. Grate-bars can be altered more easily if this should be desirable.

(6) Where high stacks are not made necessary the cost which they entail is avoided, or is obviated by a less cost of the draft-machine. The troublesome settling of massive stacks is avoided when foundations are difficult or defective.

(7) Leakage of air into the setting does not occur with forced draft on the pressure system.

129. Disadvantages of Artificial Draft. The objections to be raised against the artificial draft are:

(1) The running cost of the machine. While it takes less coal than the chimney to do a given work, the fuel is not the only expense where an engine must be run, consuming oil and other supplies, calling for repairs and supervision, and the expense of the latter may be considerable.

(2) The artificial-draft machine occupies space which can often be ill spared.

(3) Running machinery, and particularly that at high speed such as most draft appliances demand, is rarely silent, is often noisy, and is liable to breakdowns which compel it to stop.

It will be seen that chimney-draft is not liable to these disadvantages.

The machine for causing the draft may be a centrifugal fan driven either by its own directly coupled engine or by a detached engine, or a revolving shaft, or by means of an electrical motor. The positive blowers will be driven by belts, or their own direct-coupled engine or motor, whether used for pressure or suction methods, and the steam-jet, which is the third appliance, requires no moving machinery when used in either system. It will be seen that each of these offers some advantages and disadvantages of its own. The fan method, if driven by belting, increases the running cost; and if electric current must be generated, the cost of its transformation must be considered. The steam-jet plan occupies very little space and is cheap to buy in the first instance. It is in most cases too noisy. If used as a forcing system, the steam passes through the fire and is objectionable. If used as a suction system, the steam goes out with the products of combustion and does no harm.

The methods which have been used in marine practice to secure the necessary forced draft are either the closed ash-pit system, the closed

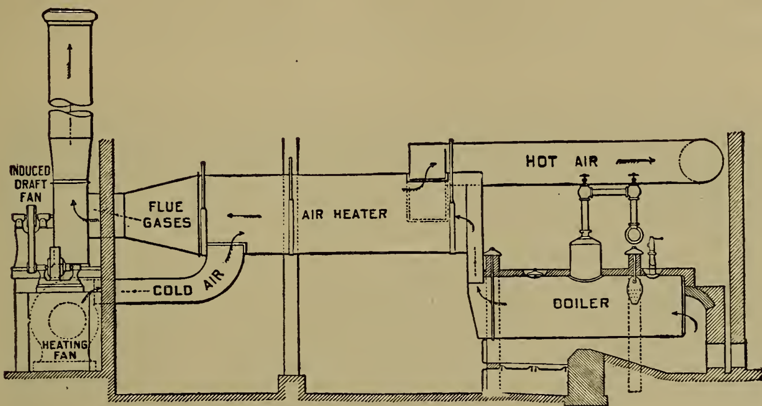


FIG. 199.

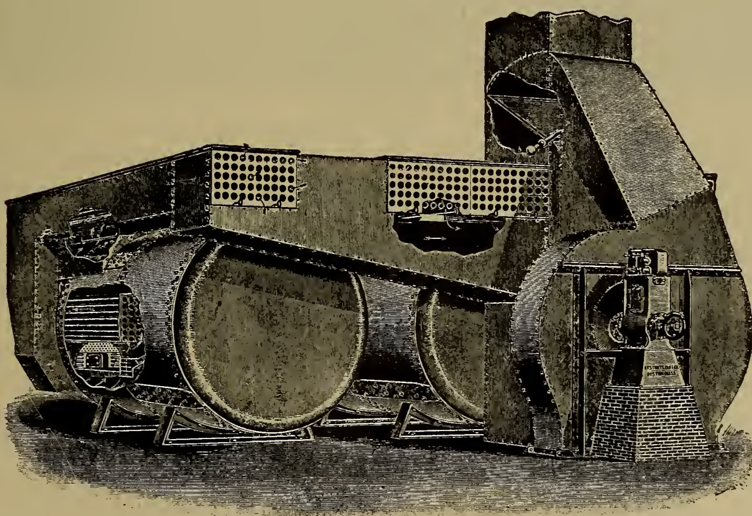


FIG. 200.

fire-room system, or the induced-draft system. The combination of closed ash-pit system with the induced-draft system enables preheating of the air to be easily done before it enters the ash-pit. Figs. 199, 200 and 201 show typical stationary arrangements.

The artificial-draft system is a feature of the automatic stoker shown in Fig. 175 and in some others, and it offers the advantage that the steam-pressure can be made to act upon the draft machinery directly and produce a more prompt and efficient effect upon it than when that pressure acts upon the chimney only and through a damper-regulator (paragraf 125). The fall of pressure in natural draft can only open the chimney wide and attain at best the full effect of the entire chimney.

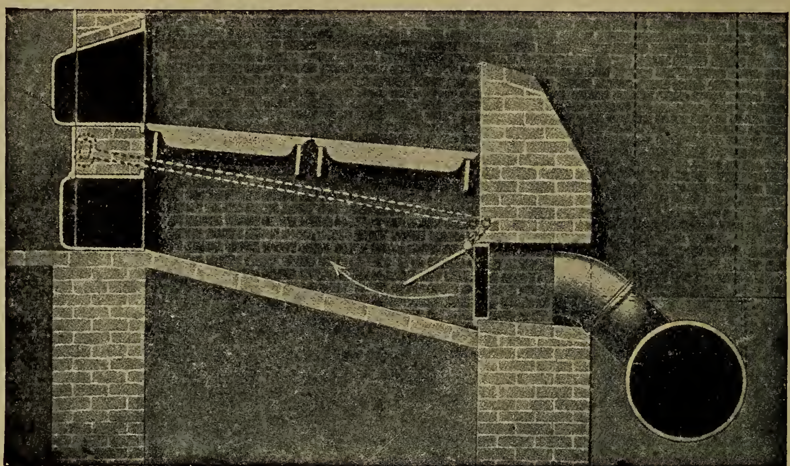


Fig. 201.

By acting on the machinery of artificial draft the fall of pressure can be made to stimulate combustion above the normal rate, and with great promptness.

130. Smoke-prevention. The preceding discussion on the liberation of heat from a fuel for motive-power purposes would not be complete without a reference to the loss of energy which occurs when combustible carbon passes out with the products of combustion, and without having undergone complete oxidation at the desired point. When this carbon goes off as carbon monoxide there is avoidable loss. When incandescent solid carbon fails to meet oxygen under favorable conditions for its union with it, the extinction of the glowing particles forms them into lamp-black or soot, which particles color the products of combustion, and cause them to darken the air and to defile the surfaces which they touch. A smoke, in its exact sense, is a current of products of combustion from a fire, in which the otherwise colorless gases carry finely divided particles of black carbon. This carbon, resulting from incandescence which has ceased, is practically incombustible at ordinary heats. It could have been burned, however, if the union with oxygen had taken place while the carbon was in the nascent or favorable state of its first incandescence, and the effort of the designer and manager of the combustion must be directed to keep up the gases to the temperature of the ignition of the carbon, and with a full supply of oxygen at sufficient temperature to satisfy the carbon. Pure hydrogen combustions are normally smokeless, because of the absence of solid matter in the flame. Such flames are usually non-luminous for the same reason.

The various methods for smoke-prevention have been grouped under the following heads:

(1) The supply of excess of air by steam-jets, inducing current which they warm, and supplying excess of warm air above the fire and behind the bridge-wall. The difficulty with these has been that, after distillation of the gas is completed, after a charge of fresh fuel is thrown on the fire, this excess of air is not needed, and the products of combustion are cooled by the diluting oxygen. Attempts have been made to correct this by graduating the supply of fresh air by chronometric or other appliances, so that the excess should be cut off after such an interval as is usually needed for the first distillation of gas.

(2) By the coking methods of firing. By these plans a large dead-plate was used, so that the gases should be distilled off from the fresh fuel before its combustion was really begun on the grate-surface proper, and when the coking was complete only fixed carbon remained to burn on the grate-surface proper when pushed back. The gas distilled from the fuel on the dead-plate passed over the hot fire, and was so warmed that it was ready to combine and burn. Alternate firing of the two sides of the furnace, or the use of two furnaces delivering into a common combustion-chamber which were fired alternately, belong to this same class (paragraf 117).

(3) The methods belonging to the principles of mechanical stoking are smoke-preventing methods in that each part of the fire always remains in the same condition, and the fresh coal which distils off gas is received in the coolest part of the grate, and passes to the hotter sections only after the volatile matter has been distilled off and burned in passing over those hottest portions.

(4) Gas and oil-firing are smoke-preventing methods, since when properly done the combustion ought to be complete, and no carbon should pass out of the setting except in the form of carbonic acid. It is to this group that those settings belong in which the actual combustion of the fuel containing volatile matter is done in a separate furnace and away from contact with the boiler. This makes a relatively smokeless and efficient apparatus, and will answer with coals which cannot be economically burned in any other way.

(5) The down-draft furnace appears to be one of the most successful appliances for smoke-prevention with smoky coals. As satisfactorily applied it involves the use of two sets of grate-bars, one over the other,

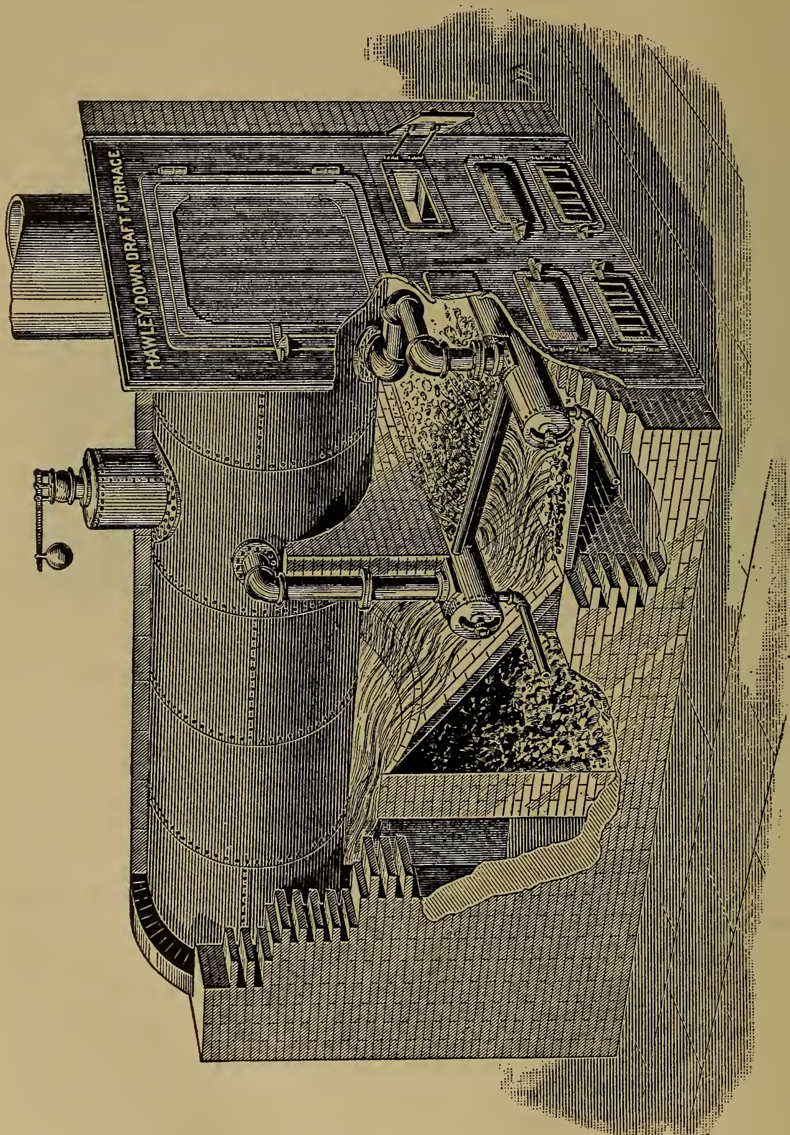


Fig. 202.

so arranged that the draft passes downwards through the upper and lower sets of bars, or else passes downwards through the upper and upwards through the lower. Each set has its own fuel, but the intention is that the gases shall be distilled off from the fresh fuel on the upper

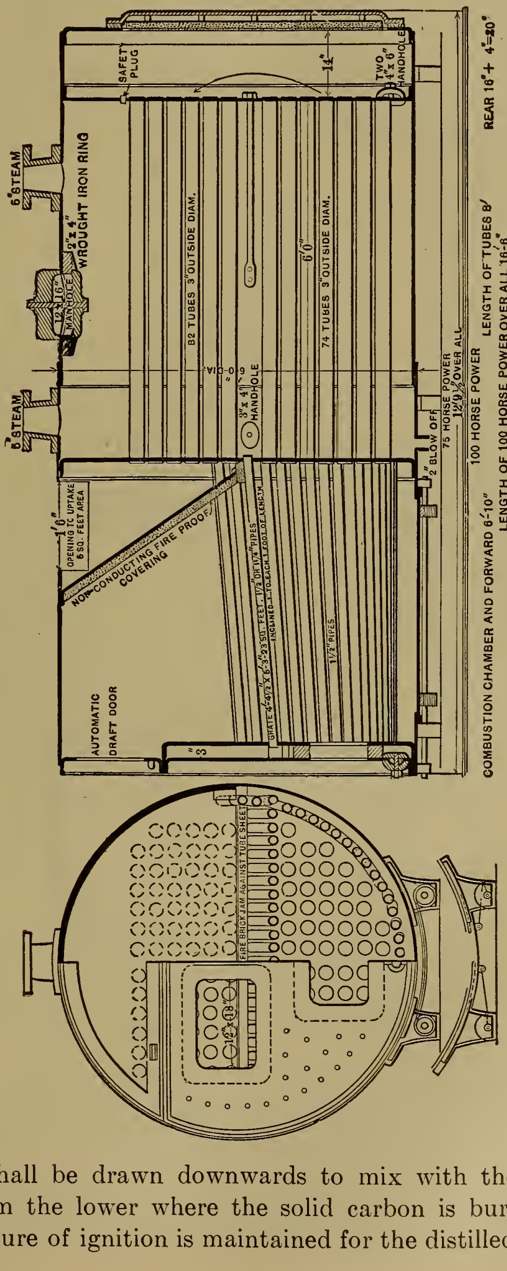


FIG. 203.

grate, and shall be drawn downwards to mix with the hot products escaping from the lower where the solid carbon is burning. By this the temperature of ignition is maintained for the distilled gas, so that it shall burn with the abundant supply of warm air admitted for this purpose. Figs. 202 and 203 show boiler-settings of this type.

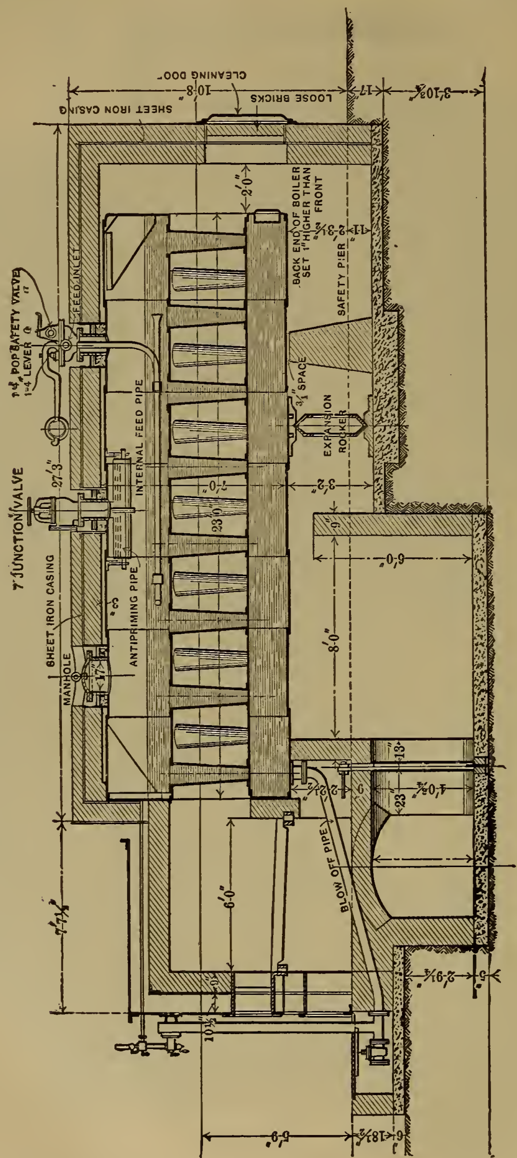


Fig. 204.

(6) The use of fire-brick or similar refractory material for the furnace or in the combustion-chamber (Fig. 204). This becomes hot by the impact of flame and gas, and keeps the temperature of the gas up to

ignition. It imparts some of its heat to the boiler by radiation after it is once brought up to full heat.

(7) Preheating of the air-supply by hollow walls or flue-boxes which the hot gases surround while the fresh air flows within them.

The objections to most of the smoke-prevention devices have been that the introduction of such appliances diminishes either the economy or the capacity of the plant as compared with what it was when the chimneys were allowed to smoke. The excess of air, diluting products of combustion, explains a loss of economy and capacity, and the superior efficiency of the yellow flame heating by radiation, as compared with the colorless flame of perfect combustion, is also responsible in part for this result. The losses seem to be about 12 per cent of power or from 7 to 13 per cent of economy.

The term smoke-consumption or smoke-burning is an improper one. Lamp-black once made is incombustible and cannot be burned. The products of combustion are often colored brown by the presence of tarry or similar combustible matters, and these will ignite if the temperature be made hot enough. It is possible to prevent appearance of smoke by catching it in water through which the products of combustion pass, and in which the carbon is thrown down.

A standard of the degree to which a chimney offends in the matter of smoke has been proposed by Professor Ringelmann of Paris, and has met with general acceptance.* He proposed six grades, of which the zero was a white surface and No. 6 was a dead black one. The intermediate degrees were to be marked by a combination of black lines on a white background such that if held 50 feet distant from the eye the network would become a tone ranging from gray to black in definite stages. The network was on the following scale:

Card 0. All white.

Card 1. Black lines 1 mm. thick, 10 mm. apart, leaving spaces 9 mm. square.

Card 2. Lines 2.3 mm. thick, spaces 7.7 mm. square.

Card 3. Lines 3.7 mm. thick, spaces 6.3 mm. square.

Card 4. Lines 5.5 mm. thick, spaces 4.5 mm. square.

Card 5. All black.

Fig. 205 shows a small area of the four intermediate grades, and in Fig. 206 is a reproduction originated by Professor Breckinridge of what these colors mean when compared to a smoke-plume.†

* Transactions A. S. M. E. 1899, Vol. 21.

† How to burn Illinois Coal without Smoke, L. P. Breckinridge, Univ. of Ill. Bulletin, 1907.

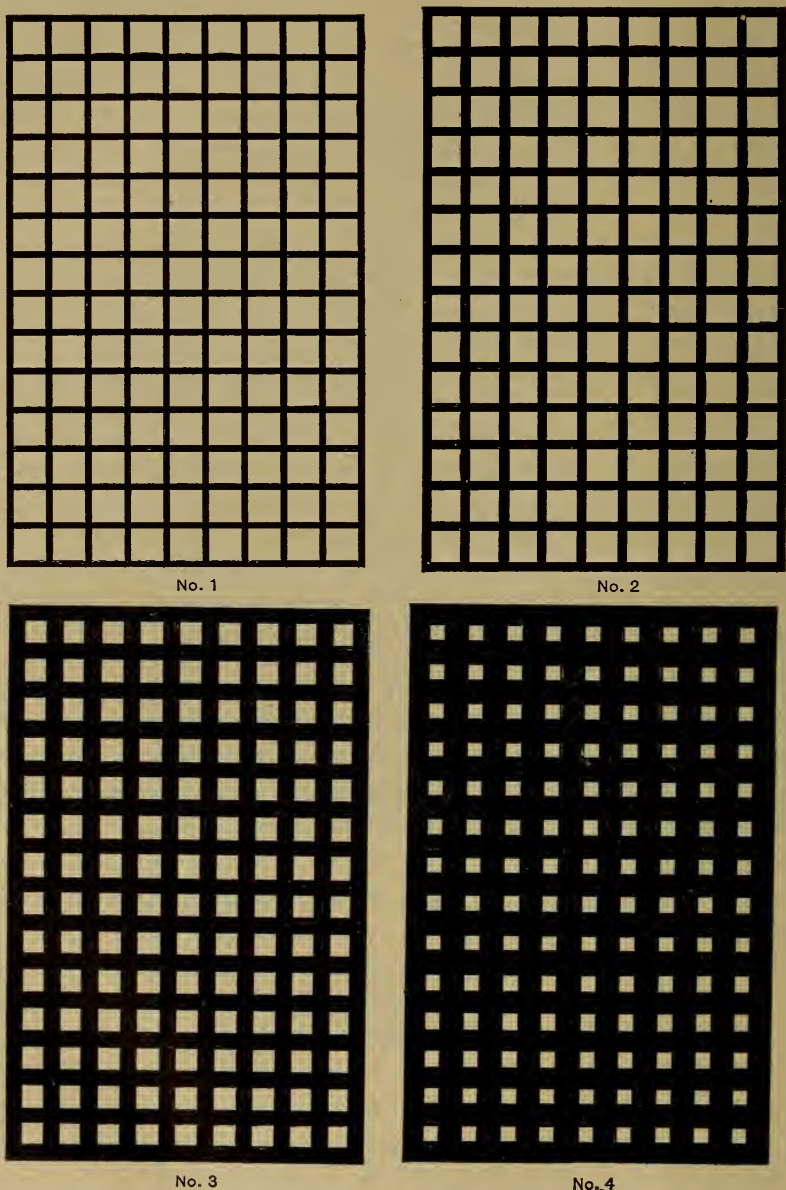


FIG. 205.

131. Boiler Setting. Side Walls. The internally-fired boilers are ready to use as soon as they are located and properly supported (the Cornish and Lancashire excepted). The externally-fired boilers and these two examples of the former class require a structure to be erected which shall support them and shall provide a proper place for the fire and some of the flues or spaces for combustion. This structure is called the boiler-setting. It must be of a refractory material to withstand heat, and of a non-conductor for heat so as to cause least losses by radiation. Its material must be easily manipulated to form flues of proper shape and character, and must be one with which it is cheap to build.

Fig. 206.

These conditions are best met by the use of brick. Those parts exposed to fierce action of heat will be of fire-brick, and the rest of the cheaper common red brick. The fire-brick may be used as an inner lining on the fire-surfaces for the more massive walls, provided proper care be taken in bonding the two grades together. The fire-brick is a little larger than the common brick, which may cause trouble in uniting them. Bridge-walls and the thin parts at the front of fire-boxes will be of fire-brick altogether. Cheapness can be secured by using fire-brick only above the line of grate-bars both in the fire-box and behind it, but it is a question whether this is worth while. The fire-brick lining need not be carried very far into the chimney with anthracite fuels, since the gases should never be above 600° F. after they leave a properly set tubular boiler. With some sectional boilers the gases may be hotter, and with bituminous or long-flame fuels flaming may occur within the chimney, making a fire-brick lining desirable all the way to the top.

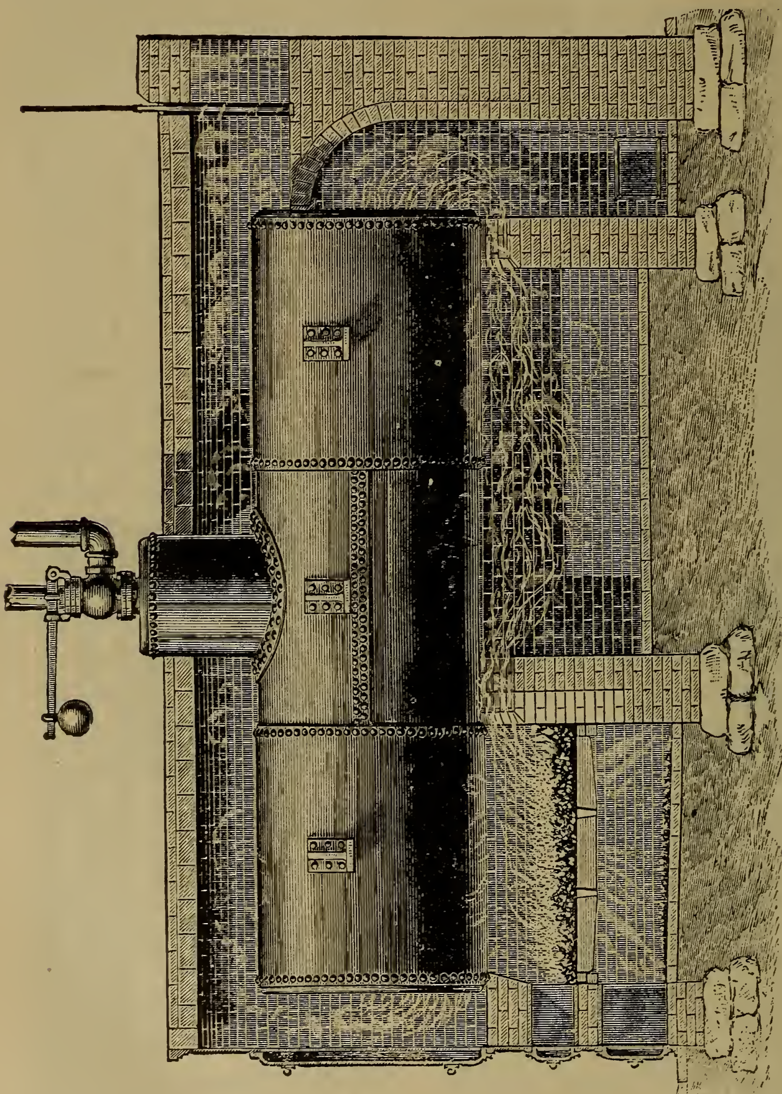


Fig. 207.

The thickness of the walls will depend in part on the methods used to support the boiler. If the setting of brick is to support the boiler and its contents, it must be at least a brick and one-half thick (12 inches), and will be more usually 17 or 21 inches for outside walls. The twenty-one-inch wall is usually made with a two or a four-inch air-space between two eight-inch walls. This makes a non-conducting wall for the sides, and with considerable stability, because at intervals a brick is laid stretcherwise to act as a buttress in the air-space, and at other intervals a header-brick in each wall is laid to project across and touch the other wall without entering it (Fig. 208). The hollow wall has less meaning in walls between the boilers in a battery. Such walls will more usually be solid, and probably twelve inches thick.

If the boiler is supported upon an iron framework independent of the brickwork as in the case of sectional boilers in the main (Figs. 134 and 216), the brickwork of the setting becomes a mere shell to retain the heat and gases, and may become an eight-inch solid wall or a twelve-inch wall with air-space.

Rear walls which form a sort of reverberating surface require to be thick and well laid, because exposed to the deteriorating effect of heat in a marked degree. It is more usual to make these solid or without air-space, and depend for coolness upon the non-conducting quality of brick.

The use of lime-mortar in boiler-settings is not to be commended. The heat tends to calcine the lime, or continually to unset or loosen the mortar bond, and the effect of hydrating the calcined lime is to injure the iron which it may touch. Fire-clay mortar is refractory and harmless, and will be used in any case with the fire-brick work.

132. Buck-stays and Tie-rods. The heat of the fire and its gases causing expansion and deformation of the setting, to which the expansion of the boiler and its supports may add their influence, makes it necessary that the setting should be treated structurally like a heating-furnace, and tied together by means other than the bond of the brickwork. For this purpose tie-rods will be laid lengthwise (Fig. 215) in side walls, and will be used also crosswise between the side walls. The lengthwise tie-rods bear at the front on the outer side of the front castings, and at the rear either on buck-stays or large washers, against which they bear by means of a nut on the threaded ends of the rod. The side-wall ties bear on buck-stays (Fig. 208).

The buck-stay is a vertical bar of cast or wrought iron, of a section adapted to resist transverse bending. They are used in pairs on opposite sides of the setting, drawn together by the tie-rods and binding together the section of the wall against which they bear. Usually there are three pairs in the ordinary length of a boiler-setting. Their section may be a T iron, with the flat of the head against the wall (Fig. 215), or any convenient structural section may be used. Old rails used in pairs will be met quite often. Tie-rods need not be used at the bottom of the setting, provided the feet of the buck-stays be let securely into the footings upon which the walls are built (Fig. 185).

133. Hanging of Boilers. — The weight of a shell or sectional boiler is considerable, and when its contents of water are added, the method of carrying this weight requires to be carefully studied.

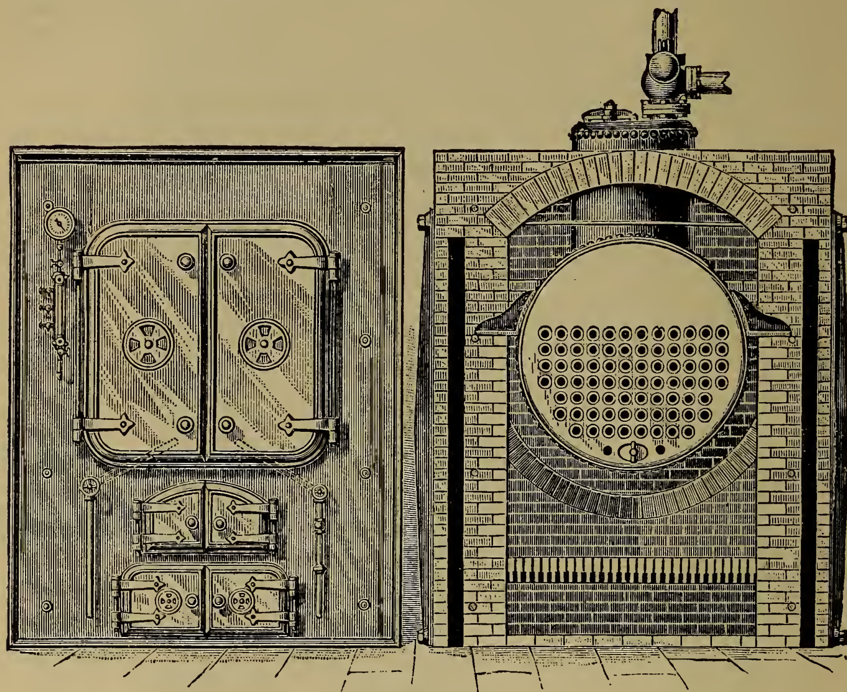


FIG. 208.

Two methods are usual. The boiler is supported at its sides at about the horizontal diameter, or it is hung from above by eyes and links attached to its upper part, and symmetrical to a vertical diameter.

The first plan calls for projecting brackets or "lugs" to be riveted to the shell along its sides, and giving strength sufficient to carry the weight. These lugs will be of cast iron or steel castings, either solid or with the projecting part fitted to slide home in a socket made for it and fastened to the shell. The number of these lugs will be fixed by the length of the boiler. Two on a side is best if possible, since then every lug carries not far from one-fourth of the load, no matter how the boiler may be deformed by expansion. If there are three or more lugs on a side, then when the lower elements of the boiler lengthen, the end lugs lift, and most of the weight is on the central pair; if the lower elements shorten relatively, then the boiler lifts off the central pair and is carried at the ends. These changes in length come from impact of cold feed-water, or of cold air when the fire-doors are opened wide and suddenly. Figs. 207, 85, 215, and others show such lugs and their forms.

Recent practice has favored the use of steel plate forgings for the lugs. Fig. 209 shows the type designed to provide for the rollers, on which the bracket is to move in expanding.

The other method of support calls for an eye on the top of the boiler (Fig. 134), or on the two sides (Figs. 70 and 212), into which a hooked link may be fitted, so as to hang the boiler to a pair of cross-beams

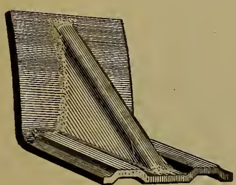


Fig. 209.

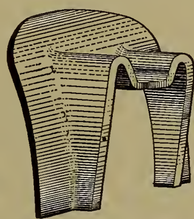


Fig. 210.

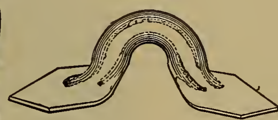


Fig. 211.

of structural material, the latter carried either upon the side walls as abutments, or by metal columns independent of the side walls (Fig. 216). Sectional boilers may be hung from the top, from the sides, or from the bottom, as may be most convenient and preferred. Fig. 210 shows a bracket designed to take the rod and nut of the suspending link from the overhead frame or gallows, and Fig. 211 a suspension eye forged up from flat steel.

The method of support by lugs usually depends upon the side walls to carry the weight. To prevent injury to the walls, the wall is fitted with plates under the lugs to distribute the load, and to furnish a surface for the motion of rollers of one-inch round iron inserted between the plate and the lug-surfaces at one end so as to allow a free end to move lengthwise in expanding and contracting without pushing the wall or deforming the shell (Fig. 179). The rollers may be omitted in light boilers, and the boiler allowed to slide on the plate. This pushes the wall about, however. The end to be fixed is determined by the convenience of the attachments to the boiler. The locomotive boiler is fixed at the front to the cylinders and frames, and is free to move at the fire-box end; most stationary boilers are fixed at the furnace end and expand toward the rear.

Expansion in suspended boilers is provided for by the suspending link, and by this their expansion is independent of the brickwork.

This is the great advantage of this method of hanging. The objection to it with large-diameter shell boilers is that the weight tends to make the flexible shell take an oval shape when pressure is low, while the internal pressure restores the cylindrical shape when it rises again. The flexure of the longitudinal joints caused by these changes of shape causes grooving near the joints, to be discussed in a later paragraf (196). With shells of small diameter this trouble is scarcely felt.

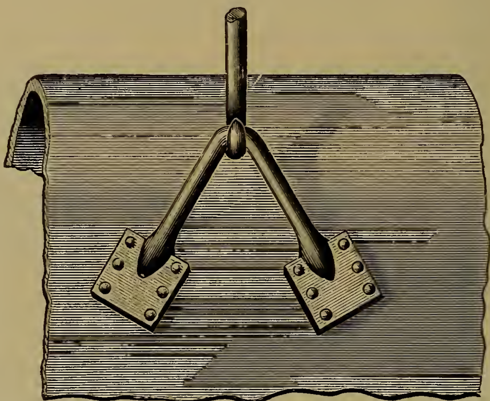


Fig. 212.

With very long cylindrical boilers the necessity for many points of support conflicts with the considerable changes of shape by heat in such long lengths. This has given rise to the methods of hanging by means of equalizing-levers over the transverse supporting beams (see Fig. 134), whereby each eye carries its proportion of load in every condition of shape; or the same result is approximated by using stiff spiral springs (like car-springs) under the nut on the suspending links which hook into the eyes. When the boiler curves itself lengthwise, the spring accommodates the excess of relief of load, without causing so much strain on the plates of the boiler itself. A skilful designer, compelled to use long boilers, has cut them into lengths and linked them togther by flexible connections of copper tube (Fig. 16), in order to meet this serious trouble.

134. Boiler-fronts. It would be possible to make the front part of a boiler-setting of brick, but it is not usual to do so, because the openings through it for access to the furnace and ash-pit, and to the boiler itself, would make troublesome and short-lived constructions in brick, and would make the door-fittings difficult. Hence the use of cast-iron fronts for settings is universal, for their convenience and cheapness, for ease of fitting, and for the effect to the eye which as completed structures the boiler-settings can be made to produce. They will be made in one or two sections set up edgewise, and held in place by the nuts on the ends of the longitudinal tie-rods through the walls (Fig. 215).

Boiler-fronts are either full fronts or half fronts. The full fronts are sometimes called flush fronts, and the half fronts are also called extension or overhanging fronts. The full or flush front will be used always

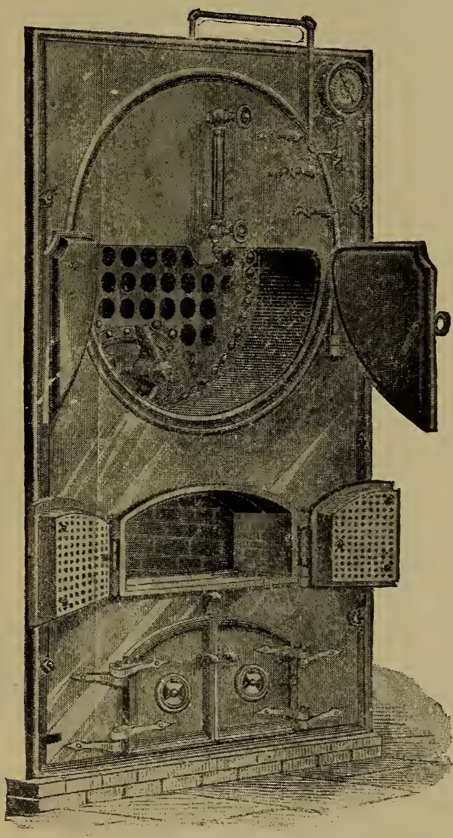


Fig. 213.

with sectional boilers, usually with tubular boilers, in which the products of combustion are to be carried backwards over the top of the shell to a chimney behind the boiler, and quite often where the gases are to be taken from the front end of the boiler to a chimney by means of a sheet-iron duct. When the full front is used the side walls will be carried up level with the top of the front (Fig. 207), and the joint between the side walls over the boiler will in this case be made either arching, or by means of filling in on top of the boiler with some non-conducting material.

The half fronts will be used where the side walls are to be carried up to the height of the supporting lugs only, and the smoke-box is to be made an integral part of the boiler itself (Figs. 215 and 185). A cylinder

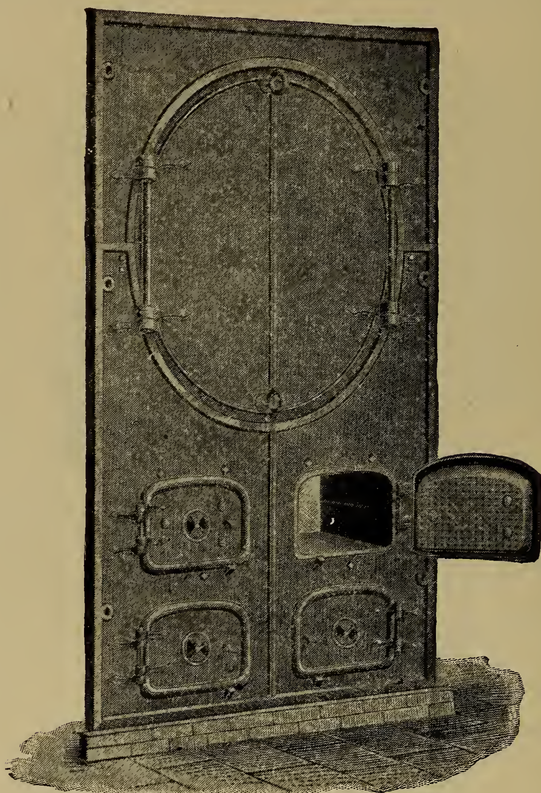


Fig. 214.

of a relatively light boiler-plate is secured to the shell of the boiler itself and, projecting beyond the plane of the front, forms a smoke-box independently of it. This arrangement is shown in Fig. 215. It implies of necessity that the gases are to be taken off from the smoke-box either directly to a chimney-stack or by means of a sheet-metal flue or breeching.

There will be three sets of openings to be made in the boiler-front, each of which must be closed by proper doors. In the full or flush

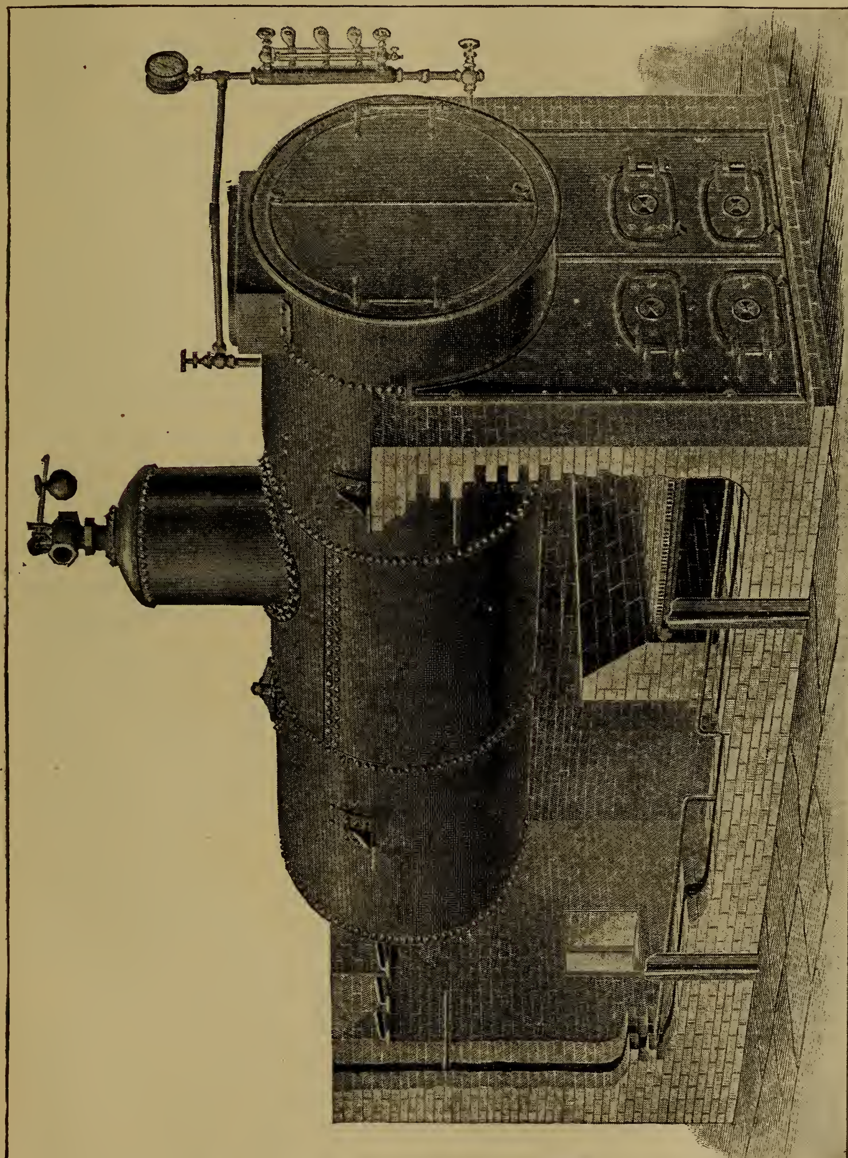


FIG. 215.

front these three openings are all made in the front proper. In the half front the two lower ones will be formed in the front, and the upper will be a part of the structure of the extension smoke-box. The top of the front is then curved to match the curvature of the shell or smoke-box which protrudes beyond it (Fig. 182). The lowest opening gives access to the space below the grates, which is called the ash-pit. The doors which close it are called the ash-pit doors. These doors are a means of controlling the draft of air which passes up through the fire, and will be closed when the fire is to be checked. From examination of Figs. 208 to 215 it will appear that such doors may be either a single large door, two smaller doors closing a large opening, or two independent openings, each with its own door. The advantage of small doors is the diminished strain on the hinges when such doors are opened, and the fact that such short doors are less in the way than long ones. It is not unusual to make register-openings in these doors, but they have comparatively small significance, since it is much easier to leave the door slightly open if but a little air is desired.

The second set of doors open into the furnace or fire-box on a level above the grates, and each will be called a fire-door. Its function is to give access to the fire for charging it with fuel and for cleaning, and it also has a use in the control of the fire, since by leaving it open cold air from the fire-room enters above the fire, lowering the temperature of the hot gases by dilution and actually serving to cool the boiler, while at the same time the easy passage of air over the fire checks the draft through it. The same considerations as to the use of one large or two smaller doors are to be noted with respect to the fire-door, but the latter requires that it should give access for coaling and cleaning to every part of the grates, and consequently with a wide furnace two doors become a necessity. They have the further advantage that in coaling and cleaning they need not both be open at once. The single door is better than two doors which overlap, when it is possible to use it, because it closes the fire-opening somewhat more tightly. The minimum width of a fire-door should permit the easy handling of an average coal-scoop, which measures for small coal 14 inches across.

The fire-door furthermore requires a special construction to prevent its becoming unduly hot by radiating heat from the fire. This is done by forming an air-space between the outer surface, which is the door proper, and the inner plate of perforated iron which is fastened to the door with distance-pieces to keep them at a fixed distance apart (Figs. 213 and 214). The inner or baffle plate receives the heat of the fire, and the circulating air between the baffle plate and the door serves to carry off some of the heat. The fire-door is often also made with register-openings to permit a certain amount of air to enter this air-space and so reach the fire above the grates. It is difficult to provide sufficient area to make these openings serviceable to supply oxygen for combustion, but the old rule used to be that such openings should be 2 square inches for each square foot of grate-surface with short-flame fuels, and 5 square inches where the fuel contained much volatile matter. The real use of air above the fire can be best obtained by leaving the door slightly open when it is required.

The third set of doors will be called in shell boilers the flue-doors, and are intended to give access to the front of the boiler for cleansing the flues or tubes and for inspection. In full fronts and with boilers of large width it is desirable to make these doors double in order to keep their weight down. They will then be arranged to open on vertical hinges, and will be held shut by a common latch. In extension fronts it becomes more convenient to make the opening to the smoke-box for inspection by a door turning upon a horizontal hinge at about the horizontal diameter.

135. Concluding Comment. Gallows-frame Supports. The typical fire-tubular boiler burning solid fuel and with return of the hot-gases has been the form chosen in the foregoing paragraphs. The sectional type is always suspended from transverse girders by tension rods, and with the increasing use of concrete foundations this so-called gallows-frame support is coming more to the front. With concrete footings going down some distance into the ground, the Francis rule for area of adequate support is:

With hardpan, allow load per sq. ft. of 8 tons;
With gravel, allow load per sq. ft. of 5 tons;
With clean sand, allow load per sq. ft. of 4 tons;
With dry clay, allow load per sq. ft. of 3 tons;
With wet clay, allow load per sq. ft. of 2 tons;
With loam, allow load per sq. ft. of 1 ton;

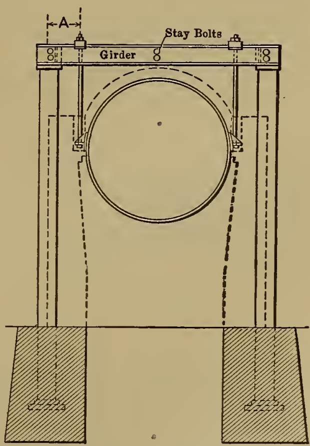


Fig. 216.

which are larger values than are permissible for chimneys (paragraf 125). The deep foundation in concrete enables the columns for the supporting girders to be cheaply made of steel pipe flanged top and bottom (Fig. 216) and buck-stays can be eliminated since the boiler does not tend to throw down the brick walls.

The costly character of the chimneys and boiler settings required for the external combustion system of utilizing fuel energy (paragraf 2) are among the arguments used in favor of the internal combustion system as applied in the gas or oil engine which has no boiler nor furnace. For further amplification of this the student is referred to other treatises.* It must not be overlooked, however, that in plants of any size using gas in the engine-cylinder made from a solid fuel, the cost of such gas-generator or producer and its process is an offset to the boiler-setting and chimney of the steam plant. The advantage is the convenient transmission of energy in the form of gas through pipes to the point where power is to be developed. The use of liquid fuel or gas under a boiler in a setting as a means of getting fuel energy is of considerable importance and will be treated in the next chapter.

* The Gas Engine: — A Treatise on the Internal Combustion Engine using gas oil or other liquid hydro-carbon. By F. R. Hutton. John Wiley & Sons, New York, p. 163. Edition of 1908.

CHAPTER IX.

FIRING BOILERS WITH GAS OR LIQUID HYDRO-CARBON OR WITH PULVERIZED FUEL.

136. Introductory. The introduction of gas or liquid fuel under the heating surface of a boiler by mechanical pressure does away with the hard labor of the fire-tender, and derives all the other advantages of mechanical stoking (paragrafs 114 and 223) and introduces some advantages of its own. In the cities where natural gas has been introduced through mains and pipes to industrial establishments, the gas does not have to be manufactured, but interest on the installation of the pipes and gas-compressing plant must be paid for. Illuminating gas is too costly to be economically used for firing: fuel gas with less lighting power is the only one to be considered.

With gas under pressure its velocity of flow can be used to draw into the burners the air necessary for combustion. The argand principle of tubular or hollow cylinders of gas with air both outside the sheet of flame and within it will be the basis of such burner design (Fig. 217). For liquid fuel installations the oil must be broken up by atomization into a spray or mist, so that air shall meet every particle of fuel for quick and complete combustion. Such atomization can be effected either by sending a jet of air or of steam through a film or thread of the liquid, or by breaking up the liquid mechanically by pressure into small threads or films. In any case an auxiliary is required to produce the pressure on the jet or burner before it can be started under the main apparatus. The same sort of burner can be used for either gas or oil, and in the discussion of advantages and difficulties, what is true of oil is also true of gas.

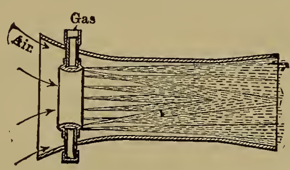


FIG. 217.

137. Gas-firing of Boilers. The ordinary process of combustion means a gas-making in the fire-box and a subsequent combustion of the gas. But if the gas is made outside and led to the furnace, it is cooled in that process, and the heat of distillation and warming up the gas is lost, so far as the boiler is concerned. The cost of such boiler firing is therefore greater than the firing cost from solid fuel. The advantages are those of convenience, which are listed in the next paragrafs, rather than of economy, except where the labor economy is great. There are some liquid systems where the oil is first made into a gas before reaching the burner, either by passage through a heated coil of pipe (Archer) or other similar method. What is known as the Dutch Oven furnace is again a gas-firing system (Figs. 139, 140 and 204) since the philosophy of this system is the making of gas in the exterior furnace or fire-chamber and having it at so high a temperature by reason of the fire-brick arch at white heat over the fire that it shall be sure to ignite if oxygen enough at temperature enough can get to it. What is here lost is again somewhat of the radiant heat of the hot solid fuel upon the heating surfaces, but in gaseous or flaming fuels this may be less than the gain from better combustion of the flames and gases. It is better than the remote or producer manufacture of the gas, and its bringing to the boiler in a comparatively cool state, since here all the radiant heat of the fire is lost, and some of that of the gas. If the gas is to be made at a distance and piped to the boiler, it is probably more economical to utilize the gas in internal combustion engines (paragraf 2) and bring the power by electrical transmission to the working point rather than to use the gas for making steam. If the heat of the combustion is desired for heating, then the gas can be led in pipes to stoves where needed, and burned in such heaters directly rather than to burn the gas to make steam and carry the steam with attendant losses in condensation to the place to be heated.

138. Liquid Fuel Burners. There are two systems for the combustion of oil: the liquid systems or the atomized vapor systems. The former is of no importance by reason of the advantages which attach to the latter. Fig. 218 shows a type approved in the U. S. Navy experiments in which the oil current is induced by steam, and Fig. 219 a form in which air is used. In Fig. 220 the moving steam jet carrying the oil draws in the necessary air by aspiration in an annular film forming a combustible mixture before it leaves the nozzle. In Fig. 221 is a type of the mechanical atomizer where the oil under pressure is forced through the spiral channels and is thus finely subdivided when it leaves the jet. This depends on meeting its air within the furnace. Fig. 222 illustrates an oil-injector burner for locomotive use. By retracting the adjusting hollow steam jet, the nozzle can be swept out clean by shutting off the oil and blowing it through. The thimble at the nozzle passes the jet through the rear water-leg of the boiler.

With respect to the use of air or steam for the inducing means to draw up and atomize the oil, it may be said on behalf of steam that it requires no air-compressing plant to bring it up to pressure for use under boilers, and there is not introduced into the flame a mass of

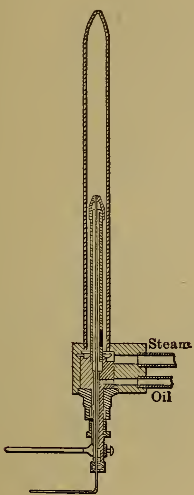


Fig. 218.

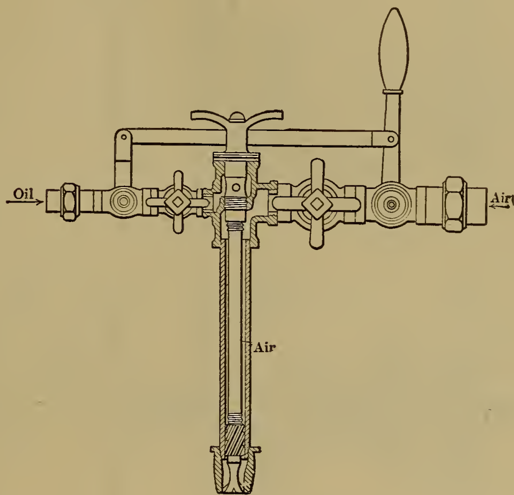


Fig. 219.

inert nitrogen which must be heated at the expense of the oil-fuel, and acts to cool it. Steam is hot, furthermore, when it enters the flame,

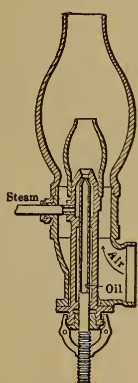


Fig. 220.

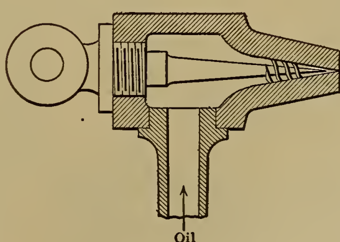


Fig. 221.

and may be superheated. On the other hand, air must be introduced for combustion, and it is best to introduce it as the spraying and subdividing medium; steam dilutes the burning gas if it is not dissociated; and if it is, the heat of dissociation is lost unless the temperature is high enough for recombination.

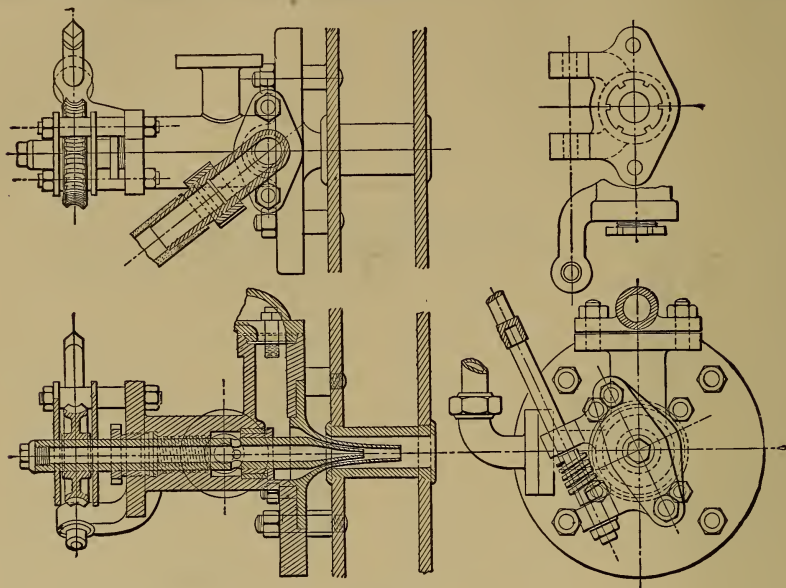


Fig. 222.

139. Liquid Fuel Furnaces. The jet of burning oil and steam or air must not be allowed to impinge directly upon a heating surface of metal. Fig. 223 shows the usual furnace arrangement where the impact and scattering of the flow is against fire-brick set on edge. The grate is covered with asbestos slabs luted with fire-clay except where the air comes in. The bricks become white hot and serve to light the spray if it should chance to blow out. Or the grates may be covered with broken fire-brick and asbestos, through which air can pass, but which become hot enough to secure ignition. Great danger of explosion and fire is present if any mass of liquid oil or its resultant gas can accumulate upon the grate or in the setting. It must be assuredly ignited as fast as it arrives. A fire of coke is often used upon the grate to start the first ignition

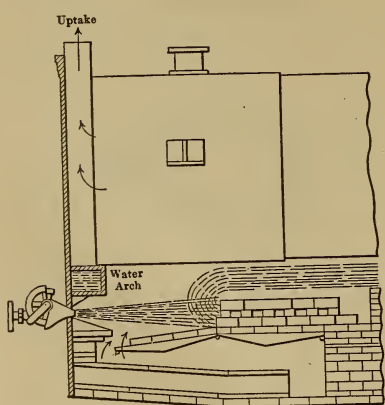


Fig. 223.

and to keep up any ignition required subsequently. Such coke fire will burn very slowly upon a closed grate.

140. Precautions in Oil-firing. Besides the danger with liquid fuel in external combustion just referred to, there are also the objections urged in many places against the storage of any large quantity of liquid hydrocarbon in one place. Some cities limit the quantity which is allowed: others compel its storage below ground: or if above ground, then with a well or depression below it sufficient to receive and confine all the contents of the tank above. Crude oil or some of the volatile hydrocarbons make a combustible mixture with the air which filters into or enters the storage tank, so that water or street gas makes a safer displacing agent than air. The supply of fuel by gravity from any overhead reservoir of large capacity is objected to because the force supplying the oil cannot be shut off in case of accident which puts the shut-off valve out of service: and yet if steam or air is used in atomizing, it must never fall so low in pressure as to fail to atomize and thus flood the fire-box with raw liquid.

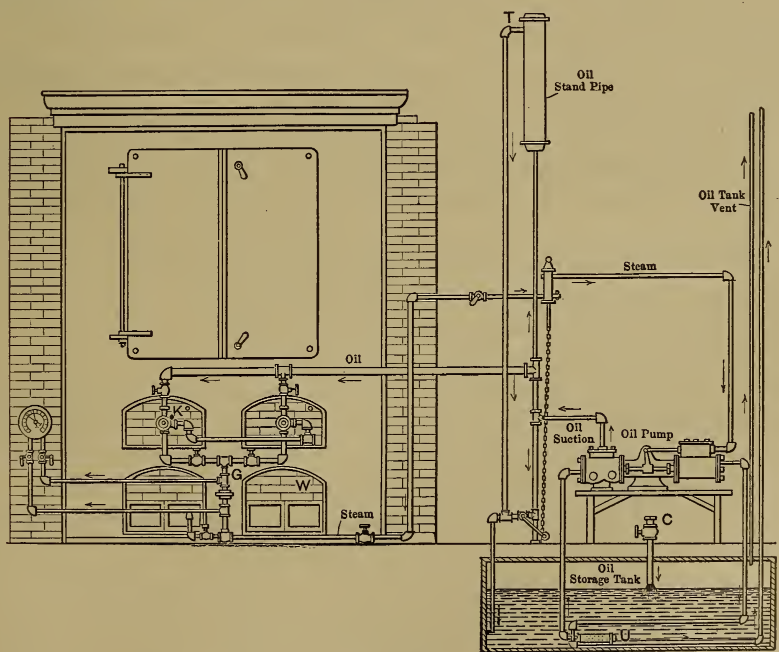


Fig. 224.

The arrangement of Fig. 224 shows a plant with the underground tank for storage and an oil standpipe to secure a constant head even with variable pump-action. The pump is steam driven at a speed to supply a little more oil than the burners can use, so that the stand pipe is always full and overflowing with a gentle pressure on the oil to keep any air from getting in the circuit. The oil is kept fluid in the storage tank by carrying the hot exhaust pipe into and through it. An air vent carries all vapor from the tank to a safe point outdoors and prevents any pressure or vacuum from establishing itself in the tank. *C* is the filling nozzle. To empty the standpipe back to the tank when atomizing steam is shut off or the pressure of steam falls below the atomizing point a weighted lever on the bottom of the standpipe leg is held up by a piston on whose lower side is the steam pressure. If the pressure on the piston falls below the safe and determined point the weight falls, opening the valve into the overflow back to the tank and emptying the standpipe down to a level below the burners, so that the pump delivers back into the suction tank instead of to the burners.

141. Advantages of Oil Fuel. Oil or liquid fuel offers many attractive advantages over the solid fuels. Many of these are those incidental to mechanical firing, to which oil lends itself easily, but besides these there are many others of its own.

Mechanical handling of oil by pumps or aspirating burners gives the following advantages:

1. Economy of labor. One fireman by handling the necessary valves can manage eight to ten or more boilers of 100 horse-power each. With hand-firing of coal, one man can never manage more than four such boilers.

2. No ashes, and their attendant labor and possible cost.

Economy and convenience in oil-firing result from:

3. No waste of fuel in ashes and cleaning of fires.

4. No waste of fuel in banking fires overnight.

5. No opening of furnace-doors for firing or cleaning. This is easier upon the brick-work of the setting, and on the metal of the boiler, by diminishing strains of sudden contraction.

6. No injury from firing-tools in fire-boxes.

7. No sparks pass out from a chimney, to set fire to combustibles outside.

8. Absence of dust and ashes in fire-room and adjoining engine-room.

9. Wide range of controllability of fire, not only within the limits of ordinary consumption, but beyond these. The fire is put out when demand for heat stops; an excessive demand for heat can be met by unusually great supply of oil. With solid fuel, a charge once made must burn itself out. In boilers, safety-valve waste is diminished.

10. The greater calorific power of oil, and its controlled combustion,

enable more energy to be gotten from a plant whose capacity has been calculated upon a solid fuel basis.

11. Smokeless combustion is more easily secured, and there is diminished loss of unburned carbon.

12. Lower temperatures of fire-rooms, and lessened physical strain upon firemen.

13. Absence of sulphur to corrode metal.

14. Fires easily started.

15. Economy of stowage and carriage of oil as compared with solid fuel.

16. Economy of fuel-stations for navy or locomotive practice.

17. No grates are required.

If the heating power of oil be placed at 21,000 B.T.U. and that of the coal used for comparison be called 13,000 B.T.U., a rule of three proportion will give 1 pound of oil as equivalent to $1\frac{3}{4}$ pounds of coal; or 1 gallon of oil equals 6.5 to 6.7 pounds of oil and will compare to 12 pounds of coal; or 190 gallons of oil will equal a long ton of coal of 2240 pounds.

142. Disadvantages of Oil Fuel. There are objections to oil as a dependence for a source of heat.

1. The use of crude oil with the volatile constituents in it is opposed by the health ordinances of some cities. In others the fire or insurance ordinances permit the use of oil only if the oil-tank is below ground, or so placed that it cannot flow out of its reservoir and carry flame to other buildings in case of conflagration.

2. The vapor from crude oil is ill-smelling and makes an explosive mixture with air. It vaporizes even at low temperatures.

3. If fuel-oil must be used, it is usually more costly than coal in most places. The problem is really to get the most heat-units for a unit of value. If the quotient of the calorific power of oil per pound divided by its price per pound at any point is greater than the same quotient for solid fuel, the oil is the cheaper.*

4. The total oil-production of the world would supply but a small portion of the demand for heat as a source of energy. This would immediately affect the price of oil, if any large number of consumers were to decide to use oil.

* If the number of pounds of oil per barrel be called B and be computed by multiplying the weight of oil per gallon by the number of gallons in a barrel, and if B be multiplied by the calorific ratio of oil to coal as above, and the product be called A, then $\frac{2000 \text{ or } 2240}{A} \times$ price of oil per barrel = equivalent price of coal per ton, and $\frac{A}{2000 \text{ or } 2240} \times$ price of coal per ton = equivalent price of oil per barrel.

5. Most of the spray burners make an objectionable roaring noise.

6. The surfaces exposed to an oil-flame usually become coated with a deposit of residue from the burning oil.

7. Oil creeps past valves and leaks in a way which is annoying and may be dangerous.

8. Explosions occur from the flame blowing out and igniting again with dangerous combinations of oil-vapor and air.

9. Auxiliary apparatus in the way of a source of steam or compressed air is required for the burners; in starting, there must be a supply available of air or steam from a boiler or reservoir.

The discovery of extensive oil-fields in the southwestern part of the United States has given fresh impetus to the use of oil in locomotives, with the special advantages of easy mechanical firing and sparkless combustion. Fig. 119 is of a boiler for this service.

Both liquid and gaseous fuels, when the mixture of fuel and air is intimate, require no excess of air to dilute the products of combustion, and hence to cool the fire temperature (paragraf 110).

143. Pulverized Fuel Systems. Designers have sought to secure for solid fuel or where liquid hydrocarbon was not available, the same advantages of rapid combination with oxygen and smokeless combustion by finely dividing the solid fuel and feeding it to the air-jet instead of the atomized oil. These are called pulverized fuel systems. The coal is ground between rollers to a fine or almost impalpable dust, and is led from a hopper through a feeding tube and forced into the furnace by the air which opens out and separates the particles so that air surrounds every one. If the temperature is high enough the combustion will be complete, exactly as in the flame from a fuel-bed in which the solid carbon particles have been separated by heat from the mass upon the grate. The condition for success is that the fuel shall be so fine that each particle shall be kept afloat in the gas current, without risk of its settling to the bottom before it has been completely burned; or that the velocity of the current of flame and gas shall not be so high that a particle of that mass shall not be completely burned before combustion must cease in tubes or around them when the latter are cooled by water. It is not convenient to pulverize and distribute to several boilers, for the material clogs and jams: it is best to pulverize at each boiler separately. The plant and labor and operation cost are serious offsets to the advantages.

144. Sundry Special Fuels. In a parallel group to the pulverized coal systems are those in which sawdust is to be used. This is in effect pulverized wood. Hollow grate bars (Fig. 173) are practically a necessity here and forced draft (Fig. 225), to keep the fire from clogging.

When the fuel in addition is wet or is not easily ignited as in the case of spent tan-bark or the woody refuse from ground sugar-cane (bagasse),

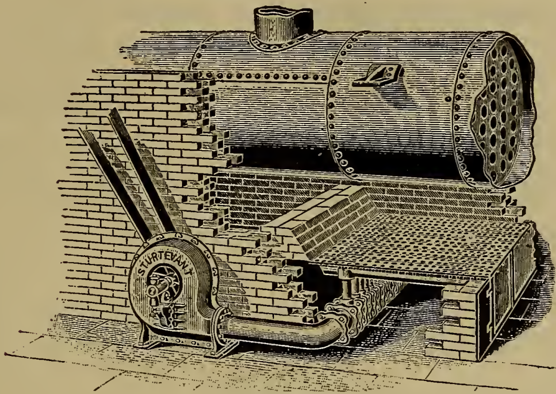


Fig. 225.

the further demand is made that the fuel be completely surrounded by heated surfaces and that distillation may proceed before ignition occurs. Hopper-feeding is an advantage for such material, giving it a slow,

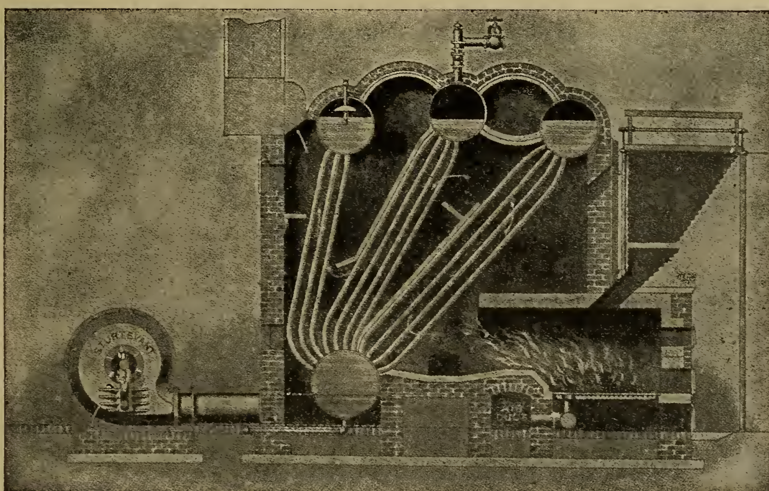


Fig. 226.

regular, gradual feeding to the furnace area. Fig. 226 may stand for a type of such furnace with a sectional boiler, and the Myers furnace has been very successful in economy where it has been convenient to use vertical boilers. The fuel is fed into annular combustion chambers surrounding the central boiler at its base; the distillation occurs successively in the various sectors or furnaces, and the fuel is always kept at a high temperature until completely consumed.

145. Conclusion. Heat Balance. In concluding this final chapter of that section of this treatise which has to do with the liberation of heat energy from the fuel and its transfer to the water in the pressure generator, it may be serviceable to call attention to a convenient form of presenting the facts of the upper lines of Fig. 2 of a power-plant analysis. It is to present the items Nos. 2, 3, 4, 5, 18 and 19 as charge items in an accounting with the boiler, and to credit the boiler with its output in heat units both productive and non-productive. Such a statement of account has been called a "heat balance" and its general form would be:

HEAT BALANCE, OR DISTRIBUTION OF THE HEATING VALUE OF THE COMBUSTIBLE.

Total heat value of 1 pound of combustible......................B.T.U.

	B.T.U.	Per Cent.
1. Heat absorbed by the boiler = evaporation from and at 212 degrees per pound of combustible × 965.7.		
2. Loss due to moisture in coal = per cent of moisture referred to combustible ÷ 100 × $[(212 - t) + 966 + 0.48\ (T - 212)]$ (t = temperature of air in the boiler-room, T = that of the flue gases).		
3. Loss due to moisture formed by the burning of hydrogen = per cent of hydrogen to combustible ÷ 100 × 9 × $[(212 - t) + 966 + 0.48\ (T - 212)]$.		
4.* Loss due to heat carried away in the dry chimney gases = weight of gas per pound of combustible × 0.24 × $(T - t)$.		
5.† Loss due to incomplete combustion of carbon = $\frac{CO}{CO_2 + CO} \times \frac{\text{per cent C in combustible}}{100} \times 10{,}150$.		
6. Loss due to unconsumed hydrogen and hydrocarbons, to heating the moisture in the air, to radiation, and unaccounted for. (Some of these losses may be separately itemized if data are obtained from which they may be calculated.)		
Totals......................................		100.00

* The weight of gas per pound of carbon burned may be calculated from the gas analyses as follows:

$$\text{Dry gas per pound carbon} = \frac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)},$$

in which CO_2, CO, O, and N are the percentages by volume of the several gases. As the sampling and analyses of the gases in the present state of the art are liable to considerable errors, the result of this calculation is usually only an approximate one. The heat balance itself is also only approximate for this reason, as well as for the fact that it is not possible to determine accurately the percentage of unburned hydrogen or hydrocarbons in the flue gases.

The weight of dry gas per pound of combustible is found by multiplying the dry gas per pound of carbon by the percentage of carbon in the combustible, and dividing by 100.

† CO_2 and CO are respectively the percentage by volume of carbonic acid and carbonic oxide in the flue gases. The quantity 10,150 = number of heat units generated by burning to carbonic acid one pound of carbon contained in carbonic oxide.

Of two appliances for utilizing heat energy, that is the more effective which most completely renders the available heat into useful work or product. Examples of the distribution of the available heat as reported by various authorities are given in the following table:

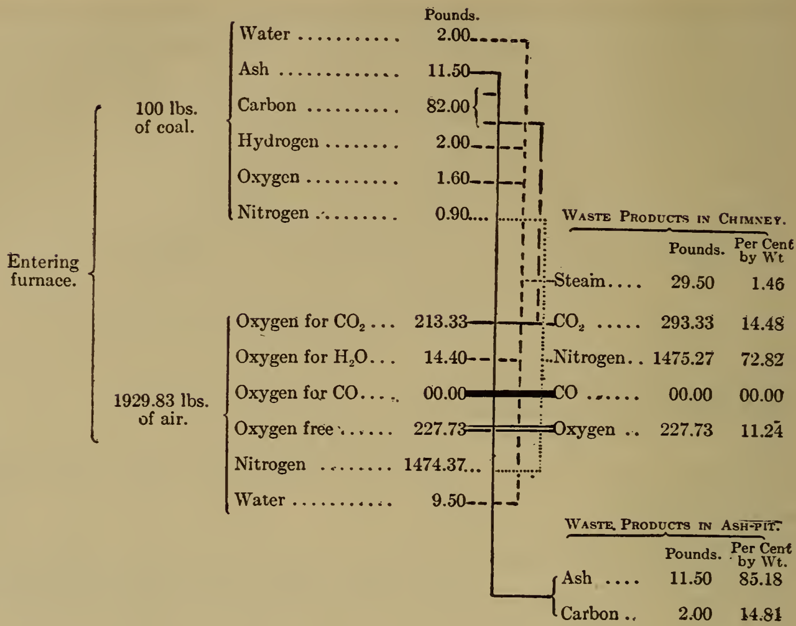
DISPOSITION OF HEAT IN STEAM-BOILERS.

Disposition of Heat	Authority.						
	Bunté.	Scheurer and Meunier.		Donkin and Kennedy.			Hoadley.
		A	B	C	D	E	
Waste in flue-gases, including evaporation of moisture in coal and heating vapor in air when these losses are not separately given	18.6	5.5	14.8	9.4	22.5	6.5	5.04
Evaporating moisture in coal	3.5	2.5	6.1	0.1	0.1	0.0	1.55
Heating vapor in air	......	......	......	......	......	......	0.18
Imperfect combustion	8.0	6.0	......	12.7	0.0	0.0	1.44
Clinker and ash	4.1	......	......	0.1	0.2	0.0	......
Radiation and heat not otherwise accounted for	7.6	23.5	13.4	13.9	11.0	15.0	4.00
Heating and evaporation of water	58.2	61.0	65.7	63.8	66.2	78.5	87.79

An interesting computation of the results and requirements with a combustion of 100 pounds of anthracite is given in the following table, where it is assumed that the coal and air have a temperature of 60° and that the chimney-gases are at 500°. Hot ashes are withdrawn at 450°, and 2 per cent of carbon goes out with them. Under the assumed conditions 21 per cent is lost, for which ash and moisture in coal and air are responsible for over 5 per cent.

HEAT-LOSSES INCIDENT TO THE COMBUSTION OF 100 POUNDS ANTHRACITE COAL.

Heat-losses.	Number of B.T.U.	Per Cent of Total Heat of Fuel.
By water = [(212 − 60) × wt.] + 965.7 × wt. + [sp. heat × (500 − 212) × wt]	37012.5	2.83
By carbonic acid = wt. × sp. heat × (500 − 60)	27994.2	2.13
By nitrogen = wt. × sp. heat × (500 − 60)	158452.8	12.07
By free oxygen = wt. × sp. heat × (500 − 60)	21973.6	1.67
By ash = wt. × sp. heat × (450 − 60)	1105.7	0.08
By carbon in ash = wt. × sp. heat × (450 − 60) + wt. × 14650	29488.3	2.24
By carbonic oxide = wt. × sp. heat × (500 − 60) + wt. × 4400	....	....
Total heat lost exclusive of loss by radiation	276027.1	21.02
Theoretically possible evaporation in pounds of water from and at 212° per pound of combustible utilized	12.73	
Theoretically possible evaporation in pounds of water from and at 212° per pound of fuel utilized	10.44	



TOTAL HEAT OF FUEL.

$$\text{Weight of C} \times 14{,}650 = 82 \times 14{,}650 = \ldots\ldots\ldots\ldots\ 1{,}201{,}300 \text{ B.T.U.}$$

$$\text{Weight of H} - \left(\frac{\text{Wt. of O}}{8}\right) \times 62{,}100 = 2 - \left(\frac{1.6}{8}\right) \times 62{,}100 = 111{,}780 \text{ B.T.U.}$$

$$1{,}313{,}080 \text{ B.T.U.}$$

HEAT GENERATED.

$$80 \times 14{,}650 = \ldots\ldots\ldots\ldots\ 1{,}172{,}000 \text{ B.T.U.}$$

$$2 - \left(\frac{1.6}{8}\right) \times 62{,}100 = \ldots\ldots\ldots\ldots\ 111{,}780 \text{ B.T.U.}$$

$$1{,}283{,}780 \text{ B.T.U.}$$

It will be apparent that the losses are due to Nos. 7, 8, 9, 13, 14 and 15 of the analysis in Fig. 2 and may be listed as

1. Losses in heat of ashes 7
2. Losses in heating fire-room 8
3. Losses in excess chimney temperature 9
4. Losses in an incomplete combustion 13–15
5. Losses in evaporating fuel moisture 14
6. Losses in heating inert matter 15

Nos. 10 and 12 are returned to the steam: Nos. 18 and 19 are partly returned, and what is not returned is not lost.

Heat energy is expended after the steam is made in running some of the auxiliaries discussed in later chapters, but for the purpose in hand and with regard to the boiler only, the above is reasonably complete. It will be for the later treatments to bring up the economy from use of auxiliary or subsidiary devices in diminishing labor expenses or diminishing heat consumption.

It is conceivable that the balance of heat should not be a credit balance, or that the losses overbalance the input. This is so rarely the case that the reverse is the general case. If the plant were called on to make a financial balance of input of capital and labor and material, the credit balance on the heat balance computation, multiplied by 778 to reduce it to foot-pounds and by the factors to change it into horse power per year, and again multiplied by the price of a horse-power per year, makes a product which is the income per year. The expense per year on the debit side will be the interest on the plant, deterioration, repairs, labor, fuel, supplies, water, and general expense. This is for the steam furnished to the engine; and the significance of the heat-balance factor is at once apparent.

CHAPTER X.

BOILER ACCESSORY APPARATUS.

146. Introductory. It will be apparent from Figs. 1 and 2 and the logic of the development in the preceding chapters that the steam boiler as a generator of pressure from heat and a storage reservoir for energy requires for its safe and economical handling as well as for convenience that certain accessory apparatus should be applied to it. The water has to be fed into the boiler against the working pressure, and a pump or its equivalent driven by power would seem essential: the water must not be pumped in excess, else the generator floods with water, nor in too small quantities, else the metal of the boiler overheats. The control of the fire, too, must be related to the consumption of energy and steam by the engine, and there must be a relief valve to prevent accumulation of excessive pressure, and provisions for draining mud and dirt from the boiler when the feed-water brings it in and leaves it behind when it goes off as steam gas., All these belong in a class of accessories which are essential and will be therefore present in any plant whatever.

In addition to these, and forming a separate group, are certain pieces of apparatus which may be called auxiliaries to distinguish them from the essentials. These are to secure economy of labor or economy of heat utilization, and are to be described as desirable but not indispensable. In this class are the coal-handling and ash-handling machinery, the low-water alarms, fusible plugs, feed-water heaters, and the like. The present chapter is to treat of the essential accessories, leaving the auxiliaries to a later chapter.

It must not be inferred from the word "essential" that no plant can be run without these. Plants were so run before they were introduced, and can be still run when these are disabled. But it is so much better to have them that it is foolish and wrong to try to do without them.

147. The Feed-Pump. It will have been obvious from Fig. 1 and from the requirements of the case that an apparatus to force water into the boiler or pressure generator in an essential. Nos. 18, 19, 20 and 21 of Fig. 2 show the place and function of the water-supplying machinery. In the beginning, when pressures in the boilers were low, water could be gotten into the boiler by gravity from an elevated tank

(Appendix, Fig. 697A): in a few cases where the city mains carry water at high pressure no separate pump for the boiler may be required; but in this latter case the water-works pump is the boiler-feed pump. Such pump as the boiler requires is to feed to it the water which it evaporates, and is hence called the feed-pump, and all its accessories are designated by the same prefix. The suction of the feed-pump comes from a well or cistern or pond or water-course or from the municipal water supply. Its quality is the main element, and this will be taken up further in Chapter XII. If it is desired to doctor the feed-water or to physic the boiler by introducing reagents for a chemical reaction, they are introduced into the suction tank, or by a branch inlet into the suction pipe. The cross section of the pipe should be such that the water should not be compelled to flow in it at a lineal rate faster than 200 feet per minute. The suction pipe rarely has valves in it, unless the supply comes under pressure or from several sources, so that at times an isolation of any source becomes necessary. The prime requisite of the feed-pump is that it shall supply the quantity of water needed, and deliver this quantity into the boiler against the pressure therein, and overcome all frictional resistances of valves and piping between the suction tank and the boiler. This quantity of water per hour is at least the maximum evaporative capacity of the boiler-heating surface, or the maximum water consumption per hour of the engine (paragrafs 8, 9, 11, and 12). It is usual to make it 125 per cent of this maximum, to allow for losses of steam in leakage at the safety valve or whistle, and to enable a quantity of water and heat to be accumulated in the boiler while the continuous demand is being made, if this should be desirable. A strong feed of cool water is an effective way of lowering pressure or keeping it down.

The feeding power may be given by a pump or by an injector. If a pump is preferred, it may be used in several ways.

148. The Attached Pump or the Independent Feed-Pump. By the term "attached" pump is meant the type in which the power for the pump piston or plunger is derived from the main engine, or the pump is attached to its mechanism. It was first driven in early vertical beam engines from one of the rods actuated by the beam of the engine mechanism: it was later in horizontal engines driven from the cross-head or crank-shaft. It is often in factories driven by a belt from the shafting of the transmission machinery (Fig. 230). The advantages for the attached plan are:

1. The feeding is constant in small quantities per unit of time.
2. The feeding is persistent or without pauses, and needs only to have quantity controlled by an attendant.

3. The feeding by such control can be kept proportional to consumption or the water level constant in the boiler. In theory, the pump should at each stroke send into the boiler the same weight of water as went out of it to the engine in the form of steam at the previous stroke.

4. The big engine cylinder will use the little extra steam required to work its pump with more economy than the little independent cylinder operating the same pump cylinder.

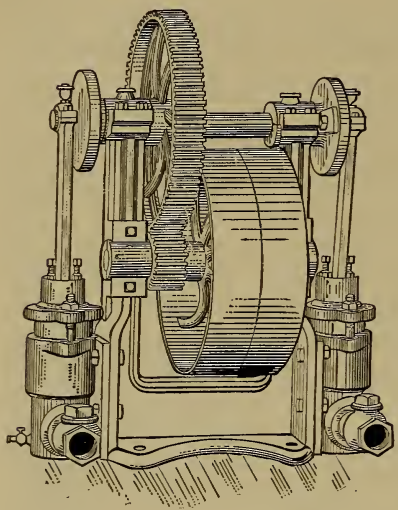


Fig. 230.

The objections are:

5. The system cannot be used in direct form on very high speed engines.

6. The pump only works when the main engine is running: or the whole power plant, as in a locomotive, must be run in order to work the pump.

7. The control of the feed cannot be on the delivery or forcing side of the pump, but must be on the suction side. The power of the main engine could break the pump and fittings if the water could not get away from the pump-barrel. To control on the suction side means a partial filling of the barrel when the suction valve is not wide open, and hence a thump or pound when the piston or plunger fetches up against solid water just as the valves open from the pressure.

When driven from the main engine mechanism directly these pumps are usually plunger pumps, giving their volume by a long stroke and relatively small diameter.

A system of control much used with the attached system is to place a branch outlet on the delivery pipe from the pump, with a valve on it, controlling the discharge through that pipe which leads back to the suction-tank or to some outfall to waste or overboard. When the branch or by-pass valve is wide open, all the pump delivery will go that way against the less resistance and none will go to the boiler. When the branch valve is closed, all the feed will go to the boiler. At partial openings some goes to the boiler and some by-passes through the valve according to the relation of resistances in the two pipes, but at no time can the resistance or pressure in the pipe exceed that due to the pressure in the boiler. Fig. 230 shows an accepted form of shaft-driven pump, capable also of being attached to an electric motor in the next class.

The Detached or Independent Feed-Pump is one driven by its own independent power, and not from the main engine directly or indirectly. This may be a steam pump or an electric pump or get its power from any source. Its advantages are:

8. It can be run at any speed of its own, not limited by the high speed or the low speed of the main engine.

9. It can be located anywhere.

10. The main engine does not have to be run to pump water. The pump can be run by hand if desired.

By running the pump at a slow speed and controlled by its power, the advantages 1, 2, 3, and 4 of the other system are secured.

11. In large plants several feed-pumps in duplicate can be operated so as to lessen danger of a complete stoppage by a break down.

If steam is used to drive the pump, the objections are:

12. The uneconomical use of steam in small cylinders:

13. The disposal of the exhaust from the steam-cylinder, and the drips.

14. The running expense of the pump in oil and supplies.

The electric motor-driven pump secures the variable speed advantages of Nos. 8 to 11, and the four advantages of the attached system. Its current costs more than the steam of an economical engine, but less than that of a wasteful one. In very large plants the feed pump becomes large enough to secure some economy in steam consumption by compounding and expansive working (paragrafs 298, 317). The electric pump is easily governed so as to be partly or largely automatic in action: it will usually be multiple-barreled to secure constant flow and equable resistance to the motor.

Independent pumps may be either plunger or piston pumps. The piston is lighter in weight and is supported through its entire traverse: it will be more usual with clean waters. The plunger is at its best with gritty waters since it does not touch the bore of its barrel on the bottom where the sand lies. Massive plungers are best used in vertical engines since they then produce no tendency to flexure as they are moved in and out. When the plunger system is made double-acting, it may either pass through a partition in the pump chamber (Fig. 231); or the two plungers may enter opposite ends of the chamber with the partition between without an opening but the plungers connected outside by a pair of rods; or the plunger may slide in and out of a pair of separate chambers (Fig. 232). With these latter arrangements the plunger packing is external and accessible: with the piston or internal partition leakage may take place from end to end. It does no special harm except to increase the "slip" of the pump, or the difference between the water actually pumped and the computed volume of the plunger displacement per minute or per hour.

The steam power may be applied to the pump in two differing systems. In the one a fly-wheel is used to control the speed and a crank and connecting rod or its equivalent to control the length of the stroke. The fly-wheel carries the piston past the dead centers so as to operate the valves effectively and prevent a stalling of the pump with the valve covering the ports to both ends of the cylinder. The other is the non-fly-wheel non-crank pump or the direct-acting pump.

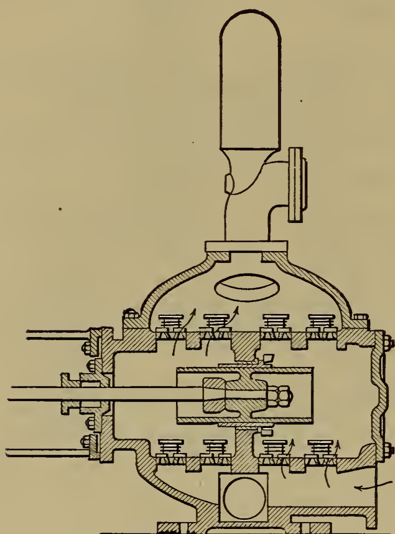


FIG. 231.

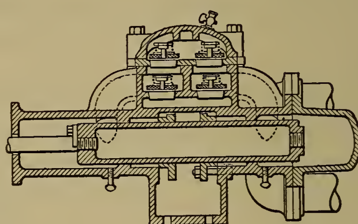


FIG. 232.

Both are direct-acting, but this term is used to distinguish the types from each other in popular use.

149. The Fly-wheel Pump. The fly-wheel feed-pump horizontal or vertical resembles a typical steam-engine mechanism, having the pump barrel on the prolongation of the steam-cylinder piston-rod (Fig. 233). Instead of the yoke scheme for eliminating the connecting rod and the room it occupies, a short rod is often used in fire-engine practice which turns the crank within a frame in the plane of the rods which gives the necessary swing for the short crank which is required (Fig. 234). The power does not go through the crank but only the forces of regulation by means of stored energy in the fly-wheel. The vertical fly-wheel pump used for feeding boilers on the Mississippi valley rivers has been called a "Doctor." The advantages of the fly-wheel principle are

(1) It is simple and positive in its action.

(2) This adapts it for use where only unskilled labor is to be had.

(3) It secures economy of steam-consumption by its ability to work the steam expansively.

(4) Its stroke is a positive length determined by the crank, and if necessary it can be worked as a hand-pump by turning the fly-wheel.

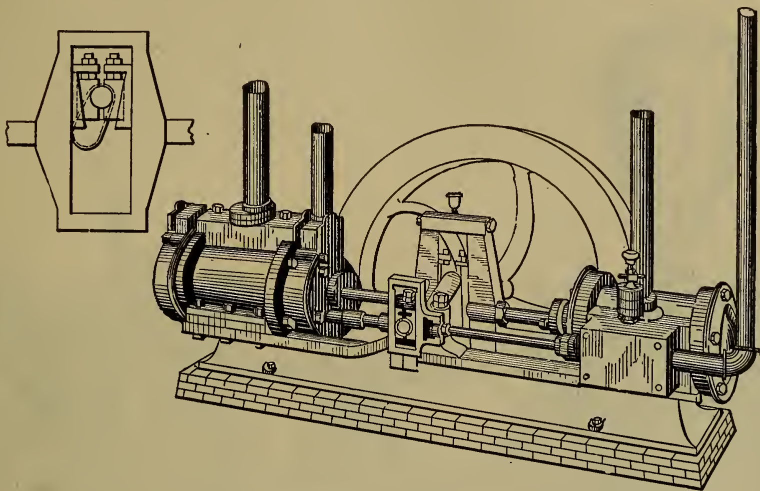


Fig. 233.

(5) Its valve gear is very simple and obvious, and its stoppage from any clogging of ports unlikely. If there is pressure in the steam pipe and valve chest, the pump must run.

The objections to the fly-wheel pump are

(6) It cannot be run slowly without danger of stopping on its dead centers, unless duplex and with cranks quartering. Then it can be kept down to 20 turns a minute.

(7) It cannot be conveniently controlled, therefore, by valves upon the delivery-pipe from it.

(8) The objection which has been urged against fly-wheel pumps that they accelerate the flow of the water through the pump-cylinder as the velocity of the piston is controlled by that of the crank, has no significance with the relatively small masses of water which these pumps are required to handle.

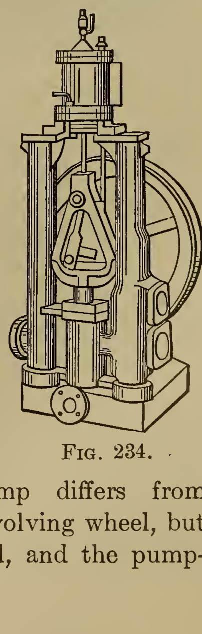


Fig. 234.

150. Direct-acting Pump. The direct-acting pump differs from the fly-wheel pump by having no crank, shaft, or revolving wheel, but simply the steam-piston on one end of a piston-rod, and the pump-piston or plunger at the other (Fig. 235). Since there is no stored energy or velocity in a revolving or moving mass to carry the motion past the end of the stroke and cause the engine to reverse, this result must be otherwise attained. The energy in the reciprocating masses

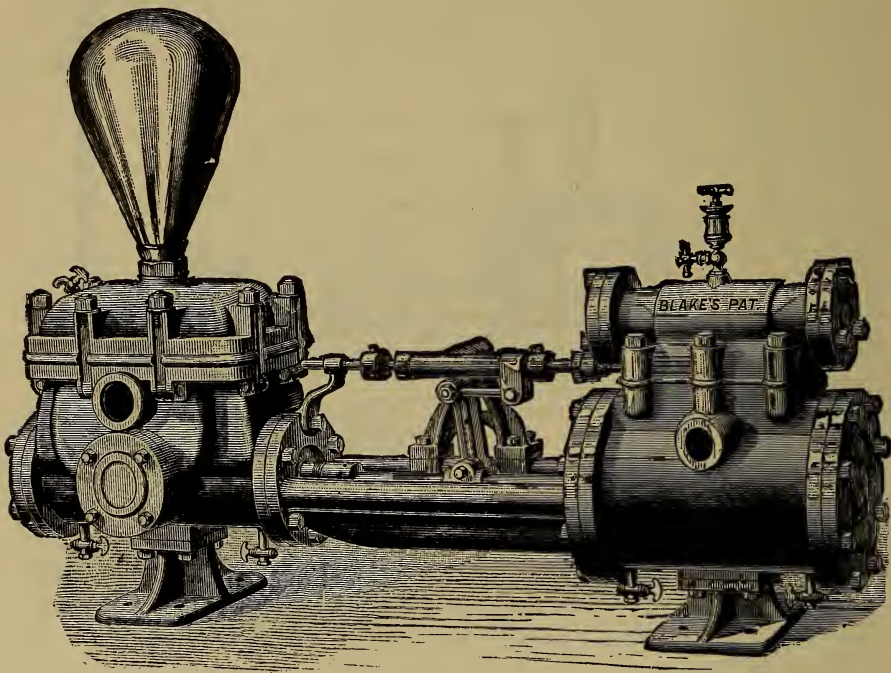


Fig. 235.

of pistons and rods is not enough, since the friction of the pump and the resistance of the water will leave the piston in such a position that the valve attached to it positively will have so reduced the steam pressure on the forward stroke that the energy remaining in it will not carry it far enough further to open the port for the return stroke. This result is best secured by having the steam valve not directly or positively attached to the piston-rod, but by having such valve thrown from the forward to the backward position by another engine piston driven by steam, the valves of the latter being operated by the main engine piston. In the design of Fig. 235 the piston which moves the main engine valve is that of a purely auxiliary engine in the little cylinder on top of the steam-cylinder. The valve-rod is that of the auxiliary and not of the main cylinder. As the piston of the pump moves to the left for example, the valve of this large cylinder is at the left and the right-hand port wide open. It stays so until the piston in its traverse by its connection to the valve-rod of the auxiliary moves this valve far enough to cause that auxiliary piston to move from left to right carrying the main engine valve from left to right and opening the main engine left-hand port wide. The main engine thereupon makes its stroke and the cycle is repeated. The main engine can never stall on its dead-centers, no matter how slow it moves, for its steam valve is either wide open at one end or at the other, and such motion occurs while the engine piston is at rest and independent of it. The mechanism of the auxiliary engine

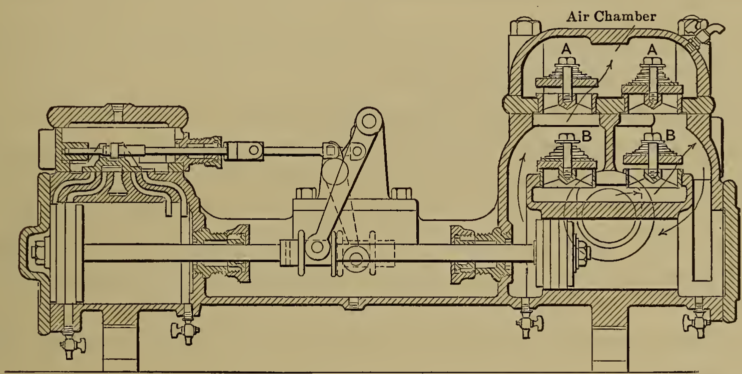


Fig. 236.

takes many forms as to its valve and the motion of its stem: the valves may be concealed in the cylinder casting and be operated by spindles pushed internally by the main piston. Or, in the duplex type, the auxiliary engine may become a second pumping engine cylinder at the side of the other one, and the valves of No. 1 cylinder be operated by the piston motion of No. 2 (Fig. 236).

In this the steam valve shown is operated by the dotted lever; and the lever in full lines is connected to the valve stem whose end is dotted. The duplex pump gives a momentary pause at the end of each piston traverse while the other piston is moving its valve for it; and this pause permits the water valves to seat quietly and without thumping and lessens slip.

The advantages of a direct-acting or non-fly-wheel pump are

(1) The velocity of the delivery is proportional to the resistance offered by the water. Hence it is possible to control the feed-pump of the boiler when of this class by the opening and closing of valves upon the delivery. When the resistance to the delivery exceeds the forward effect of the steam-pressure the pump stops. The forward or feeding

force is secured by making the area of the steam end of the pump three or four times the area of the water end.

(2) The pump has no centers, but will start from rest as soon as steam is turned on to it. This property is the result of the steam-thrown valve, because the main valve is open at one end of the cylinder until reversed and opened wide at the other end.

(3) The pump can be run as slowly as suits the convenience or the requirements of the feeding.

(4) Water in the cylinder from condensation causes no danger to the mechanism. The pump will start with water as well as with steam if the pressure is there.

(5) The velocity of flow is not accelerated by the connection of the piston to revolving mechanism.

The objections to the non-fly-wheel pump are:

(6) That the steam-thrown valve does not encourage expansive working, nor at ordinary speeds can it be secured when there is no fly-wheel to store up excess of work at one part of the stroke to give it out at the second part. In large water-working pumping-engines this has been secured by devices which are not considered worth while for feed-pumps.

(7) The stroke is not positive in length, as there is nothing to compel it to be so.

(8) This compels an excessive clearance-volume in the steam-cylinder in order to guard against the piston fetching up against the head when running at high speed.

(9) The ports or passages of the auxiliary engine in the single-cylinder type are small, sometimes tortuous, and are liable to clog or be made impassable. This comes from the lubricating oil, or from oxide of iron; or in making gasket joints between parts the port opening has been overlooked, and remains closed when the joint is made up. The pumps stop for no apparent or obvious reason, and this lends an appearance of complexity and mystery to their operation.

The advantages where intelligent labor is to be had which belong to the slow running and easy control of the direct-acting pump have made it a very popular form for boiler-feeding. The fly-wheel type remains in general preference in Western river-boat practice and for fire-engines.

It is not desirable to control a direct-acting pump by a valve on the suction, since the barrel of the pump should be perfectly filled at each stroke. Otherwise, when the pump reverses, part of the stroke will be made against little or no resistance, and as there is no controlling mechanism of crank and revolving shaft, a serious jar will occur when the pump-piston encounters solid water after part of the stroke is completed.

For similar reasons when a pump is to handle hot water it should receive the water from a height caused by gravity, and not be compelled to lift it. The vaporization of the hot water under the reduced pressure caused by the sucking action of the pump will prevent the barrel from filling, entailing the same difficulty from jar. When pumping hot water, furthermore, the pump will require to be fitted either with metallic valves or hard rubber resistant to the action of heat. With cold water the ordinary soft-rubber valves closed by springs are cheap, convenient, and tight (Figs. 231, 232).

These figures and Fig. 236 show also the usual arrangement of such valves, the suction entering by atmospheric pressure or gravity into the chamber under the lower set of valves at each end alternately and being forced out by the displacement of the piston or plunger through the upper valves at the other end. These illustrate also the two practices of a small number of larger valves, or a large number of smaller valves. Inertia effect is less as the masses of the valves are less. The use of a feed-water heater to save the waste-heat in the exhaust of the feed-pump will be discussed under Boiler-room Auxiliaries.

151. The Feed-pipe and its Valves. From the pump to the boiler, the water must be carried in a pipe. This pipe must be strong enough to resist pressure and must not be easily corroded. It is often made of copper by reason of its flexibility and ductility under the changes of temperature to which it is exposed, and because bends are easily made in it, and because the solid matter precipitated from the water does not adhere to it. Iron pipe, which is often used in stationary practice, has the advantage of being cheap and that the fittings which are required are easily made and attached to it, which is not the case with the copper feed-pipes. The diameter of the pipe should be chosen first with respect to having the velocity of the water through it not exceed 200 to 400 feet per minute of linear velocity. It is desirable also that the feed-pipe should be large enough so that even if it should become somewhat stopped up with scale, as has occurred in the example shown in Fig. 237, it may be possible to get the scale out, or to leave still space enough through which the water can be forced.

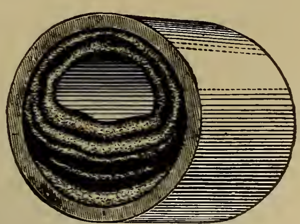


Fig. 237.

Upon the feed-pipe will be the necessary valves. The first of these is one for controlling the flow of feed-water into the boiler in question if a number of boilers are supplied through a common pipe. This will be through a cock-valve, which is preferred in English practice, or a globe valve, which is more usual in American practice. The cock-valve is not liable to clogging from precipitated scale, which is a difficulty connected with the globe valve, but with modern forms of globe valves they are easier to keep tight than a taper plug. The latter also gives trouble sometimes by expansion, although the packed-stem plug-valves are not open to this difficulty. Close to the boiler where the feed-pipe enters it will be a check-valve. This is imperatively necessary where several boilers are connected to a common feed-pipe, but is desirable in every case. The check-valve lifts by excess of pressure on its lower side, as compared with the pressure in the boiler, which bears upon its upper side. Its object is to prevent water which has once gotten into the boiler from getting out again back into the feed-pipe. This serves to keep the scale out of the feed-pipe, to prevent siphoning of water from one boiler into another, and to prevent hot water from working back to the pump where it would be troublesome. These check-valves are made to work in horizontal or vertical pipes. The difficulty to which they are liable is a tendency to leak through abrasion of their seats, or by being held off the seat wholly or in part by some solid matter in the feed-water which gets caught in the valve. All such check-valves have an opening to permit access to the valve for inspection and for repairs (regrinding of the seat, or renewal of the valve-face) (Figs. 238 and 239), and in order to permit this repair or inspection without emptying the boiler of pressure and of water it is desirable to interpose a gate or stop valve between the check-valve and the boiler, so that the latter can be cut off from the check-valve when it is to be inspected.

152. Introduction of the Feed-water. The water to be evaporated by the boiler is fed to it as a rule cooler than the water within the boiler. It should therefore be introduced at such a point as to favor and not to impede the currents of circulation and convection within the boiler; and furthermore, if it can be persuaded to deposit the solid matter which is contained in the feed-water immediately on entering the boiler, it is desirable to have regard to this in selecting the place at which the water shall enter. In sectional and most of the shell boilers this indicates that the water should commence to flow within the boiler at or near the surface, and at the back of the boiler or where the heaviest water is descending. By having the feed-water enter at the surface there is also met less danger from the siphoning of the water in a boiler out through the feed-pipe either to another boiler or to waste, if anything is wrong with the check-valve which should prevent this action. Boilers may empty themselves through a feed-pipe which enters at the

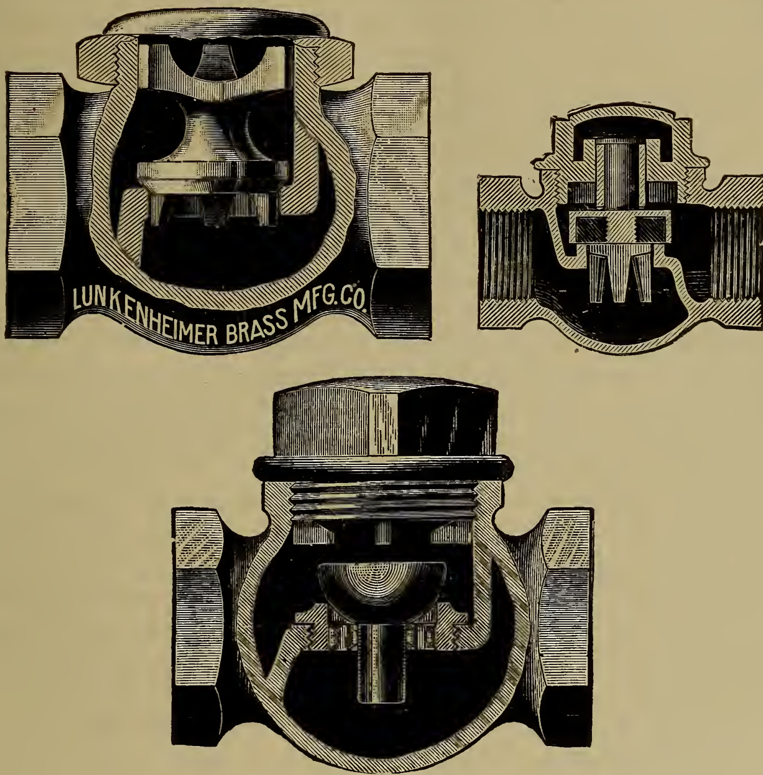


FIG. 238.

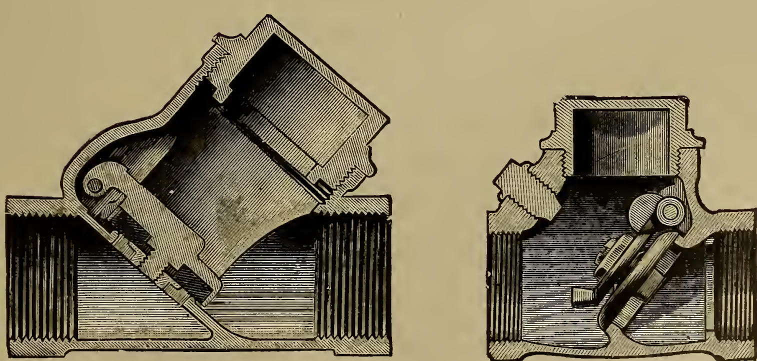


FIG. 239.

bottom; but where the feed-pipe is near the surface of the water, the water below its level can only get out by evaporation. This further produces less injury to the metal of the boiler near the feed-inlet from sudden change of temperature.

It is more convenient, however, to have the valves control the flow of feed-water into the boiler at the front rather than at the back. This has given rise to a very prevalent practice of carrying the feed-pipe through the front head, and along the length of the boiler to that point farthest from the fire at which the water shall actually mix with the water in the boiler. This serves to bring this entering water up somewhat nearer the temperature of the boiler-water before it strikes the shell-plates. This inner feed-pipe is sometimes perforated along its length with the idea of causing the cold water to enter in fine streams rather than all at one place. The objection to the interior pipe, and particularly to a perforated one, is its liability to become stopped up by matter precipitated from the water by heat.

The introduction of the feed-pipe and its details may be studied with advantage from Figs. 73, 74, 96, 126, 131, 148, 153, 178, 183, 185. Figs. 178 and 183 show the internal distribution arrangement. In the coil boilers the feed will be universally at the bottom end of the coil so as to make the current of circulation an unbroken one, and to permit the feed-pump to effect in part a mechanical circulation (Fig. 161). The exception is the White semi-flash coil, where the cooler water is at the top and the circulation from above downward (Fig. 165).

153. Blow-off Pipe. The boiler requires to have a pipe connected to its lowest and coolest point to allow the boiler to be emptied for inspection and cleaning, as well as to be used for the removal of part of the contents of the boiler into the drainage system of the plant while at work, if this is desired. Such a pipe will be called the blow-off pipe and will have in it, and as close to the boiler as convenient, the blow-off cock or valve. From its location at the lowest and coolest point the solid matter, mud, and precipitated salts will gather in its neighborhood, so that when opened with pressure on the boiler a rapid rush of the hot water out through the valve and pipe will carry away some of the material of this sort. For this reason the blow-off valve is located in the mud-drum of boilers which have one. A gate or cock-valve is to be preferred for the blow-off valve, because it is not liable to become clogged from the precipitation of salts which may harden about it, and for this same reason also it is desirable that the pipe should be of generous size, so that it may easily free itself of the accumulations which may take place within it. It should rarely, even in a small boiler, be made less than 2 inches in diameter. In brick-set boilers, where the blow-off pipe must pass through the combustion-chamber, it is particularly liable to become burned out by the overheating to which it is liable if scale gets into it. It is for this reason quite customary to cover it with some incombustible and non-conducting material in that part of its length where it is exposed to flame and hot gas (see Figs. 70, 74, 137, 140, 148, 178, 183, 185, 188).

In boilers using salt water the blow-off cock must also be used frequently in order to reduce the percentage of salty matter which is forced into the boiler with the feed-water, but cannot go out with steam. This opening of the blow-off valve is called blowing down, and permits the concentrated solution to be diluted by pumping in water to replace that which has blown to waste. Modern practice admits little or no salt water as such to the boilers, but distills sea water in separate apparatus and pumps the pure distillate into the boilers to replace what is lost in whistles, sirens and other non-condensing apparatus, and from leakage. Flash boilers should be blown off every night. In the open country or at sea, the blow-off pipe may be open to the air or to the drains, or to waste overboard. In cities it is not desirable to blow hot water into brick and cement sewers, and in many places a blow-off tank is specified by law or ordinance. This has sometimes and improperly been made a simple brick cistern. It is better to make it of cast iron or of quarter-inch steel plate. In good water districts where blowing out or down is not frequent, its capacity may be as low as five per cent of the gross bulk of water in the boilers to blow into it: several tanks will be better than one large one in a large plant. The blow-off pipe comes in at the top of the closed tank, and the outflow from the top also reaches down to near the bottom. This makes a seal, so that the cold contents of the tank are forced out first before the new hot water comes. A vapor pipe from the top leading to the open air at the roof or at some height prevents pressure accumulating, or any siphon action.

154. Loss in Blowing off. The loss from blowing hot water from a boiler is found by the following:

$$\begin{array}{lr} \text{Weight evaporated} \times \text{(total heat} - \text{feed heat)} & = x \\ \text{Weight blown out} \times \text{(sensible heat} - \text{feed heat)} & = y \\ \hline \text{Total} & = x + y \end{array}$$

$$\frac{y \times 100}{x + y} = \text{loss in per cent by blowing down.}$$

155. The Injector. If the water for the boiler is not forced into it by a pump (paragraf 147) the alternative apparatus is an injector. Its fundamental theory was first propounded by Giffard, and his name is often attached to all embodiments of the principle however far these may differ from his original. The injector differs from a feed-pump for this same purpose, because the latter is based upon easily understood principles of pressure and resistance, and a greater volume of steam displaces a less volume of water where the head of steam-pressure and the resistance-head of the water-cylinder are the same. The injector, on the other hand, depends upon a direct conversion of heat energy into dynamic energy, and by a process not so obvious or plain as in the case of the steam-pump.

The injector may be defined as an appliance or apparatus whereby a jet of steam moving at a velocity due to its pressure is made to impinge upon a mass of cool water, to which it transfers its energy to such an extent that the combined jet of steam and water will overcome a resistance-head equal to or greater than the pressure which actuated the original jet of steam. That is, a jet of steam issuing from a boiler into an injector will pick up a quantity of water, and will be able to force that water into the boiler against the pressure which actuated the jet, and will carry the steam in the original jet also back into the boiler from which it started.

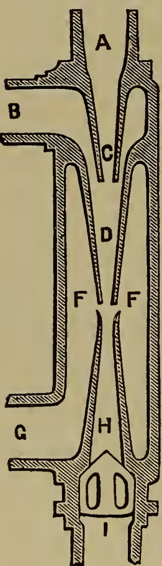


FIG. 240.

The injector conforming to this definition consists of a hollow, somewhat tubular casting, usually of brass, into which are made three openings. The first one (A, Fig. 240), which usually enters the top of the instrument, is for the delivery to it of hot, dry steam from the dome of the boiler or other convenient place. The second opening, B, is the inlet for the water to be fed, which is usually delivered to it from below. The third will be the feed-outlet, HI, opening towards the boiler, through which the feed-water impelled by the steam will pass to overcome the pressure on the check-valve and enter the boiler.

The injector, properly so-called, has the cross-section of its tube proportions diminishing from the nozzle of the jet, in order that the velocity of the stream may increase from the point of meeting the water until the current streams into the vessel in which a high tension is maintained. If, on the other hand, the tube proportions are arranged to flare or increase in cross-section beyond the combining point, the velocity of flow is decreased, less resistance-head will be overcome, and the instrument becomes properly designated an *ejector*. The term injector may therefore properly be limited to instruments operating to force water against considerable resistances and in comparatively small volumes, the water becoming considerably raised in temperature in the process.

The ejector, on the other hand, is adapted for handling large volumes against low resistances, and by virtue of the greater mass of water handled the latter becomes only slightly warmed. Both injector and ejector as appliances for moving water are wasteful of heat as compared with a good pump; when applied as a boiler-feeding appliance where heating of the feed-water is convenient and desirable, the injector does what can only be done with a pump by adding to the latter much of complication in the way of heat-saving devices.

The injector depends on three sets of principles. Two of these are physical or mechanical, attaching to it because it is a jet and impact apparatus; the third are the thermal or heat principles, resulting from the heat-transfers when steam and water are the fluids concerned.

156. Mechanical Principles underlying the Injector. The Induced-current Principle. The injector must be capable of lifting the water which it is to feed into the boiler from a level in a tank lower than the instrument itself. If circumstances permit the water to flow to it from a higher level, it does not require to use this capacity, and will be called a non-lifting injector. Where the machine must raise the water to its level it will be called a lifting injector.

The principle on which the injector depends for its capacity to lift water is known in pneumatics or hydraulics as the principle of induced currents. If a jet of steam or air or water is made to move with a considerable velocity in a line parallel to the axis of a second or larger tube which surrounds the issuing jet, the impact of the matter issuing from the moving jet upon the matter within the surrounding tube will cause the contents of the latter to move with the jet in its direction. The action may be an impact of particles, or it may be a frictional entrainment of the one fluid by the movement of the other. If the cross-section or profile of the inner jet and the outer tube be adjusted to each other in the light of experience, the jet will induce a continued flow, tending to exhaust the contents in the space in the outer tube B which lies behind the orifice of the inducing jet C. This principle is a familiar one in the exhaust-steam blast in the locomotive, in the atomizer of the chemist and physician, and in many much-used applications. As applied in the lifting-injector it requires that the space behind the nozzle of the operative jet be connected by an air-tight pipe to the suction-tank, within which it must be so immersed as to be water-sealed. In front of the nozzle of the jet must be an orifice opening to the air or to some waste-connection. With this latter orifice open, if the operating-jet be started, it will carry with it the air behind the jet until the pressure between the jet and the suction-tank becomes so much less than the pressure of the free atmosphere that the latter forces water up the suction-pipe to maintain the equilibrium. If the suction-pipe be not too long — probably less than 20 feet of the possible 32 feet — and the water in the suction-pipe is not so warm as to form a vapor in it under the reduced pressure, the induced current of air will cause the water to rise and meet the impact of the steam at the issue from the nozzle.

The second mechanical principle is the principle of impact of a small mass of steam in the jet against the greater mass of the water which the induced current has lifted. This will be discussed in a succeeding paragraf.

157. Heat-transfer, Work, and Efficiency in the Injector. The injector problem usually comes in the form that a certain weight of water W_w is to be forced into the boiler, and to do this work a weight of steam W_s must be expended. The weight of water will have its temperature raised from T_4, which it had in the tank, to the temperature T_2, which will be the temperature of the hot feed-water after leaving the injector, or $W_w (T_2 - T_4)$. The weight of steam condensed by this weight of water loses an amount of heat per pound which is represented by the equation

$$L = C (T_1 - T_2) + xr,$$

in which the specific heat may be called unity, and the percentage of vaporized steam is 100, or x equals unity, and r is the heat of vaporization at the temperature T_1 in the boiler, at which temperature the steam enters the injector. For a weight of steam W_s this becomes

$$W_s h = W_s [(T_1 - T_2) + r].$$

The heat given by the steam must equal that absorbed by the water; or

$$W_w (T_2 - T_4) = W_s [(T_1 - T_2) + r],$$

whence

$$W_s = W_w \frac{t_2 - t_4}{t_1 - t_2 + r}$$

in ordinary temperatures, since differences alone are used. The temperature t_2 of the water in the pipe leaving the injector must be low enough to have the condensation complete, and yet the hotter it is the better so far as the boiler is concerned. It is likely to be about 160° with small or light feeding and 120° to 140° with heavier feeding. According as the upper or lower values are taken, a calculation within ordinary ranges of pressures will bring a relation of steam weight to weight of water ranging between 10 and 13. That is, the steam supplies from 10 to 13 times its weight of water. In the absence of tables for r, it may be calculated from the formula

$$r = a - bT_1 = 1114.4 - 0.7\ T_1.$$

The water and the steam unite and flow together through the feed-pipe. The work to be done in the boiler where the water passes the check-valve will be the displacing per unit of time of a volume in cubic feet which is that of $W_w + W_s$ pounds against the boiler-pressure. Since the pressure of the atmosphere is exerted within the injector (p_0), the effective pressure to be overcome at the boiler is $p_1 - p_0$. The volume of $W_w + W_s$ in cubic feet will be

$$\frac{W_w + W_s}{62.4} = 0.016\,(W_w + W_s);$$

hence the work will be, if pressures are in pounds per square inch,

$$\text{Work} = 144\,(p_1 - p_0) \times 0.016\,(W_w + W_s) \text{ foot-pounds.}$$

From this it is easy to pass to the efficiency,

$$E = \frac{\text{Work done}}{\text{Heat expanded}} = \frac{144\,(p_1 - p_0) \times 0.016\,(W_w + W_s)}{778\,W_s\,(T_1 - T_2 + xr)},$$

which as a rule works out a small value only.

158. Mechanical Principle of Impact in the Injector. The mass of steam which meets the water in the combining tube of the injector has to act upon the latter by impact of the condensed water upon the feed-water. These masses being related to each other in a high ratio, such as one of steam to ten or sixteen of water, it follows that the principle of the conservation of the motion of the center of gravity will bring about a resultant velocity when they meet which will be to the velocity of the steam as W_s is to W_w. That is, if the mass of the steam be one and that of the water be ten, then the center of gravity of the two bodies will lie nearest the water and at one-eleventh of the distance separating them, and the velocity of the combination after impact will be one-eleventh of that of the steam. If the ratio of masses be 15, then the resultant velocity will be $\frac{1}{16}$ of that of the steam.

The accepted formulæ for the velocity of flow of a perfect gas from a reservoir within which is a pressure p_1 into another chamber where the pressure p_2 prevails are those of Joule and Thomson. When the weight of a unit of volume is denoted by w_1 at the pressure p_1 and the cooling during discharge is adiabatic, then

$$V^2 = 2\,g\,\frac{k}{(k-1)}\,\frac{p_1}{w_1}\left[1 - \frac{T_2}{T_1}\right] = 2\,g\,\frac{k}{k-1}\,p_1 v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right],$$

in which k will have the value appropriate for the observed adiabatic expansion of the gas in question. For steam it is 1.111 according to Rankine, and according to Zeuner the value

$$k = 1.035 + 0.100\,x,$$

in which x is the initial proportion of dry steam-gas. Solving, with the assumption that the steam is dry,

$$V = 23.2687\sqrt{p_1 v^1\left[1 - \left(\frac{p_2}{p_1}\right)^{0.1189}\right]}.$$

The velocity of the mixture of steam and water will be less than the one-tenth or one-sixteenth of the foregoing values, because the impact of all particles cannot be in the line of propulsion. Calling this water velocity V_2, the cross-section of the water-tube inside the injector will be

$$\text{Tube area} = \frac{0.016\,(W_w + W_s)}{V_2}.$$

The area of the steam-nozzle should be

$$\text{Nozzle area} = \frac{\text{Volume of steam corresponding to } W_s \text{ per sec.}}{V},$$

from which the diameters will follow, since the area will be πr^2.

It will be found, on making the calculation for any case, that the expenditure of heat-units resulting from the condensation process is so much greater than the equivalent expenditure in lifting the water by suction and of forcing it into the boiler, that these latter quantities are negligible by comparison.

159. Double-tube Injector. The Inspirator. — The necessity of adjusting the weight of water to the weight of steam in the jet, and the variation in the latter with varying steam pressure brought about the self-adjusting types of injector, where the combining tube was moved forward or backward, enlarging or constricting the water-area as the pressure outside of the tube might vary. This was a feature of improvements by Sellers & Co. as far back as 1865 (Fig. 241). In 1876–7 Körting of Hanover, Germany (Fig. 242) and Hancock of Boston introduced the use of two jets of steam to secure this same result. The first nozzle is the smaller and acts upon a relatively large mass of water to deliver it under a slight pressure to the second or larger steam-nozzle, which forces the supply to the boiler. The first jet will be the lifting-jet when the injector requires to raise its water.

Inspirator was the proprietary name given by Hancock to his double-tube design.

160. Restarting or Automatic Injectors. — An injector which will establish its action as a boiler-feeder automatically after the continuity of the combined jet has been broken by a stoppage of either the steam

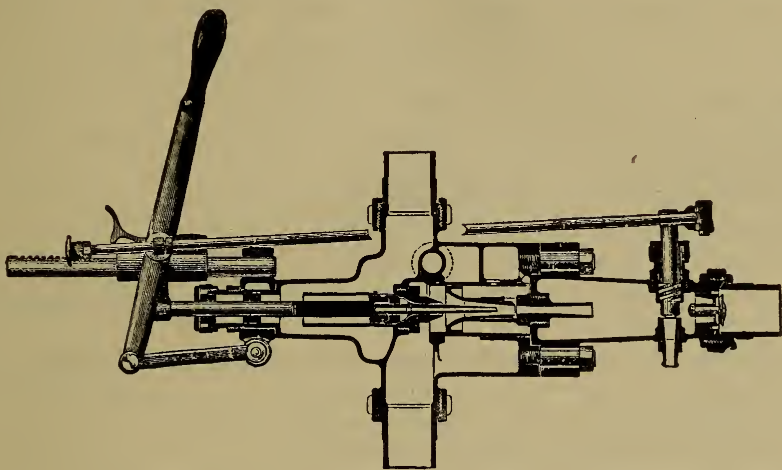


FIG. 241.

or water-supply is called an automatic or restarting injector. The usual method of securing it is to have two steam-jets, one for lifting and one for forcing. When the continuity of the combined jet is broken by a failure of the water-supply, the discharge of steam from

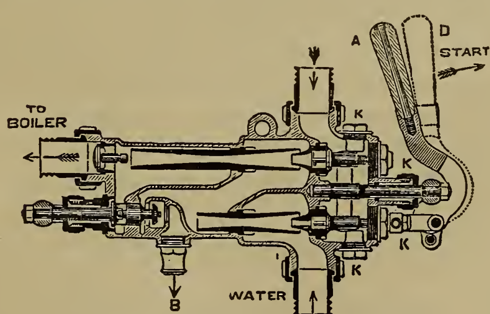


FIG. 242.

the forcing-jet finds its way to a waste-pipe through a check-valve, while the lifting-jet keeps up maintaining a vacuum in the feed-pipe and draws up the water as soon as it can be supplied. The adjustment of tubes and nozzles is also so made as to favor a wide variation of steam-pressures before a break in the flow shall occur. The advantage of the restarting principle is the simplification of the apparatus by doing away with adjusting spindles and the like, so that a simple valve on the steam-pipe to the injector is all the regulating appliance required.

161. Exhaust-steam Injectors. The steam-current for an injector may be derived from exhaust-steam when the back pressure head is less than 75 pounds. The feed-water should flow to the apparatus and should be as cool as possible — never over 100° F. If a supplemental live-steam jet be added to revivify the exhaust-jet, the injector will feed up to 150 pounds.

162. Advantages of the Injector. The advantages to be claimed for the injector are:

1. From its construction, it is cheap.
2. It is compact, and takes little space in proportion to its capacity for moving water.
3. It has few or no moving parts, and hence a small running cost for repairs.
4. It delivers the water warm to the boiler.
5. It has no exhaust-steam to dispose of.

163. Disadvantages of the Injector. The disadvantages of the injector are those which belong to the apparatus as a class, and those which belong to certain forms of the instrument only.

1. The impact of the jet on the water is not an effective method of pumping. As a pumping appliance the injector is about $\frac{1}{5}$ as efficient as the equivalent steam-pump. Its duty is about 2 million foot-pounds, against 10 million for the pump.
2. It heats its water with live steam, while to utilize the heat of its exhaust a pump with feed-water heater should show a superior coal economy of 12 per cent from the saving of waste heat.
3. The feed-water must be cool enough to condense the steam-jet, and this limit is about 100° F. Hence the injector cannot be used upon hot water.
4. It will not start with pressures much lower than those for which it was designed.
5. If it is not a restarting instrument, it will stop working after the limit of co-relation of feed-water and steam are passed. Then it must be started anew by the operator. Often when it has become hot by the interruption of the water-jet, it can only be restarted by complete cooling with water.

164. Water-gauges. Since the water rate or steam consumption of the engine is varying from minute to minute, and the pump rate can neither be constant nor uniformly varying, a piece of apparatus should be supplied such that the operator in charge can see whether the water is being supplied to it at the same rate that steam is being withdrawn from it, and regulate the supply of feed-water accordingly.

A secondary use of the water-gauge device will be to enable the attendant to see whether the quantity of water, or the level of water, in the boiler is falling so low as to expose heating surfaces to the action of gas or flame without water on the other side, and also whether the water-level is rising to a point at which the priming or mechanical entrainment of water would be feared. The danger from low water-level in the boiler, whereby overheating is caused, would be that any or all of three injuries would follow. First, a general overheating would cause a corrosion or wasting of the iron by oxidation. Second, if there were any lack of homogeneity in the plate from cinder or defective welds, a blister would be caused (Fig. 26); and third, when the overheated plate was cooled suddenly by filling the boiler with cool water the reduction of temperature would cause a sudden shrinking which, if not general and easily yielded to, might strain some joint of the boiler beyond its point of resistance when the boiler was already under considerable strain from the internal pressure. It is this last danger which makes low water so often a contributing cause to a boiler-disaster (paragrafs 191–208).

165. The Glass Water-gauge and Water-column. The simplest form of water-gauge is a glass tube about a foot long or a little over, which is connected by proper fittings so that its bottom shall be below the lowest water-line, and its top above the highest (Fig. 243). These fittings are screwed into the head of the boiler directly in boilers which are not set in brick, and the water will stand in the glass tube at the same height that it stands behind the head of the boiler. A simple inspection by eye is all that is necessary to see whether the water-line is above or below the normal. In boilers set in brick, where the head is not exposed, the gauge-glass will be carried upon an independent vessel which will be connected by proper pipes above and below the water-line respectively. This fixture is called a water-column (Fig. 244), and the water in it should stand at the same level as that in the boiler, and therefore make the gauge-glass show the water-level in all three vessels. Care must be taken in connecting the water-column that it shall be easily cleansed of deposit or other material which might clog it, and prevent its giving the same indications of level as are correct for the boiler itself. The connections shown in Fig. 244 will also illustrate those used for the high and low-water alarm columns (see paragraf 168), which are required by law as fixtures

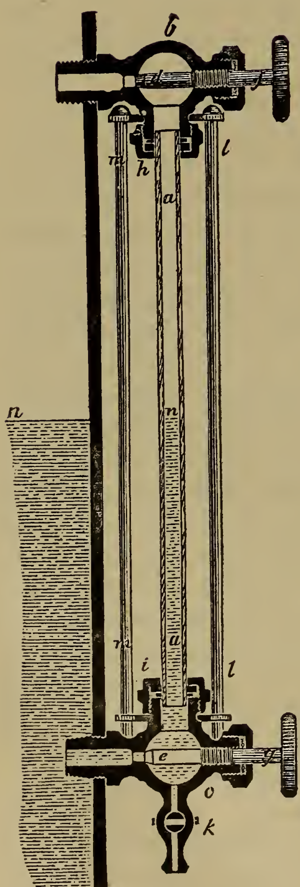


FIG. 243.

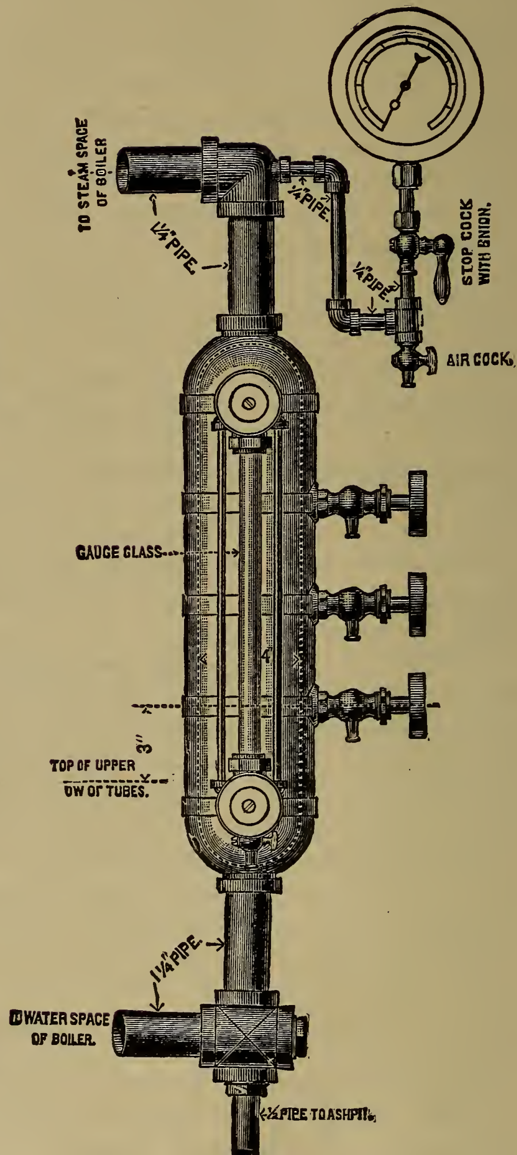


FIG. 244.

for boilers in some of the states, using either floats or fusible alloy disks to operate the alarm function (paragraf 168).

The advantages of the gauge-glass are its simplicity, its cheapness, and that it is easily observed.

The objections to it are

(1) Its fragility. It may be broken by accidental blows in spite of the brass-wire guards *ml* shown in Fig. 243 to prevent this accident.

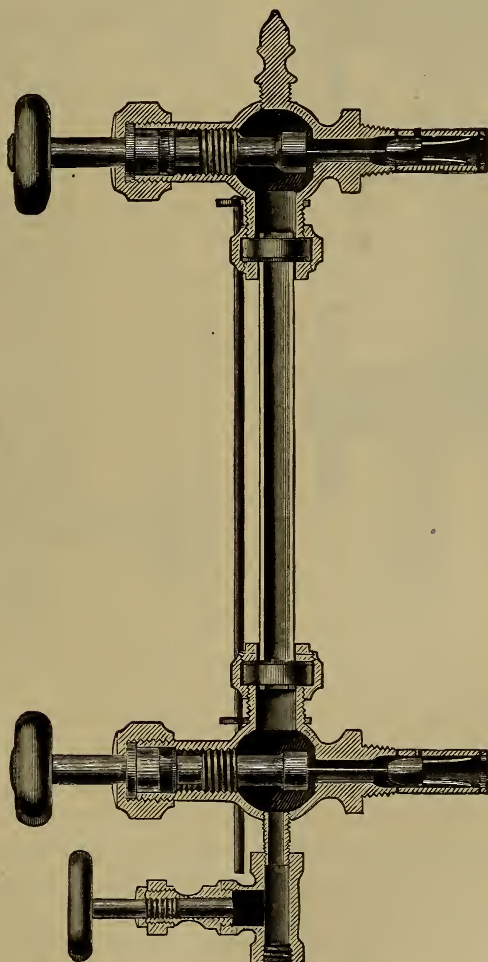


Fig. 245.

Furthermore, it is liable to break from a defective alignment of the two fixtures cramping the glass and causing it to crack, and also from a deterioration which the glass undergoes in service, particularly with waters containing any alkali. In locomotive practice the jarring tends to break the glass, and in confined spaces it is specially liable to accidental injury. When the glass breaks under pressure it will be apparent that two very powerful jets of hot water in one direction

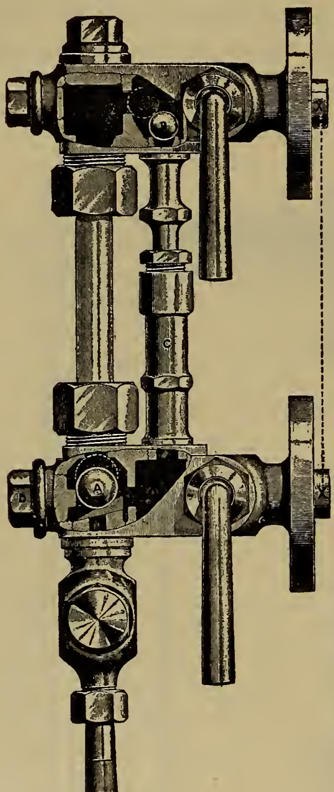


Fig. 246.

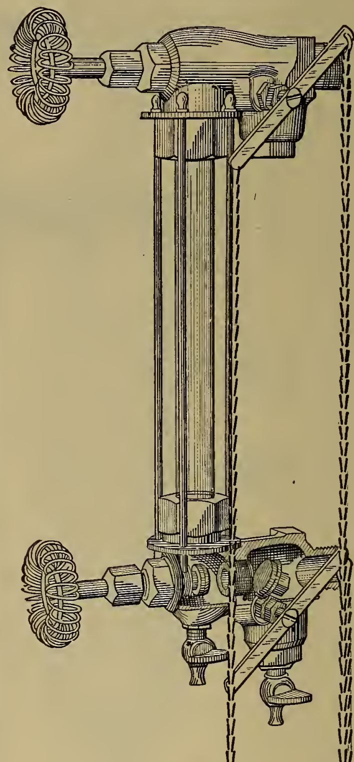


Fig. 247.

and of steam in another are thrown out from the fixtures, and under high pressure will fill the room instantly with hot and irrespirable steam. To prevent this difficulty a form of gauge-glass fixture has been devised in which the two attachments have an automatic valve opening inwards, and held in that position by a spring (Fig. 245) or by gravity (Figs. 246, 247). When there is equal pressure on both sides of the valve the opening is clear into the glass-tube. When the tube breaks, the outrush of steam and water will close the valves against the spring or gravity and thus automatically shut off the broken tube. Fig. 246 shows this device with a ball as the valve to be closed when breakage occurs. The pin on the end of the spindle of the hand-valve (Fig. 245) forces the automatic valve away and opens the connection through the glass when the spindle is withdrawn. The spindle should be withdrawn slowly, so that the pressure may equalize in the tube without drawing the valve to its seat. In Fig. 247 the valve is of the flap-type and while the chain pulls open the valve, the latter closes by itself.

(2) The gauge-glass may give false indications. This may happen because the somewhat tortuous passage in the lower fixture has become stopped with scale. This can be guarded against only by frequent opening of the blow-cock at the bottom of the fixture, so as to wash it out and be sure that the passage to it is free. The indication may be deceptive because the lower valve is closed, preventing the water from descending in the glass when it descends in the boiler. This is best guarded against by making the valves of the fixtures to be cocks operated with a handle whose position indicates whether the valve is open or closed (Figs. 246, 248). The third objection is the invisibility of the water in the glass when the water is clean. This can be obviated by having a colored strip made in the glass at the back which will be visible through the steam above the water, but which will be made invisible by diffraction caused by the water within the glass. The water usually is slightly colored, and it can be detected with care even if it is clean. The freedom of the connections between the gauge-glass and the boiler can also be insured when the boiler is steaming by observing the motion which the operation of steaming always causes in the water in the boiler from the presence of waves. Fig. 244 shows a standard water-column with the connections for cleansing at the bottom.

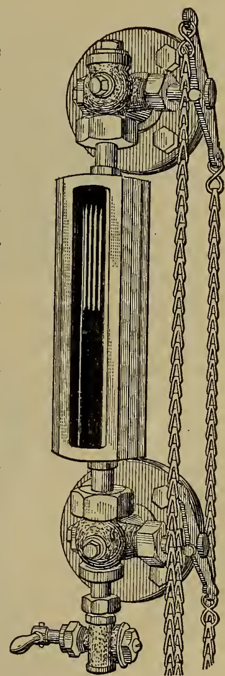


Fig. 248.

An Austrian invention known as the Klinger gauge-glass uses a flat, thick glass of trapezoidal section held in a brass frame. The back-plate behind the glass is fluted and at such angles for the sides of the ridges that when the water is in the space between the two the refraction which it causes sends the rays of light at such an angle that they do not reach the eye from the part where the water is. From the upper part the light is normally reflected (Fig. 248.) Hence where the water is appears black, while the back plate shows bright above the water. Breakage is a cracking at most and does little harm but the chain shut-offs enable the gauge to be promptly isolated from pressure.

166. The Gauge-cocks. By reason of the difficulties attaching to the gauge-glass, another form of water-gauge is usual without the glass or in addition to it. This involves the making of three or four openings into the boiler which can be opened by valves, and through which openings a sample can be taken from the level into which the openings are made. Such valves are called gauge-cocks or try-cocks, and will be fastened by screwing into the head of the internally-fired boiler, or into the column-pipe of the brick-set boiler. It will be apparent from Fig. 244 that if there are three of these cocks, the middle one should be at the normal water-level, the upper one above it, and the lower one below it. If the cock be opened above the water-line, it will permit dry steam to flow out, and its quality will be revealed to the eye and to the ear. The passage through the small opening will tend to superheat the steam, so that it will be an invisible gas for an inch or two from the nozzle, and will give a sound like the escape of a true gas. From the lowest cock water will be drawn, if it is below the water-level, but this water on reaching atmospheric pressure at the outlet will at once become saturated steam, which will be a white cloud from the very outlet of the valve, and will reveal itself to the ear by the difference in sound of the escape of water as compared with the sound of escaping gas or air. The middle cock, when the water-level is practically opposite its opening, will withdraw both steam and water, which will make the characteristic sputtering noise of air and water escaping through an outlet. The appearance to the eye will be the same as from the lower cock, since the water is the visible thing. Large boilers often have four cocks.

These gauge-cocks require to be opened by hand and therefore must be modified for boilers whose water-level is higher than convenient reach. Different forms of try-cocks have been introduced, operating either by a weight which has to be lifted to open them, or else they are made like cock-valves which can be easily turned by an extension which comes down to convenient reach. Figs. 244, 249, and 250 show types of gauge-cocks.

167. Float Water-gauges. — It has been sought for many years to find a satisfactory method of indicating the water-level by means of floats within the boiler whose position should cause the motion of a convenient indicator without. Some very early boilers had water-gauges of this class. The objection to them is the friction, which is a variable quantity and which acts upon the means used to transmit

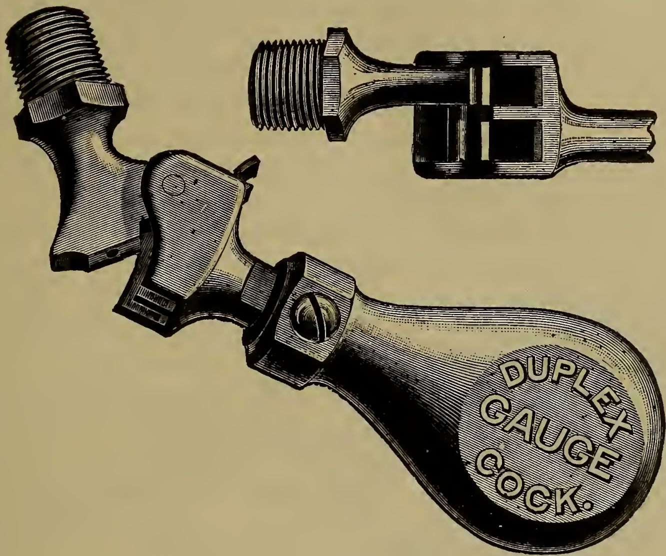


Fig. 249.

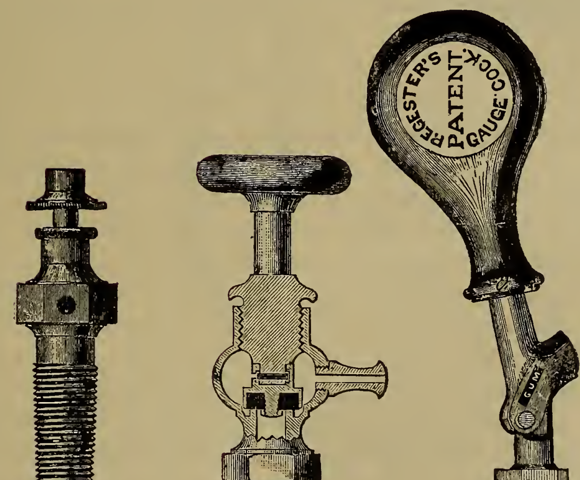


Fig. 250.

the motion of the float inside to the indicator outside the boiler. The float is apt to catch and be held by such friction, and to fail to indicate the real changes in water-level. The other difficulty is that no float material has been found which does not ultimately become affected by heat and pressure, so as to absorb water from the water in which it stands and to become either partly or entirely filled. It seems to be a general idea that it is not safe to put sole dependence upon float-gauges for these reasons.

168. Low-water Alarms. — It is quite possible, however, to use the float as a means of giving warning that the level in the boiler has been allowed to fall too low. This use is justified from the fact that they are not depended on, or should not be, for the normal working of the boiler, but are present as a safeguard if they fulfill their purpose in emergency. The usual plan is to allow these floats to control a valve where steam shall be admitted to a whistle. The normal rise and fall of the float within the limits of safe working is without effect on the whistle-valve, but it opens it if the float is too high or too low (Fig. 251). A similar device can be arranged, depending on the difference of expansion of metals in steam or in water. When a spindle is surrounded with water it is short enough to hold a valve shut, but when the water falls below the opening into the tube within which that spindle stands, the expansion of the spindle will open the valve.

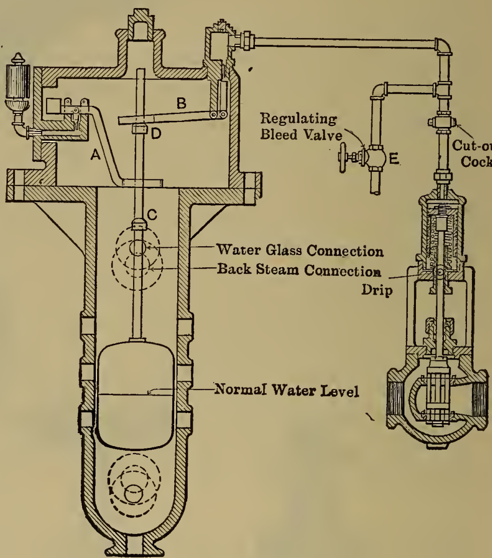


Fig. 251.

169. Fusible or Safety Plugs. — As an additional safeguard to prevent injury from low water it has been the custom of many engineers and of some state legislation to demand that a plug shall be inserted into the boiler, at or near the dangerous low-water line, made of Banca tin or of some of the cadmium alloys, which have a relatively low fusing-point. When the disk or plug of such fusible metal is covered with water the heat is transferred so rapidly that it should not melt. When the water leaves the plug, the lowered specific heat of steam prevents the rapid withdrawal of heat, whereupon the plug melts, and steam blows out through the opening to give warning of trouble. Fig. 252 shows a construction of such fusible plug in which a brass shell is fitted with a core or disk of fusible metal. The objection to such fusible plugs is, first, that the melting-point of most of these alloys changes with time and is not always certain. Secondly, when covered with a crust of boiler-scale they may not be properly cooled by the water, and will fuse when everything in the boiler is normal. On the other hand, they sometimes fail to act, either from the first difficulty or from some unknown cause, and in any event, when blown out, it is annoying to replace them. The location of such fusible plug in a tubular boiler is shown in Fig. 178.

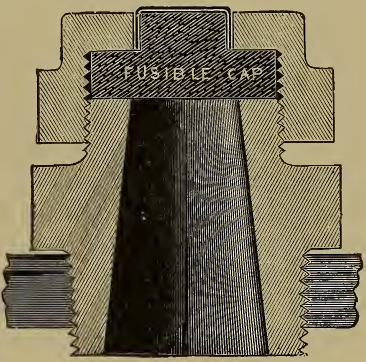


Fig. 252.

The fusible-plug alloy has been applied as a safety or low-water alarm by inserting a disk or diaphragm of metal of this fusible quality in a pipe which admits steam to an alarm-whistle. When the pipe is sealed by the boiler-water, the plug does not become hot enough to melt. When the fall of the water-level permits the water to flow out of the tube, steam replaces it and has a sufficient temperature to melt the plug and blow the whistle.

170. Automatic Feeding Apparatus. It has long been sought to arrange a mechanism which should be operated automatically, and as the level of the water in the boiler might vary, to have this change of level operate the feeding mechanism without human intervention. If automatic feeding in a reliable form could be combined with automatic stoking, the labor of the fire-room would cease to be manual and become supervisory only.

It has been sought to obtain automatic feeding by several methods. They all make use of the direct-acting pump, and provide that the variation in water-level shall operate its steam-valve, so that the rise of the water-level above the normal shall shut off the pump, and a fall below shall turn on more steam and speed it up. This has been secured, first, by the expedient of having the steam-valve operated by the pressure of a column of water against a flexible diaphragm. When the water-level was normal, or above it, the bottom of this column of water was sealed in the water and thus kept full by the steam-pressure. When the water fell below its opening into the water-space of the boiler, the column emptied its water into the boiler, and thus withdrew its pressure from the diaphragm, which yielded to an exterior weight and opened the valve. A second method has been to insert in a pipe a rod of some metal with a high coefficient of expansion. When the outlet from this pipe into the boiler was below the water-line, water was forced up into the tube, and its high specific heat and radiation kept the rod cool. When the pipe was emptied by the fall of the water-level in the boiler, steam replaced the water around the rod, caused it to lengthen, and turned on the valve. A third plan has been by means of floats whose rise and fall within the boiler transmitted motion outside through a proper stuffing-box to slow down or start up the pump. The newer and better form of this to avoid the stuffing-box and its uncertainties of resistance, is to inclose the whole apparatus in a casing, and have a float open and close a valve leading to the controlling piston or diaphragm (Fig. 251). This piston operates the pump valve. When the water rises in the column and lifts the float, the pin valve is opened and the spring is overcome and the pump slows down. When the float falls, the pin valve closes, the spring forces steam in the connecting pipe out through the bleeding valve and the pump speeds up. But the piston must never stick. A fourth plan, which has been used since the development of the electric motor for pumping, has been to make the rise and fall of the water-level operate a float to throw out or in a switch or a resistance-coil, in the circuit driving the pump, whereby the action or speed of the pump should be made to vary.

The fifth method has been to have the feed-pump operating continuously, and to arrange that its suction should draw from the supply of fresh feed-water only when the water-level was below a certain point, determined as before by the seal of a pipe by the water in the boiler. When the water was above the opening of the sealed pipe the pump simply circulated the boiler-water without drawing in a fresh supply. A modification of this principle used in flash boilers (paragraf 100) is to have the pressure in the boiler act upon a flexible diaphragm controlling a valve in the pump discharge. A spring resists the steam pressure, and when the pressure exceeds the determined point the feed is by-passed into the suction. This must work in connection with automatic control of fire-temperature, so that too little water shall cause excessive superheat and shut down the fuel supply, while a slight excess of water causing excess of pressure energy shall operate the by-pass and prevent flooding.

The idea of an automatic or magazine feed-pump is a very old one, and will be found applied to early boilers (Fig. 697 A in Appendix), for which the feed was supplied from an elevated reservoir at such a height that the light pressure within the boiler could not balance the water-column, but water would flow in when a valve was lifted by the descent of the water in the boiler through the operation of a float.

The objection to the automatic-feed principle as thus far applied has been that it is not entirely to be depended upon, or is not automatic in the true sense. The apparatus will work satisfactorily for six months or a year, and then, failing suddenly, causes a disaster or compels such repairs to the plant as to be a serious offset to the previous advantage.

171. The Steam-gauge. It is entirely possible to operate a low-pressure boiler without an apparatus to indicate the pressure in it. Boilers were so operated before gauges in their modern form were cheap and portable. The relief valve (paragrafs 174, 377) could be relied on to open if the pressure went too high; the engine slowed down if the pressure fell too low. But this is neither safe nor economical with high pressures; it is not economical because it is important that the fireman in charge of a boiler should know whether the fire is supplying heat-energy to the water faster than it is being withdrawn in the form of steam, or slower, or just at the proper rate. The most convenient indications of the heat-reactions are given by an appliance which shall record the pressure in the boiler, since if the pressure is rising, heat is being stored by the water, and if the pressure is falling, heat is being given off faster than it is being supplied. Such an apparatus, therefore, becomes a reliable guide to the firing process. If the pressure is stationary, the water-feed, draft and heat supply from fuel are rightly related to each other; if the pressure is rising the rate of combustion controlled by the draft needs lowering, or this is the time to pump in an extra cooling feed supply; if the pressure is falling, the draft needs stimulating and perhaps the feed-pump may be putting in more cold water than the boiler can digest with the fire in that condition. In locomotives or motor vehicles, the steam-gauge is full of information as to whether with that load on that grade with that condition of track or roadway the speed then in use can be maintained, or can be increased or must be reduced. This strictly and properly is the principal function of the steam-gauge. A secondary but sometimes very important function is to indicate whether the pressure and the heat-supply are rising so rapidly as to endanger the structure from excess of internal pressure.

The first and simplest form of steam-gauge was a U tube or manometer. The size which this appliance has to receive with high pressure precludes its use as a pressure-gauge, although it remains the standard for all of its more convenient substitutes. It has been found most convenient to replace the weight of the mercury-column by a spring which shall undergo a known deformation for each pressure, and which shall indicate its deformation by the movement of a needle over a dial. Such spring may either be a flat disk or diaphragm, Fig. 253, or it may be a hollow brass or steel tube which is bent into an arc of a circle, and the sides permitted to come together by this bending, so that the

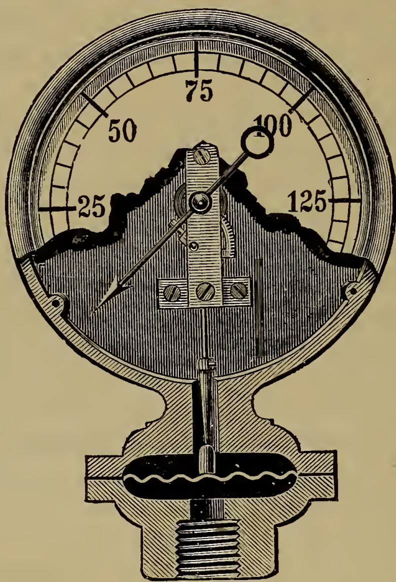


Fig. 253.

section of the tube is that of a very much flattened oval. When pressure is admitted on the inside of this tube the parallel sides tend to separate, and in separating they must increase the radius of curvature with which the tube was bent within the circle. The tendency to straighten out by internal pressure can be indicated by a multiplying gearing which shall cause an indicating-needle to traverse an arc. Fig. 254 shows the ordinary Bourdon gauge with flattened tube availing of this principle. Pressures below that of the atmosphere can be observed by similar appliances. The aneroid barometer is a vacuum-gauge.

Fig. 255 shows a form of Bourdon spring in which the two arms have been shortened so as to prevent their shaking disagreeably when exposed to the jarring in locomotive service. Sensitiveness or a considerable motion of the needle is secured by a double connection to the multiplying device. This arrangement is also of advantage for use in gauges which are to be exposed in portable boilers to temperatures below freezing. The two arms can be drained of the water which they will contain from condensation, whereas the form of Fig. 254 will always hold the water

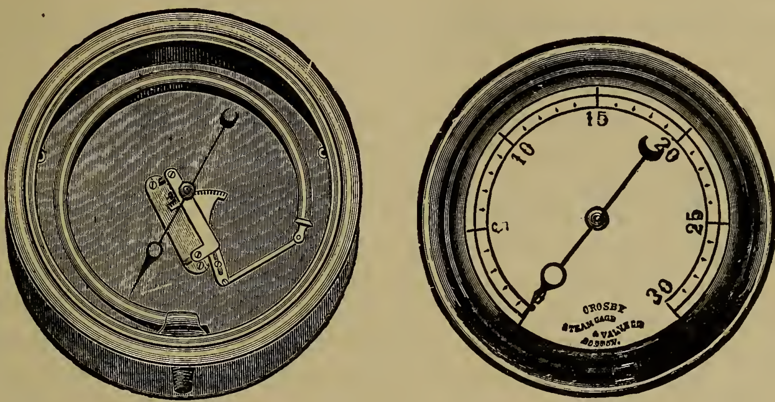


FIG. 254.

which has once entered it. It is usual to secure the separation of hot steam from the gauge-spring by an intervening water-column, for which provision is made by connecting the gauge to the boiler through a

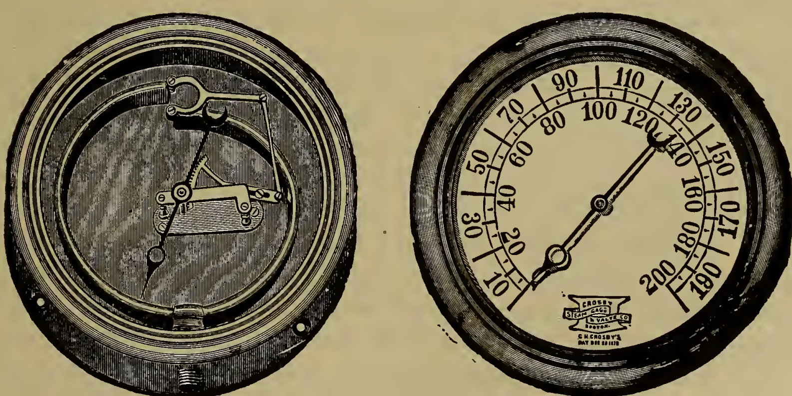


FIG. 255.

U tube; which will make a siphon (Fig. 244). Fig. 256 shows a device which produces the same effect as a water-seal. Provision, however, should be made for draining this siphon both for cleansing indoors and to prevent freezing without.

The defects of such spring-gauges are:

(1) The spring loses its original resilience by use and heat. If this is due to a permanent change in the structure of the material, the gauge is useless.

(2) Rust and improper treatment may cause the friction of the needle mechanism to prevent the recording of the full pressure against the spring.

Fig. 256.

(3) The needle may slip so as to change its relation to the spring, which will cause the gauge to record permanently above or below its proper pressure.

172. Standardizing or Calibration of Steam-gauges. To ascertain the accuracy of a gauge it is to be compared with a standard. It is usual to compare it in practice with a test-gauge, which is one kept specially for this purpose and not exposed to the conditions of service. The test-gauge, however, requires to be itself standardized, and for this purpose three methods are usual. The first is to connect the gauge which is to be tested upon a pipe or similar apparatus within which hydraulic pressure can be admitted to come also upon a valve closing an opening which is made exactly one square inch of area. By loading this valve with known weights, and observing the pressure recorded by the gauge when the water-pressure lifts the valve and weights, the gauge is calibrated. This can also be done with a piston which can be loaded with known weights moving without friction, or with a minimum friction, in a cylinder.

The ultimate standard is the mercury-column. The gauge to be tested is connected on the same pipe which opens into the short leg of the mercury-column, and the pressure recorded in the gauge when the mercury in the long leg stands at the heights which correspond to the real pressure indicates its error or its truth.

173. Recording-gauges. It is convenient to have a continuous record of the variations of pressure in the boiler or in the other appliances in which pressure is to be observed. It is very simple to connect the spring or piston mechanism of the gauge or testing appliance to a link carrying a pencil-point which shall move in one direction by variation of pressure over a piece of paper which is made to travel in a direction at right angles at a known rate by means of a clock. The pencil-point traces a line as the pressure directs, and the intensity of that pressure is measured by the vertical ordinates on the diagram, while the time at which it occurred can be found by a horizontal measurement. The same result may be compactly attained by giving a radial motion to the recording pointer receiving pressure, while the card or receiving paper moves circumferentially under it by a clock movement. Such records are not only a check upon the fireman's fidelity and competence, but are means of avoiding controversy as to responsibility where pressures are factors in the satisfactory operation of any plant.

174. The Relief or Safety-valve. The steam boiler being a reservoir of heat energy in the form of pressure, it is plain that an attempt to store too much heat will produce such a stress in the material enveloping the steam and water as to exceed its resistance to rupture from such internal pressure. A relief valve opening outward and held shut by an adequate and adjusted force can easily serve to secure safety for the vessel by guarding against such accumulation of pressure energy. It does not necessarily secure safety unless some steps are at once taken to check the flow of heat and energy into the water; that is, a valve of this relief-type is not a safety-valve in the true sense, unless as it lifts it has sufficient area to allow all the steam to escape through the opening, which will be made as fast as the boiler can make steam with all other outlets closed. In other words, the pressure in the boiler should not be able to rise above that for which the valve is loaded, even if all other outlets are closed and the fire burning with its normal or even its maximum capacity for steam-making. Comparatively few safety-valves are of this capacity, for reasons of cost and convenience, but the presence of such a loaded valve acts as an alarm to give warning of the passage of the known pressure-limit, so that means may be taken to stop the generation of steam and an accumulation of pressure.

Furthermore, as the pressure does accumulate under the valve when open and blowing, it has a tendency to lift the valve higher and enlarge the outlet in many of its forms. Most of the legislative requirements concerning boilers compel a safety-valve of an accepted construction.

175. Forms of Safety-valve. The safety-valve for boilers is likely to be in one of five forms. Historically the first, now practically not used in America, is the method of weighing the valve down by a direct weight, resting on its back or suspended to it from below (Figs. 96 and 98). The difficulty of this form is that with large valves and high pressures the weight to be used becomes considerable and inconvenient. In English practice, where the direct weight is still preferred, the inconvenience is mitigated by using a number of smaller valves to secure the necessary area and subdivide the weights.

The second form is to replace the weight with a spring whose intensity can be graduated. This avoids the bulk of the direct weight, but is open to the serious objection that as the valve lifts the resistance of the spring increases. The springs are also liable to corrosion, which makes them stiff. This form was used in many cases where jar from motion was to be experienced, but has practically been entirely superseded by the fifth form.

The third is a very frequent type, in which the spindle of the valve is held downwards by the action of a lever carrying the resisting weight with a long arm, while the effort of the valve to overcome the weight has but a short lever-arm (Fig. 257). This does away with the inconvenience of a great weight, and makes adjustment of the pressure

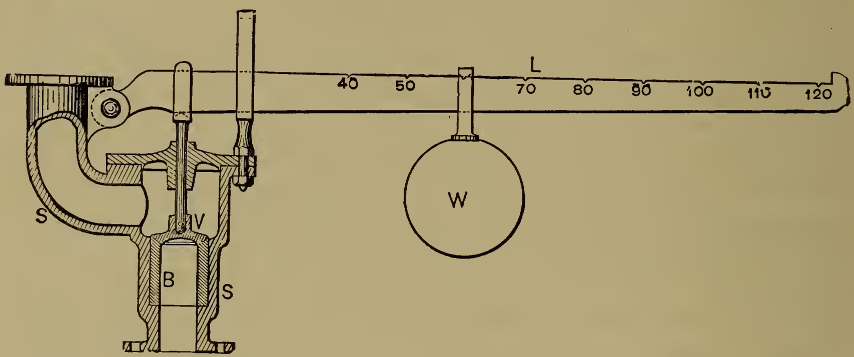


FIG. 257.

to hold the valve shut both easy and rapid. It is probably the most widely prevalent form of safety-valve for stationary boilers. The objection to it is the tendency of the lever in its rise to cause the valve to become jammed from the oblique motion around the fulcrum of the lever. It may also be urged as an objection to it that the weight may be so easily increased by sliding out the regular weight or by hanging other weights upon the lever. It is also easy to stop the valve from operating by wedging it so that it cannot open under any pressure whatever. The lever-valve construction lends itself to a desired use in river-boat practice. By attaching a rope or chain to the end of the lever and leading it up over a pulley, with a weight on the free end, that weight acts negatively and takes weight off the valve, and lightens the pressure at which the valve will open when the engine is at rest. By hanging this second weight up so as to leave the rope or chain slack by which it is attached to the lever, the entire counterweight comes on the lever, and full pressure is restored for regular running.

For locomotive use before the fifth form was introduced it was usual to replace the weight by a spring which acted at the end of a lever to hold the valve down. This spring was arranged so that the tension upon it could be varied by the engine-runner. It never became widely used outside of locomotive practice.

The fifth form is what is called the pop or reaction safety-valve, which is practically universal in locomotive practice, and is widely extended elsewhere. The principle of the pop-valve is that, as the valve proper lifts from pressure, the escaping steam, instead of passing out directly from under the valve, must find its way out, after undergoing a change of its direction in an annular groove formed in the valve outside of its inner bearing. The force due to the reaction of the steam in escaping adds an additional effort to lift the valve, increases the opening thereby, and with a given loading the valve will remain open until the pressure within the boiler has fallen perhaps 5 pounds below that at which the valve lifted. The additional area exposed to pressure when the valve lifts causes it to open with a sudden motion which has given it its ordinary name, and it also closes suddenly when the pressure has fallen. Figs. 257 and 258 show types of lever and of pop safety-valve.

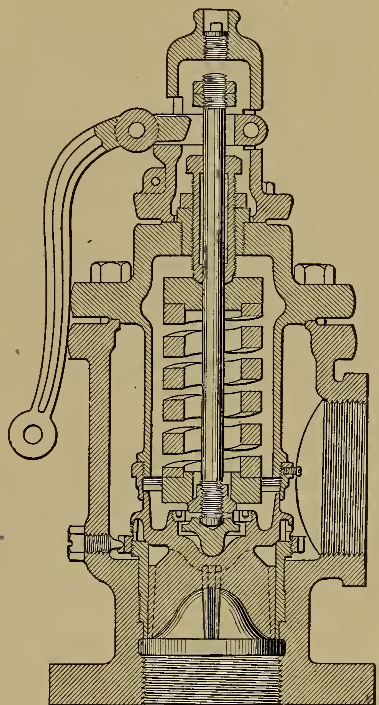


Fig. 258.

A failure of the safety-valve is often due to corrosion either of the valve upon its seat or of the guiding-spindle in its guides. The safety-valve, therefore, should be frequently lifted by hand in order to be sure that corrosion has not made it worthless, and a further safeguard is secured by the use of metals in the valve or seat which do not rust together. Nickel has been applied for valve-seats with success by reason of its being a non-rusting metal, and certain bronze alloys are used for the same reason.

176. Safety-valve Formulæ. "Experiments on Flow of Steam," by R. D. Napier (see *Engineer* of London, September to December, 1869), gave for discharge from a conoidal nozzle per inch of area per second, provided the pressure into which steam flows is less than three-fifths of that in the boiler from which it flows,

$$w = \frac{p_1}{70};$$

or, for an area of a square inches,

$$aw = \frac{p_1 a}{70}, \quad \text{whence } a = \frac{70w}{p_1},$$

in which w should be the entire weight of water which the heating-surface can evaporate in one second of time. Some other formulæ for safety-valve areas are

$$A = \frac{\begin{bmatrix}\text{grate-area}\\ \text{in sq. ft.}\end{bmatrix} \times \begin{bmatrix}\text{rate of combustion in}\\ \text{pounds per sq. ft.}\end{bmatrix} \times \begin{bmatrix}\text{pounds of water}\\ \text{evap. per pound}\\ \text{of coal per hour}\end{bmatrix} \times 70}{3600\,P}$$

$= \dfrac{\text{coal burned per hour}}{5.14\,P}$ (assuming that 10 lbs. of water are evaporated per pound of coal); or

$$A = \frac{\frac{1}{2}W}{p + 10};$$

or $\quad A = 1$ sq. in. to 25 sq. ft. H. S.;

or $\quad A = 1$ sq. in. to 1 sq. ft. G. S.

The lift of a safety-valve is usually a very small quantity. The standard experiments on a conical seated valve four inches in diameter are those of Burg in Vienna. His values were, with a pressure below the valve of

	12	20	35	45	50	70	90
the lift of the valve in inches was	$\frac{1}{36}$	$\frac{1}{48}$	$\frac{1}{54}$	$\frac{1}{65}$	$\frac{1}{86}$	$\frac{1}{168}$	$\frac{11}{168}$

The area of the opening is the cylinder or cone included between the valve and its seat when the valve is open. Its area will be the circumference of the valve multiplied by the foregoing small lift.

177. Computations for the Lever Safety-valve. The equation for a safety-valve of lever type is

$$(P \times A) \times l = WL,$$

in which A is the area in square inches; P the pressure on each square inch; W the weight in pounds on the long arm of the lever, and strictly should cover the weight of the lever-arm, applied at its center of gravity; L is the length in feet or inches from the fulcrum to the weight; and l is the length in the same unit from the fulcrum to where the spindle of the valve presses up against the lever. The U. S. law compels l to be more than 4 inches, and L cannot be over 40 inches.

178. Concluding Comment. In the foregoing chapters the power plant in its generating function or in the boiler-room has been considered in its simplest and most uncomplicated form. Additional apparatus for increasing economy and lessening cost of production will be taken up in a later chapter under Boiler-room Auxiliaries. At this point the designer or seller or writer of the specifications for the plant turns it over to the buyer or owner or operator to be run. The problems facing such owner and operator will be discussed in the next chapters under the headings of care and management.

CHAPTER XI.

CARE AND MANAGEMENT OF BOILERS.

180. The Firing. The firing of a boiler-furnace is to be done in accordance with the general principles of combustion, and the application of these to fuels which differ so widely makes it difficult to give anything but the most general suggestions. References also have been made in other connections which bear on this subject. The three usual methods of firing are the spreading method, the side-firing or alternate method, and the coking method. The spreading method is to keep covering the fresh and incandescent coal on the grate with thin layers of fresh coal thrown in at short intervals. This is the usual and most successful method with anthracite, where best results are secured when the fire is least disturbed. Side-firing is to divide the furnace into two halves lengthwise and charge the fresh fuel on one, while the other is in its best state of incandescence. This has been referred to under the double Cornish or Lancashire method as a means of keeping up the temperature of combustible gases, and is especially applicable to bituminous coal. The coking method is to divide the fire crosswise instead of lengthwise, and charge the fresh coal containing gas at this front part or on the dead-plate, and push it backwards when the gas has been distilled by the radiant heat of the fire behind it. The thickness of the fire will be determined by the draft and the quality and size of the fuel. Anthracite fires will be as a rule thinner than bituminous, and small coal will require a thinner fire than the larger sizes. With thin anthracite firing from 4 to 8 inches is accepted good practice, and with bituminous coal from 6 to 14 inches.

The starting of fires in the boiler-furnace is also a matter which varies with the fuel and the conditions of draft. If the chimney is reluctant to draw from its being cold, it can be helped by starting a little wood-fire in the base of the stack and beyond the boiler-setting, so as to create the first action of the chimney before the resistance of the setting is interposed. It must be remembered that anthracite ignites reluctantly and large quantities of wood are necessary to get it well started.

181. Cleaning Fires. The interval between cleaning of fires will depend on the rapidity of the combustion and the quality of the fuel with respect to ash. With anthracite fires it is usually only necessary to clean fires in stationary practice about four times in twelve hours. With bituminous coal it must be done more frequently, and often the best results are obtained by pulling the fire about at short intervals, which is fatal to the satisfactory working of an anthracite fire. The cleaning is done by means of slice-bars which break up the clinker and separate the combustible from the incombustible matter, and after the fire is thoroughly broken up the aggregations of incombustible matter are removed by a rake or hoe. What remains is then spread evenly over the grates, and a fresh charge of fuel thrown on the fire. The ashes and clinker drawn out from the furnace will then be extinguished and cooled by a jet of water from a hose, and will then be removed. Care must be taken in handling the extinguishing water that it should not strike by accident any of the hot castings about the ash-pit, which it would be certain to crack.

182. Banking Fires. It is usually the least trouble and expense to bank the fire at the close of the day, or when the fire is to be kept over for some hours during an interval of inaction. After the fire has been cleaned, what remains on the grates, instead of being spread evenly, is piled against the bridge-wall and upon the back half of the grates, leaving the front part bare. Fresh coal is then charged in a thick layer over the banked fuel, and the fire is left with the ash-pit doors closed, the fire-door open with anthracite fuel, but closed with bituminous, and the damper closed, or nearly so. The closure of the ash-pit and the access of cold air above the fire make the ignition of the bank of fresh coal very slow, so that several hours will elapse before it has become ignited, and even then it burns slowly and not actively. At the end of the time, determined by the quantity of fresh coal used in banking, the fire is cleaned and spread, and is ready for a new campaign.

183. Regulation of the Fire and Pressure of Steam. The regulation of the fire is done by controlling the access of air to it whereby combustion is stimulated or checked. The closure of the ash-pit and the damper check the fire, and to open them stimulates it. The fire-door opening into the furnace above the grates is a further means of controlling the fire in part, since by opening it cool air comes in to lower the temperature in the fire-box, and without passing up through the coal it does not stimulate combustion. The cool air further checks the making of steam by cooling the products of combustion, and acts by contact as a cooling medium passing over the heating-surface and through the tubes. It has already been noticed that this is not a desirable thing to do, by reason of its effect on the metal of the boiler, but it is an efficient method of control.

The steam-pressure is controlled by the fire principally, but it can also be regulated by the use of the feed-water. The introduction of cool water cools the contents of the boiler, and checks partly or altogether the formation of steam. The presence of additional water, furthermore, makes additional material to absorb heat-units, so that

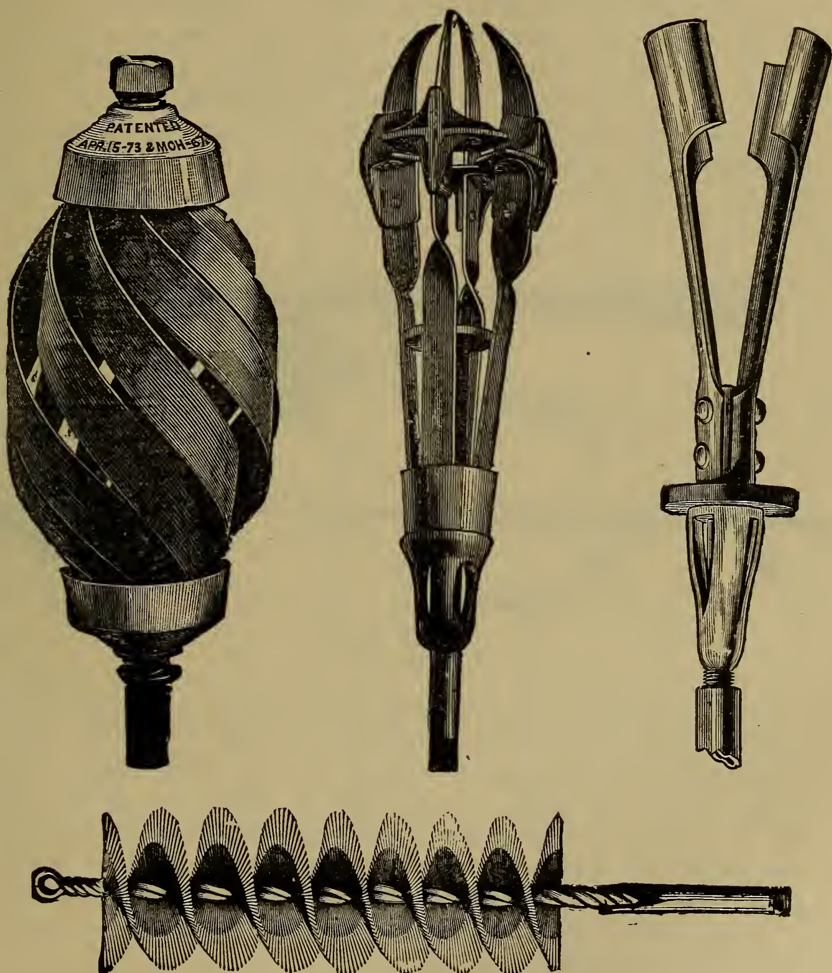


Fig. 260.

great skill is to be shown when known variations for demands for steam are to be expected, by so controlling the times for feeding and the amounts fed that this storage of heat shall be utilized to its best extent.

184. Cleaning the Heating-surface outside. The metal of the heating-surface in most boilers becomes coated with a scale of some non-conducting character caused by the light dust and ashes attaching themselves to the plate, and particularly when tarry matter is present in the products of combustion. Within the tubes of a tubular boiler a deposit of dust and ashes will also take place, perhaps choking the tubes, or at any rate rendering them less efficient (Fig. 8). The cleansing of these exterior surfaces from the soft scale is done either with scrapers, or by brushes (Fig. 260), or by means of a jet of high-pressure steam or air directed upon the surface to be cleansed from a nozzle (Fig. 261). The tube brush or scraper is passed through the tube and scrapes the surface clean, but the steam-jet acting at high velocity seems to have a special cleaning effect, and is used either independently or in connection with

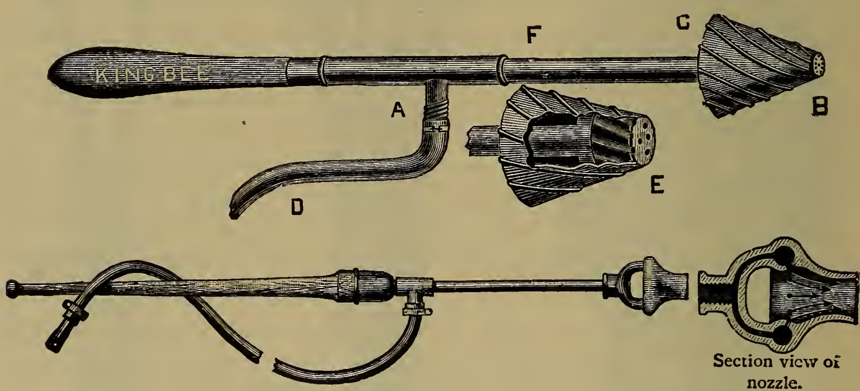


Fig. 261.

the scrapers. Special forms of such steam-cleaners are used, in which the jet receives an annular form and a spiral motion (Fig. 261), but very good results are obtained by means of a simple short-length of pipe coupled to the steam-space in the boiler by means of a flexible hose. The settings of sectional boilers usually have openings made through their walls in which pipes are built, and through which pipes the cleansing jet of steam can be inserted at different levels, and so keep the surfaces up to their efficiency. The tarry deposit sometimes refuses to be moved by the steam-jet, when, of course, scrapers must be used.

Locomotives are more often cleaned by air-jet, because of the danger to the person cleaning if any accident should occur to the hose joints while he is confined within the fire-box. With steam in use under these conditions, the cleaner would be burned before he could escape, while with compressed air there is no such danger.

185. Boiler-scale or Incrustation. It will be apparent that any solid matter in solution or present in suspension in the water fed to a boiler

TABLE X.

MINERAL MATTER IN SOLUTION IN FEED-WATERS. GRAINS PER U. S. GALLON.

	Charles River.	Long Pond.	Schuylkill River.	Lake Michigan.	Mississippi River.	Missouri River at Council Bluffs.	Mississippi River at Keokuk.	Riverside, Ill. Well.	Downer's Grove, Ill. Well very Bad.	Rockford, Ill. Artesian Well.	Dead-sea Water.
Silica (SiO_2)	....	....	0.0800	0.306	0.863	1.522	1.190	0.484	0.741	0.624	....
Calcium carbonate ($CaCO_3$)	0.1610	....	1.8720	4.461	6.870	8.847	4.673	5.237	17.091	8.141	....
Calcium sulphate ($CaSO_4$)	0.2624	....	....	0.309	0.484	2.251	....	0.776	14.037	....	....
Calcium chloride ($CaCl_2$)	0.0420	0.308	....	....	....	....	....	....	....	....	29.220
Magnesium carbonate ($MgCO_3$)	0.0399	....	0.3510	2.200	4.006	1.866	0.857	4.023	....	7.336	....
Magnesium sulphate ($MgSO_4$)	....	0.1020	0.0570	....	0.338	3.505	....	....	25.422	....	....
Magnesium chloride ($MgCl_2$)	....	0.0764	....	....	....	....	....	....	....	....	50.950
Magnesium bromide	....	....	....	....	....	....	....	....	....	....	7.950
Sodium carbonate (Na_2CO_3)	....	....	....	....	....	....	2.129	....	....	....	....
Sodium sulphate (Na_2SO_4)	0.3816	....	....	....	....	....	....	....	....	0.554	....
Sodium chloride ($NaCl$)	0.1547	0.0323	0.1470	0.225	....	....	0.100	....	....	0.362	78.650
Potassium carbonate (K_2CO_3)	....	....	....	....	....	....	....	....	....	....	....
Potassium sulphate (K_2SO_4)	....	....	....	0.283	....	....	0.430	....	....	0.525	....
Potassium chloride (KCl)	....	0.0380	....	....	....	....	0.489	....	....	....	....
Ferrous carbonate ($FeCO_3$)	....	....	....	0.029	....	....	2.682	....	....	....	....
Alumina (with ferric oxide)	....	0.0800	....	....	0.233	0.233	....	0.146	0.192	0.087	....
Organic matter, etc.	0.5291	0.5295	1.6436	....	....	....	....	....	....	....	....
Suspended mineral matter	....	....	....	....	....	....	1.802	....	....	....	....
Suspended organic matter	....	....	....	....	....	....	2.455	....	....	....	....

will remain behind in the boiler when the water is evaporated, because the steam will carry none of this material with it. If the salts in the water are of a soluble character, the process of evaporation will tend to concentrate the solution which remains in the boiler, and if they are insoluble they will gradually fill up the water-space. Concentration of soluble solutions is prevented by the process of blowing down (paragraf 153), but for the removal of the insoluble matter some special procedures or appliances must be used.

The solid matter which gives trouble inside of boilers is introduced either in suspension or in solution. When it is in suspension the water is called a muddy water, and the proper procedure is to filter the feed-water for the removal of such suspended matter. This gets rid of the difficulty from this mud outside of the boiler altogether by preventing it from getting in. If this is inconvenient, the mud-drum will be a necessary feature of the boiler, and blowing off the accumulations must be practiced at frequent intervals.

A class of salts enters the boiler in solution, but is precipitated as an insoluble precipitate on boiling. These are among the most troublesome, because they are really formed within the boiler itself, and consequently can only be prevented from getting in by chemical reactions of some magnitude. The salts of this class are the carbonates of the alkaline earths, lime and magnesia, and the sulphate of lime, which is the most troublesome of all. The feed-waters which are usually drawn from fresh-water sources are not likely to contain much sodium or potassium, which form the soluble salts, but in sea-water the chloride of sodium and magnesium, which are both soluble, are elements which give it its salty taste. Silica, alumina, and organic matter are to be found in some of the Western waters, or where the wash of surface-water may have come into the source. Typical water analyses from various sources are given in Table X.

Great difference in the difficulty of the problem of dealing with boiler-scale results from the form which the scale takes. The carbonates of lime and magnesia are a mud — white or grayish in color when pure. They have no cementing tendency and can be treated like suspended impurities. Silica and the sulphate of lime, however, are crystallizing bodies in the water which form into a hard, adhesive crust, and not only this, but they have the property of causing the carbonate scales to crystallize with them and add to the extent and thickness of the adhesive coating. The following table shows the properties of these most prevalent scales, and their degree of solubility at various temperatures. They enter the boiler as the bicarbonate, which is soluble, but on boiling one part of carbonic acid is expelled, and the protocarbonate which remains is the insoluble form. The carbonate of magnesia is a light, flocculent powder which usually floats at or near the surface of the water, rather than sinks to the bottom. Organic matter is apt to act in the same way, especially when it is of a vegetable character.

TABLE XI.

Salt.	Temperature. Degrees Fahr.	Authority.	Parts by Weight of H_2O to dissolve 1 Pint of Salt.	Grains to Gallon.
$CaCO_3$ and $MgCO_3$	62	Bucholz	41,600 to 62,500	1.4 to 0.9
Do...........	212	Do	16,000 to 24,000	4.25 to 2.75
Do...........	285 to 300	Cousté	Insol.	0
Do...........	62	B.	461	126
$CaSO_4$.............	95	Regnault	393	178
Do...........	212	R. & B.	460	126
Do...........	290	R.	Insol.	0

186. Inconveniences due to Boiler-scale. The presence of the solid matter in the water of a boiler may do harm in one or more of four ways.

(1) If it forms hard and solid over the heating-surface, it adds a non-conducting thickness to the evaporating-surfaces, so that an excess of fuel is burned to make the required quantity of steam.

(2) This non-conducting covering causes the metal of the boiler to be overheated because the water does not cool it. This may produce an injury which is general or local. The general deterioration all over comes from an oxidation of the plate on the outside, because its high temperature makes the oxygen reactions more rapid than they would be if the plate were cool. The local injury comes from the presence of a thickness of scale at points exposed to intense action of fire, whereby they become practically red-hot and softened, so as to yield under the internal pressure. Bags or blisters result from this trouble, which is aggravated if grease has become mixed with the scale at the point in question (Fig. 26). A lump of scale is sometimes carried by circulation and dropped in a special place, and becoming attached there, a local overheating begins underneath it.

(3) The scale which crystallizes, accumulating in feed-pipes, blow-off pipes, water-gauge connections, and the like, is occasion for trouble in the use of these appliances (Figs. 237 and 244). In sectional boilers, besides the annoyance from the first two causes, the presence of scale impedes the rapid circulation, and increases the troubles which are met from this difficulty (Chapter VI).

(4) The presence of the floating mud or flocculent precipitate causes the boiler to prime, because the steam-bubbles must force their way

through the scum at the disengaging-surface, and in doing so water follows upwards with the steam, and is entrained mechanically through the steam-pipe (paragraf 49).

The ill effects or injuries caused by scale are to be mitigated or avoided by methods which can be grouped under three heads. The first is the removing of the scale which is allowed to form. The second is the preventing of the solidification of the scale either by changing its character or by other means, and then causing its removal by methods of the first class. The third is the purification of the feed-water from its impurities before it enters the boiler.

187. Removal of Boiler-scale. When the scale is a mud and without tendency to cake upon the heating-surface:

(1) The boiler can be allowed to cool down full of water, and when cooled emptied through the blow-off pipe. By removing the manhole and entering the boiler with hose-jet and brooms, the accumulations of soft scale can be washed out and the boiler is clean.

(2) The mud can be prevented from accumulating and with an efficient mud-drum can be removed, by blowing the boiler down at short intervals during the day, and blowing it out completely at the end of a week or oftener. The objection to this method is that the scale which is not thoroughly washed out by the outflow through the blow-off pipe will dry on the heating-surfaces in a cake which it is difficult to remove when it has once solidified.

When the scale is of the character which cakes on the metal of the boiler, due to the presence of sulphate of lime, two methods can be used:

(1) The boiler being emptied of water and cooled empty, a brisk fresh fire is started under the empty boiler. The effect of this is to expand the iron at a rate faster than the scale, and causes the latter to crack off in flakes, which are then swept out after the boiler is cooled again. The objection to this is a fatal one, in that it is very hard on the boiler and injures it.

(2) A more usual plan is to allow the boiler to cool, empty it, and enter it through the manhole with what is called a scale-pick. This is a species of hammer with both faces formed to a wedge, and with it the scale is struck and broken very much as a film of ice is broken off the exposed stones of dwellings or pavements in Northern cities. The objection to this method is that the forcible removal of scale carries with it the film of oxide of iron which is formed on the inside of a boiler, and which adheres to the scale rather than to the iron. The ultimate effect is to thin the iron by this continual removal of the oxide film.

Belonging to this same class of methods is the use of an apparatus in the form of a trough or false bottom inserted within the boiler, and so arranged as to catch the precipitate which is moving with the currents of circulation. When the solid matter has fallen into such pan or trough it no longer is exposed to circulation, but lies where it has fallen, and therefore does not have a chance to get to the real heating-surfaces. This, of course, is a method available in shell boilers only. For the flocculent or floating type of scale a blow-off connection at the surface of the water in the boiler has been found convenient (Fig. 262). This has been arranged to have a trumpet-shaped mouthpiece whose ampli-

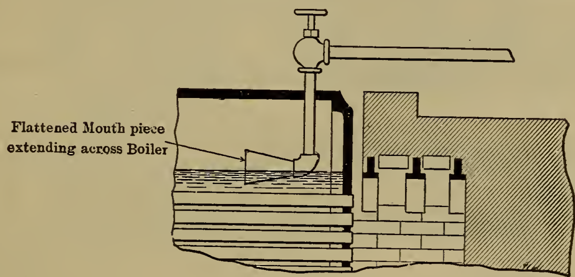


FIG. 262.

tude is greater than the normal range of the water-line in blowing down, so that when the attached valve is open, surface-water flows into the trumpet mouth and out of the boiler, carrying with it the floating scum. In sectional boilers the main dependence against the adhesion of scale within the small units which make it up is the rapid circulation (see Chapter VI), but special cleansing tools have been devised to meet this problem, in which an appliance driven by steam or air can be introduced within the tube. It has cutting or impact tools which break up and loosen the scale so that it can fall out or be swept away. (Figs. 263 and 264). The impact will loosen some scales upon the outer surfaces of fire-tubes, but not all. They are mainly directed to work in water-tubes.

188. Prevention of Scale-formation. There are three great methods which are used to prevent the scale from forming a hard, adhesive crust or coating. The first of these is to introduce in a boiler some reagent or material which shall prevent the scale from hardening or crystallizing by a sort of mechanical reaction. This is the basis of methods which have been used involving the introduction of sand, sawdust, malt-grains, and similar material which shall form the nuclei around which the scale is to solidify in the form of balls or larger grains and remain in a form easily removable.

The second method has been to make use of some material in the boiler which shall act as a varnish caked upon the surface of the boiler, so that the adhesion of scale shall be made more uncertain. The introduction of kerosene, starch, or the real varnishing of the surface with some suitable composition all operate in this way. Perhaps kerosene is one of the best known of the reagents of this class, and for many waters seems to be the best to be used in this way. It is introduced either gradually by a small connection to the feed-pipe suction, or a charge is put in at intervals. Mineral oil or grease does not meet the case, by reason of the tendency which it has to adhere itself to the heating-surface, and, by keeping water from contact with the metal, to cause overheating as badly as the scale itself, if not worse.

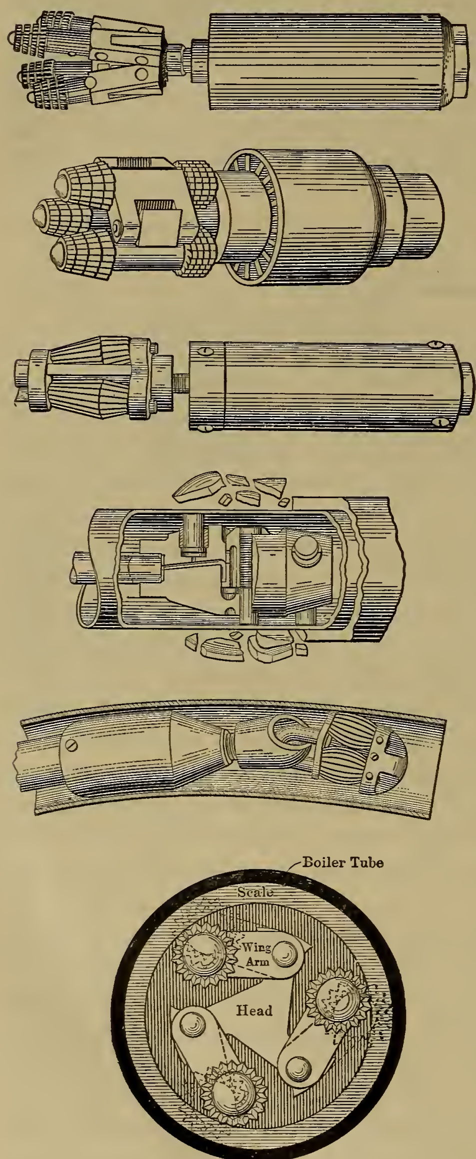


Fig. 263.

The third method is to introduce a reagent which shall act chemically on the precipitate either to change its crystallizing or solidifying character, or to change an insoluble into a soluble salt. The reagent may be either an acid or a salt. Among the acids, tannic and acetic acid are perhaps the most usual and preferred by reason of their reaction upon the sulphate of lime, and the relatively mild action which they have upon iron if the acid should not find sufficient base in the feed-water and remain free. These acids are introduced in the form of brewer's grains, molasses, oak-bark, etc., or in the form of a liquid or crystalline acid. The difficulty with this method is the danger to the metal of the shell itself. In some few places where an acid water has been at hand it has been alternated or mixed with the basic water so as to oppose them to each other's action.

The salts which are used are either the carbonate of soda, the chloride of barium, the tannate of soda, or the sodium triphosphate. The reactions of these with the sulphate of lime form a non-adhesive precipitate, and the soluble salt which results from the reaction with the lime

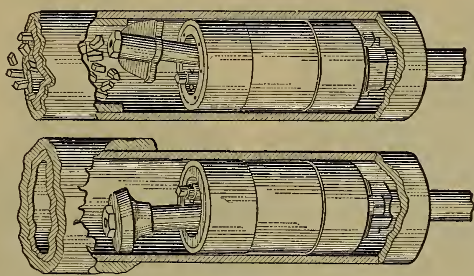


FIG. 264.

is removed by blowing down. The proportions of salt to be used with any water are to be determined on the basis of chemical analysis. To this class belong also the methods of which the use of zinc suspended in

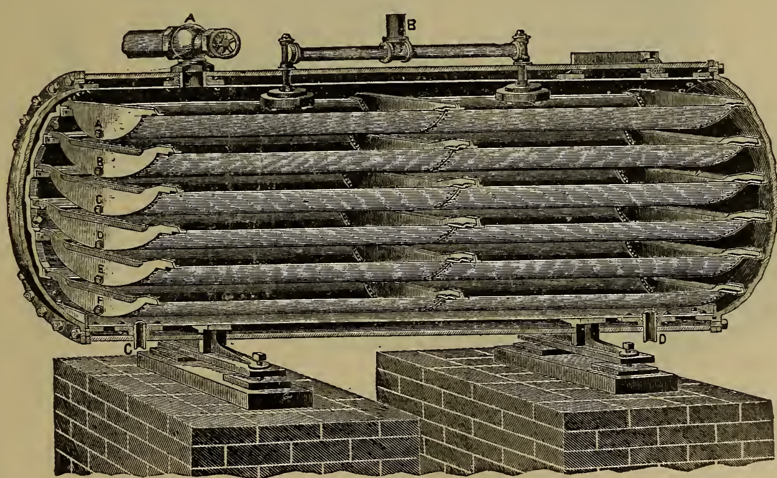


FIG. 265.

the water is a type. While the scientific basis for the practice is not clear, its practical success in many cases for the purpose desired cannot be questioned.

189. Previous Purification of Feed-water. The method which stands on the highest scientific plane with a boiler plant which must use a bad feed-water is to prevent the solid matter from getting inside the boiler at all. There are several methods for attaining this object.

(1) The use of such forms of feed-water heaters as may properly be called lime-catchers (Figs. 265, 281, 282). The feed-water is heated

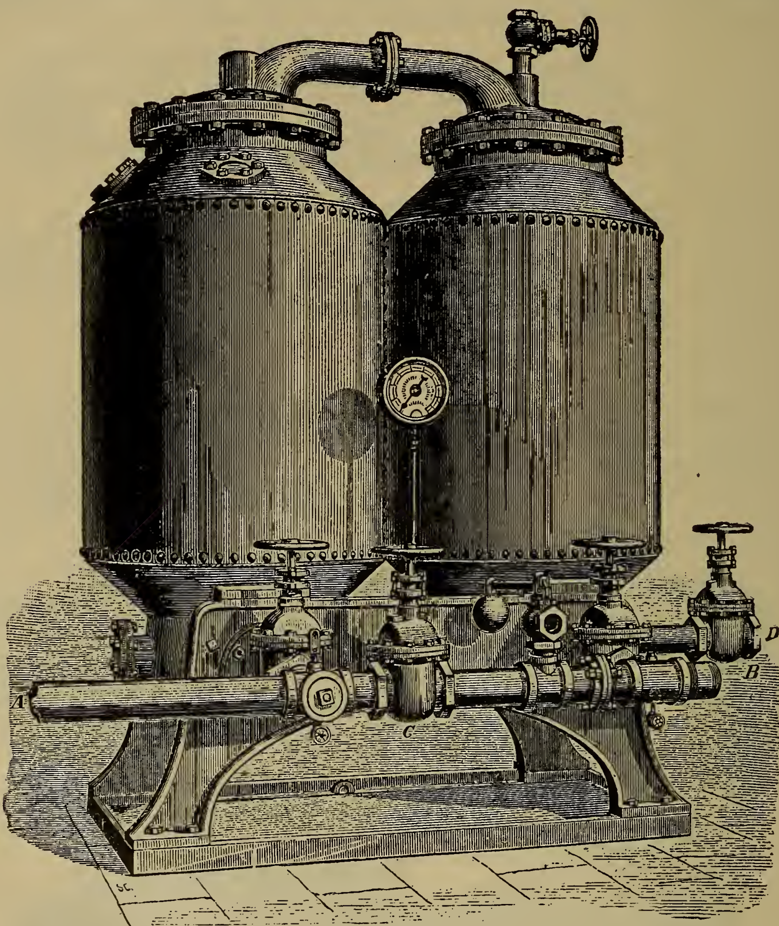


Fig. 266.

in the heater to a point at which it precipitates all or most of its solid matter (paragraf 185).

(2) The use of the methods of surface condensation which prevail in marine practice, whereby the same distilled water is used over and over again and no additional solid matter is introduced with the feed-

water, except with so much of the latter as may have to go in to supply leakage and waste (Figs. 663, 664, and 665). The use of impure water to cool the surface condenser is entirely admissible.

(3) A previous purification of the feed-water by chemical means. This means that the feed-water to be used in any day is introduced into a tank, and into such tank is thrown the necessary reagent to throw down the solid matter in the water. The milk of lime or hydrate of lime will transform the soluble bicarbonate into the insoluble protocarbonate, and the chloride of barium will form the sulphate of barium with the sulphate of lime. The precipitate thus formed can either be filtered out, or it can be allowed to settle and only the clear liquid is pumped into the boiler which contains the soluble sulphate constituents which remain after the reactions. The disadvantages of this method are obvious in the cost of the tankage, and the room which it will occupy, and the cost of the reagents used in the process.

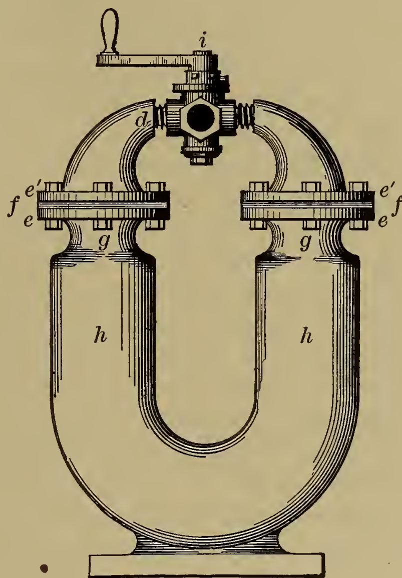


FIG. 267.

190. Filtration of Feed-water. The filtration of feed-water for the removal of either suspended solids or precipitates can be done in open filter-basins, or in close or pressure filters. The open filter-basin is the usual water-works method, whereby the water is made to pass through layers of gravel, sand, and charcoal in succession, and in each of which a certain proportion of the suspended material is caught and only the clear liquid passes through. The pressure filters operate on the same principle of forcing the feed-water through layers of successive fineness, but this will be done in a closed tank and under pressure instead of depending on the simple head due to gravity. Most of these filters operating under pressure are arranged to be reversible either by three valves, or a system of equivalent pipe-connections, so that the accumulated mud in the layers of the filter can be washed out by such reversed current. Otherwise provision must be made at intervals to remove the filtering material, cleanse it, and replace it, during which the water either goes unfiltered or is filtered through a duplicate or reserve apparatus (Figs. 266, 267). Consult also paragraf 530 on oil-filtration.

191. Deterioration or Wear and Tear of Boilers. The conditions to which a boiler is exposed in service tend to wear it out. Many engineers have felt so strongly on this point that they have proposed to limit the life of a boiler in use, and to specify that a boiler is good for ten years, may be run at reduced pressure after fifteen years, but should be thrown out at the end of twenty. The causes which tend to wear out a boiler are partly inherent and unavoidable, and partly accidental so as to be avoided by care. The avoidable ones are usually acute forms of the sources of deterioration which are inherent and unavoidable. Deterioration of boilers is caused by overheating, by unequal expansion and contraction and by corrosion.

192. Overheating of Boilers. An injury to the heating-surface of a boiler from overheating is usually due to carelessness either in permitting the water-level to get so low as to expose the heating-surface uncooled, or to the presence of scale or grease. These have been already referred to in the previous paragrafs. The furnaces of internally-fired boilers and the tubes of sectional boilers are particularly liable to injury from overheating caused by grease. In sectional boilers, besides the oxidation due to overheating, a strain of great magnitude is set up in the straight-tube boilers, where the tubes tend to become of unequal length. In some forms of sectional boilers also, in which disengagement is inadequate and circulation impeded, a section may become overheated because the water cannot reach the metal. Sectional boilers whose units cannot be cleaned are especially open to the danger of overheating.

193. Unequal Expansion and Contraction of Boilers. The intense action of the fire upon boilers tends to raise the temperature of the metal forming them, while an impact of cold air or cold water produces a tendency to cool and contract that metal. Where this action is local the boiler has a tendency to stretch out of shape, and strains are brought upon its structure which act to wrench and destroy the boiler, causing leakage and deterioration. These changes of shape may produce several consequences.

(1) In fire-tube boilers they cause a leakage of the joints where the tubes are expanded into the heads.

(2) The boiler has a tendency to change its shape, and therefore to alter the distribution or the proportion of strain which comes on its various lugs or supports (paragraf 133). In shell boilers this change of shape may produce such a super-position of strains as to cause a boiler to give way under them at a point where such combined strains may be concentrated.

(3) If there are any defective welds in the plate of which the boiler is made, contraction of the layers of skin causes that lamination to

extend, and finally to develop a blister or bag (Fig. 26). In steel plate a blow-hole will be the occasion for similar action.

(4) The contraction and expansion of a boiler with lapped joints produces an effect which has been called "grooving." The effort of the two contiguous plates to flex into line with each other when they are pulled lengthwise (paragrafs 32 to 40) causes the protecting scale of oxide of iron to be broken off at the point of greatest flexure. Fresh rust forming there and again broken off by the flexure of the joint results ultimately in the erosion of the metal and the formation of a groove at this point, whereby the strength of the plate is gradually reduced. Figs. 268 and 269 show a groove of this sort and its location. They are often deep enough to take the blade of a knife, and are to be detected in most cases by its use. They are also revealed frequently by the presence of a stain of oxide of iron upon the scale removed from the plate around the joint. Grooving is made worse when there are acids in the water, and where the flexible part of the shell is joined to a stiffly-stayed or inflexible part, so that the bending action is concentrated. Such a place is the joint of the flange of the head of a boiler with the flat surface of the head.

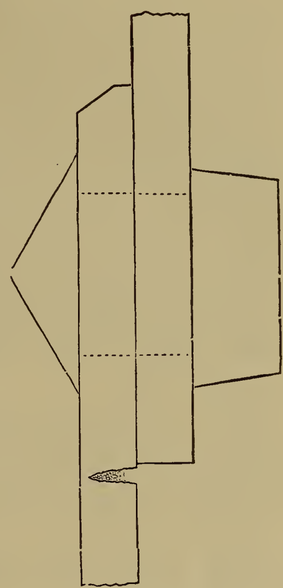


Fig. 268.

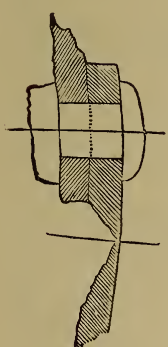


Fig. 269.

194. Corrosion External. The third source of deterioration of boilers is the corrosion to which they are exposed from the conditions of their use. This corrosion takes place from the inside of a boiler and from the outside.

External corrosion may be the result of any or all of the following conditions:

(1) The action of the hot gases upon the heated plate which forms the heating-surface. If the fire is forced, or if the surface is covered with a thin scale which prevents rapid transfer of heat, the metal will be heated so that it will react with oxygen in the gases and become rusted or corroded. This difficulty is aggravated by the presence of moisture in the gases, either from the coal, as water present mechanically,

or from the combustion of hydrogen to water. This condition is favorable to rapid action on the iron from carbonic acid, or sulfurous acid resulting from the oxidation of carbon, or sulphur in the fire-box.

(2) From leakage. This may occur from seams, around rivets, where the tubes enter the tube-sheets, around the dome-joints, and at the joints of the hand-holes or manholes in shell boilers. In sectional boilers, in addition, will be the leakage caused around joints of the caps and that which is caused by unequal expansion of tubes in their headers. In internally-fired boilers, where the water-legs are closed by massive rings at doors and bottoms, leakage is apt to occur from differences of temperature due to defective circulation. The leakage from valves which are not thoroughly packed or tight upon their seats is also a further occasion for corrosion. This moisture not only corrodes of itself, but is the occasion for forming an active corrosive agent with carbonic or sulfurous acid; and if the water exerts any mechanical action upon the rust, scales tend to loosen and expose fresh surfaces to corrosive action.

(3) The presence of lime in the setting of brick-set boilers is another occasion of external corrosion. The heat of the fire and its gases causes the lime to become calcined to the hydrate, and in this form it is likely to exert a corrosive influence where it touches the metal.

External corrosion is to be detected by careful inspection, and the setting of the boiler should be such that this inspection should be possible.

195. Corrosion Internal. The corrosion which takes place inside a boiler is more rapid and injurious than the external corrosion. It may be due to one or all of several causes.

(1) The presence of acid in the water undergoing evaporation. The source of this acid in the water is often determined by local conditions. In the mining districts where sulfur prevails in the coal or in the surface-water and, worse than that, in the mine-water, the sulfurous acid which results is very actively corrosive upon iron. Nitric acid in the form of decomposable nitrates and nitrites is present in waters which have been contaminated with sewage or which contain organic matter. Water from bogs or peaty deposits containing vegetable matter in decomposition will contain the earthy or humous acids, formic, etc.

(2) Perhaps the most trouble in boilers is caused by the corrosive action of the acids due to decomposition of the lubricants. The reaction on boiling animal oils, tallow, etc., breaks such material up into stearic and oleic acids, both of which are corrosive to iron. The oil comes into the boiler with the feed-water from condensing engines where pains are not taken to prevent it (paragrafs 246, 307, 528), and will undergo

this trying-out process. The active element of corrosion in sea-water and water used in marine boilers is hydrochloric acid, which results when the chloride of magnesia present in sea-water is boiled. The heat decomposes the chloride into the hydrate of magnesia and hydrochloric acid, which latter attacks the iron.

(3) From galvanic action between the iron of the boiler and some metal which is electropositive to iron. Such metals are copper and brass, which form with iron a galvanic couple, and in waters containing even weak acids, like carbonic acid, the iron undergoes oxidation and corrosion. Such metals for galvanic action would be found in copper stay-bolts, tubes, ferrules, and even in brass mountings of fixtures for feed-connections. Sea-going boilers are particularly liable to this kind of corrosion by reason of the presence of the acids in sea-water, and it has been found a convenient thing to hang a piece of zinc in the water-space of such boilers in order that by its presence, which furnishes a lower electric potential, the zinc might be the element attacked rather than the boiler itself.

(4) Distilled water containing carbonic acid seems itself to be corrosive of iron, under the conditions which prevail within a boiler. Laboratory experiments have not always been conclusive on this point, except with respect to water containing no air but carrying carbonic acid.

(5) The water seems to have an erosive action mechanically against surfaces upon which it is thrown violently by the currents of steam in which the water will be carried in drops.

Spattering followed by drying of the spattered water seems also to wash off and loosen the scale of oxide of iron and produce the effect of corrosion.

The corrosion due to water is to be expected below the water-line or where the mechanical action of water may make itself felt near the water-line in the steam-space or in the pipe-connections.

Corrosion, however, is often met in the steam-space of the boiler and manifests itself with somewhat of capriciousness. It has been found that a boiler in the steam-space may be kept quite hot by the non-conducting covering over it, and sometimes causes the corrosion to manifest itself more rapidly by reason of the high temperature producing a considerable expansion, and at a rate different from that of the oxide of iron, so that the oxide is cracked off and fresh surfaces exposed.

196. Pitting, Wasting, and Grooving. The corrosion of a boiler on its internal surfaces usually takes place in one of three forms. The wasting is a gradual thinning of the plate all over, due to a uniform acid action whereby the iron is dissolved. It is not always easy to detect this, except by close inspection of the joints, and the indications around the rivets which show what the original and unreduced thickness of the plate should have been.

Pitting is a curious and capricious eating of the plate in spots. The reasons for local corrosion of this sort are not easy to find. It is doubtless often due to lack of homogeneity in the plate, so that it has been more disposed to yield to corrosive influence where cinders or similar impurities are present. Mechanical erosion is apt to produce the effect of pitting.

Grooving is the corrosion which has already been referred to in paragraf 193, where the changes of shape cause a mechanical breaking away of the oxide of iron formed, so that fresh surfaces are continually exposed. When corrosive tendencies are present in the water, grooving goes on so much the more rapidly.

It is a matter of discussion as to the best preventive of corrosion in boilers which are to go out of use. Some advocate the plan of preventing access of air by filling the boiler full of water. Others dry out the boiler by putting a charcoal fire in a brazier within it which disposes of the oxygen also, and any remaining moisture is absorbed by hydrate or chloride of lime. Then the boiler is closed. This is the English naval practice. Others again fill the boiler full of water and inject some oil which rises to the top. The water is then withdrawn slowly from the bottom, leaving a film of oil laid on by capillary action over the entire previously wetted surface.

Minute and painstaking inspection of the interior surfaces of a boiler is necessary if danger from corrosion is to be guarded against.

197. Repairs. General. The repairs to a boiler are of the same nature as the operations in its construction so far as leaky seams or tubes and joints are concerned. The leakage at a tube-joint can be prevented by expanding once or twice, but after that the metal becomes hard or brittle and further expanding cannot be done. Locomotive-tubes are particularly liable to trouble of this sort, and the custom has prevailed of cutting the tube off at its two ends and inserting a snort length at one end, which should bring fresh and unfatigued metal to make the joints at the tube-sheets without renewing the entire tube-body. The piecing out of the tube is done by making the two ends to be joined into a, male and female cone, and then welding the lap of the two surfaces over a mandrel. In sectional boilers the repairs are usually renewals of the tubes in detail, the regrinding of joints between the caps and headers, and the like. All boilers with manholes will require that the gasket shall be renewed periodically, and usually at each time that the manhole-lid is removed for purposes of cleansing and inspection. On shell boilers, however, it may be necessary to apply a patch.

198. Patches. The failure of a part of the metal in a shell boiler where the entire plate does not have to be renewed may be repaired by putting a patch on the defective part. Such patches are of two kinds, the hard patch and the soft patch. The patch will be put on a boiler by cutting away the metal which has deteriorated, leaving a hole where the defective metal has been and including enough to come to the solid and unaffected metal around the edge. A piece of boiler-plate is then shaped to the surface which it is to cover, and of a size to cover the hole and lap over its edge so as to be riveted to the shell in lap-joint. The necessary holes are then drilled and the patch is riveted on in place and calked. Such a patch is as good as can be made, but a patched boiler is never as good as the unpatched plate, and the presence of patches usually indicates either defective material or hard usage.

The hard patch just described will be used wherever possible, but it can only be used where riveting can be done. If the patch must be made at a place where riveting is impossible, the patch will be secured in place with bolts, but they will not make the joint as tight as the rivets, and consequently a packing of some sort must be inserted between the two plates and also around the bolts. This packing is usually for the ordinary soft patch a cement of red lead mixed to a paste with oil, and held by being formed into a rope or gasket by working it into unwoven lamp-wick. A rope of this paste and wicking is laid around on the inside of the bolts in the lap of the patch, and is compressed to fill the joint and make it tight. Such a patch, however, is not as reliable as the hard patch, and is liable to blow out under heat and deterioration combined with pressure.

CHAPTER XII.

BOILER-INSPECTION AND TESTING. BOILER-EXPLOSIONS.

200. Boiler-inspection. The steam-boiler being an engineering construction and exposed to known strains, it becomes necessary that the person responsible for it should be able to inform himself concerning its condition and ability to withstand these strains. This is to be done by means of inspection by the eye of experience and skill. It involves a knowledge both of accepted practice and of the causes which tend to wear out a boiler, and judgment in deciding how far they have acted either to render the boiler unsafe at its former pressure or unsafe to use under any conditions. A proper and full inspection, therefore, covers all the points which have been made the subject of discussion in Chapters III to XI hitherto, particularly with respect to corrosion in its various forms, the effects of overheating, and proper care and design with respect to bracing and staying, and also the use of satisfactory appliances for the management of the boiler and the relief of any excess of pressure. Further than this, the inspector should satisfy himself that the boiler is able to withstand its working pressure by exposing it to a pressure somewhat higher than that which it is expected to carry, and then observing whether under such pressure the boiler shows any signs of weakness, deformation, leakage, or similar failure. It is usual to expose the boiler to a pressure-test equal to one and one-half times its ordinary working pressure. This is entirely safe with normal conditions, since the boiler was probably designed with a working pressure of one-sixth of its calculated bursting pressure (paragraf 26), so that if exposed to three halves of one-sixth of its bursting pressure, it is only tested to one-quarter of the ultimate pressure. There are three ways of making this pressure-test.

201. The Steam Pressure-test. The steam pressure-test is to close the orifices of the boiler, increase the safety-valve weight, and build in a fire under the boiler to make it test itself to one and one-half times its working pressure. The advantage of this method is that it exposes the boiler to the conditions of service with respect to strains caused by heat as well as by pressure. The objection to it is evident; that if the boiler is to fail under its test, its failure, by reason of the presence of a volume of hot water, will be the occasion of a disaster. It should only be practiced, if ever, where proper public safeguards can be applied.

202. The Hot-water Pressure-test. The second method is to fill the boiler completely full of water, and with all outlets closed start a fire in its furnace. The water will expand more rapidly than the iron forming the shell, so that the expansion of the water will bring a strain upon the shell from within which can be graduated to the required amount. When the pressure is reached the fire is withdrawn. Water expands $\frac{1}{224}$ of its volume in passing from 60° to 212 F., and the boiler, being full, is subjected to this expanding strain. This method has somewhat the advantage of having the boiler warm or hot, and in case of failure or rupture of the shell the water escapes without doing great harm, since but little energy is stored in it. The heat condition, however, is not favorable to the inspection of the shell for deformation and leakage, and consequently the third method is more usual.

203. The Cold-water or Hydrostatic Test. The hydrostatic test as usually made is to fill the boiler completely full of water, and then by means of a pressure-pump, operated either by hand or by power, to raise the pressure of the water in the boiler to one and one-half times the working pressure. This is done in the cold; and while the boiler is subjected to this internal pressure it should be carefully examined for bulging of the heads or other deformations, and for leakage which can be attributed to this tendency to go out of shape under pressure. Leakage will be manifest by the rapid lowering of pressure, since the comparative incompressibility of water makes a slight leakage release pressure very rapidly. By putting a test-gauge upon the connections of the pressure-pump the boiler-gauge can be tested for accuracy at the same operation (paragraf 172). The only objection which has been urged against the cold-water test is that it is a severe one and may injure the boiler by overstrain, and that the pressure due to the water brought by the action of a pump is a different and more exacting one than would be brought by the pressure of the steam. The rejoinder to this is that if a weakness is to be developed, it is immensely to be preferred that it should be developed while the test is on than in service, and the large mass of water in most boilers precludes any very great concentration of the pump-pressure. The steam-pressure is a fluid pressure, and the water is as flexible and mobile hot as cold. The laws of most cities compel a hydrostatic test to be made once a year at least of all boilers which are under municipal control.

204. The Hammer Test. In addition to the hydrostatic test for the resistance of the boiler to pressure and the detailed examination within and without by the eye of an inspector, much information as to the quality of the boiler as a construction will be given by means of a careful and exhaustive examination with a light hammer. The blow upon a loose rivet or a stay which is not doing its work will reveal by the difference in the resonance the difference in its condition, its looseness, or its overstrain. The hammer will also indicate and reveal defects in the metal of the shell, the presence of cracks and similar weakness which may lead to a failure. Where the plate has begun to laminate and the beginnings of a blister have occurred, the hammer-blow will show that the spot is no longer solid at the point struck.

205. Boiler-explosions. General. The disaster which is most feared in connection with a steam-boiler as a reservoir of accumulated energy is that which is called an explosion. An explosion results from a very limited number of immediate causes, but a large number of secondary causes may be looked for as bringing about the primary cause.

The primary cause is a failure or rupture of the enveloping shell of the boiler due to a pressure or strain greater than the metal could resist. This interruption of the equilibrium or the destruction of the reserve of metal strength may come about by two different ways.

(1) The boiler may be too weak for the pressure, so that it ruptures at its working pressure.

(2) The pressure may become too strong for the boiler to withstand, and it ruptures at some point above working pressure.

206. Boiler Ruptures because too Weak. The boiler may fail because it cannot withhold the pressure within it by reason of one of four conditions:

(1) The original pressure at which the boiler has ordinarily been worked may have been fixed too high for a structure of that material, design, thickness, and construction. Such a boiler never was safe from the very first day it was used.

(2) The boiler, originally strong and able to withstand the working pressure, may have become weakened by age, wear and tear, corrosion, or abuse.

(3) The boiler, exposed to normal working strain, may have superposed upon such strain an extra strain of contraction, local and sudden. This may come by low water and sudden introduction of fresh cold water on overheated plates, or cold air acting similarly. This is the rupture of which low water is apt to be the occasion.

(4) A defect of workmanship or material which escaped inspection when the boiler was new may develop in service, and particularly under abuse. The boiler may not be as strong as it was supposed to be, and fails.

It is obvious that the more familiar the inspector is with the construction and sources of deterioration of boilers, the more reliable is his judgment with respect to fixing the working pressure upon an old boiler. It will be seen presently that if the rupture of the shell is caused by working pressure, the disaster called an explosion will or will not follow according to the combination of conditions under which that rupture takes place.

207. Boiler Ruptures from Excess of Pressure. The boiler being strong, perhaps new, may have the pressure within it raised to such a point that it is unable to withstand it, and fails at its weakest point. This is the condition in explosions above the working pressure or at pressures approaching the computed bursting pressure. This group of conditions has been the favorite field for erroneous theories with respect to the disaster called a boiler-explosion. Most of them have been based on the idea that an explosion of some sort takes place within the shell, creating a pressure suddenly within that envelope, which it could no more withstand than if a powder-explosion were to bring suddenly an enormous pressure upon such a flexible envelope. Those who have upheld this idea have explained the explosion from the union of oxygen and hydrogen, which are the gases of the water, supposing them to have been dissociated by an overheated plate occurring with the condition of low water. A second theory of this sort has been that by reason of low water and overheated plate a sudden rise of pressure results from the coming in of the feed-water, causing an explosive sort of pressure within the shell which it could not withstand. A third and similar theory is that the water in the boiler gets into the condition called the spheroidal state, in which the bubbles of water are kept away from the red-hot plate by a film of steam, and which bubbles form steam with a concussive rapidity when that film of steam is forced out. This condition is met in forging or rolling where water is used on red-hot metal and then struck with a hammer.

It is difficult to realize the above conditions in a boiler, and when an explanation without recourse to them is to be found from accepted principles it does not seem necessary to search for less obvious causes.

Furthermore, it can be proved (paragrafs 213 and 214) that in a boiler containing a relatively small weight of water, and particularly with ample heating-surface, the time required to pass from a low pressure to a higher one, which may be called the dangerous pressure, becomes surprisingly short, so that in the absence of an efficient safety-valve, or where it is inoperative, the steam pressure to rupture the boiler may be reached very soon after the outlets from it have been closed, unless proper precautions be taken with respect to checking the fire.

208. Theory of Boiler-explosions. When it is remembered that the specific heat of water is unity, so that a large quantity of heat is absorbed in raising its temperature one degree, it will be apparent that when a large mass of water under high pressure lies within a boiler an immense reservoir of available energy is at hand.

The boiling-point of water increases with the pressure, so that if the pressure on the surface of the water in a boiler is suddenly released and drops to atmospheric pressure, or nearly to it, a large quantity of water will form steam-gas upon the release of such pressure without the addition of heat. When, therefore, a boiler ruptures under pressure and full of water, permitting the escape of steam instantly or with great rapidity, the tendency of the stored heat in the water is to cause it to form steam under the reduced pressure with equal rapidity. The formation of steam-gas from water is easily comparable under these conditions to the formation of carbon, sulfur, and nitrogen gases in the combustion of gunpowder. Hence the rupture of the shell causing a release of pressure on the water brings about a condition analogous to the touching of a flame to a gasifying substance like powder, and the water flashing into steam-gas is the thing which explodes. If this operation is retarded, the energy is gradually released in forcing the water out through a small hole, and no disaster follows the rupture. This is the element of safety of the so-called sectional boilers.

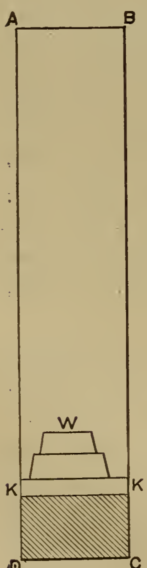


Fig. 270.

209. Energy Resident in Hot Water under Pressure. It can be shown by a simple diagram that an enormous quantity of energy is stored in a cubic foot of water at high pressure. If the base of the cylinder in Fig. 270 be supposed to have one square foot of area, and at its bottom a cubic foot of water is inclosed below the piston, so that upon the application of heat to that water the piston would have a tendency to rise under one atmosphere of pressure on it, the water would form steam to fill 1700 cubic feet. If, however, the pressure be increased upon the piston, the specific volume of steam at such higher pressure diminishes, while the amount of heat necessary to make the water into steam increases. The cubic foot of water at seven atmospheres or 103 pounds, instead of occupying 1700 cubic feet, would occupy but 274, because the piston would be held down by a pressure of 14,832 pounds. If it be conceived that similar water in the bottom of that cylinder was not quite hot enough to make steam at that pressure, but was hot enough to make steam at a somewhat lower one, it will be apparent that the water when the pressure was released would be able to lift such weighted piston through a good many feet. A simple calculation shows that if this energy be represented in foot-pounds, it is able to carry the weight represented by a boiler-shell, or a part of it, through a good many hundred feet, and to produce the effects which have been observed to attach to the most disastrous explosion.

210. Reaction in Boiler-explosions. It happens not infrequently that when the rupture of the shell from excess of pressure or weakness has permitted a partial escape of pressure, the water in the shell seems to be lifted by the sudden release of pressure on that side of the boiler, and the unbalanced pressure produces a reaction; or the water itself falls back against the part opposite the rupture, producing a strain which the already weakened shell cannot withstand, and thereby makes so large an opening for the release of pressure that the formation of steam-gas is almost instantaneous, and the boiler is driven out of its setting as a rocket is driven by the reaction of gas behind it. As a rule the light portions of a boiler after rupture are found in the direction of the initial rent, while the more massive pieces are driven by the reaction of unbalanced pressure in the opposite direction. This reaction-phenomenon resembles concussive ebullition (paragraf 236) in that a stress almost like that from a solid blow is brought by it against that part of the boiler which remains in place.

211. Procedure when a Boiler is in Danger of Rupture. It would be manifestly unwise to release the pressure suddenly from a boiler which was already under a great strain and in danger of rupture. To do so would be to invite the reaction caused by the lifting of the water, and the possible superposition of strains from this cause. The opening of a large throttle-valve or of the safety-valve may act in the same way, and it is doubtless this combination of strains which explains the frequent failure of boilers when the day's work is just beginning and steam is turned from a boiler into a cold pipe, where it condenses and makes a reduced pressure, so that the steam rushes from the boiler at higher velocity than usual. A large valve should be opened with exceeding caution, slowness, and care under these conditions.

The proper procedure is to withdraw the fire and permit the boiler to cool off gradually, and so permit the dissipation of pressure by these means. The fire can be checked by dumping it or by throwing ashes or dirt upon it. If great confidence is felt in the ability and strength of the boiler, the blow-off valve can be cautiously opened for the relief of pressure slowly through that opening. The heat stored in the water is thus disposed of, and after the boiler is empty of water it is comparatively safe. The escape of pressure through the blow-off valve is also unlikely to cause difficulties from reaction.

212. Heating Effect of Steel Plate and Cooling Effect of Water. If 10 square feet of plate one-quarter of an inch thick be overheated so as to be at 1000° F., it will represent 100 lbs. weight of iron, with a specific heat of 0.112. If water come on that plate at a temperature of even 300° F., it will cool the plate by a transfer of heat to the water; whence

$$Q = w \times c' \times (t_1 - t) = 100 \times 0.112 \times 700 = 7900$$

units of heat received by the unknown weight of water. It takes about 1000 units of heat to vaporize a pound of water under the pressure corresponding to 300° F., or that plate would vaporize about 7.9 lbs. of water only, or less than a gallon, in being cooled to the temperature of the rest of the boiler. The volume of one pound of steam at 300° is 6.28 cubic feet, so that this steam would occupy but 7.9 × 6.28 or 49.6 cubic feet in the boiler.

213. Time required by Boiler to absorb Heat and Pressure Energy. The following formula, due to Zeuner, shows the time to be allowed to a boiler to pass from one pressure or temperature to another. The lower pressure (t) may be that of the cold feed-water, in which T will give the time required to get up the steam-pressure corresponding to any higher temperature (t_1); or t may correspond to the working pressure, and t_1 be the temperature corresponding to a pressure which will endanger the shell.

Let T = time in minutes elapsing between the period when a lower temperature (t) prevails and that at which (t_1) will be the temperature when all outlets are closed for steam or discharge of heat;

t = temperature corresponding to the lower pressure;

t_1 = temperature corresponding to the higher pressure;

W = weight of water in the boiler;

Q = quantity of heat in B.T.U. transferred to the water in the boiler per minute.

Then
$$T = \frac{W(t_1 - t)}{Q}.$$

The quantity Q for any boiler is found from the expression

$$Q = \frac{\left(\begin{array}{l}\text{heating-sur-}\\ \text{face of boil-}\\ \text{er in square}\\ \text{feet}\end{array}\right) \times \left(\begin{array}{l}\text{pounds of water}\\ \text{evaporated per}\\ \text{hour per sq. foot}\\ \text{of heating-surface}\end{array}\right) \times \left(\begin{array}{l}\text{the quantity of}\\ \text{heat absorbed}\\ \text{in evaporating}\\ \text{1 lb. of water}\end{array}\right)}{60}$$

The third factor in the numerator is 966 at atmospheric pressure. For higher pressures it may be called 1000, to make round figures.

Illustrations of the application of this formula would be:

CASE 1. *Locomotive Boiler.*

W = 5000 lbs. of water;

Grate-surface = 11 sq. ft., and each square foot burning 60 lbs. of coal will evaporate 7 lbs. of water per pound of coal per hour, or 77 lbs. of water per minute;

t = working pressure of 90 lbs. = 319° F.;

t_1 = dangerous pressure of 175 lbs. = 371° F.

$t_1 - t$ = 50°+ F.

Hence $$T = \frac{5000 \times 50}{77 \times 1000} = 3.2 \text{ minutes.}$$

CASE 2 *Marine Boiler, Flat Surfaces.*

W = 79,000 lbs. of water;

Heating-surface = 5000 square feet, evaporating 3 lbs. of water per hour, or 250 lbs. per minute;

t = working pressure of 37 lbs. = 262° F.;

t_1 = dangerous pressure of 60 lbs. = 291° F.

$t_1 - t$ = 29° F.

$$T = \frac{79000 \times 29}{250 + 1000} = 9.1 \text{ minutes.}$$

CASE 3. *Fire-engine (a) Boiler to get up Steam.*

W = 93 lbs. of water, or about $1\frac{1}{2}$ cu. ft.;

Heating-surface = 157 sq. ft., evaporating 1 lb. of water per hour, or 2.6 per minute;

t = atmospheric pressure, or 212° F.

t_1 = working pressure of 329° or 100 lbs. pressure;

$t_1 - t$ = 117°

Then $$T = \frac{93 \times 117}{2.6 \times 1000} = 4.2 \text{ minutes.}$$

CASE 4. *Same Boiler (b) to become Dangerous.*

W = 338 lbs. of water;

t_2 = 200 lbs. pressure, or 388° F.

Then $t_2 - t_1 = 49°$,

and $$T = \frac{338 \times 49}{2.6 \times 1000} = 6.4 \text{ minutes.}$$

It will be apparent that the danger increases with the heating-surface, and diminishes with the greater weight of water contained in the boiler.

214. Steam Boilers as Magazines of Explosive Energy. Nystrom, p. 393, gives dynamic work of gunpowder at 150,000 to 200,000 foot-pounds per pound of powder. Even at atmospheric pressure, the energy resident in one cubic foot of water heated to form steam-gas at that pressure is

$$1700 \times 144 \times 14.7 = 3{,}598{,}560 \text{ foot-pounds,}$$

which if all released at once as gunpowder gasifies would bear a ratio of destructive energy of $\frac{3598560}{200000}$, or nearly 18 times that of such powder.

PART III.

CHAPTER XIII.

BOILER PLANT AUXILIARIES.

220. Introductory. In the development of the previous chapters the logical basis has been the simple problem outlined in Fig. 1 where from the pile of coal and the water tank two streams of potential energy have been directed into the boiler and there transformed into a mechanical available force. These elements are present in the simplest power plant: they are never absent from the largest and most complicated. The effort has been heretofore to keep to the universal principles, which are not affected by the size of the plant whether large or small; and where alternative methods have been presented, these were coördinate in importance, no matter on what plane the plant is to work.

When the plant becomes a large one, burning a large weight of fuel per day or per hour, and where the problem of the number of men and their cost becomes financially significant, it may be worth while to plan and to expend in first cost for a group of auxiliaries in the way of steam machinery whose purpose will be either a diminution of the demand for human labor by replacing it by mechanical appliances, or an increase in economy and efficiency in the use of fuel, or both of these. Such machinery is called boiler plant auxiliary machinery or boiler plant auxiliaries. In the diagrams of Figs. 2 and 3 such auxiliaries are presented either as driven from the main engine by transmitted power, either electrically or through shafts and belts, or as independent auxiliaries, each having its own steam cylinder receiving steam from the common main boiler plant, or from a donkey-boiler if the former is not in operation. If the plant is small or temporary, it may not be worth while to provide these auxiliaries: if large or permanent it usually pays to do so.

Such auxiliaries belonging in the boiler plant will be:

1. Mechanical coal-handling plant.
2. Mechanical ash-handling plant.
3. Mechanical stokers.
4. Heaters for feed-water.
5. Special auxiliaries for a particular plant.

The term auxiliaries must also be applied as used in Fig. 3 to the essential apparatus required for the boiler plant as:

6. Boiler feeding,
7. Artificial draft,

and for the engine-room auxiliaries proper such as:

8. Condensers.
9. Circulating and air pumps.
10. Lubricating oil pumps. Special pumps.
11. Special engine-room machinery.

The middle series, 6 and 7, being essential to any plant or as alternates for an essential, have been already referred to; the last four will have a chapter of their own in later treatment. The present chapter is to enlarge upon the first five.

221. Coal-handling Machinery. The labor of handling the fuel for a locomotive developing about 1000 horse-power is so severe that one man cannot stand the strain for more than three or four hours without a period of rest. When this same power is distributed in several fire-boxes and with fixed grates, and where the fireman must also look after the ashes and work for a ten-hour shift, the limit is from 250 to 300 horse-power, depending on the fuel and its quality.

If the fuel has also to be handled by human labor from bins or pockets to the boiler front, additional labor is called for; and if the coal must be carted from yards or boats or cars and delivered to the local bin an added charge in its price is incurred.

Large metropolitan and other plants therefore in recent years have been seeking the water front of navigable waters, and have been grouping themselves along the railway lines where there is no water, with spurs from the main line into the yard of the plant. The purpose of this is to handle their fuel in car or in boat-load lots. The principle of adequate storage has also been considered, so that in case of ice in the river or snow on the rail, or where for any other cause (such as a labor dispute) the regular weekly or daily delivery of coal was interrupted, there should be a reserve on hand to prevent a shut-down for lack of fuel.

From the boat at the wharf or bulkhead, the coal must be lifted in skips loaded by hand in the hull and then run on rails to where they will be dumped or emptied. Or, elevators of the type used for grain may be lowered into the hull and there charged, and the coal passed on to conveyors. For the boat and wharf combination, the coal may be piled upon the wharf or adjacent land, or put in elevated pockets. If piled, it must be handled again by elevators and conveyors. If the pockets are outside the building of the boiler plant, they can deliver into cars on grade and the latter be run in to the boiler-fronts on industrial indoor railways.

Probably the best way of all is to deliver by elevator or conveyor or combinations into elevated bins or pockets within the building of the boiler plant on levels above the firing-floor. In cities this can be done by dumping by gravity from the street carts into steel cars below the street level: these cars are run on tracks to a vertical elevator within the building or outside of it, and raised to the top of the pockets. They are then run over the pockets and dumped. These pockets have a sloping or hopper bottom (see Fig. 695) whose angle is greater than that of the angle of friction of coal, and these hoppers converge to a descending pipe — say of 12 inches diameter — through which the coal is led to the desired boiler-front. The foot of the pipe may be closed with a valve, parting like the two halves of a clam-shell digger, or the bottom may be left open, coming down so near the floor that the cone of fuel shall not have an inconvenient base with the pipe orifice as its apex. When the cone is in place and untouched by the shovel or stoker no more coal can flow down.

The modifications of this plan by special conditions are numerous. For example, with car-load delivery from a hopper-bottomed coal car on an elevated trestle, the pocket will be below the track, and usually outside the building. A conveyor will bring the coal to an elevator within the building, and deliver in turn to a second conveyor, which fills small pockets over each boiler or stoker. The type of elevator and conveyor must be chosen with regard to the special service required. With the masses of the coal lumps, the bucket-types on chains are usually better adapted than the faster-moving belt-conveyors. Such bucket-chain types combine both elevating and transporting functions in one system. Instead of conveyors, industrial coal cars can be used on indoor railways; or the pockets may be indoors and elevated over the fire-room floor, etc. In recent large metropolitan installations where ground is costly and the batteries of boilers have been put over each other in successive stories, the plan of an overhead storage in pockets of perhaps 10,000 tons capacity seems to be by far the best (Figs. 695 and 696).

222. Ash-handling Machinery. A coal which has only ten per cent of ash and solid matter to be rejected from the bottom of the grates will be considered above the average. This means that 200 pounds of refuse must be disposed of for every ton burned.

This should also be handled mechanically and gravity utilized as far as possible. It means a transverse tunnel or duct running on a level below the firing-floor under the ash-pits of all boilers, and with metal hoppers forming the bottoms of such ash-pits. From these hoppers after cooling and quenching the ashes may fall on opening the hopper bottom either into cars or upon a conveyor which shall carry them to a central point from which they shall be elevated and discharged into carts or cars for removal. In many cities such removal must be contracted for, as the quantity exceeds the normal capacity of municipal service.

At sea, the ash is elevated and thrown overboard by what are called ash-ejectors, utilizing either atmospheric pressure or a flow of sea-water from a steam jet to lift and discharge.

In shore plants the use of forced draft in the ash-pit compels that the hopper and its closure be practically air-tight, and no water can be used in the ash-pit. The tunnel or duct under the floor-level is also an expensive adjunct, but it is practically always justifiable in a large plant.

223. Mechanical Stokers. The same arguments advanced in paragraf 221 for handling the coal from without into the plant upon the firing-floor apply also to the firing itself or the delivery of the fuel into the fire-box. There are also some additional points applicable. By mechanically supplying the fuel to the fire the supply of heat units is constant and uniform, and hence a complete and smokeless combustion is much more probable. Men prefer to fire hard and then to rest after it, rather than to fire easily and continuously without stopping to rest. Particularly in bituminous coal firing and with rapid combustion, the mechanically fed fire secures the initial coking and subsequent incandescent periods with more certainty and regularity than hand firing is likely to do, with the type of man who keeps to firing as an art and vocation.

With mechanical stoking, the fire as a whole is at all times in the same condition. In certain cases, where a peak or maximum in the load curve appears, the regular supply of heat units may need to be exceeded for a time, but this can be met by special hand service, or particularly by the use of forced draft, increasing the rate of combustion for the time.

The fuel is led down from the pocket or bin overhead through a tube to a flat or wedge-shaped hopper in front of where the firing-door used to be. The delivery on to the grate from this feeding hopper may be through a space which is open to the grate on one side, where the angle of friction of the coal lets just so much coal pass per minute under a regulating plate. Or a cylinder with radial vanes or wings turning within another hollow cylinder at a determined rate may measure off from the hopper the quantity which fills one of these segments. The traveling-grate principle may be used (Figs. 175, 271) or the alternately reciprocating bar (Fig. 272) or the step with tipping effect (Fig. 273) or the underfed principle of Fig. 274. The latter possesses considerable interest for special grades of coal, inasmuch as the distillation takes

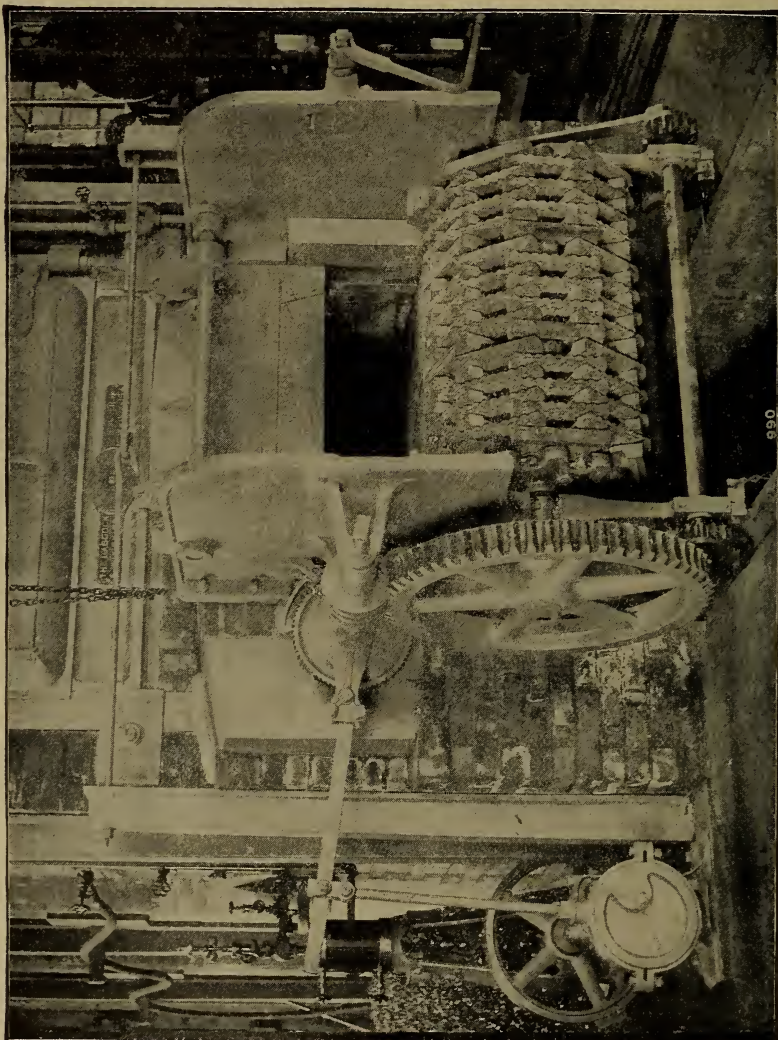


Fig. 271.

place from the bottom upward, and the gas without admixture of air passes up through the burning mass above it, becoming heated and ready to ignite when it escapes from the top. The coal flows laterally downward to the sides and combustion is complete when the coal reaches the limit of motion, as in all the designs. If now the motor driving any of the stokers be controlled by a valve operated by a diafram or piston upon which the steam pressure is balanced by a weight or spring, the variations in pressure can directly and automatically be made to vary the speed of the grate-motion. This system for a constant load or one which varies within narrow limits has the advantage of replacing hard and exhausting human labor by substituting for it

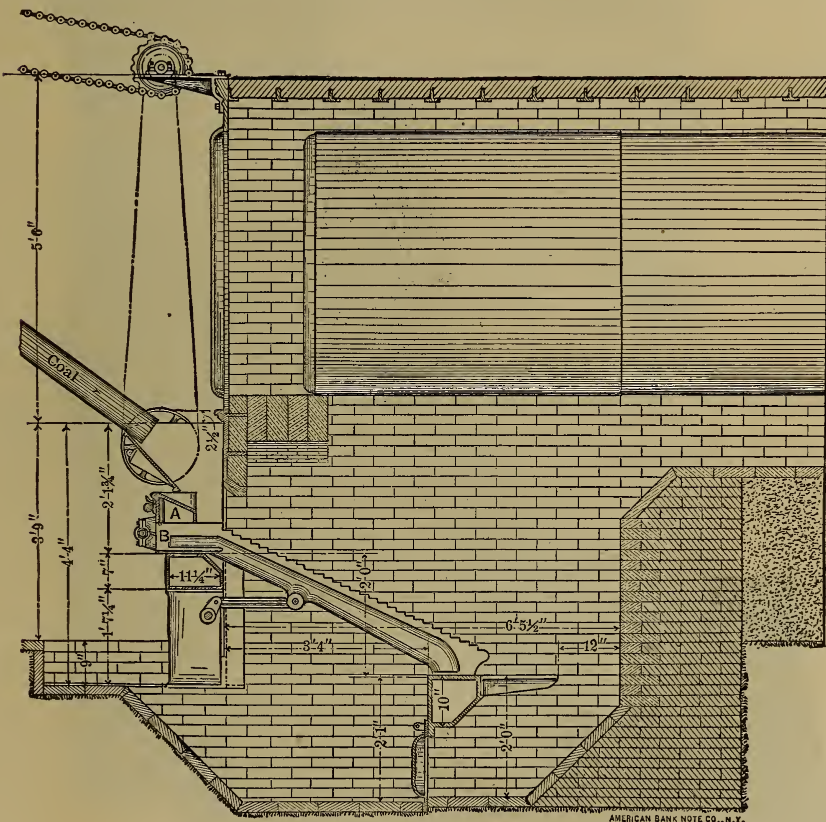


Fig. 272.

intelligent control of inanimate force. A form is also in use where the inclined or step bar is operated and the fuel fed down the bar from each side toward the center, where it escapes as ash at the bottom.

Mechanical stoking has not done its best work with the hard varieties of anthracite coal with which the fireman's labor is the least. Fig. 274 is at its best in comparison with the others with those bituminous coals which are fat and which melt and cake with the heat.

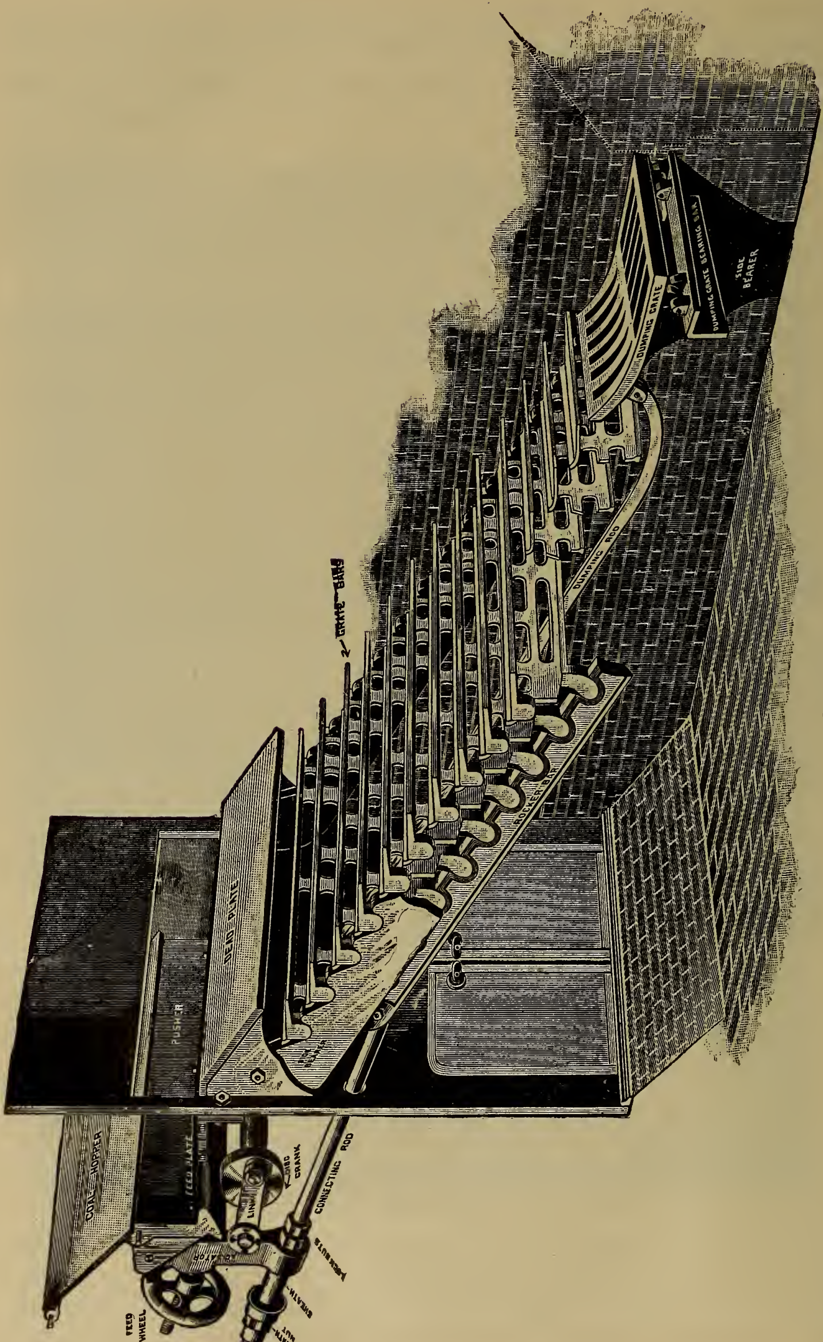

COAL HOPPER
FEED PLATE
PUSHER
DEAD PLATE
GRATE BARS
ROCKER BAR
DUMPING ROD
DUMPING GRATE
DUMPING GRATE BEARING BAR
SIDE BEARER
DISC CRANK
CONNECTING ROD
FEED WHEEL

Fig. 273.

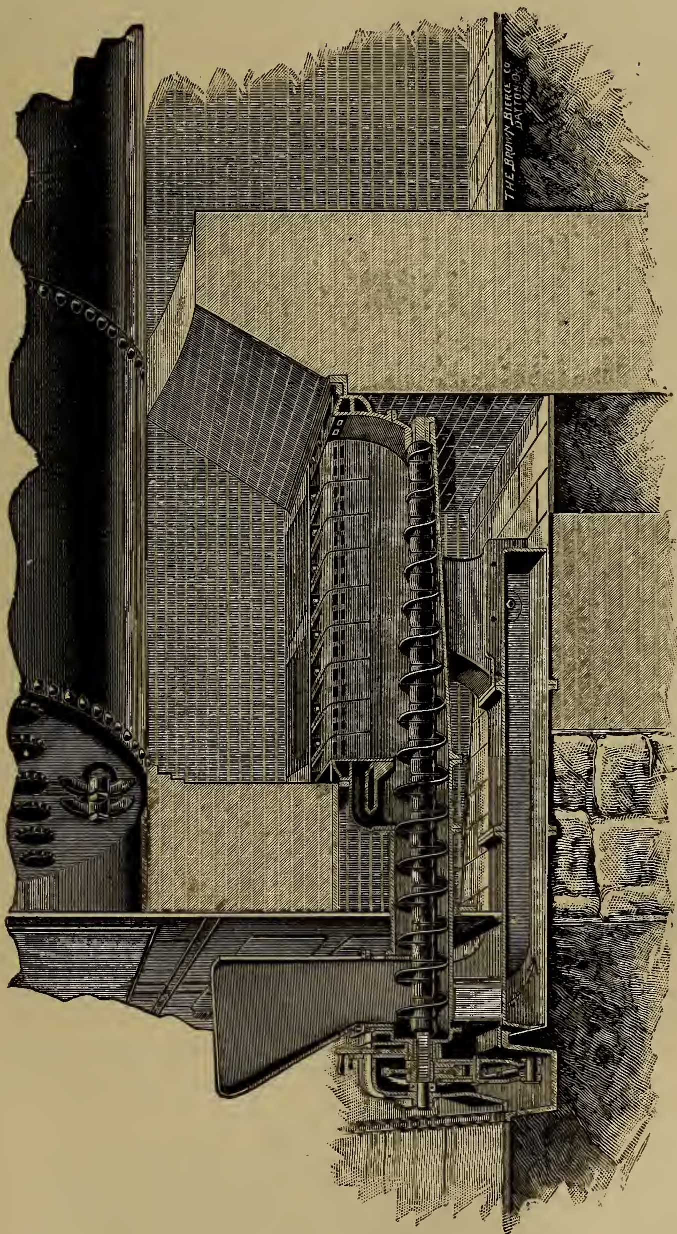


FIG. 274.

The objections to the traveling type of grate in stokers has been that only a part of the nominal grate area is covered with heat-giving fire. The front inches are covered with black coal, and if the travel is slowed

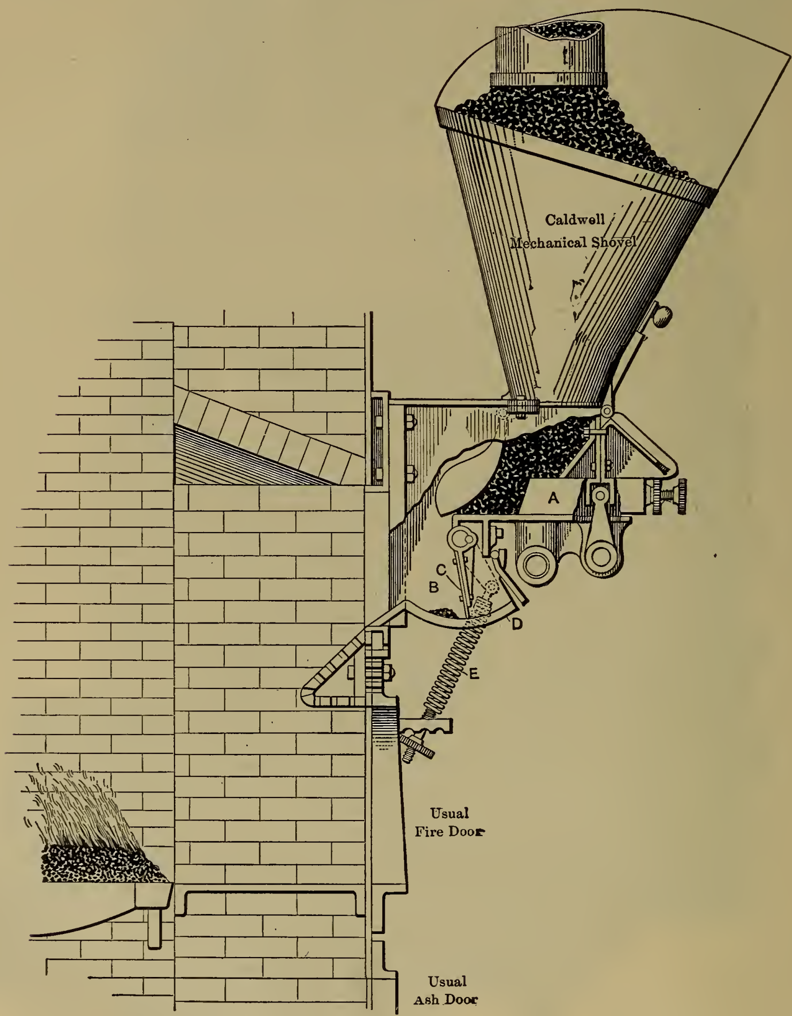


Fig. 275.

down to favor the front ignition, the back end is not effective unless the fireman in charge is specially attentive. The grate also gets no protection from any ash on its surface, and burns out sooner than the stationary grate with hand-firing. A type of mechanical shovel stoker is in use on locomotives and a form for stationary use with overhead pocket and feeding-tube is shown in Fig. 275. The coal is mechanically forced forward by the ram A into the hollow cylinder D and from the latter projected forcibly by the spring action of E through the swing of C when the spring is released. The impeller C has a graduated sweep, and throws the coal a short distance when the swing is limited, but all the way to the back of the grate when the swing is ample. Grates eight or more feet deep can be covered. The fireman cleans and slices through the ordinary door, so that some of the advantages of closed front stokers are sacrificed in order to work with the less costly grate.

224. The Forced or Induced Draft by Fan and Motor or Engine. In this same group of boiler plant auxiliaries is the engine or motor for driving the fan or blower which furnishes air to the fuel in the furnaces as discussed in paragraphs 127–129. In electric power plants these fans will be motor driven; in others they will have their own engines.

225. Preheating the Feed-water. Since the fire is the source of all heat energy (paragraf 105) the heat to raise the feed-water from its temperature in the suction-tank to that at which the water boils at that pressure must be supplied, as well as that necessary to make the water into steam. If the total heat of each pound of steam be called T and the temperature of the unheated feed-water be called t, the total units of heat furnished to that pound may be called $T-t=x$. If, now, by means of heat which would otherwise be wasted and lost this feed-water be preheated before entry into the boiler to a temperature t', then the heat to be supplied from the fire will be the quantity $T-t'$, which may be called y. Then the quantity $x-y=t'-t$ will be the saving of heat per pound of water due to such preheating. Then for any time in which a quantity or weight of water W is fed to the water, the percentage of saving from such preheating will be

$$\frac{W(x-y)}{Wx}=\frac{100\,(t'-t)}{L+T'-t},$$

where L is the heat of vaporization of water at the pressure corresponding to the boiling temperature T' attaching to that pressure.

This waste heat utilizable for preheating the feed-water is to be gotten either from the products of combustion in the flues on their way to the chimney, or from the heat rejected in the exhaust steam from the engine cylinder. Flue heaters are usually called economizers. When the engine operates condensing (paragraf 500) the pressure in an exhaust steam-heater will be less than atmosphere, and special conditions must be regarded, to be referred to in paragrafs 227 and 306.

226. Flue Heaters or Economizers. Flue heaters are of special value with water-tubular or sectional boilers of short period of contact between tubes and gases; or when for any reasons the flue temperature is high because of a high rate of combustion. Fig. 276 shows the usual plan of arranging the economizers with a by-pass flue to the chimney: Fig. 278 the general lay-out, and Fig. 277 the end elevation. The ordinary proportion of pipe is to give $4\frac{1}{2}$ to 5 square feet of heating surface for each boiler horse-power, using 4-inch tubes about 9 feet

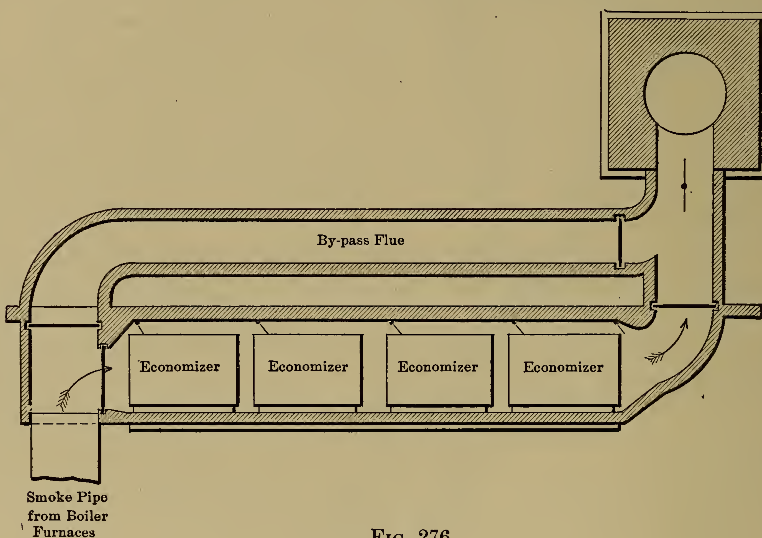


FIG. 276.

long. The scraper outfit is necessary to keep soot and tarry matter from making a non-conducting cake on the tubes by condensation.

The advantages of the flue-heaters are that they heat the water to a high temperature, and higher than the exhaust-steam heaters when other things are equal. The exhaust-steam is apt to be at a temperature not much above boiling-point, whereas the flue gases may easily be hotter than 500° F.

The objections to them are:

(1) That the difficulties from unequal expansion are very great, and unless special care is taken both in manufacture and design these will cause leakage.

(2) They are exposed to corrosion on the outside by the gases from most of the fuels, and particularly when a light covering of soot has coated the outside of the coil with an absorbent covering which holds moisture and acids in contact with the metal.

(3) When feed-water is not circulating through the coils so as to keep them full of water, they will make steam which will escape through the check-valve into the boiler, leaving a part of the heater exposed to overheating.

(4) They require to be cleansed from soot or tarry deposit by careful scraping in order to be kept efficient. The formation of scale within the pipes will take place with waters having solid matter in them. Figs. 277–8 show the usual form of economizer with provision for external scraping, which becomes particularly easy with the vertical type.

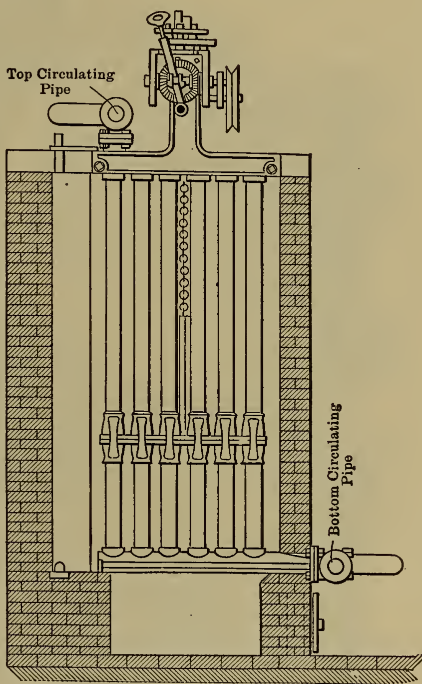


Fig. 277.

It is safe and convenient either to put a relief or safety valve in connection with an economizer, or else to see that no stop valve intervenes between it and the relief valve on the boiler. The economizer is in fact a boiler, and if the pump valves are tight, the only escape of pressure of steam formed in the heater during a stop must be forward through the boiler check. If this outlet is closed the pressure may accumulate dangerously.

Feed-heaters are always placed between the pump and the boiler, to give solid water for the pump and to keep its water cool as possible.

227. Exhaust Steam Feed-water Heaters. In a steam-engine which discharges its exhaust steam at the pressure of the atmosphere or above it, there remains in that steam at least the amount of heat which made water at 212° F. into steam at atmospheric pressure. This is 966 units of heat per pound. If this heat could be abstracted by bringing the steam into contact with the feed-water, the heat otherwise wasted would be economized, and this is the basis and the limit of heaters of this class. For if the feed-water be put at 50° F. as an average temperature for the year, and it be assumed that in an efficient heater the temperature of the feed could be raised to 200° F., or through a range of 150°, then the entire weight of exhaust steam from the engine W with

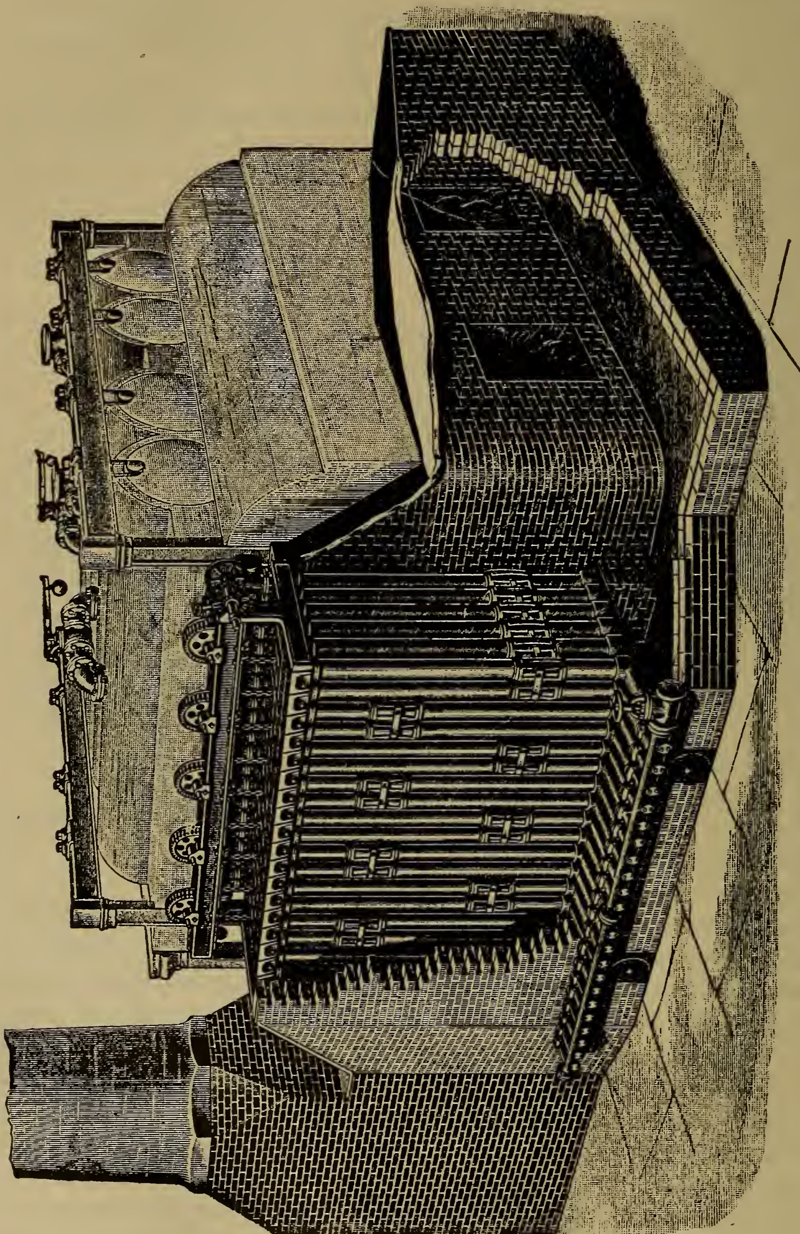
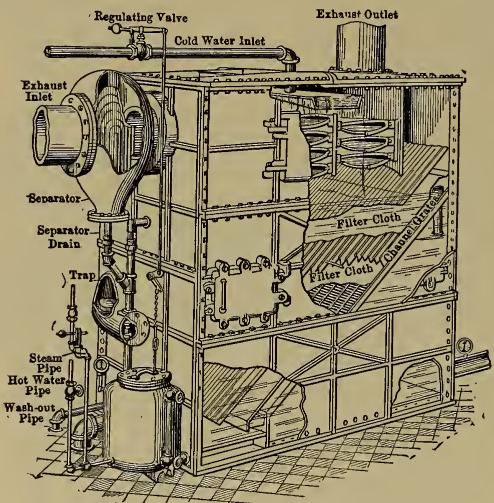


Fig. 278.

a heat-carrying capacity of 966 units per pound could heat a weight of feed-water w through the 150° range, since the equality must prevail that $W \times 966 = w \times 150$ less losses by radiation and in transfer. If the relation between 150 and 966 be called one-sixth to allow for some loss, then the weight of water undergoing heating in the heater should not exceed six times the weight of steam passing through the heater in the same time, else the heater will not heat it so hot as above computed. Stated otherwise, if the weight of steam be called the same as the weight of water which is to replace it in the boiler, then the heater can only utilize one-sixth of the heat of the steam exhausted, and the rest must be saved otherwise if at all. If the engine is to operate condensing, with an exhaust pressure below atmosphere and a temperature below 212°, there will be fewer units of heat to be imparted to the feed-water than in the foregoing computation. If, however, the auxiliaries are operated, as is often the case, with an exhaust to atmosphere while the main engine is condensing, it may be worth while to arrange two feed heaters in tandem for the feed-water. The first one is in the main engine exhaust circuit, heating the feed-water part way; the second receives the feed-water from the first and is heated by exhaust from the auxiliaries. Should it happen, however, that the weight of the atmospheric exhaust from the auxiliaries exceeds the above one-sixth of the total feed-water used in the plant, then the first or main engine heater would be an unnecessary complication and expense, since the auxiliary exhaust would handle all the feed-water which the plant required, and effort should be directed to improve the vacuum and lower the temperature of the main engine exhaust. The condensing engine exhaust will give from 110° to 120° as the feed-water temperature.

Feed-water heaters are usually designed without specific reference to any one set of conditions of feed-inlet or exhaust steam temperature. Hence an average proportion gives one square foot of heating surface in closed-circuit heaters for each 90 pounds of water per hour passed through the heater when the steam is above atmospheric pressure. If the water rate be called 30 pounds per hour (paragrafs 8 and 10), this is one-third of a square foot per horse-power. The ratio of one to thirty-five or thirty-six (paragraf 9) which this bears to the heating surface in the boiler results from the fact that the heating range is much lower in the heater than in the boiler, and the ease and speed with which steam parts with its latent heat as compared to its reluctance and slowness to absorb that same heat. In condensing engine heaters, the heating surface should be increased to one and one-half square feet for the same weight of water. As the heating and absorbing masses draw near to each other in temperature the rate of transfer of heat to the cooler body grows less.

There are two great classes of exhaust-steam heaters. The first are known as open heaters, in which the feed-water comes in direct contact with the exhaust-steam and withdraws its heat by direct condensation. The other class are called closed heaters, in which the steam and water are in separate circuits of pipes or coils which have the steam on the outside and the water within, or the reverse. The open heaters are in some respects the more efficient, since the steam and water come together, and since sufficient heat is often imparted to the water to bring it to that point at which it will precipitate the



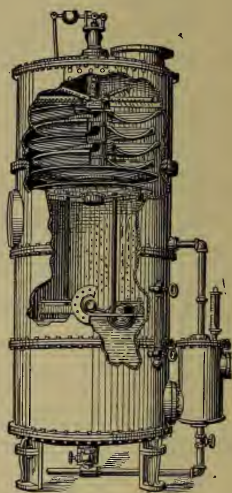


FIG. 279.

solid matter which it contains. This class of heaters have been called lime-catchers. Fig. 265 shows a form of this class of heater in which the water passes over the set of trays within the heater which are surrounded by the exhaust-steam. The hot water deposits its solid matter most rapidly in the thin films in which it escapes over the bottoms of the trays, making removal of such material complete, and the cleansing of the trays easy and rapid. Figs. 279 and 280 show other forms. Figs. 281 to 284 show types of the tube or coil-heaters of the closed class. The tubes are apt to be of copper or brass, in order to be rapid conductors, and are curved so as to yield easily to the condition of rapid expansion and contraction to which they are exposed. It is convenient to pass the steam through the inside of small coils, because the only deposit in the small tubes is the lubricant, which is not so difficult to remove. The arched or flexible forms given to the tube-plates and corrugation of the tubes also provide for these inequalities of expansion.

Such steam-heaters will act partly as surface condensers if they have an abundance of surface, but care should be taken that the resistance offered to the exhaust should not impose a back pressure upon the engine-piston, which should cost more coal to overcome than the saving of fuel caused by the heater. This is a matter of simple calculation when the back pressure is observed with the heater in action and out of action. Feed-heaters for condensing engines with the steam circuit below atmospheric pressure must necessarily be closed heaters.

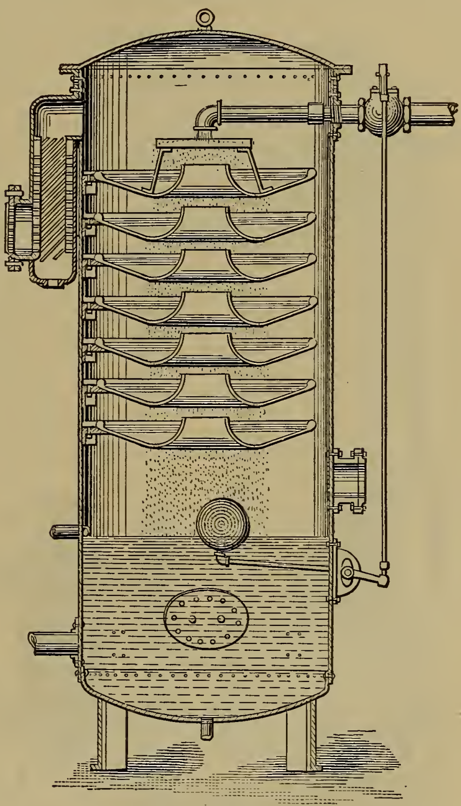


Fig. 280.

228. Superheating of Steam. The historic researches of Regnault which are embodied in all steam tables (paragrafs 23 and Appendix 566), have proved that for every pressure of steam there is a corresponding temperature and boiling point. This equilibrium between pressure and temperature is therefore a precarious one in any mass of steam and liable by change in either to be disturbed from that previously existing. If a pound weight of steam be enclosed in a vessel of the proper volume, and this volume is full of dry invisible steam gas — that is, if this volume be saturated with steam at that pressure, so that it will hold no more — the equilibrium of temperature and pressure can be upset in four ways:

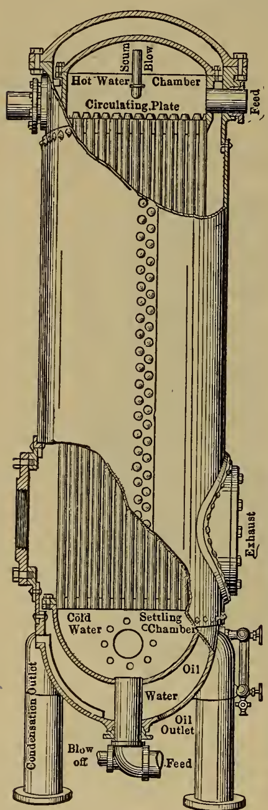
1. The pressure can be mechanically increased from without. There is not temperature enough to meet this higher pressure, and some of the weight of steam must go back to water in the form of a mist or vapor — even drops.

2. The temperature can be lowered. At once there is not heat enough to keep all that weight of steam as gas, and some of it goes back to water, or condenses, as in case 1.

3. The pressure can be lowered without cooling. The temperature is now too high for the pressure. If there was any water present in the steam it would evaporate; if not, the steam, being at a temperature higher than the tabular value for the pressure, is superheated.

4. The temperature can be increased. The pressure will go up a little, since the steam acts like any gas receiving heat. As in case 3 it is too hot for its pressure, or is by definition superheated. It can be cooled from this state without condensing, down to the condition of dry saturated steam, and condensation only sets in as in case 2, after the cooling down to the equilibrium point has taken place.

In an engine operating on the most economical principles in the use of steam (paragraf 303), the final temperature at the end of the piston-stroke is lower than at the beginning when steam entered from the boiler. Hence when new fresh steam comes from the boiler it finds itself exposed to the conditions of case 2, in contact with the piston and cylinder walls. Some weight of steam is at once condensed to mist or to drops of water upon the metal. As the piston advances, the pressure falls after cut-off of the admission; then the condition of case 3 is present, and the mist or drops evaporate into gas, taking the heat necessary for this transformation from the walls or from the steam. All steam and water go out together as mist or gas when exhaust-valves are opened. This condensation and reëvaporation is therefore twofold: "initial," due to cooler masses of metal, and secondly, that due to expansion and the doing of work. It explains part of the cause of missing water in the engine water-rate (paragraf 8).



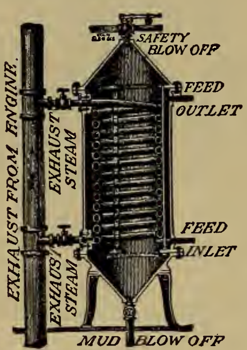


Fig. 281.

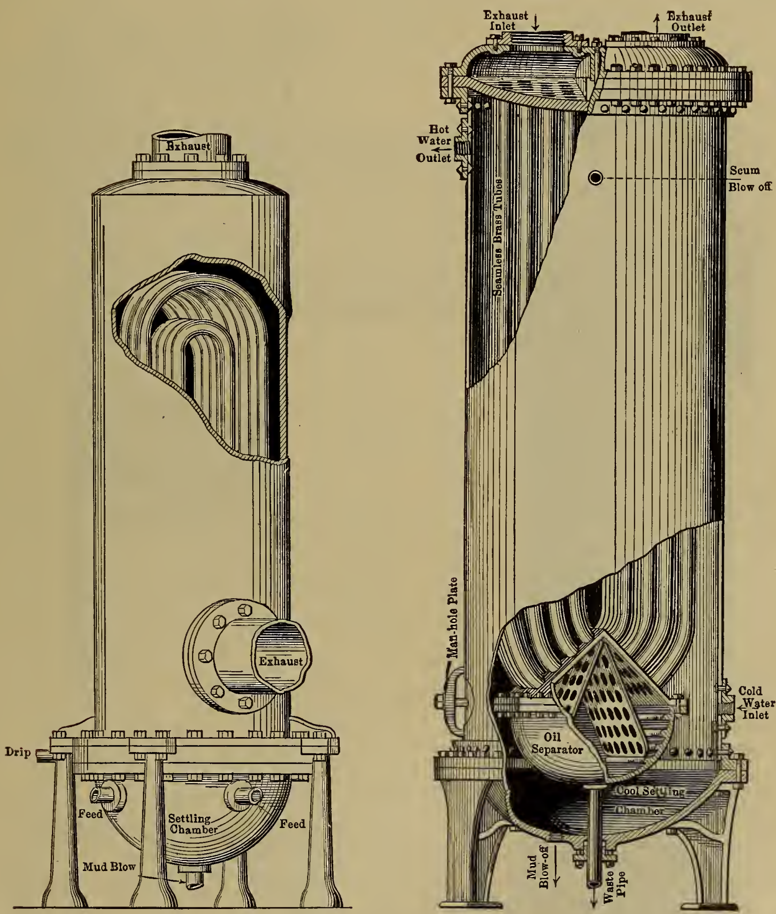

Exhaust
Exhaust
Drip
Feed
Settling Chamber
Feed
Mud Blow
Exhaust Inlet
Exhaust Outlet
Hot Water Outlet
Seamless Brass Tubes
Scum Blow off
Man-hole Plate
Cold Water Inlet
Oil Separator
Cool Settling Chamber
Mud Blow-off
Waste Pipe

FIG. 282.

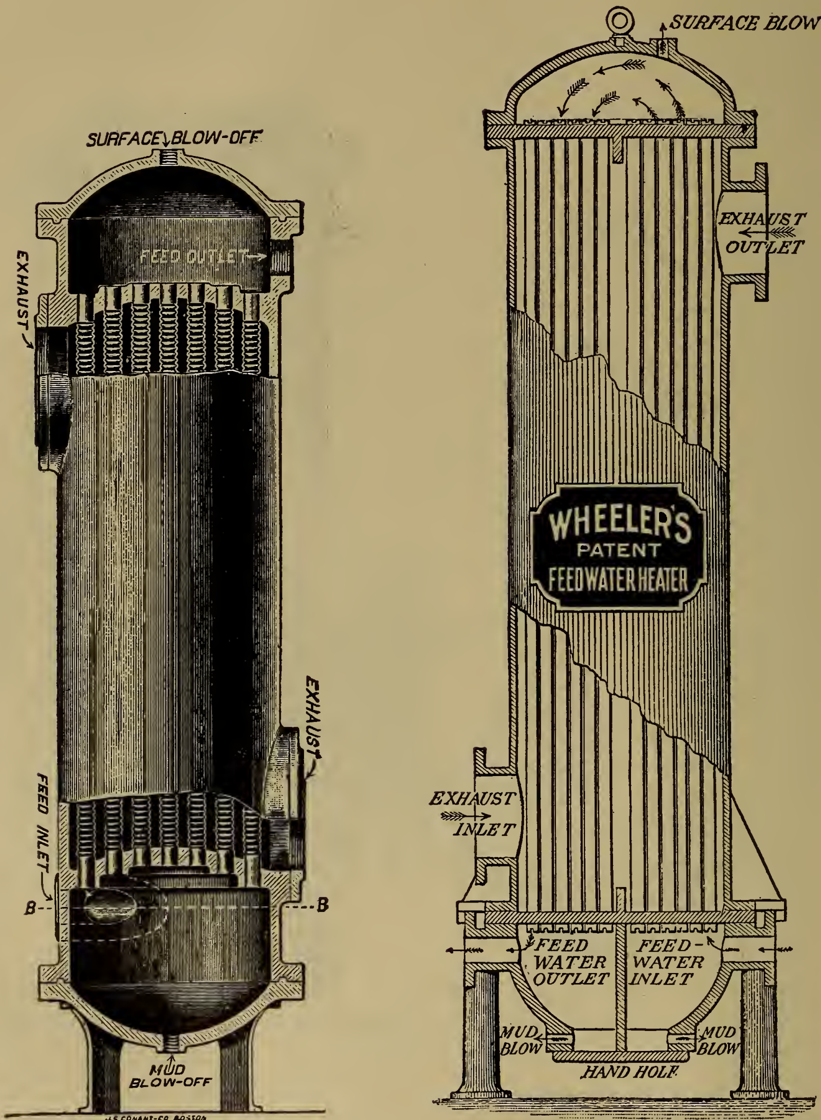

SURFACE BLOW-OFF
FEED OUTLET
EXHAUST
EXHAUST
FEED INLET
B
B
MUD BLOW-OFF
SURFACE BLOW
EXHAUST OUTLET
WHEELER'S
PATENT
FEEDWATER HEATER
EXHAUST INLET
FEED WATER OUTLET
FEED-WATER INLET
MUD BLOW
MUD BLOW
HAND HOLE

Fig. 283.

229. Methods of Superheating. If the steam on its way from the disengagement area of the boiler (paragraf 48) which it leaves as wet steam or as dry saturated steam, according to circumstances, can be made to pass over or through an additional heating surface where it may undergo heating as in case 4, the initial condensation loss can be

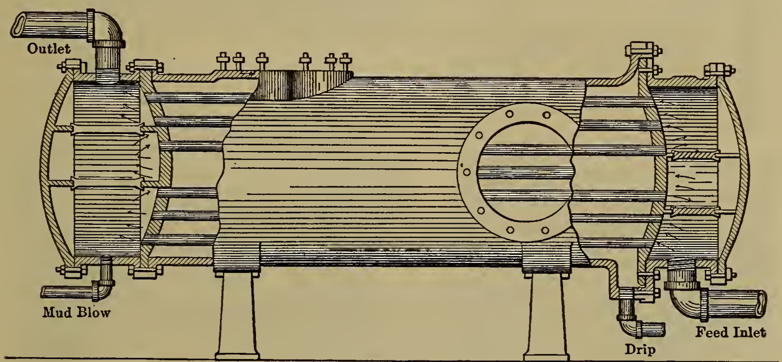


Fig. 284.

prevented largely or wholly. The steam can be cooled enough to heat the metal to the steam temperature, and leave a margin of heat above that temperature at which condensation begins.

This superheating can be effected in two ways. The steam can be passed through pipes in the combustion chamber of the boiler, so as to use heat which would otherwise be wasted. This is called indirect superheating: the other plan is to use a special furnace separately fired and for this purpose alone. This is called direct superheating. The rise of steam temperature above that of saturation of from 75° to 150° F. is called low superheat; if from 150° to 225° it is medium superheat; above 225° and rarely exceeding 300° will be called high superheat. If the steam be at 150 pounds pressure, and a corresponding saturation temperature of 354°, the total temperatures will be from 430° to 650°, which are so high as to give trouble in the directions to be referred to later in paragraf 231.

Fig. 285 shows indirect systems of superheating where in a boiler of shell or sectional type an additional heating surface of pipe has been introduced to absorb heat otherwise wasted. Fig. 286 shows a direct system where the heat is derived from a special fire and grate. In the direct system there is better control of the degree of superheat and it can be higher than in the other system, where just in proportion as the boiler proper is well designed, there should be little excess of heat available

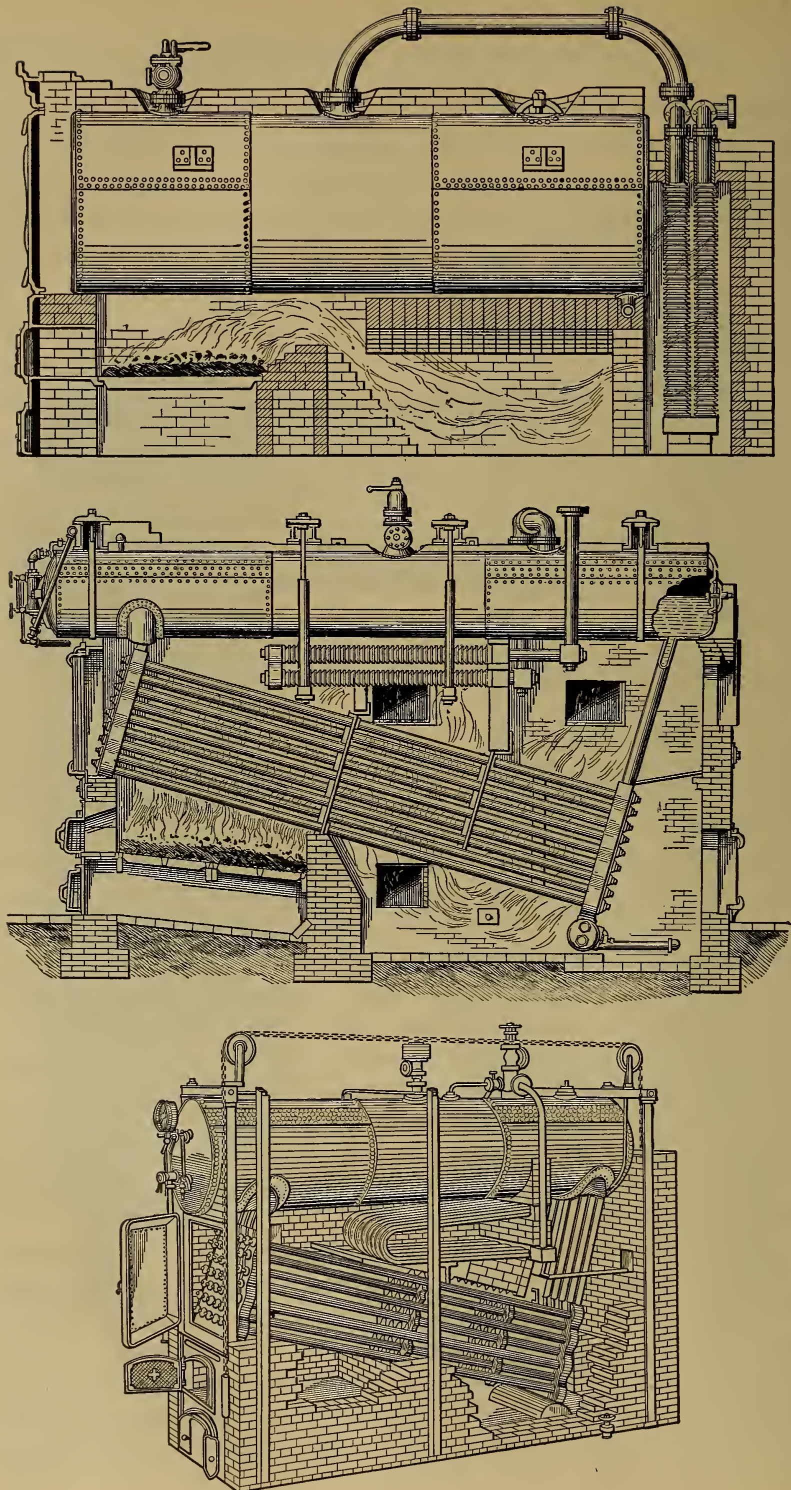


Fig. 285.

for superheating, and the temperature of the gases on leaving the heating surface should not exceed 600° (paragraf 107). Fig. 287 shows a locomotive superheater and reheater in the smoke box.

Superheating has been considerably applied in the locomotive where the conditions of use in the external air make the condensation losses excessive. The apparatus is introduced into the smoke-box and in one design certain of the upper tubes enlarged for the purpose. Tubes of one and one-fourth inches diameter lead from a header connected to the dry pipe in one design nearly back to the rear tube sheet and then

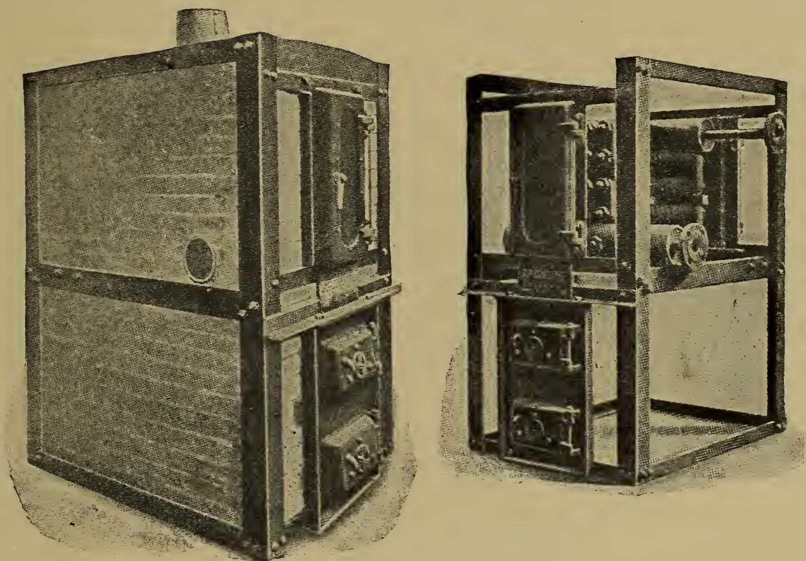


Fig. 286.

forward again to a second header leading the steam to the cylinders. In a test of these superheating coils, with 193 square feet of superheating surface, a superheat of 122° was obtained when the boiler was evaporating 7 pounds of water per square foot of heating surface. At an evaporation rate of 15 pounds of water per square foot, the superheat was 188°. In some boilers of coil or sectional type, superheating occurs in such part of the boiler as is exposed to heat of flame or gas above the water line, or beyond it (paragraf 100).

230. Advantages of Superheating. The advantages of superheating the steam above the temperature of saturated steam at that pressure are

1. A lowering of the steam consumption or water rate by lessening

or eliminating the initial condensation in the cylinder from the relatively cooler piston cylinder head and side walls.

2. A reduction in the amount of condensation during an expansion after cut-off. This is more surely attained by jacketing the cylinder and head with hot dry steam, but the heat to eliminate such condensation during work is supplied from the heat of the jacket steam made by the fire, and condensation in the jackets should be charged to the

Fig. 287.

consumption within the cylinder also. If superheating is done by heat otherwise wasted, and the steam can be thoroughly dry when exhaust occurs, the expense of the jacket construction and its consumption have been saved to the fuel burned. Adding these together, an average economy of steam would appear to be represented in tests where the water or steam rate per I.H.P. was reduced by superheating from 24 to 27 pounds to 20 to 22 pounds per hour. This is an economy or saving of 17 per cent.

3. It is better to attack condensation by superheating than by the hot steam jacket, since the latter furnishes heat continuously, and

must therefore be heating the exhaust steam on its way out and to waste. The cooling takes place from the exhaust in any event, but the loss is less.

4. The adjustment of the point of cut-off with saturated steam has to be earlier than with superheated steam, in order to secure the same water rate in order to offset the condensation loss. It is of advantage to reduce the degree of expansion to get equal distribution of piston effort.

5. The superheating expands the volume of the steam for a given weight at any pressure. Hence the cylinder volumes can be increased in the later cylinders of multiple type and secure the same degree of expansion as with saturated steam.

6. Due to this, the engine can be speeded up, since the lowered density or weight per cubic foot enables it to attain higher velocities through pipes and passages without throttling effect. This increase in velocity may be from 40 to 60 per cent.

7. A cheaper engine or one with fewer cylinders can secure the same economy with superheated steam which is secured with saturated steam only with a continuous expansion in several cylinders. The multi-stage expansion has for one of its objects the lowered temperature rate of change due to the less pressure range in each cylinder.

231. Disadvantages of Superheating. To secure the superheating, certain difficulties must be entailed.

1. Deterioration and expense of maintenance of the superheating surface, exposed to high heat on one side, and only inadequately cooled upon the other. These metals oxidize, waste, and give trouble at the joints.

2. Valves and their casings become deformed and leak, or by expansion of certain types, they stick. Corliss valves do not work at their best with superheats of over 120° or total temperatures of 500°. Poppet valves of two or even of four seats avoid the troubles from defective lubrication. Piston valves are the alternate type.

3. Packing for the piston rod cannot be of the fibrous sort or contain rubber or any carbonizable material. Metallic packings of alloys resembling Babbitt metal — say 80 per cent antimony and 20 per cent tin — give best results.

4. Lubrication cannot be done by mixing the oil with the incoming steam. The latter makes a gas of the oil and it produces little effect. The oil should be of the mineral or non-oxidizing class, and should be introduced mechanically by a pump into the cylinder and upon the rods and valves. Graphite in some flocculent form avoids these troubles.

5. The castings of the engine must not be of complicated form, liable to warp by heat. Prolonged exposure of cast iron in particular but also of all forms of iron to the temperatures of superheated steam causes them in time to grow permanently larger, due probably to some molecular rearrangement of the crystalline or other structure.

6. The heat used to superheat the steam from an extra fuel supply must be charged to the fuel rate. The economy of superheating if stated in terms of fuel economy is usually less than when stated in terms of steam-consumption and water-rate. That is, a steam economy of 20 or even as high as 30 per cent will be cut down to an economy of 10 and 20 per cent in terms of fuel.

7. The offset of advantage No. 7 is that in engines which have been bought and paid for, the economy of superheating is not as great in expensive multi-cylinder engines previously run on saturated steam as it will be in cheaper simple engines, non-condensing.

It is not advisable to give capacities of superheaters within too close limits, since so many factors enter which would influence the computations in any given case. In a general way, however, about two square feet of heating surface go to each horse-power in indirect systems aiming to get 150° of superheat.

The materials which resist heat best do not resist pressure equally well. Cast iron, which stands heat with slowest oxidation, is unreliable by itself. Hence an ingenious combination has been made of cast iron washers forming flanges which are slipped on and shrunk over a steel pipe. The flanges become red hot and conduct heat to the inner tube while protecting the latter. The steam is led in a thin film along the superheating surface by the use of an inner tube which acts as a displacer or filler of the center of the main tube. The other features of successful design must be adequate support to prevent deformation by the weight of the superheating coil in case of high heat; accessibility for renewal or repair; free expansion under varying heats. To save the superheating surface when the engine is not running and steam is not circulating, some designers have arranged to have the superheater flooded at such times. This is objectionable from the care which is required when the superheater is to be started anew, lest the engine get the water in objectionable amounts. It also gives trouble from scale therein.

232. Combustion Indicators. There should also be included under auxiliaries of the boiler-plant as apparatus but not as machinery the means used to inspect and control the effectiveness of the combustion in the furnaces. The complete laboratory equipment for volumetric analysis of the products of combustion is known as the Orsat apparatus,

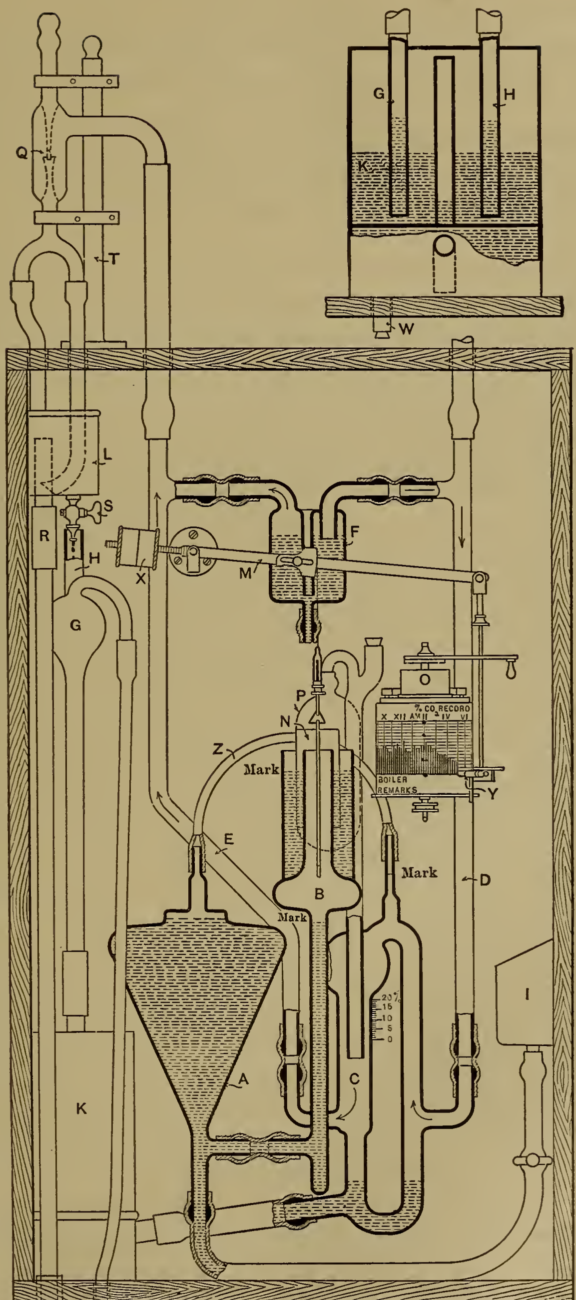

G
H
K
W
Q
T
L
S
R
H
X
M
F
G
O
P
N
Z
Mark
% CO2 RECORD
BOILER
REMARKS
Y
E
B
Mark
Mark
D
I
A
C
K
20%
15
10
5
0

Fig. 290.

and includes three vessels for the absorption of carbonic acid gas (CO_2), carbonic oxide gas (CO), and oxgyen. This has been fully described elsewhere * and depends on the successive withdrawal from a given volume of gas of the foregoing elements, measuring the resulting volume after each element has been withdrawn. For practical use and guidance of the fireman, a simpler and continuous apparatus with a recording apparatus has been successfully devised in Germany and America, in which only the percentage of CO_2 is recorded by a curve, on the ground that if the CO_2 is kept high, the other losses in excess of oxygen due to holes in the fire and from incomplete combustion to CO due to too little air, will be kept low. The hydrogen in the fuel is sure to burn to H_2O. If the fuel is low in hydrogen, and be called carbon, and is burned with a dilution of air of 100 per cent excess, the maximum carbonic acid will be about 14 per cent of the products of combustion. For the proportion of CO_2 in theoretically perfect combustion with just enough air is 3.66 pounds of CO_2 in 12.3 pounds of CO_2 and nitrogen mixture in the products of combustion, or the CO_2 is 29.7 per cent by weight. If double the weight of air is added, for air over the fire, then the products should have CO_2 in the proportion of 3.66 to 26.26, or at 13.9 for each unit weight of coal. Too little CO_2 in the chimney gases in the volume of the sample indicates excesses of air from leakage through holes in the fire, or through the setting: or if smoke is also present to the eye, that the carbon monoxide is in excess. Usually the oxygen or free air is in excess when the draft is powerful. Fig. 290 shows the apparatus used in one of these CO_2 recorders which has had widespread acceptance. The chimney gas is drawn in at Q by a water-injector: the caustic potash absorber of CO_2 is in the vessel A; the sealed bell N receives the unabsorbed gas under determined pressure and determines the height traced by the pen Y upon the chart driven by the clock mechanism in O. Then the tubes are scavenged by siphoning action of the water, and a new charge of fresh gas is drawn in and sampled. The flow of water through the cock S determines the frequency of the sampling and recording process. The fireman or others interested can inspect the curve of the chart and see exactly what needs to be done. At the 59th Street power house of the Interborough subway system of New York City, an economy was effected in reducing the stack loss of 22.7 per cent by 19 per cent of itself in hand-firing conditions, and with stoker firing of 12 per cent.

233. Water-meters. The equipment of measuring apparatus to give the water consumption of the plant is an important auxiliary. If the water must be paid for per unit used, the meters are a check upon

* "The Gas Engine," F. R. Hutton, p. 103, John Wiley & Sons, 1908.

waste. Increase in the water rate above the normal or standard consumption indicates some disarrangement or mal-adjustment; and every pound of water unnecessarily fed to the boilers means so much extra coal burned to evaporate it.

Self-registering or recording water-meters are therefore the most convenient and satisfactory; but if by reason of their cost they are not realizable, the records of the common type can easily be plotted from their readings.

Water-meters are of two classes: the shunt form, where only a part of the water passes through the meter, and the holotype class, in which all the water used goes through the meter. The objection to the first type is the uncertainty whether the water passing the meter is actually the computed fraction $\frac{1}{n}$ of what passes in both the main pipe and through the shunt or by-pass. The advantages are the smaller size and cost of the meter and the fact that the disarrangement of the latter does not stop the flow in the larger cross-sections.

Water-meters are positive or displacement instruments; or they are velocity indicators or recorders. The displacement type are in effect piston or plunger water-motors, whose piston strokes empty and fill a known volume at each reciprocation. The water temperature remaining constant, and the pistons having no leakage from wear, such meters should be closely accurate. The recording mechanism registers the number of strokes, which can be read in cubic feet and gallons when the displacement volume is accurately known. They usually cannot go at very high speed, and for large cross-sections are bulky and heavy. The velocity meters of small capacity have a propeller wheel which is turned by the flowing current of water. The speed of such revolution recorded by a counting mechanism gives the lineal rate of flow, and this multiplied by the cross-section gives the volume. Friction is the bane of such meters, retarding and rendering variable the response of the wheel to the real velocity of the water. Both types require strainers to keep grit and mechanical obstacles from the measuring chamber.

The Venturi-meter which has no moving elements or mechanical details, is the best water-meter for large cross-sections, and for very large pipes and considerable velocities of flow is the only possible or practicable one. Its principle is a theorem of Bernouilli.

Let a pipe of circular section have two cross-sections of differing areas in a section of limited length. The section should be made tapering from the larger area a_1 to the smaller section or gorge whose area is a_2, and then gradually be increased again to the full original size a_1. Let there be inserted in the pipe a pressure-measuring or observing apparatus such as a gauge or manometer at the point where a_1 is observed and another at the smallest section or throat. Let the head at a_1 be called h_1 and the head at a_2 be h_2. Then, since the quantity Q in cubic feet of water passing in the pipe per second is the product of the area in square feet into the linear velocity in feet, and since water is incompressible, it will be true that

$$Q = AV = a_1v_1 = a_2v_2 \tag{1}$$

where v_1 and v_2 are the respective velocities at a_1 and a_2.
But the incompressibility of the water and the consequent velocity variations at a_1 and a_2 will make the heads of pressure at these points to differ so that the relation will hold

$$h_1 + \frac{v_1^2}{2g} = h_2 + \frac{v_2^2}{2g}, \tag{2}$$

whence

$$v_1^2 - v_2^2 = (h_2 - h_1)\, 2g. \tag{3}$$

From equations 1 and 2 may be written

$$\left[v_1^2 = \left(\frac{a_1}{Q}\right)^2\right] - \left[v_2^2 = \left(\frac{a_2}{Q}\right)^2\right] = 2g\,(h_1 - h_2), \tag{4}$$

whence

$$Q = \frac{a_1a_2}{\sqrt{a_1^2 - a_2^2}} \sqrt{2g\,(h_1 - h_2)}\,. \tag{5}$$

But since the areas a_1 and a_2 are constant and known, the fraction in the above equation can be computed once for all and multiplied by the $\sqrt{2g}$; if this quantity be called K, then

$$Q = K\sqrt{h_1 - h_2}\,. \tag{6}$$

From the readings of the pressures at a_1 and a_2 the quantity flowing can be known. If the two gauges be so connected that the reading observed or recorded is the difference $h_1 - h_2 = h_3$, then the quantity is $Q = K\sqrt{h_3}$.

234. Concluding Comment. It may easily happen that auxiliaries listed in the later chapter under engine-room auxiliaries will be located in the boiler room of a particular plant. This will be done for convenience and the logic of its location there will be entirely defensible. But the more usual attachment of these to the engine economy and efficiency and the advantage of familiarity with the engine before discussing them are reasons for placing them in the second group (paragraf 21). The pumps and the forced-draft machinery are also called auxiliaries in general speech and in the accounting of the plant, but by the logic of Fig. 1 and the method of treatment these have been called essentials as they are, in fact, in themselves or in alternate form. Certain topics also concerning the plant as a whole, the testing for economy and efficiency of the boiler plant, and the economic questions are reserved also for later treatment after the engine plant has been made familiar.

The energy of the fuel having been liberated in the furnace and transformed into pressure of steam gas is now to be transmitted through piping to the engine mechanism which is to make it available in mechanical form.

PART IV.

CHAPTER XIV.

THE PIPING OF PRESSURE TO THE ENGINE AND ITS ACCESSORIES.

235. General. It is usual in important power-plants to isolate the engine from the boiler plant in a separate room or in a separate structure. The reasons for this include:

1. Better insurance rate by concentrating the fire and explosion risk.
2. Avoidance of injury to costly machinery from the gases and dust of the fire-room.
3. The higher standard of maintenance and upkeep of the machinery favored by such separation than is possible or worth while in the firing room.
4. Convenient development of each department by adding units, without interference.
5. The separation of the differing grades of service.
6. The easier differentiating of the accounting, where the generation of power and its consumption are separate.

There will doubtless be other reasons of weight in special cases.

It will be apparent, therefore, that the pipe which conveys energy in the form of pressure from the boiler and grate where it has been liberated and stored is as essential and vital a link in the chain of power production as either the boiler plant or the engine plant. Its disablement or breakdown stops the industry or productive processes dependent upon the power of the engine as completely as the annihilation of the boilers or the wreckage of the engine. If the financial loss or personal discomfort of those dependent on the continuity of the plant's working are sufficient, the designer may well expend thought and money upon the engineering of his transmitting pipe.

In small plants such as a portable contractor's outfit, which is none the less a complete plant, the problem may be simple and inconsiderable. In larger installations both pipe and necessary fittings become costly, and time for repairs increases with the sizes of the units, making such upkeep both costly and an economic loss as well from the loss of salable product. Sometimes such loss from a shut-down far outweighs in two days the entire additional first cost of a more elaborate and more carefully installed proposition.

236. The Stresses in a Steam Pipe and its Requirements. The stresses to be resisted and the difficulties to be avoided or reduced to their least values in the design of a pipe system include:

1. Internal static pressure from the steam tending to rupture it along the cylinder elements. This pressure is Pd as in the boiler, and is resisted by $2\,tf$ (paragrafs 24 and 25).

2. Internal static pressure from the steam tending to pull it apart at the joints. This pressure is $P\pi\frac{d^2}{4}$ as on the head of a cylindrical boiler and is resisted by the flange-bolts, or the threads of the screws at the ends of the pipe-lengths, and by the greater strength of the pipe ringwise, measured by πdtf. Failure of good pipe in the body of the material from this stress is unknown. The joints always fail first.

3. The static tension lengthwise of the pipe due to its expansion by heat of the steam within it. If the pipe was put up cold, or at 65° F., and when at work is exposed to steam of 150 pounds pressure whose temperature is 365° F., the steel of such pipe will be longer when hot by .00000689 of itself per degree per unit of length. If the length be 100 feet and the temperature rise 300 degrees, the increase in length will be 0.21 of one foot or two and one-half inches in that length. The force exerted by this heat change is measured by the completeness of the anchorage of the pipe to any fixed object. If the fixation is complete, it is πdtf, or the tensile strength of the hollow cylinder whose thickness is t and whose diameter is d, in pulling, and the compressive or buckling resistance of such a column if it is pushing. Expansion should not be resisted, but should take place freely without pushing or pulling any anchorage or tearing apart any joints.

4. A dynamic stress of unknown and scarcely calculable intensity (the quantities being unknown) resulting from a water-hammer phenomenon in the pipe. When by condensation of steam to water which is not removed by drainage a mass of condensed but hot water under pressure gathers in a pocket or recess in the pipe during a period of arrested or retarded flow of steam, there is great danger on a reëstablishment of flow that this water will be picked up bodily by the rapid flow of steam gas over it. By its inertia and mass it will resist diversion of its direction of flow by elbows or tees, but will be brought up against such fittings with a blow whose intensity is measured by the mass of water multiplied by its velocity. If the mass is large enough,

and moving fast enough, it has force enough in that blow to crack fittings or break them into pieces if of brittle material, or to split the pipe lengthwise. The noise of such blow is alarming and when continuous becomes impossible to endure. Water hammer may also occur in the warming-up process. Condensed water gradually rising in temperature in pockets of the pipe as hot steam flows to it from the boiler reaches the critical temperature at which it is ready to flash into steam at any lower temperature than that of the steam pressing upon its surface. The opening of a valve or any sudden cooling beyond the water pocket causes a rush of steam towards the lower pressure which is not met from the boiler by reason of friction in the pipe. The heated water changing suddenly and in part into steam explodes the water pocket as it were, carrying the unevaporated water with it and causing the water-hammer disaster if the masses are large enough. This phenomenon is the same as the " concussive ebullition " alleged to be a danger in the boiler itself. The only way to prevent this disaster is adequate drainage of all pockets, so that there can be no water accumulations in the pipe at all. The pipe cannot be made strong enough of itself.

5. A thermal loss of heat and pressure by radiation to the air around the pipe, increasing the amount of the condensed water above, and by friction in the pipe making the pressure at the engine less than that at the boiler. This drop of pressure is reduced by effective non-conducting coverings surrounding the pipe.

6. A frictional loss from resistance to flow in the straight lengths of varying cross-section, and at the bends, outlets and controlling valves. Such loss appears as a difference between the initial pressure at the boiler, p_1, and the resulting terminal pressure at the end of the pipe, p_2. This difference $p_1 - p_2$ will obviously vary directly as the length of the pipe in feet L; as the density or weight per cubic foot of the steam at the initial pressure p, since this is in effect the moving force which keeps the contents of the pipe in motion, or is equivalent to a pressure head in a hydraulic problem: since the quantity of matter undergoing the retardation is the cubic feet of steam per minute Q multiplied by w. The loss will vary inversely as the diameter d in inches in some power, and will vary also as that diameter multiplied by a coefficient to take account of experimental data that the loss of head is proportionally greater in small pipes than in large ones. One of the most widely accepted formulæ for flow in pipes is D'Arcy's, which takes the form

$$p_1 - p_2 = \frac{Q^2wL}{c^2d^5},$$

from which

$$Q = C\sqrt{\frac{(p_1 - p_2)d^5}{wL}},$$

or, since

$$W = Qw,$$

$$W = C\sqrt{\frac{(p_1 - p_2)d^5}{L}}.$$

If the D'Arcy coefficients be introduced for c the values of c corresponding to each d must be taken from such a table as Table XII.

TABLE XII.

COEFFICIENTS FOR RESISTANCE TO FLOW IN PIPES.

For diameter d in inches	$\frac{1}{2}$	1	2	3	4	5	6	7	8
Corresponding value for c is	36.8	45.3	52.7	56.1	57.8	58.4	59.5	60.1	60.7
For diameter d in inches	9	10	12	14	16	18	20	22	24
Corresponding value for c is	61.2	61.8	62.1	62.3	62.6	62.7	62.9	63.2	63.2

A more usual avenue of approach to the problem for steam is by an assumption of a linear permitted velocity in feet through the pipe. Such velocity in feet per minute V will be

$$V = \frac{Q \times \pi d^2}{4 \times 144},$$

and experience shows that 100 feet per second or 6000 feet per minute is not too high so as to cause excessive resistance or loss of head on the one hand, or make so large a pipe as to be of prohibitory cost on the other. For example if the gage-pressure be taken at 100 pounds and the value of w found to be 0.264 pounds and L be called 100 feet, then

$$d = \frac{8.8}{p_1 - p_2}.$$

This means that a 9-inch pipe (or 8.8-inch) 100 feet long carrying 100 pounds pressure would lose one pound of pressure: or, that a one-inch pipe would lose 8.8 pounds. These values are within the limits of good judgment.

7. A final requirement is that the hollow pipe cylinder considered as a girder, having a wall-thickness as determined by (1) and a diameter determined by (6), carrying its own weight, and that of contained water if any, and the weight of non-conducting covering added so as to meet (5) shall not flex between its supports enough to bring an extreme fiber stress on the elements of the pipe itself, and, even more important, that there be no tendency to open the joints. The pipe is therefore in the condition of a continuous girder resting upon many supports and uniformly loaded or nearly so. Heavy valves make a concentrated load on the two adjacent supports, but the load is fixed, and its effect should not go beyond them. This is a problem of the supports rather than of the pipe.

237. The Material for a Steam Pipe. There will be four standards in the engineering of steam lines. High-pressure work — say above 150 pounds pressure — and low pressure work and exhaust piping form two of these, and the other two are large diameter work — say over 6 inches diameter, — and small work. In small work the engine builder often fixes the steam-pipe size by the steam inlet into the cylinder. Knowing the cylinder volume and the proposed number of traverses per minute, the total steam volume is found: this volume divided by the limit of 6000 gives the pipe area and hence the diameter in feet or fractions of a foot.

For small diameters and low pressures the standard lap-welded steel pipe is safe enough. This can be bought up to eighteen inches in diameter without extra price in standard sixteen-foot lengths. It has been tested up to 2000 pounds pressure for its weld but should not be used at one-tenth of this. For higher pressures and smaller diameters, the extra heavy or hydraulic pipe will be specified, the greater thickness giving better and more reliable welds. For larger diameters and medium and high pressures the pipe will be of riveted steel plate (Fig. 291), lap- or butt-riveted on the longitudinal seam, and flanged and welded or riveted by hand or power at the ends where the joints are to be made. Spiral welded pipe has not come into general use for heavy work in spite of the advantages of long length of its units. Spiral riveted pipe for light work and exhaust pipe is much used (paragraf 245 and Fig. 331). Cast iron is not reliable for high-pressure work by reason of the high modulus of elasticity, and its relatively low tensile strength. This makes

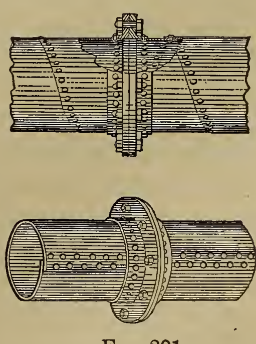


Fig. 291.

the pipe thick and heavy, and it should not be exposed to stresses of unequal temperature and cross-bending. Cast-iron fittings will still be used for joints and branches and for valve-bodies, even where steel is used in the run of the pipe. Drawn steel or alloy tubes are used in high-priced small diameter work for heavy pressures, since there is no weld in such pipe as it is drawn down over a mandrel from a solid cast ingot of proper weight and shape. Copper pipe has been used in marine practice where pressures were not great, because it could be curved so readily with gentle sweeps to get round deck-beams and past corners in contracted space. Such pipe was made for light work by taking sheet copper, bending it into a cylinder over a former and brazing the longitudinal joint, using brass filings as the solder. The joints were made by brazing flanges on the pipe. The next stronger stage was the use of electro-deposited copper, deposited on a former or core until the desired thickness was reached. The pipe was then hammered and annealed to secure ductility and homogeneity. The third step has been to wind a copper pipe with steel-wire outside of it to give the strength of the steel while retaining the ductility and flexibility for curving. After the winding, the reinforced pipe could be again placed in the electro-plating bath and an exterior shield of copper deposited outside the reinforcing wire. Copper has much less strength than steel but expands and contracts easily without injury to itself and resists corrosion.

The modern tendency toward higher pressures and their economies compels the pipe to be made like the steam and water drums of the sectional boilers, and to follow the lines of their practice in butt-riveted joints and the use of high-class steel of specified quality (paragraf 26). Cast steel is not much used except in fittings.

238. Steam-pipe Joints and Fittings. The inconveniences of manufacture and shipment of extra long single lengths of pipe usually make such units costly to buy. Twelve-, sixteen- and twenty-foot lengths will be the usual standard. How shall the successive lengths be joined together strong and tight? In small-diameter low-pressure work the end of the pipe will have a thread cut into the metal of the pipe by a die or threading machine. Such thread will have a taper of one in sixteen or three-quarters of an inch to the foot, and for a length sufficient to get six or eight of the threads of such a screw into a corresponding female thread upon the next element in the line. By screwing up until the thread refuses to enter further under the compulsion of a pipe-wrench of sufficient leverage the joint is presumably tight as well as strong when both threads are standard. Such threads, however, being incised, leave the pipe-metal thinner at the bottoms of the threads than elsewhere and therefore the stretch lengthwise due to heat and pressure will be concentrated at these threads. The stretching loosens any scale of sesquioxide of iron which may have formed, and the pipe corrodes more rapidly here than elsewhere, and then the joint leaks under the thread.

If the threads are right-handed, or make up by screwing the turned element in the direction of clock-hands, the closing of the last joint between fixed objects like an engine and boiler is a problem. The introduction of a left-hand screw is one solution but is not used above 2-inch pipe. The union is the more satisfactory device (Fig. 292), giving a closing nut with a female thread at one end, and with a finished sliding contact surface internal to its other face. These make up on special fittings screwed on the pipe at the two ends to be joined. No springing of the pipe lengthwise is necessary with the union to get the

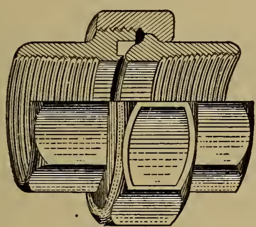


Fig. 292.

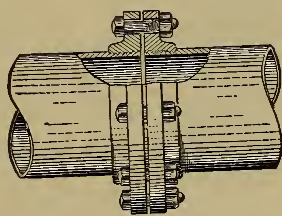


Fig. 293.

two threads into the closing fitting as with the right-and-left screw, and no difficulties appear from unequal entry of the two threads into such fitting, whereby one thread refuses to go further while the other is not yet tight in place. The contact faces of the two union elements exposed to pressure and leakage are best ground surfaces; where this is not desired some compressible material of soft metal or fiber or rubber composition must be introduced as a gasket. New designs show an inserted contact ring of brass or alloy on the hollow face. The friction of large union nuts, and the difficulties of getting adequate strength even with malleable iron have kept these down to smaller sizes. The machine work on them makes them costly. Hence the flange joint will be used in all important work.

Fig. 293 shows a typical pair of flanges with the pipe ends screwed into them. The faces are bolted together with a gasket between, either of flat soft metal or of fiber or of manufactured composition with rubber and graphite or asbestos as probable elements. Corrugated or ribbed gaskets of copper or soft metal have wide acceptance, either plain or in combination with fiber or special material as in Figs. 294 and 295. The two halves of the flange are made up by putting a couple of taper steel pins into the bolt holes on opposite ends of a diameter, and then by a steel bar bearing against the sides of the

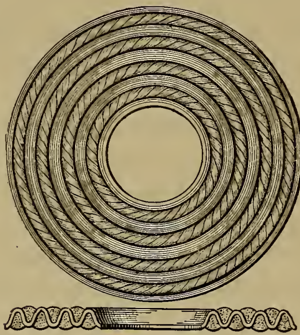


Fig. 294.

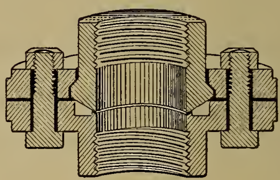


Fig. 295.

projecting pins on the face, the threads are made up to refusal. In Fig. 295 a special metal face is used, which can be ground. To improve the permanence and quality of the screwed flange joint, a number of modifications of the type of Fig. 292 have been made and have worked well.

1. To run the pipe through the flange until it projects beyond the face by a sixteenth of an inch, and then face the projecting end and face off in a lathe.

2. To form a recess in the face at the end of the threads, and to upset the end of the pipe by hammer-peening down into this recess, and face off in the lathe.

3. To form a projecting ring on the face of one flange, which fits into a corresponding annular groove in the other face (Fig. 296). The same purpose is secured by raising a part of one face and depressing an area of the same center and diameter upon the other, so that they shall interlock. These both are to lessen the stress upon a soft or non-metallic gasket to blow it out radially (Fig. 297). The interlocking, while tending to preserve alignment, is yet open to the objection that when a gasket is to be replaced or any repair to the joint is required in heavy pipe systems, a jack must be used to spring the joint open after the bolts have been released.

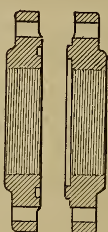


Fig. 296.

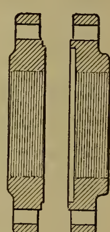


Fig. 297.

The more recent development of the flange joint has been towards the " loose flange " principle in which the faced joint-forming elements are not fast upon the lengths of pipe. In Fig. 295 is shown the type in which the pipe is screwed into a special end-fitting, and the two halves are drawn together by the loose flange bearing against a shoulder on the fitting. This is a derivative of the union, and is, in fact, a bolted union. In Figs. 298 and 299 are the loose flange types, American and British, in which the end of the pipe section is flanged over after the half flange has been slipped over the end. The contact is thus made with the pipe ends and the flange bolts draw them close and collars or recesses in the flange cover in the radial joint in the British forms.

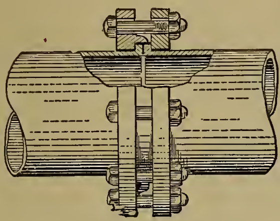


Fig. 298.

Again the pipe may be expanded into the metal of the flange as a boiler tube is expanded into a tube-sheet (paragraf 57). Or finally the end of the pipe is flanged over and then welded to a flange ring of sufficient thickness to make a strong and stiff permanent joint. This is an admirable but costly way, and if anything happens to the integrity of the faces the whole pipe section must come down and be taken to the forge and shop (Figs. 300, 301). Fig. 302 shows welded nozzles with flanges, and heavy flange joints. Fig. 301 is what is called the Van Stone joint, with pipe end thickened first, then flanged over. The thickening is effected by a welded ring. Pipe-fittings include all special devices for connecting pipe and permitting branch outlets to be taken off at angles. In small work these will be screwed on the outside of the male thread on the pipe: in larger work flange joints with bolts will be used. In small work these fittings will be of cast or malleable iron: in larger of steel castings. In small work the branch may be taken off at 90° from the main stem, or the corner be turned with a 90° elbow: in large work the angle will be changed to 45° or the curve produced by a long pipe sweep curved while hot to an arc of several feet radius and always over five diameters of the pipe. The tee for a

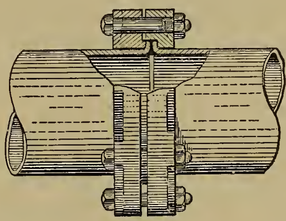


Fig. 299.

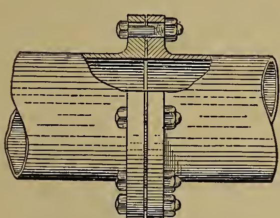


Fig. 300.

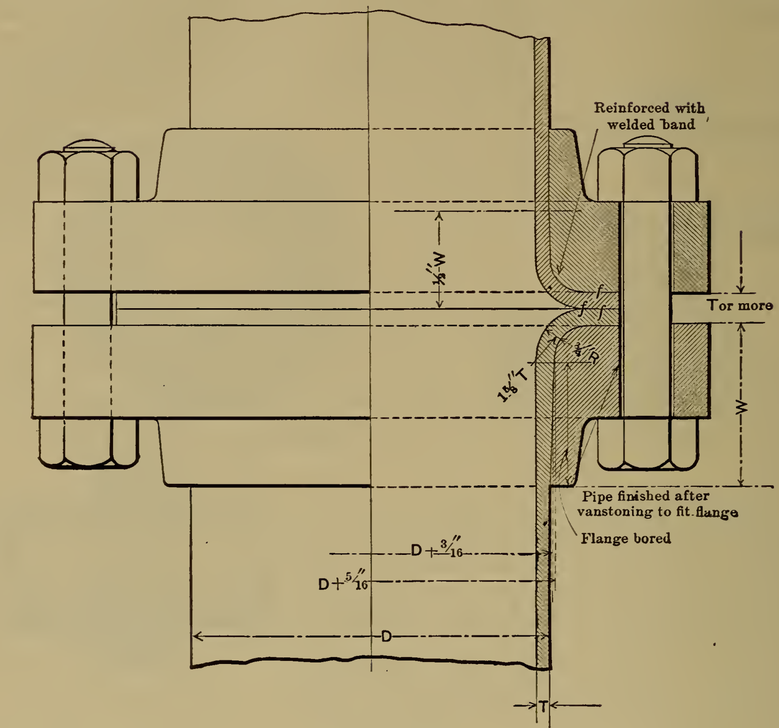


FIG. 301.

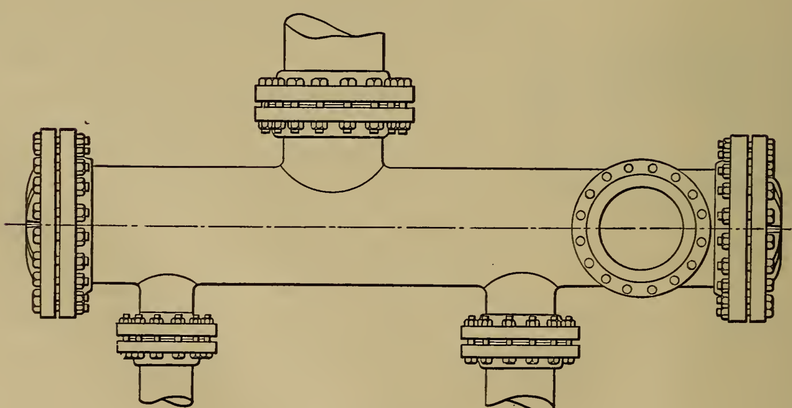


FIG. 302.

right angled branch in small work becomes the *Y* when the branch starts at 45°. Fig. 303 shows the standard types of fittings, with short and long radii, and outlets at 90° and 45°. The proportions given by the letters vary slightly with different makers, but are always furnished by them in their lists. Recent practice and particularly with superheated steam has favored the replacing of tees by welded nozzles formed on the pipe itself (Fig. 298), lessening the number of joints to give trouble and the time to erect the pipe but at the expense of first cost.

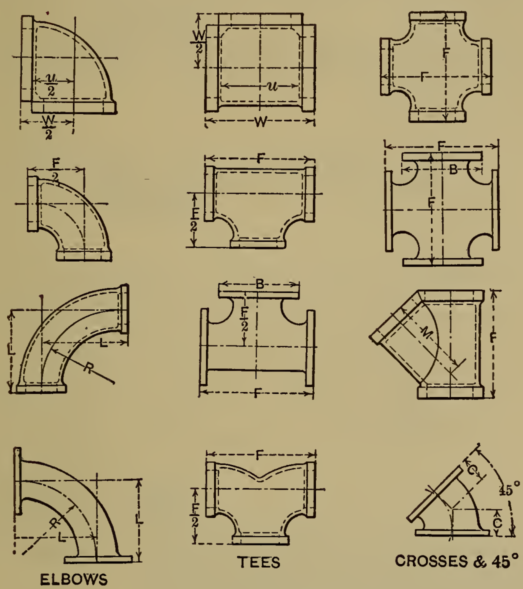


FIG. 303.

239. Expansion of the Steam Pipe by Heat. Since the force which expands a steam pipe by heat from within is practically irresistible (paragraf 231), the best plan is to let it take place freely, but control the direction and starting points of such expansion. It is very undesirable to have such expansion produce cross-strains on fittings or tend to open joints. The pipe may be anchored at one end and expand toward the other: or it may be anchored in the middle and expand both ways. It cannot be anchored at both ends: it will either buckle and set its joints to leaking, or it will push or pull over one or both of the anchorages.

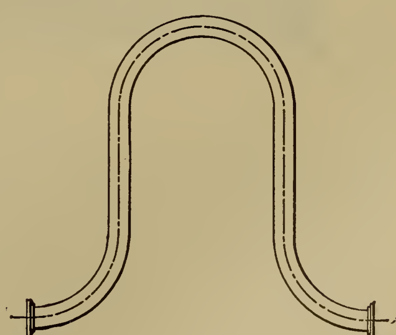


FIG. 304.

In small work with light pipe, the loop of Fig. 304 has been much used, either in copper formerly or in steel latterly. The bending is not enough to go beyond the elastic limit in the extreme fibers. The same result is better attained by the rectangular offset to a parallel vertical plane as in Fig. 305. Expansion spends itself in twisting the two vertical riser nipples, and in changing the direction of the center line BC and its angle between the axes of the two pipes A and C. The length of the verticals should be long enough to allow for this torsion

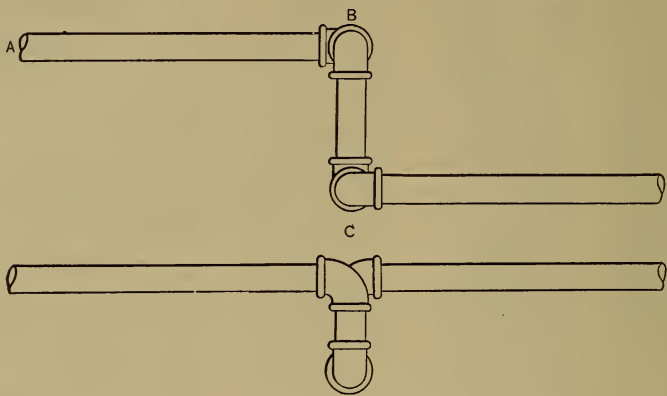


Fig. 305.

without starting the threads in the elbows if screwed fittings are used. This can be worked also in a pair of horizontal planes, and advantage can be taken of a sudden change of level to introduce this type of provision or expansion.

A third type is the slip joint (Fig. 306), providing that the two lengths

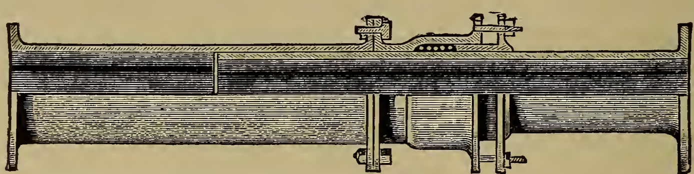


Fig. 306.

may bring their ends together or separate. One length, usually the comparatively fixed one, carries a stuffing-box with gland, and the other end a brass sleeve which slides steam-tight in and out of the stuffing-box as the pipe expands or contracts. These slip-joints are troublesome from leakage when the packing deteriorates, and from a tendency to seize and become hard and fast from corrosion and defective alignment of the pipe. Care must be taken also that the pipe is not allowed freedom of movement sufficient to blow the slip-tube out of the stuffing-box from end pressure of steam within the pipe.

To prevent this, the balanced type of slip-joint has been made (Fig. 307), in which the slip-tube carries an annular flange exposed to steam pressure and of an area equal to that of the pipe. Hence the tube is forced inward with the same pressure as the end to which it is attached is forced outward. The flange is also a mechanical stop transferring pressure to the holding bolts which are not present in Fig. 306. In Fig. 308 the leaking slip-tube is replaced by the corrugated copper or steel or alloy which is long enough to allow motions of the flanges towards each other or away without producing excessive flexure in any one groove.

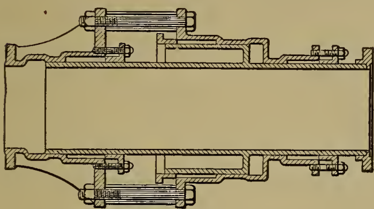


Fig. 307.

A fourth form is used for high pressures and temperatures and consists in making a flexible flange-joint of steel plate or of copper of wide diameter. The pipe is expanded into the middle of this flange and the two edges are bolted or riveted together ringwise. The flange opens and closes like a bellows under changes of length (Fig. 309). This same type of expansion-joint is used in river-boat engines to connect the side pipes to the upper steam-chest, which is a part of the cylinder-casting.

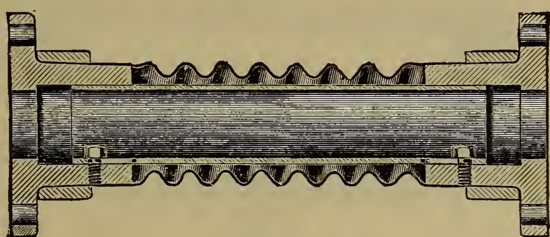


Fig. 308.

240. Hangers and Carriers for Steam Pipe. The changes of length and motion of the pipe under heat must be provided for in arranging to support the weight of the pipe. It is best to have two points of support in each length between joints: then no bending or flexure comes upon such joints if properly alined, and erection of the pipe in place is made easy. The best way if there is head-room enough overhead is to hang the pipe by pendant rods from the girders or from brackets above it, with length of rod sufficient to allow for the come and go of the pipe. If there is little head room, then roller hangers of the type of Fig. 310 have been used in mills with wooden beams, or the type of Fig. 311 is standard for bracket supports in tunnels or upon side walls. There must be no tendency for the pipe to cramp in its lengthwise motion.

241. Valves for Steam Pipe. In the simple and elementary power-plant of Fig. 1 there must be a valve in the pipe between boiler and engine to control or arrest the flow of pressure energy from the former to the latter. When the number of boilers and engines multiplies, the valves required to enable any boiler to be cut out for repairs or inspection increase proportionately. When the engines or boilers or piping are any or all of them in duplicate as a precaution against shut-down of the plant the valve system becomes complicated and costly, and yet each valve is again a vital unit, which must not be liable to fail at a critical moment.

The valve which throttles the flow of steam from a single boiler to a single engine as in Fig. 1 will be called a "throttle-valve". Any valve between boiler and engine performs this same function in a complicated system, but in this case the valve which is placed close to the valve chest of the engine and serves to isolate that engine from the piping system is called its throttle-valve. These are often special respecting outlets or length of stem, and are supplied with the engine itself. The rest are standard, and are bought with the pipe. The throttle-valve of the locomotive is required to be quick-acting, particularly as respects closing; and the engineer is always standing by close to it. Hence it is quite usually of the balanced poppet type, with two disks on a common lifting stem with equal pressure below and above. In low-pressure work the throttle is often a disk-valve, swinging on an axis through a diameter of the disk, securing balance as in the damper of Fig. 193 or 194, and quick-action. But both the disk and poppet type require to be locked from undesired change of position, and for high pressures the advantage of the power of a screw movement and its property of self-locking in place have given screw-valves the preference. The disk-valve is also liable to inconvenient deformation by heat and pressure.

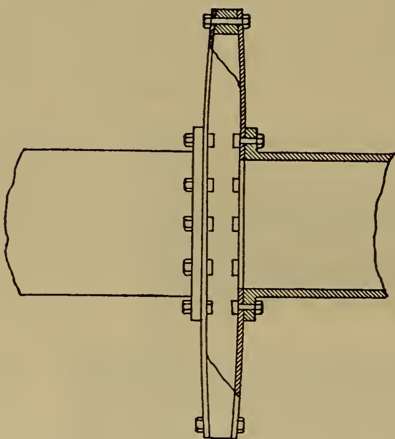


Fig. 309.

As in the matter of the pipe itself, the valves will fall into groups by sizes and by pressures. For small work and for all-round use with light pressures, the globe-valve of Fig. 312 will be found in the lead. The

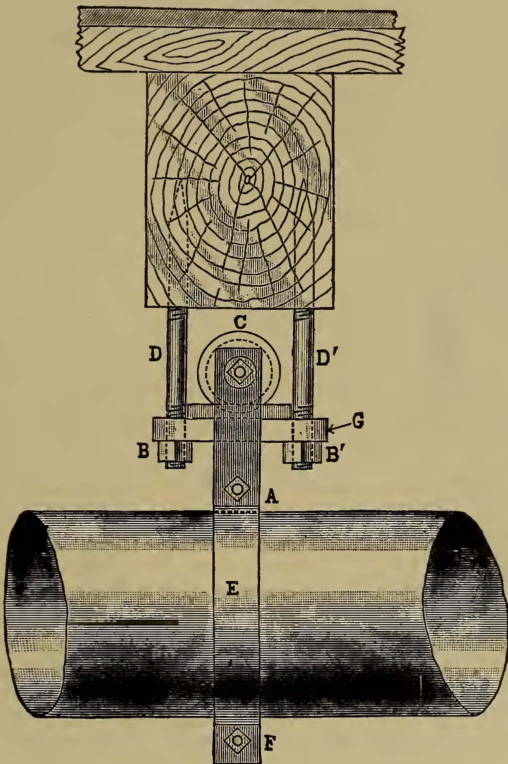


FIG. 310.

seat is in the plane of the axis of pipe and body, with a partition above and below to compel the steam to pass through the round hole in the

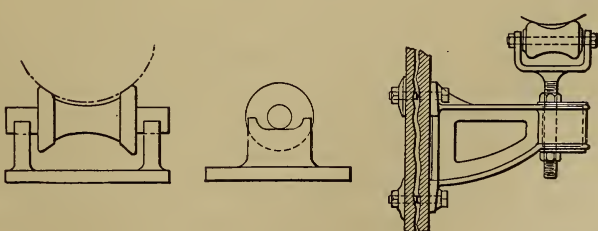


FIG. 311.

seat. The valve is conical on its face in small sizes, flat in the larger. Special metal or alloy disks or rings are used in the valve face or on the seat face or both, so that these can be renewed on wear without disturbing the globular body made up on the pipe. The spindle works on a square-threaded screw in a removable cap which carries also the stuffing-box. The cap when removed takes valve and spindle out for repair. The screw is universally right-handed in globe-valves, so that the valve is closed by a clockwise motion of its wheel, and opened by reverse motion. The objection to the globe-valve is its tortuous passage with the pocket below the seat where water will gather in horizontal runs of pipe, and give chance for the water-hammer dangers of paragraf 231. Scale and sediment may also accumulate either above or below the seat, causing trouble in opening or closing. The trouble from water can be minimized by putting the valve with its spindle in the horizontal plane of the axis of the pipe, instead of vertically as in Fig. 313. Such valves also should be put with the bottom or lower face of the valve toward the pressure to be resisted, both because the packing of the valve can be renewed when pressure is on the line by the closure of the valve, and because if the seat or disk becomes detached from the spindle, the valve remains held down by the pressure if the latter is on top, and no indication is given of the inoperative state of the valve.

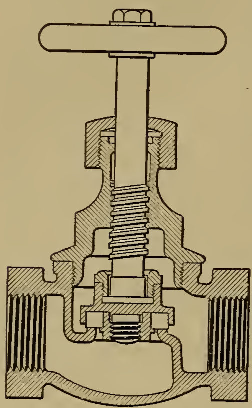


FIG. 312.

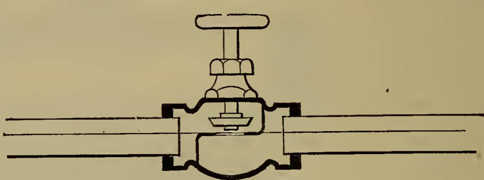


FIG. 313.

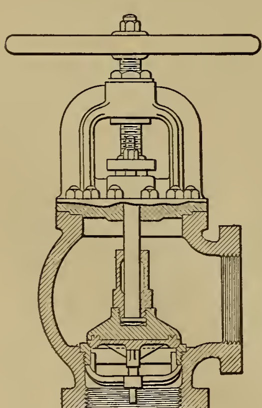


FIG. 314.

In addition, the globe-valve becomes of inconvenient size and weight in large sizes of pipe if it is to offer anything like the opening of the pipe area itself. Hence straightway valves are preferred for large work. The globe-valve in the form of an angle-valve is a convenient and excellent type where it can perform the double service of the turn of the corner and the regulation of pressure (Fig. 314). There is no tortuousness of passage, and can be no accumulation below the seat. The angle-valve is at its best with a vertical spindle and as before with the pressure on the under face.

The straightway-valve is usually of the gate-valve type. Such valves have the opening through them coincident with the pipe-axis, and the closure is effected by sliding a door or gate across the opening from one side, or from above. Since the gate has to resist pressure over its area and must be tight against this pressure, which strives to force it away from the seat, it is usual to give the valve a wedging action as it reaches its closed position, using the face of the opposite opening as the abutment for such force. This is done in two general

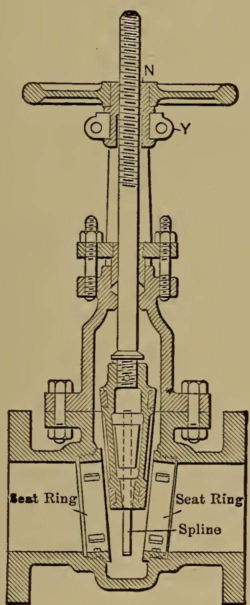


Fig. 315.

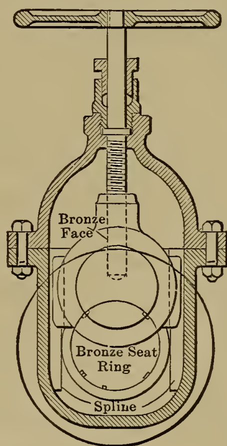


Fig. 316.

ways: either by inclining the faces of the seats toward each other (Fig. 315) or by keeping the faces of the gate parallel and separating them by a wedging effect of inclined planes between the backs of the two face plates of the valve, which are carried down by the spindle motion after the valve faces have reached their final vertical position. The advantage of this last principle is that the contact faces do not move over each other laterally after they have received nearly the full pressure required to close the valve tight, as is the case with the solid gate construction. In opening, the reverse action takes place, as the wedges or inclined surfaces release first, letting go contact with the faces.

The loose faces rattle under certain conditions, and the other construction is cheaper. Gate-valves require a lateral recess into which the gate may recede and leave the thoroughfare through the valve wide open. The spindle may work in a screw formed in the gate, to move the latter in and out (Fig. 316). The screw is then altogether inside the case, and the stem is called a " non-advancing stem." There is nothing external to indicate by position of wheel or stem whether the valve is open or closed, but the stuffing-box does not have the spindle working spirally through and keeps tighter. If the thread on the gate and spindle is made right-handed, the valve is opened by clockwise turns of the wheel, which is the motion to close an " advancing " stem-valve when the nut threads are a part of the valve casing and not of the valve gate (Fig. 315). Hence it is desirable, to prevent confusion and error, to put an arrow on the rim of the wheel and the word " open " to indicate the direction to turn the wheel to open. An increasing favor attaches to the outside screw-valve of Fig. 315. The nut threads are here in the hand-wheel, outside the stuffing-box, and held from end-motion by a yoke. The stem is held from turning from its connection with the gate, but " advances " as the gate is lifted from its closed position. As before right-hand threads on the stem will make the valve open by clockwise motion of the wheel and close by anti-clockwise turning. Arrow and word should be used here also, but the visible advance of the projecting screwed stem informs the operator what he is doing. The stem has only straight motion through the stuffing-box without twist, and the threads are easily lubricated. The gate-valve should be placed with its spindle vertically upward or on its side: if the spindle is downward, water gathers in the gate recess and gives trouble afterward when expelled. Fig. 317 shows the types of small gate-valve with bronze body and screwed and flanged outlets, and " advancing and non-advancing stems."

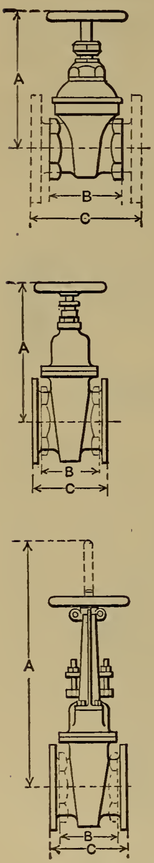


FIG. 317.

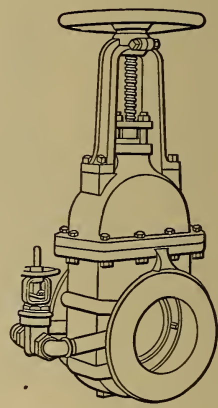


FIG. 318.

In large work and with high pressures the force pressing the gate upon its seat on the side opposite the pressure before the pressure reaches that opposite section of the pipe is very considerable. It may be enough to make it very hard to move the valve to open it. It is further undesirable to open any large area from the high-pressure to the low-pressure side of the valve until the pipe has grown warm and the provisions for the drainage of the pipe have gotten rid of the condensation due to the heating process. This has resulted in the placing of a by-pass and valve of comparatively small diameter to lead steam around the gate when the by-pass is open, while the main gate is still closed. Even carelessness cannot get steam too fast round into the cold pipe beyond the gate: and as pressure and temperature are established equally in both sides, the main gate can then be opened both with care and safety (Fig. 318). Partial opening of a large gate, and particularly of one with loose faces, is not desirable: it injures the face contacts and makes the joint leaky. Mechanical erosion and dirt in the flow through the valves are injurious to both faces, as well as rust. The contacts should not be of the same metal, to lessen the danger of their rusting together when closed for a considerable interval. Bronze and iron are the usual combination.

A large gate-valve cannot be quickly closed with a screw-motion. The gate must be moved through more inches than the diameter of the opening or pipe, and the pitch of the spindle thread should not be too steep, lest power be sacrificed, or the valve be too hard to move. Combinations have therefore been proposed where, on emergency for throttle-use, the thread and nut can be cut out, and the valve closed by straight push. This can also be made a safety stop (paragraf 494) by attaching a weight to the spindle and the emergency mechanism to the detent. When the detent is released the valve is shut quickly by the weight, while in normal use the gradual opening by the wheel and screw and nut is available.

In very large valves the cross section becomes so great that two spindles will be advisable, working in unison by being geared together. Here the hand-wheel will be replaced by a ratchet and lever; but the hand motion will be exceedingly slow. This has given rise to the motor-driven valve, in which the hand-wheel of the smaller sizes becomes a large-diameter spur gear, which is driven by the small pinion on an electric motor armature shaft. This principle of valve-operation leads very simply to that of valve-control by switches for these valve-motors, the wires leading to the motors from the switches which are located at a point of centralized control. Valves can be instantly operated in case of emergency even if at considerable distances or elevations in the plant.

There are also special valves in a modern plant having to do with requirements of varying pressures, functions and services, such as relief-valves, back-pressure valves, quick-closing valves and the like. Some of these will be referred to in connection with the exhaust-piping.

242. Grading of Steam-pipe. Experience shows that when steam is moving in a pipe at high velocity it is impossible for water of condensation to move against it. It will even be carried along upwards in a vertical pipe. Hence the steam-pipe should be graded downwards towards the engine from the boiler, and provision must be made near the engine to catch or dispose of this water. Trouble is often made in pipe systems using the ordinary fittings where horizontal branch outlets are made in the plane of the axis of the fitting, by reason of accumulations of water below the level of such outlets. Such pockets should either be carefully drained, or, better still, a special form of fitting should be used, such as shown in Fig. 320, in which the condensation will flow out of a main pipe with the steam which flows through the branch. A few pounds of water moving several hundred feet with a velocity of nearly a mile a minute, represents a mighty store of energy and is capable of most disastrous results. The presence of water in such pockets is also the occasion of mechanical erosion of the pipe similar to the action of the sand-blast, and ample provision must be made to get rid of even small accumulations.

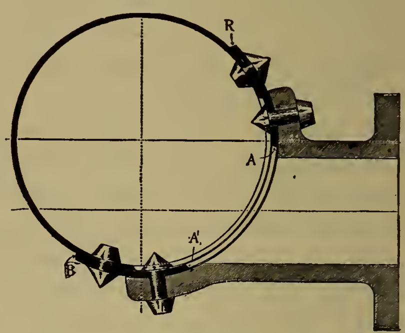


Fig. 320.

243. Drainage of Steam-pipe. Separators. To get rid of accumulations of water drawn mechanically or entrained by the steam from the boiler and those which result from condensation, at least five different methods may be used.

The first is to tap into the pipe, wherever pockets and elbow-joints occur, small pipes with the necessary valves which can be left partly open and draw off water as fast as it gathers. These pipes may all converge towards a closed reservoir or tank from which the accumulated water may be pumped back into the boiler. These drip-pipes will vary in size with the quantity of water to be taken care of, but nothing is gained from having them too small, since they are likely to become clogged and inoperative. From one-half inch to one and one-quarter will be the usual range, according to the size of the pipe and engine in small sizes and two-inch or over in large ones.

The second method is to diminish the losses of heat from the flowing of live steam through such tubes by the use of a steam-trap. A steam-trap is a pot within which is a device which is usually intended to act upon a valve or opening when water is to pass through it, but to refuse to act when water changes to steam. This result is secured in many ways in different designs of trap. The simplest plan is to have the trap inclose a float which is acted upon by water and raised, but which falls in steam (Fig. 321). It will be seen that when the drip-pipe connected to the trap is filled with water, the float will lift and open the connection through which that water can escape. When trap and pipe are emptied of water the float will fall, closing the outlet from the trap, and shutting off escape until the trap is again filled. The discharge from these traps may either be back into a closed tank to be pumped into the boiler, or it may be wasted into drains (which is not to be commended). Many traps instead of using a float are operated by differences of expansion of one or two metals in steam and water, and there are many practical ways of applying the floatation principle. In Fig. 322, for example, are three ways of using the ball float. The upper left one makes the float actuate a sliding-valve: the valves in the others lift off their seats, but the lower one subdivides the valve area so that the opening is progressive and graduated to the amount of water coming in, and the principle of the water-seal is used to keep solid water only in the outlet pipe until the outlet valves have been shut by the descent of the ball. The principle may also be used of the counterpoised pot, in which the empty vessel swings up on its trunnion or pipe near its center of gravity when empty; as it fills the end further from the trunnion grows heavy and overweighs the counterpoise, causing the pot to fall through an angle ample enough to work the valve (Fig. 323).

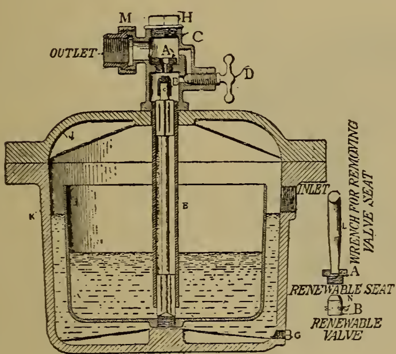


Fig. 321.

The third method is to introduce a receiver or catch-water tank in the pipe close to the engine into which all condensation shall be made to flow, and out from which only dry steam will go to the engine. Such a receiver may be a pipe (Fig. 324), or a simple vertical cylinder

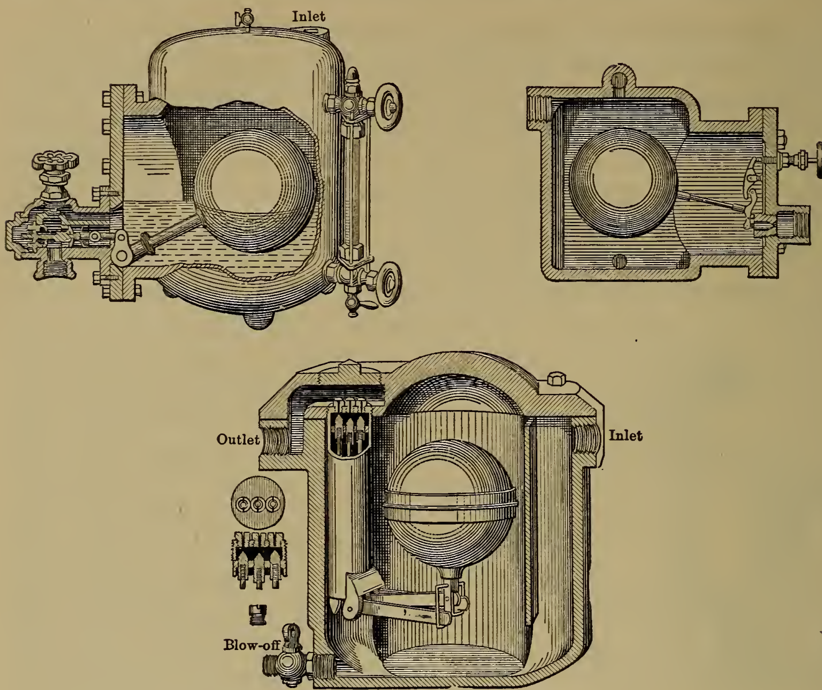


Fig. 322.

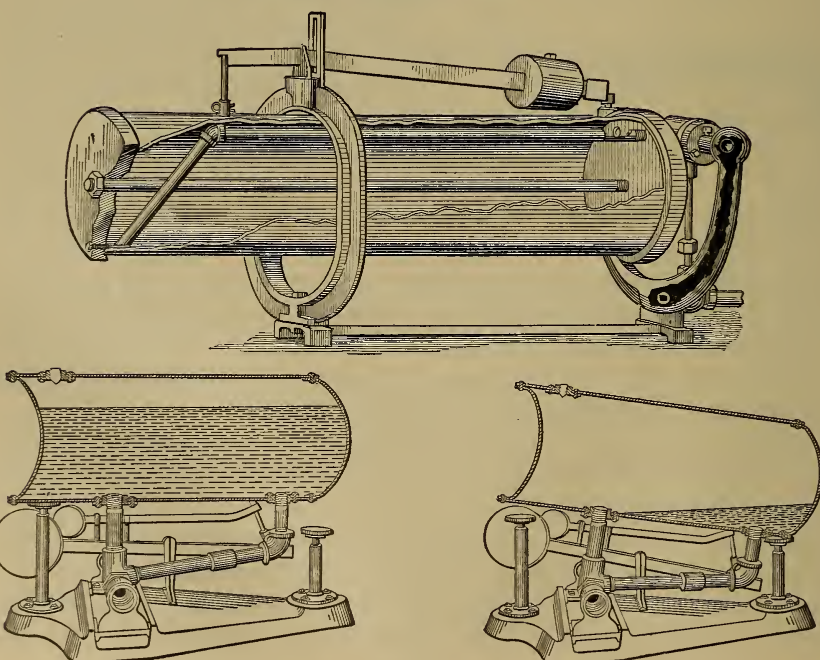


Fig. 323.

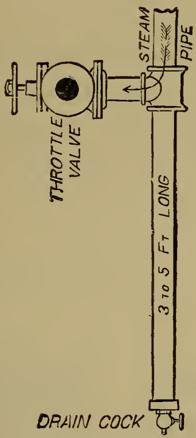


Fig. 324.

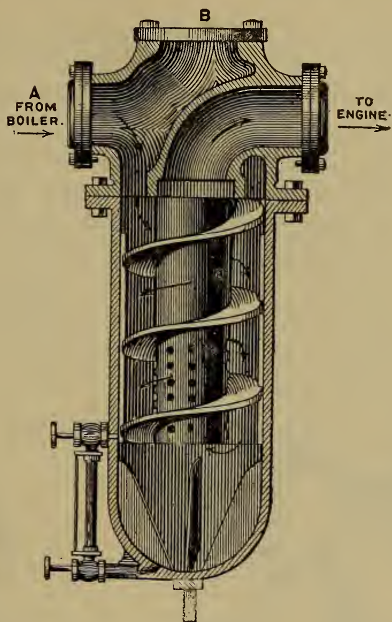


Fig. 325.

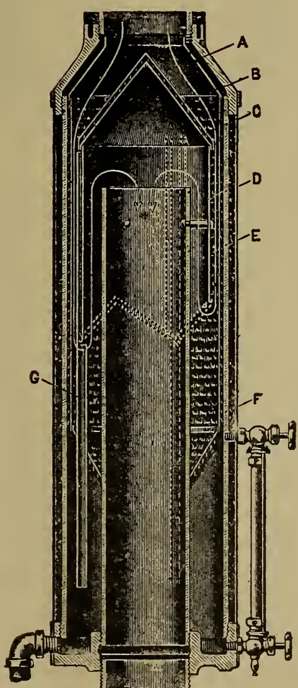


Fig. 326.

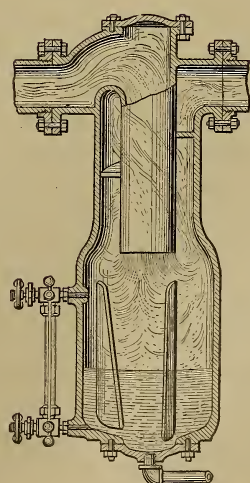


Fig. 327.

of boiler-plate (Fig. 326), into the top of which the steam-pipe enters and passes down part way. The water which is carried by the steam falls to the bottom by its inertia or by gravity, and will there be taken care of either by a drip-pipe or by a trap. The steam going to the engine leaves the receiver from a point near the top, and the enlarged diameter which diminishes the linear velocity of the steam prevents the outflowing current from drawing out the entrapped water. A glass tube can be attached to the side of the receiver so that the level of the water caught in it can be easily observed and its discharge governed accordingly (Fig. 326).

The fourth method is easily derived from the foregoing. It is the use of a separator to withdraw by mechanical means water which

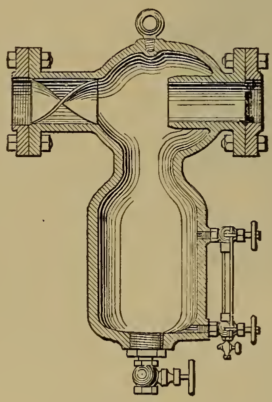


Fig. 328.

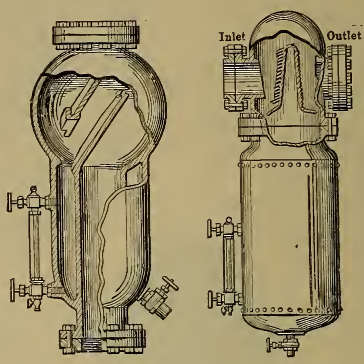


Fig. 329.

the steam has entrained. There are several forms of such separators. Several of the most successful ones are shown in Figs. 325, 327, 328 and 329 and their methods of application are obvious. It will be seen that the principle involved is that of giving a spiral or centrifugal motion to the water and steam as they enter the separator. The superior density or weight per cubic inch of the water causes it to yield most strongly to this centrifugal tendency, and it goes to the outside of this chamber, while the lighter steam, being less affected by this tendency, will remain nearer the center, from which the outlet to the engine is taken off. The presence of metallic-surface perforated deflecting or baffle-plates and similar constructions increases the efficacy of the separation since water divided in drops has a tendency to attach itself by capillary action to such surfaces. The separator requires to be, in the form shown, of a diameter at least twice that of the pipe, and the depth or length at least three or four times its diameter. The larger the separator, the more efficient, since it combines in this case the natural separation by differences of specific gravity with the mechanical separation by centrifugal action. The separator acts as a receiver when accidental quantities of water are thrown over with the steam. With reasonably dry steam (having five per cent of water in it or less) an effectual separator should allow less than one per cent of water to pass it and reach the engine. The discharge from the bottom of the separator may be taken care of by a trap, or it may be freed by hand as above described. Figs. 328,

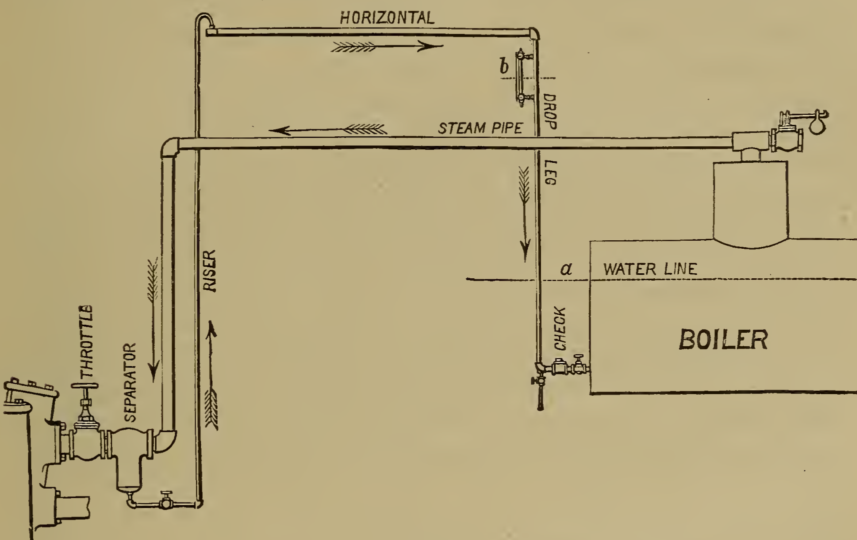


Fig. 330.

329 show centrifugal separators for a horizontal and for a vertical line of pipe. The heavier water is thrown radially and caught in the receiver below (see also Fig. 333).

A substitute for the trap, forming a fifth system, has been worked out for use in places to which it can be satisfactorily applied. It is called the steam-loop and is shown in Fig. 330. The pipe from the bottom of the separator becomes a species of siphon, whose length of leg is depended upon to move the water in it by the differences of density of the water in the two legs. In the drop-leg of the loop, which is connected to the boiler below the water-line, the water is comparatively still and solid. In the other leg or riser the water is mixed with steam in bubbles ascending up through it, and this difference in weight will maintain a continual discharge into the nearly horizontal member which is slightly graded towards the boiler. The steam-pressure is nearly in equilibrium at the level of the water-line through the system, but the weight of solid water in the drop-leg gives the dynamic head to feed the condensed water continuously from the drop-leg into the boiler.

239. Non-conducting Coverings for Steam Pipe. To meet the fifth requirement of paragraf 231 respecting loss of heat by radiation and contact of the air around the pipe, a material of low thermal conductivity must be used as a cover or jacket around the pipe. It must keep air-currents from the pipe and must resist by its properties the tendency of the pipe to radiate. These two requirements must be kept in view in selecting the material to be used. The material must furthermore be resistant to combustion and deterioration under heat, and must give off no disagreeable odors. It must be easily applied, must be cleanly and not attractive to vermin, and it is desirable that it be so made that repairs and alterations to the pipe may be made without destroying the non-conductive covering and making its renewal necessary. These latter conditions point to the use of what are called sectional coverings. It should also be as cheap and light in weight as is consistent with effectiveness.

Since air undergoes no heating by radiation, but is heated by contact only, it has been found that materials of such porous or fibrous character as to shut in or occlude a considerable quantity of air, finely subdivided, make the best non-radiating coverings. The air is easily heated by contact, so that care must be taken to prevent this air from circulating, and it is best to keep it from actually touching the pipe. These peculiarities form the basis for the excellence of many combinations which use hair-felt. The porous or fibrous quality of the hair-felt holds a large quantity of air while circulation is precluded, and injury to the hair is prevented by first wrapping the pipe with asbestos-board. The hair is held in place by a canvas covering sewed over it, and if desirable bound by sheet brass or nickel-plated rings for appearance' sake. The fiber of asbestos or of blast-furnace cinders comminuted by blowing air or steam through it while fluid and known as mineral wool, possesses the same qualities as hair-felt, and for the same reasons. Other materials of successful use as non-conductors belong to the class of the earths. Infusorial earth largely composed of the silicious shells of minute diatoms, magnesian earth, ashes, and the like, made into a plaster with some binding material like asbestos-fiber or hair, form a group of non-conducting coverings often to be met with and which form the plastic class. The sectional coverings or removable coverings are combinations of asbestos-paper, hair, and canvas molded into split cylinders which are sprung on over the pipe, closed together, and held by decorative bands. It is apparent that to increase the thickness of the coverings in order to diminish loss of heat through them is to increase the cost of such coverings and the weight on the pipe. The presence of the covering compels the hanging appliances to adjust themselves for expansion without disturbing the covering.

The standard method of testing the conductivity for heat of such proposed coverings has been to take a given length of the bare pipe in an open basement or space of still air of observed temperature, and admit steam of known observed quality as to moisture temperature and pressure at one end and to observe the weight of water condensed and drawn from the other end in a given time. The heat loss in units from the latent and sensible heat of that weight of water is then computed and called one hundred. The pipe is then covered in succession with the coverings to be tested, and the same measurements and computations are made. The covering giving the least heat loss in units is then put at the top of the list as having the greatest negative conductivity. There can plainly be no absolute standard. Other methods have been to replace the steam heat by the heat in an electric resistance coil, and observe the heat transfers through one square foot or unit area to absorbent water outside of the covering, and eliminate the variable effect of air currents in a pipe test.

Some experiments of the late Chas. E. Emery have always appeared to have some special practical value and are given in Table XIII. They refer to hair-felt as a standard, used in a layer of two inches thick.

TABLE XIII.

RELATIVE EFFECTIVENESS OF 2-INCH LAYERS IN PREVENTING LOSS OF HEAT. (Emery.)

Material.	Effectiveness.	Material.	Effectiveness.
Hair-felt	1000	Loam (peat moss fiber)	0.550
Mineral wool	.832	Slaked lime	.480
Mineral wool and tar	.715	Gas-house charcoal	.470
Sawdust	.680	Wood charcoal	.630
Mineral wool (inferior)	.676	Asbestos alone	.363
Charcoal	.632	Coal ashes	.345
Pine wood (cross-grain)	.553	Coke alone	.277
		Air space (circulation unchecked)	.136

Similar researches have also been made by Ordway, Barrus and Brill*

* *Consult*
Emery, Transactions A. S. M. E., vol. II. p. 34.
Hutton, Transactions A. S. M. E., vol. III. p. 228.
Ordway, Transactions A. S. M. E., vol. V. pp. 73 and 212.
Ordway, Transactions A. S. M. E., vol. VI. p. 168.
Brill, Transactions A. S. M. E., vol. XVI. p. 827.
See also Kent's Mechanical Engineer's Pocket Book.

245. Exhaust-pipe. The computations of paragraf 231 apply also to the design of an exhaust-pipe, except that the pressures will be lower, and it is not desirable to demand from the engine-cylinder that it should exert as back-pressure upon the working piston the force necessary to accelerate the used steam out of the cylinder and to the atmosphere or to the condenser. Such pressure upon the exhaust-steam is a subtraction from the forward pressure upon the other side of the piston which is doing the work of that stroke. Hence the practice is usual to make the velocity in the exhaust-pipe two-thirds of that permitted in the high-pressure steam-pipe, and compute 4000 linear feet per minute as the permitted velocities. Bends should be with long sweep and as few in number as possible.

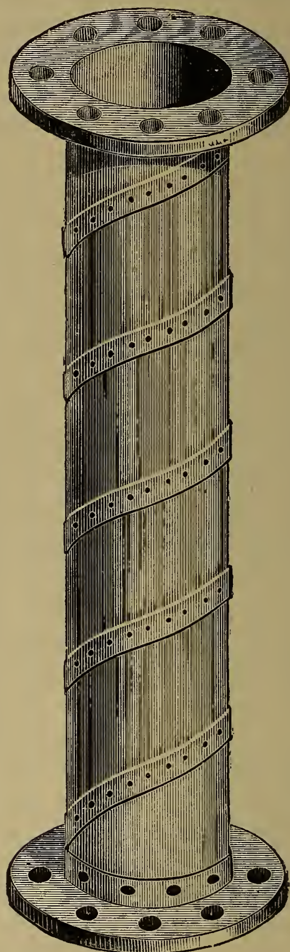


FIG. 331.

In the general or ideal case, where exhaust-steam is not to be used for heating, the pressure in the exhaust-pipe will be that of the atmosphere, or at most a pressure ranging from that up to three pounds per square inch. Hence it need not be of the same strength as the steam-pipe, which explains the use of spiral-riveted or spiral-welded pipe in long lengths (Fig. 331). Lightness and cheapness are thus secured. In city conditions the exhaust-steam must be taken to the roof of buildings or factories to be discharged, which compels a considerable ascending length of pipe. In power-plants where this does not have to be considered the engine may exhaust into the open air at its own level. Where the noise of the exhaust is of no consequence as it escapes into the air, the end of the pipe may be bare. Where noise must be prevented, and where the discharge of condensed water in the exhaust-current carrying oil from the cylinder is objectionable or harmful to roofs or structures, provision must be made to meet both of these difficulties. The exhaust-pipe must be carefully drained to remove the trouble from pulsations and noise from such water in pockets where the motion of the steam can impel the water in masses against bends, and yet be unable to expel it. The oil from the lubrication of the cylinder is also objectionable in condensing-engines, where the condenser first gets it and later it enters the boiler with the hot-feed from the hot-well. The exhaust-pipes will therefore have oil-separators and an exhaust-head.

246. Oil-separators. The oil injected into the cylinder with the steam by any system to lubricate the pistons, valves and rings (paragraf 528) will most of it be carried out with the exhaust. There only

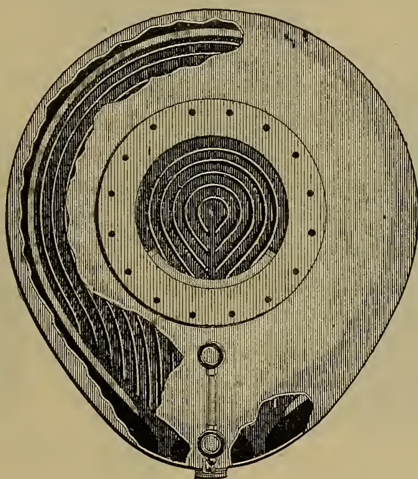


Fig. 332.

remains behind what adheres in a thin film to the metallic surfaces. With high pressures and superheated steam the oil may be made into a vapor or a mist in such fine division that it will not form again into drops on lowering of temperature. In general, however, the oil retains its character and structure and its property of adhering to metal surfaces against which it impinges. The first principle of oil-separation is therefore to baffle and buffet the current of outgoing steam and make it easy for such oil as is in drop form to reach an adherent surface where it can be caught.

The action seems to be a capillary one, the oil adhering but the steam passing on. Figs. 332, 333, 334, and 335 show the baffle-corrugated systems, and Fig. 338 the inertia principle. Here the steam passes over trough-shaped projections as it is deflected, and in these troughs is enough water at all times to entrap the oil and hold it till it overflows.

The wetter and cooler the steam the more effective are separators of these types. With hot and high-pressure steam the oil is in so fine a state of division that it does not become entangled at the high temperature of the baffling-surfaces, and here a species of filtration seems the only practical way.

The filtration scheme means a leading of the exhaust through a box or

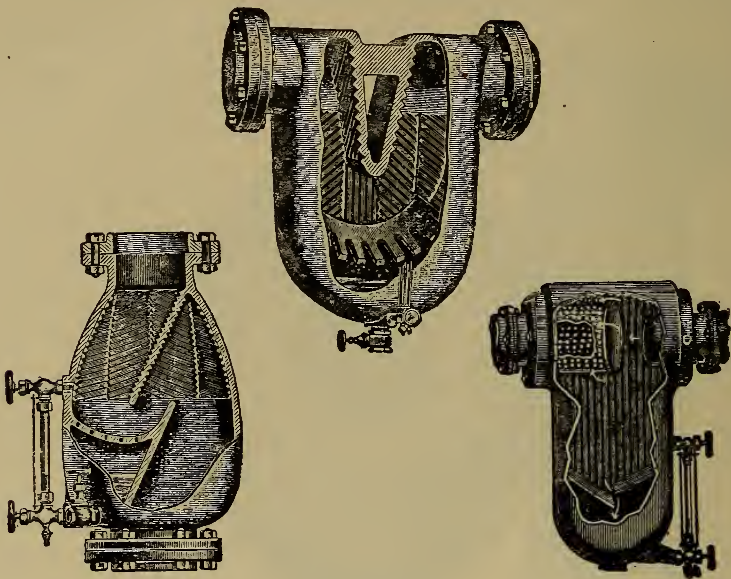


Fig. 333.

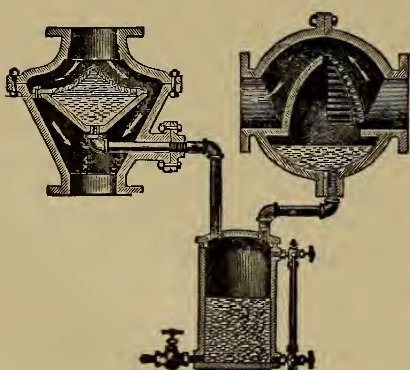


Fig. 334.

Fig. 335.

enlargement of the pipe in which shall be installed some material with a large contact surface and open porous structure. Hay or straw has been much used in marine practice and with condensing-engines for the filtering material: compressed sponge or even sand. These must of course be frequently renewed and it is a dirty and offensive job to clean the filter. The design of Fig. 336 is the Edmiston cheese-cloth filter which has been much used by reason of its convenience. The disks 1, 2, 3, 4 are frames which can be inserted through the hinged lid, and are held in place by the spider set up from outside by the screw *a*.

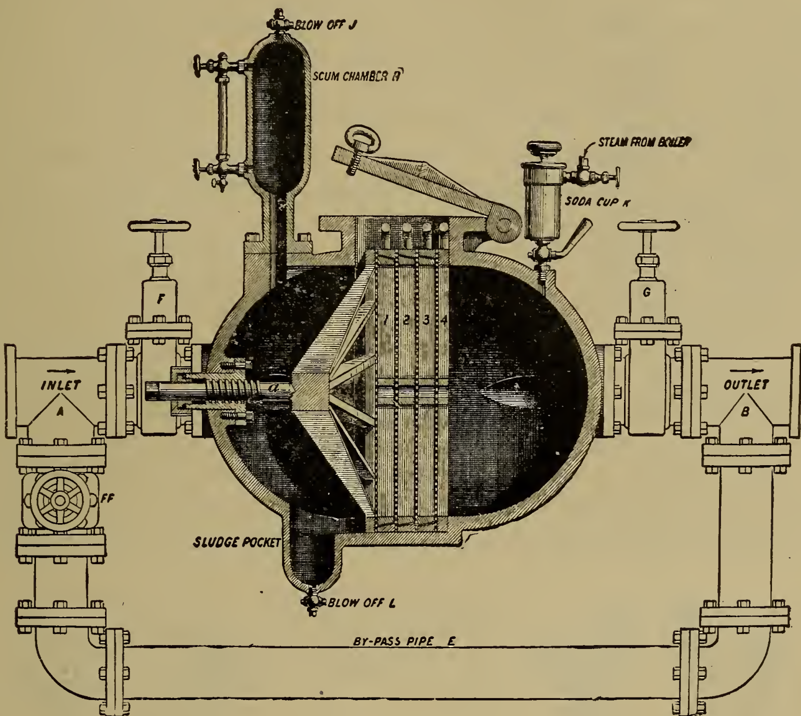


Fig. 336.

Soda can be introduced by the cup to make a soap of viscous oil for cleansing at intervals. The steam can be by-passed if renewal of the filter-cloth is desired without stopping the engine.

The oil which has been through the cylinder is usually so lowered in lubricating quality by the heat and abrasive contact that it is not worth while to try to save it and use it again except for the roughest and most unexacting service; it is usually run to waste. If saving is worth while, the oil is filtered again in special oil-filters to remove dirt which has been gathered from pipes and elsewhere, and quite often some unused oil is added to it to bring up its quality. But the heat and use has produced an oxidation of the oil-elements and brought it nearer the state of gum in which it is useless, and hence with hot steam the inexpediency of the trouble and expense of its redemption.

In the condensing engine, the oil from the exhaust-steam which the separator does not entrap goes forward to the vacuum chamber or condenser which is kept cool by the cooling injection water by which condensation of the exhaust is effected making hot water of it. Some oil therefore gathers upon tubes or other condensing surfaces and by forming a coat or crust interferes with the effective action of such surfaces, or clogs the passages when these are small tubes. The condenser must therefore be cleaned at intervals long or short according as the separator is effective or not. What is not stopped in the condenser goes forward to the hot-well from which the feed-pump takes its suction and pumps water into the boiler. The oil floats on the surface of the water of such hot-well, and if the suction can be always taken from the bottom of the well, this works as a further separator. What oil is churned up in the hot-well and does not separate here by liquation, is sent forward into the boiler where it probably remains except so far as blowing-off removes it (paragrafs 50 and 187). What remains, adhering to the metal of heating-surfaces, keeps the water from intimate contact with the steel and causes overheating (paragraf 192).

247. Exhaust-heads. In the non-condensing engine where exhaust goes to the atmosphere, the oil may be expelled from the end of the pipe in cases where its fall to the ground or overboard is not objectionable.

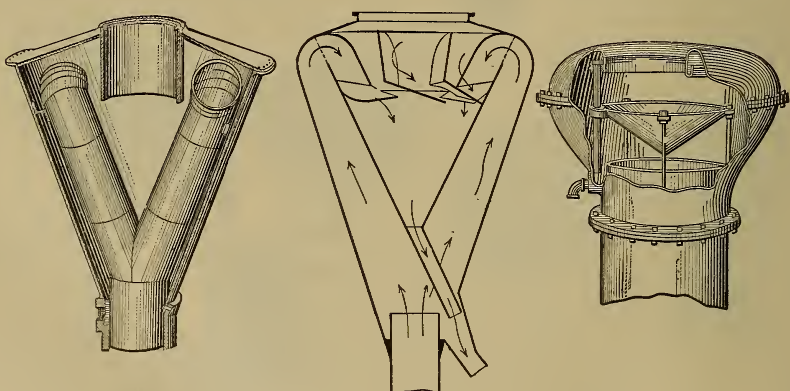


Fig. 337.

In cities, however, and when the exhaust-pipe goes vertically through or above a roof, such rejected oil is a menace and an annoyance. It is a nuisance from the defilement of it; a fire menace where it accumulates; a detriment to the roof covering. In cities, moreover, the noise of pulsating exhausts is an objection. Hence the design of devices for the end or outlet of the exhaust-pipe which shall have an enlargement of volume sufficient to break up pulsations and substitute for them a continuous flow at reduced pressure, and which shall also here entrap the oil at the time of such reduction of velocity. The exhaust-head as a muffler or silencer will have a section of a cone with its large base upward. The exhaust will rise into this enlarged area inside and will there be deflected either by the orifice of the pipe or by baffles, so that expansion, reduction of pressure, cooling, and inertia opposition shall all be secured. What escapes from the center of the conical head will therefore be water above 212° rather than steam-gas, and will flow softly away: the water below 212° and the oil will be gathered from the sides and from the baffles and will be led down to disposal at the ground or into drainage. Figs. 337 and 338 will suggest types of such exhaust-heads. There is a great variety of designs in use.

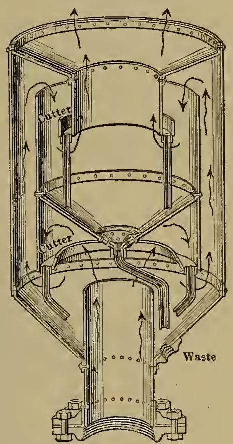


FIG. 338.

Chemical means for coagulating and eliminating oil from tanks and receivers using alum most often are open to the objection that they need continual care and skilled superintendence to secure efficient working.

248. Back-pressure Valves. The exhaust-steam from a non-condensing engine carries heat units enough to do a considerable share of the heating of buildings in mild winter weather if led into the heating radiators of a steam-heating system. Such steam may also heat hot water for circulation and heating when the steam would be inconveniently hot. In any case where heat is required for industrial or other uses, and steam would have to be made directly in boilers for the purpose if the exhaust were not so used, it is a measure of administrative economy to utilize the exhaust in this way. The net economic result may be that the plant gets its power for nothing in the way of fuel expense, the engine operating as a pressure-reducing valve between the boiler and the heating-coils or tanks (paragraf 244). In such systems and under other circumstances it is desirable that there should never be an excess of pressure in the utilizing coils or elements due to any increased use of steam in the engine and increased volume rejected, or by reason of any shut-off of the heating-coils. This will be met by what are called " back-pressure " valves, which are loaded check-valves in effect, opening in one direction and not in the other, and held shut by a known and adjustable pressure. When the set pressure is exceeded, the back-pressure valve automatically opens and acts as a relief of excess pressure as long as it lasts. When the pipe and valve are large, it becomes of advantage to use differential plans, and to use such systems of loading the valve for control as shall increase the opening as the differences of pressure increase. Fig. 339 shows the spring-loaded check-type at the right and the differential or partly balanced poppet type on the left. The weighted lever in the latter need provide only for such pressure as will supply the desired difference between the higher pressure at the right-hand inlet below the large seat and the

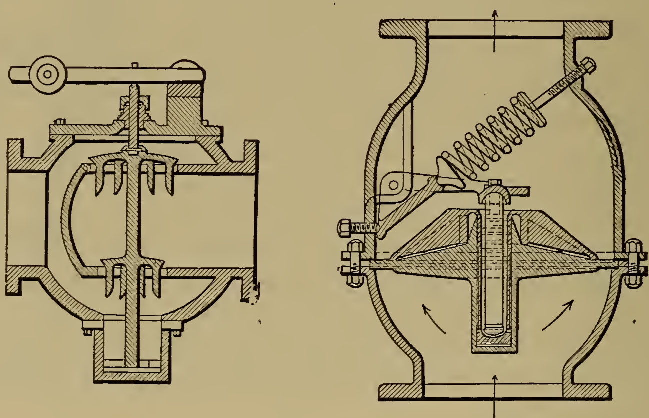


Fig. 339.

lower pressure above the upper and below the lower, and keep the valve down in place until the pressure at the right gets too high. Back-pressure valves of these types will be used also where a vacuum might be caused in an engine exhaust connected to an open heater at any time, resulting in a danger of back flow of feed-water into the engine and consequent danger to its cylinder: or where two systems of different pressures are cross-connected and the undesired higher pressure might establish itself in the lower system without such relief.

249. Reducing Valves. In severe weather in the northern climates, the exhaust-steam may not furnish heat enough for the heating-coils, and it becomes desirable to bleed from the high-pressure main pipe into the low-pressure heating-pipe. Or in storage systems of compressed air or other energy, it may be desirable to put the storage tanks under very high pressure and allow only a fraction of such great pressure to reach the motor. There are many other cases where the pressure on the lower side of the valve should be proportionately lower than on the high, either in a ratio or in an amount, and that this should be fixed so that no carelessness or ignorant handling of valves should release the high pressure. Such regulating-valves are called reducing-valves and any back-pressure valve with controllable load becomes a reducing-valve in principle if adjusted for the desired difference of pressure. When the reduction requires to be to a determined lower intensity and not to a ratio between the higher and lower pressures, the valve must have the passage for the flow of steam varied by the intensity of that reduced pressure, and such designs as Fig. 340 result. The low pressure and the weighted lever act together to operate the valve in opposite

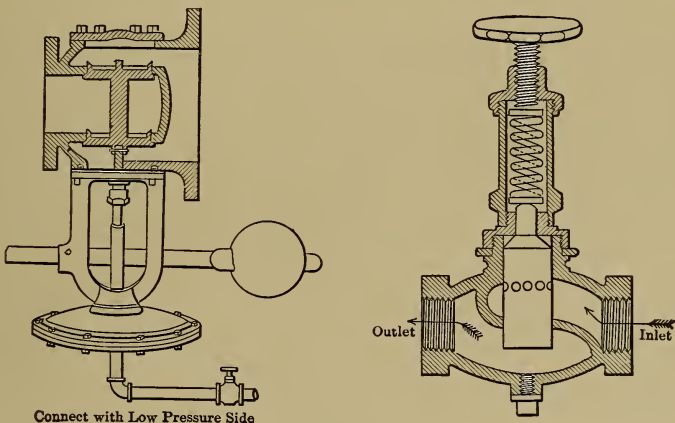


Fig. 340.

senses at the left hand, and the balance between the two determines the amount of opening. In the right-hand cut the balance is between the spring and the low pressure over the area of the plunger.

250. Drip Connections. In addition to the considerable piping required for the drainage of the main pipe line discussed in paragraf 238, there will be also pipe connections to the engine itself and its attachments, jackets, reheaters and the like for the removal of water and condensation. These are not usually continuous in their operation, but will only be used in starting or to relieve excessive accumulations, and may or may not be fitted with traps. Neither do separators attach to them as they carry solid water only. Their discharge is either wasted, or led to the hot-well or condenser of condensing-engines or into a receiver tank from which pump suctions draw. They will be therefore referred to again under the engine treatment.

251. Concluding Comment. The piping of auxiliaries offers no problems not covered in the general foregoing treatment. There are often special features such as the by-pass in compound engines through which boiler-steam can be led past the first cylinder to a later in series, in case the piston in the former should be upon the dead-center of its mechanism or its valve-function; the spindle or stem of valves may be lengthened to bring the wheel at convenient heights above the flow, and special casings may be used to inclose such spindles for appearance and for steadiness in large engine installations. There will also be water-piping for supply of water to auxiliaries, but these will not be under high pressures and the usual standards of low-pressure work will apply.

Where steam-pipe lines are led in duplicate from the boilers to the engines to minimize the danger of shut-downs the duplicate lines must come together at the boiler outlet and at the engine inlet. There will be valves close to both boiler and engine so that both lines may be cut off in case of repair to either boiler or engine. Any failure in the last link and between boiler and first valve or between engine and last valve is credited to boiler or engine and puts it out of commission.

In this chapter the energy derived from combustion and heat has been transmitted in the form of pressure to the throttle-valve at the engine, by which its delivery to the latter is controlled or arrested. The next step forming the next part of the treatment is the machine receiving this pressure-energy and transforming it from pressure in static form into work, or energy in dynamic form, of force moving under control through a space. Such a machine is called an engine.

PART V.

CHAPTER XV.

THE ENGINE.

254. Introductory. The pressure in the steam-pipe is a static force or only a potential energy until such force measured in pounds per unit of area pressed is made to overcome a resistance moved through a space in feet. So long as the throttle-valve in the pipe from the boiler is closed and tight, the heat energy of the burning fuel is either being stored in the water or being wasted. What is needed is some device to receive this pressure, to move under that pressure so that a product of feet into pounds shall give a number of units of work.

This pressure can be transformed into work in two ways. The pressure can be exerted over an area so confined in a vessel or chamber that motion of that area is possible only in one direction. This area is called a piston. Its confinement is in a cylindrical chamber which is called the cylinder. The energy or work of one traverse of the piston is PA (Fig. 4, paragrafs 3 and 6). The other way is to let the steam escape through an orifice or jet into a more open chamber where the pressure is lower. The weight of steam passing the orifice of the jet has a mass M and a velocity in feet per second v. The product Mv is equivalent to a work in foot-pounds per second, and if such mass impinges upon a surface capable of absorbing such impulse completely, the second type of engine results. The first is a pressure-motor; the second is an impulse-motor. No other ways are known for making the steam-pressure energy do mechanical work. The one system gives piston engines; the other system the turbines.

The only continuous motion is the motion of rotation, like a wheel around its axis. Straight-line motion cannot be continuous in the same direction; the best which can be done is to make the direction alternate first in one direction and then in the other. Such alternating straight-line motion is called reciprocating. The turbine motors acting by impulse of the steam mass at high velocity can be continuous in rotative effect if the blades or organs receiving the impulse can be made to revolve. This is easy to do. Pressure motors can also be made rotary if the piston receiving pressure and fitting its surrounding casing can be made steam tight and the motion of the piston around an axis be transferred to the shaft in the axis of rotation. The pistons are blades upon the arms of a wheel fast to the shaft. Ordinarily, however, the piston is made to reciprocate in a straight line in a straight cylinder: and some mechanism of practical sort is necessary to transform this intermittent reciprocating motion of the piston into a continuous rotary motion or revolution of the engine-shaft from which the power is taken off.

There would appear therefore to be three classes of engine possible:

1. The velocity or impulse type, continuous rotary; represented by the steam-turbines.
2. The pressure type, continuous rotary; represented in all rotary piston engines.
3. The pressure type, intermittent reciprocating transformed by mechanism to continuous rotary motion at the shaft only; represented by all straight-cylinder and connecting-rod crank engines. These will be called reciprocating engines.

The turbines and the rotary steam-engines would appear to have every antecedent advantage in their favor. There must be some considerations from the behavior of steam in the cylinder or from a mechanical point of view which have given to the reciprocating engine its vogue and acceptance in competition with the others.

Historically the first steam-engines of practical sort were for pumping water. In the Savery form the steam-pressure was exerted directly on the surface of water in a chamber to force it out and upward, although the piston had been proposed by Denis Papin in 1680 to keep the hot and cold masses apart. The Newcomen and Leupold designs used the separate pump driven through a lever or beam, and as the pump was a reciprocating element in a pump-barrel, the engine was made on the same lines. Watt followed Newcomen; and subsequent development hesitated to abandon the excellences of the reciprocating type. These excellences are twofold: the mechanical perfectness of the crank for its purpose; the thermal advantages of expansive pressure working in a cylinder of definite volume and low clearances. Hence the reciprocating type of piston motor will be first treated as the form pictured to the mind when the word "steam-engine" is used, leaving the simpler rotary engine and the steam-turbine for later discussion.

255. The Ordinary Steam-engine. When the word "steam-engine" is used without qualifying terms, and in advance of special study of

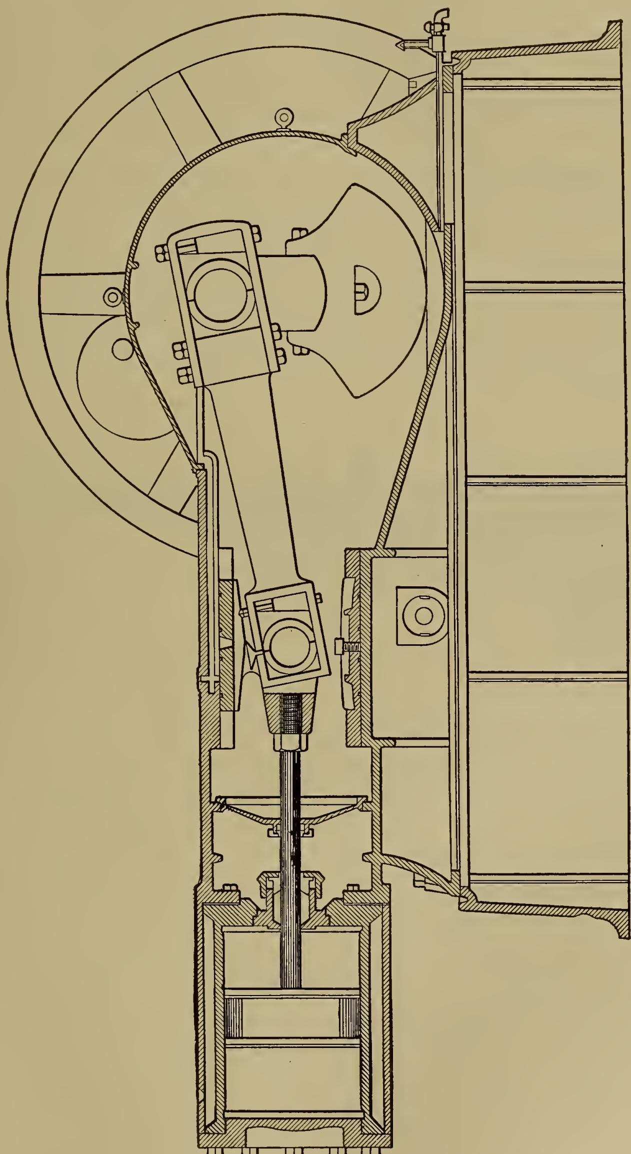


Fig. 345.

form and arrangement it brings before the mind a machine such as Figs. 345 and 346. Here at the left hand is:

1. The cylinder. This has (2) a "cover" fastened on with studs and nuts or with tap screws, and is surrounded with a hollow space or (3) "jacket" in which is either a non-conducting material (paragraf 239) or else it is an air-space, or a steam-jacket. In this cylinder, is (4) the piston. This fits the bore of the cylinder reasonably close, but not so perfectly as to run any risk of its being seized by the walls of the cylinder in case the piston expands by heat and the walls do not. To make it tight so as not to let pressure leak past its faces of contact, the piston has (5) "piston-packing rings" which are flexible and yield radially but prevent leakage from one side to the other. The piston must not be in any danger of hitting either cylinder-head, and hence there is a fraction of an inch of (6) "clearance" at each end.

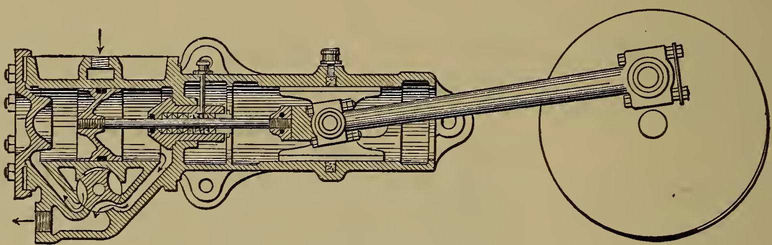


FIG. 346.

The cylinder being of limited length, the piston must move back and forth in it first in one direction and then in the other. Steam-pressure from the pipe and boiler should be admitted to the cylinder and piston alternately, first on one end and then upon the other. If so, the engine is double-acting. If steam-pressure comes only on one face of the piston — always the end away from the crank or towards the head of the cylinder, and hence called the "head-end" — the engine is single-acting. The end of the cylinder which is toward the crank and shaft will be called the "crank-end." The steam which has driven the piston on one stroke must be expelled or "exhausted" upon the return of the piston on the next stroke. This function requires openings into the cylinder called "ports" and a "valve" (Fig. 346) and its valve-gear to operate it in phase with the piston-stroke. The area of this piston $\left[A = \frac{\pi d^2}{4}\right]$ in square inches and the pressure P on each square inch give when multiplied together the quantity PA of paragraf 3; and the length of the traverse in feet L multiplied by PA, or PAL, is the foot-pounds of work done in each complete stroke (paragraf 6). The length of the cylinder inside will be the sum of the stroke, the piston depth and twice the clearance allowance.

The energy represented by PA is developed inside the steam-tight cylinder. It must be made available outside, and the motion of the piston suitably controlled. Hence there will be rigidly fastened to the piston

(7) The piston-rod. This is pushed out of the cylinder and pulled in alternately. It comes into the atmospheric pressure outside the cylinder through a construction called the (8) "stuffing-box," allowing the rod to slide through the inside cover or crank-end of the piston without leakage or loss of pressure, and yet with a minimum of needless friction. The area of the crank or rod-face of the piston is less than the head-face by the area of the cross-section of the rod. Hence the inner stroke is less powerful than the outgoing one by this difference in the value of A when P is the same on both sides.

The piston-rod though stiff is not inflexible: and it is apparent from the illustrations that at the moment chosen there is a considerable side-thrust on the right-hand end of the rod. Hence to guide this free end of the piston-rod will be required

(9) The cross-head. This is compelled to move in a line parallel with the axis of the cylinder by (10) guides. The cross-head and guides prevent the piston-rod from flexure (if the guides do not yield themselves) and keep the stuffing-box from wearing out of round and leaking. In the design shown in Fig. 345 an extra partition-plate is introduced on account of the method of oiling chosen, but ordinarily the guides come close to the crank-end of the cylinder. Connecting the (11) "cross-head pin" to the crank-pin at its other end is

(12) The connecting-rod. This is a requirement of the mechanism to convert the reciprocating motion common to the piston, its piston-rod and cross-head, into the rotary motion desired for the shaft. It will have provision to take up wear on the pins at each end, (13) "brasses," and will be twice the length of the crank on the shaft or over. Its presence is necessary but it is of no dynamic advantage. The connecting-rod transfers the alternate push and pull of the piston-rod to (14) the crank-pin on the (15) crank of the (16) crank-shaft. The crank-pin has to receive the entire force PA (less the modification due to any angularity of the push of the connecting-rod) and turn the crank against the resistance upon the crank-shaft, and must do this by a rubbing contact of the brasses upon the cylindrical pin. The leverage of the crank to revolve the shaft may be called its "torque" or turning-moment. In the form shown the crank is counter-

weighted opposite the pin to balance the inertia effects of the massive connecting-rod.

Beyond the (17) "main bearing" of the crank-shaft just behind the crank is (18) the fly-wheel, acting as a regulating device for uniform rotation, and whatever may be the form of the application of the resistance, either by a belt-wheel or gears or generating dynamo. In locomotives and marine practice there is no fly-wheel as the driving- or water-wheels supply this regulating influence as well as the mass propelled.

The typical skeleton mechanism is therefore:

1. The cylinder.
4. The piston.
7. The piston-rod.
9. The cross-head.
10. The guides.
12. The connecting-rod.
14–15. The crank.

The crank is the controlling element, as well as the power transmitter. Hence the radius of the crank determines or fixes the unit for all else. Because the length of

1. The stroke L is exactly two cranks.
2. The cylinder is two cranks and allowances.
3. The piston-rod is two cranks and allowances.
4. The guides are two cranks and allowances.
5. The connecting-rod is 2–2½–3 cranks.

From the head end of the cylinder to the tip of connecting-rod when at its furthest point is therefore a total of seven cranks and necessary allowances.

It will be serviceable to examine the kinematic principles underlying this mechanism.

256. The Kinematics of the Crank-connecting-rod Mechanism. Some Engine Mechanisms. The combination of the crank-arm, the connecting-rod, the constraint of the cross-head in its guides and the bed-plate which keeps the center of the crank-shaft in a fixed relation to the path of the cross-head, shows that kinematically the typical steam-engine is a four-bar or four-link mechanism (Fig. 347). Of this the crank is No. 1, the connecting-rod is No. 2; an arm with a center at an infinite distance at right angles to the guides is No. 3, and compels it to a translation in a straight line at right angles to such swinging link of infinite length. The bed-plate fixing the center of the first link and the path of the cross-head, is in effect the fourth link, joining the center of the crank-shaft with the center of No. 3 at an infinite distance (Fig. 348). If now the fixed link in the chain be made No. 2 of the series, then keeping No. 1 still the crank-arm, No. 3 and No. 4 must adjust themselves to the rotation of No. 1 around

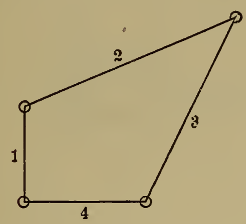


Fig. 347.

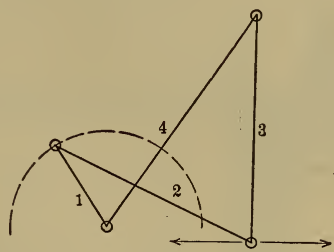


Fig. 348.

its other end, and the chain appears as in Fig. 349 with the power applied to turn the crank along No. 4. The link No. 3 must therefore swing around the other fixed end of No. 2 and an engine with oscillating

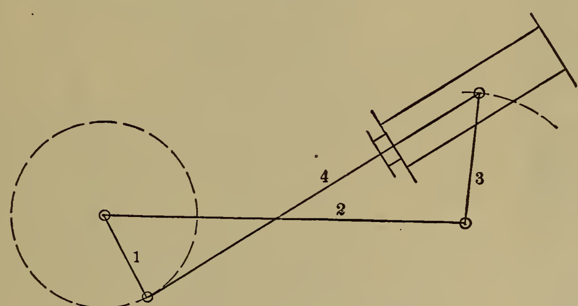


Fig. 349.

cylinder results. If instead of making No. 3 of considerable length it has the radius of the trunnion of the oscillating cylinder, the engine

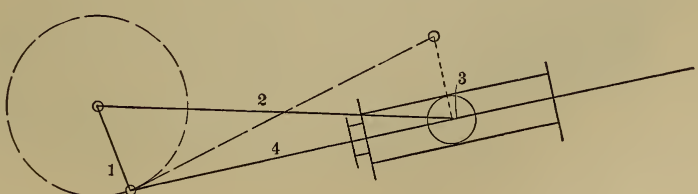


Fig. 350.

appears in its more recognizable form of Fig. 350, and the piston functions as connecting-rod also.

If now the piston-rod-element be eliminated from the type-design to secure lightness in the moving mass for high speed, and shorten the length of the mechanism and yet avoid the oscillation of the cylinder mass, the kinematic chain remains the same, but the swing of the connecting-rod or link No. 2 must be allowed to take place within the cylinder in part. Fig. 351 shows this type in the shape much used in motor vehicles where the piston is made long enough to prevent cocking or jamming from the oblique thrust or reaction of the connecting-rod, and where the piston is of small diameter. When a large turning moment is desired as where pressures are low the trunk engine takes the form of Fig. 355A in the Appendix, designed for a screw-driven monitor type to trim the weight in the hull and keep all of the engine below the water-line. (See also Figs. 403–5.)

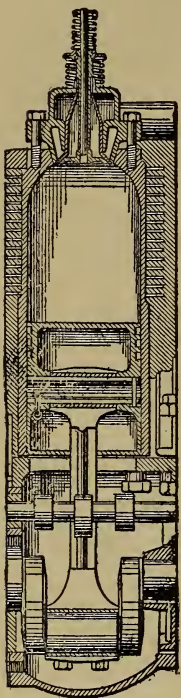


Fig. 351.

Finally to keep the engine short and yet have a long connecting-rod, the link No. 2 can be actuated from a cross-head behind the cylinder-head instead of having the cross-head between cylinder and crank. The connecting-rods for linking such cross-head to the crank-link No. 1 will be doubled for symmetry and pass outside of the cylinder on each side of it. Such an engine becomes a "back-acting engine."

The types of mechanism are, therefore,

1. The normal, or direct-acting engine.
2. The oscillating engine.
3. The trunk engine.
4. The back-acting engine.

Figs. 345 and 346 will stand for the normal direct-acting horizontal engine. Figs. 352 and 353 show the simplicity and directness of the oscillating or No. 2 type in small sizes; and the particular design is selected because the trunnion on which the cylinder oscillates as in the skeleton of Figs. 349–350 is made of a full size to reproduce the link 3 in that diagram. Such engine will work equally well bolted to the ceiling, or as it were, upside down. It is more usually constructed in the larger sizes to meet the condition of Fig. 354, which is taken from European practice in side-wheel or paddle-boat engines. Here the overhead shaft carries the wheels. Steam enters at the left hand through the hollow trunnion and passes successively through three cylinders in series, escaping at the right. The short distance from outer cylinder-head to crank-shaft center enables such an engine to be gotten into shallow hulls for light-draft conditions. Oscillating engines were much used in warships before the high-speed screw design came into vogue. The friction on the trunnions resulting in wear, resulting in leakage, and the slow rotative speed compelled when the cylinder

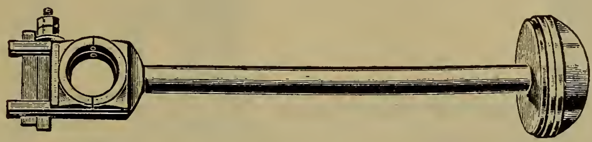


Fig. 352.

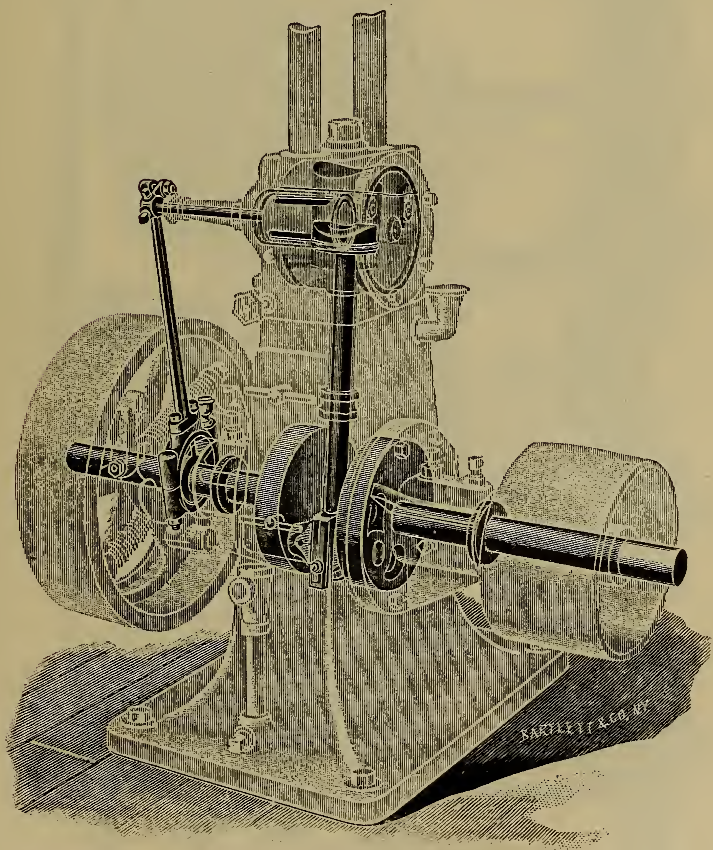


Fig. 353.

masses are large has made the large oscillating engine of historic interest only.

The third type or trunk-engine of Fig. 351 has been developed from

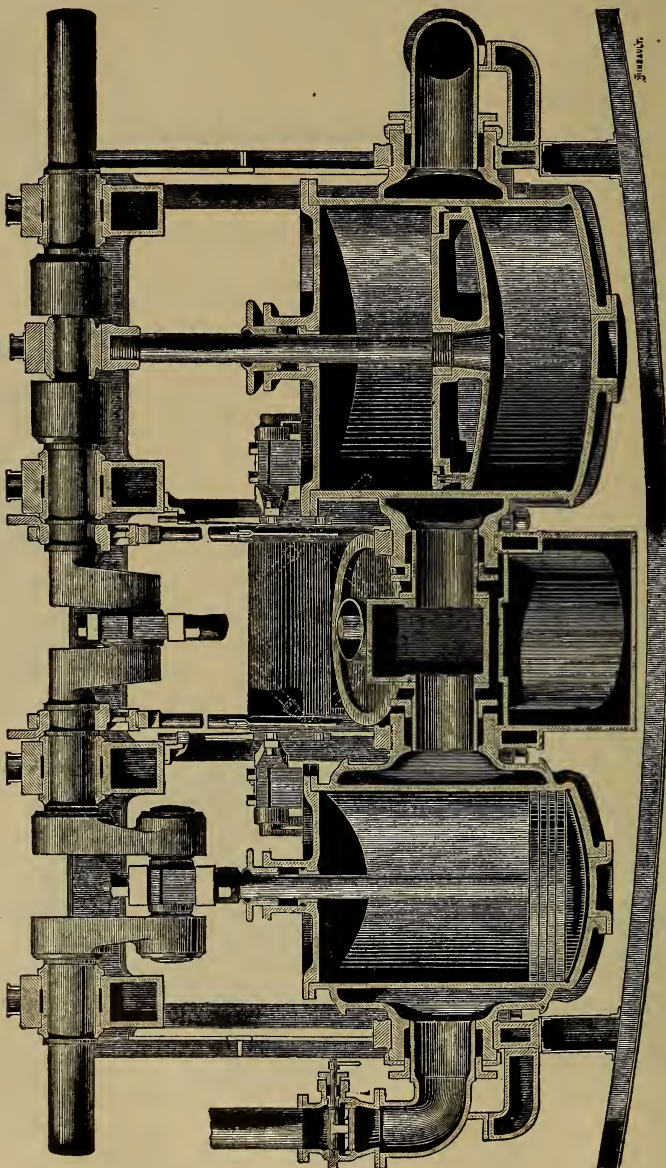


FIG. 354.

that simple form intended to be single-acting into more complete forms where it has been sought to make it double-acting. In Fig. 355A, for example, in the Appendix is the true trunk-engine, double-acting. The piston area is the annular space surrounding the trunk which slides in and out through stuffing-boxes as though it were a hollow piston-rod. The internal diameter of the hollow trunk must be sufficient to let the connecting-rod clear the upper and lower elements when in the position of maximum departure from the axis nearly as shown.

Such engines have an inconvenient diameter for the piston and a comparatively short stroke. The trunk at the back or head end is not

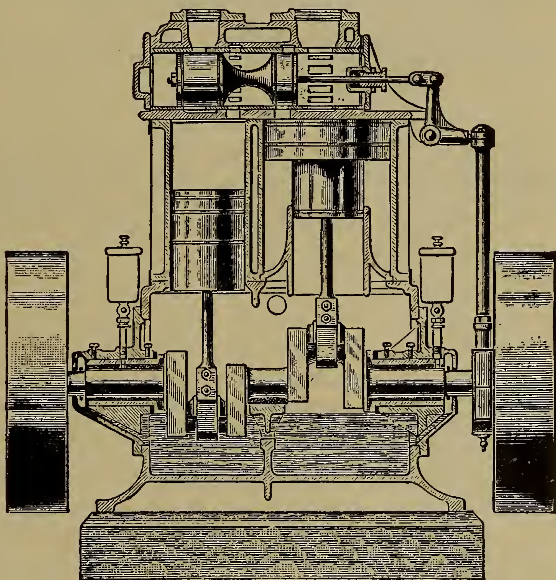


Fig. 356.

called for and is only put to equalize the values of PA on the outer and inner faces. If the rear trunk is left off, the engine is a half trunk such as illustrated at the right hand of Fig. 356. The left hand is the usual single trunk. In Fig. 357A in the Appendix is again a derivative type where mechanical effectiveness was sought by an extra long connecting rod, and the designer used the wasted area in the rear trunk to secure such length. It is applicable only to very short crank types and of small size.

In Fig. 358A in the Appendix is also a transition form where a half trunk of rectangular or long oval section was used to reduce inequality of effort up and down. This is much more troublesome to keep steam

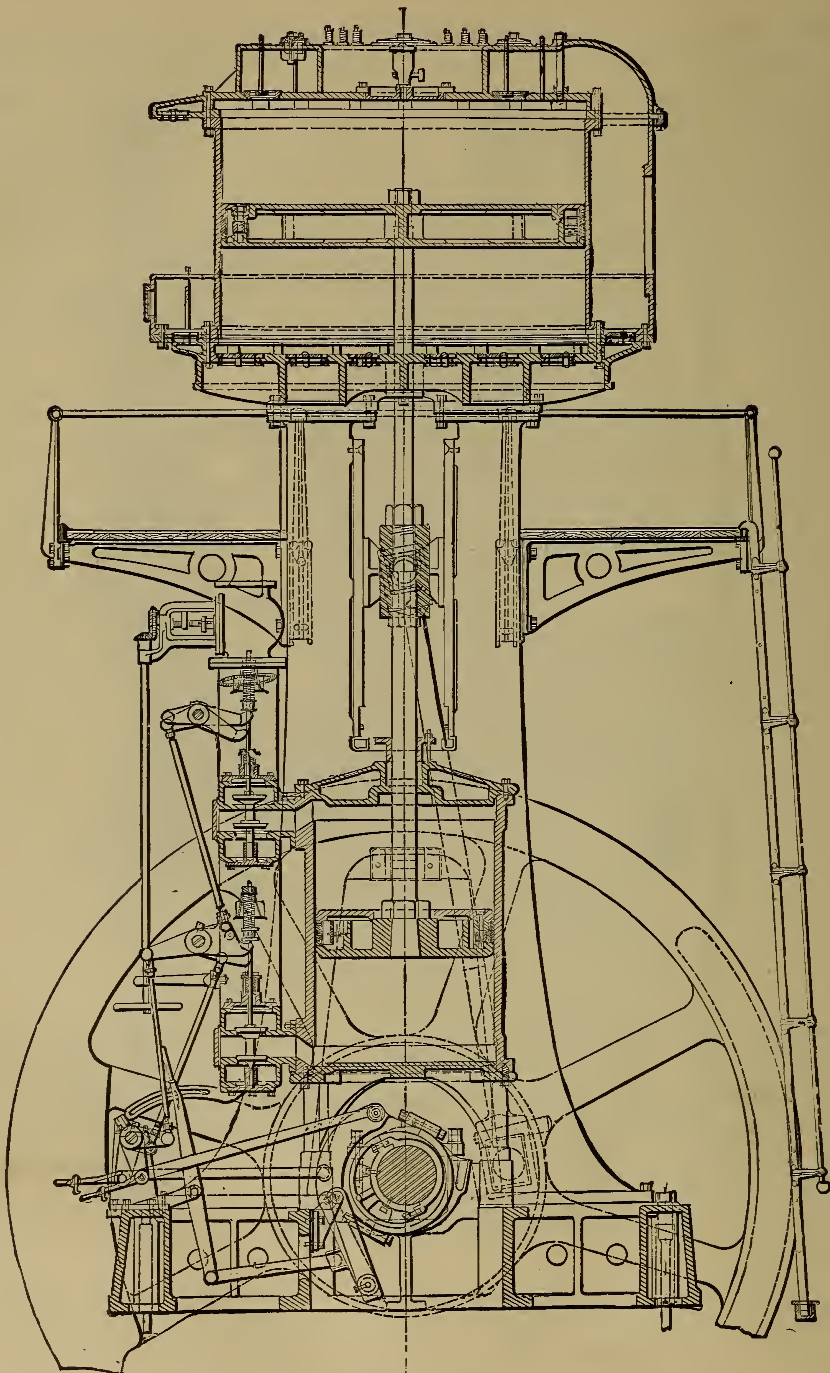


FIG. 361.

tight and will not be used where it can be avoided. The normal trunk types are those of Figs. 351 and 356.

The back-acting type of mechanism is an outgrowth of special conditions where the slow speed makes the mass of metal in the mechanism of secondary consequence. Fig. 359A in the Appendix is a type of such an engine derived from the earliest river-boat mechanisms, where the location of the crank-shaft above the keelson is at a determinate height, with no room for the cylinder below the shaft and a vertical cylinder is preferred. Two connecting-rods come down from the cross-head one on each side. The crank-shaft may or may not be continuous across the hull under the cylinder. Fig. 360A shows the horizontal type for warship conditions, and where, as in Fig. 355, it is desirable to trim the ship by putting weights symmetrical with the keelson and below protective water-levels. The piston-rod is made double and the two rods straddle the shaft in the first and fourth quadrant of the piston; the short connecting-rod bends back from the cross-head to reach the crank of the propeller-shaft. On the left hand are the condenser and pumps to balance the mass of cylinder and piston.

The back-acting design occurs most frequently in pumping-engines or blowing-engines for air. Fig. 361 is such a blowing-engine for low pressures and great volumes of air for blast-furnace practice. The steam-cylinder is just above the fly-wheel shaft, and from the cross-head at the top of the cylinder two back-acting connecting-rods descend outside the twin fly-wheels on each side to pins upon the large hubs of the wheels. The piston-rod is prolonged beyond the cross-head to receive and operate the blowing cylinder-piston above the platform. If the power were taken from the fly-wheel, this would be a true back-acting engine like Fig. 359. The fly-wheel function, however, is only of control and regulation, but it is desired to have its bearings low down and near the ground. Hence the back action. This engine shows some details of piston construction and the use of poppet valves to be further referred to later. In Fig. 362 the same set of conditions is repeated in horizontal form for an air compressor of compact arrangement. The steam or power-cylinder is between the fly-wheels, and the crank-shaft behind it. The two connecting-rods and two cranks make it a back-acting engine at its power end. The air-compressing end on the prolongation of the engine-piston is entirely free of the steam-engine mechanism, as in Fig. 361. In both cases, the cross-head should not be so rigidly fitted or keyed to the piston-rod as to be strained unduly if one of the rods wore more than the other at its bearings, so that they became of unequal length from center to center of bearing-pins

It may be said in general, however, that these more complex mechan-

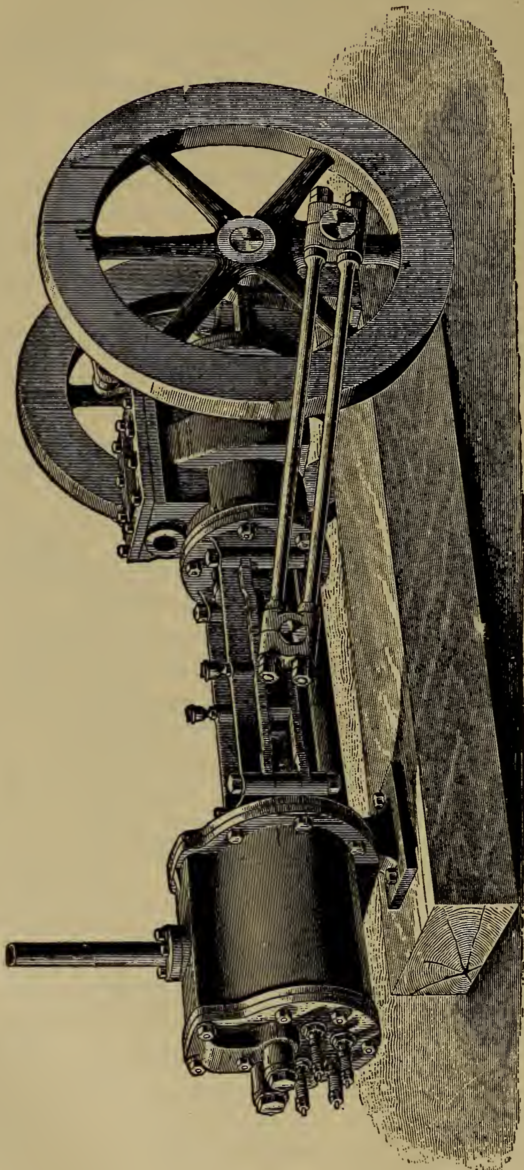


Fig. 362.

isms are becoming of less and less consequence as the advantages of standard proportions become better realized, and the advantage of reducing the mass of material to be started, accelerated and stopped in each traverse of the piston.

257. Deductions from the Kinematic Chain of the Steam-engine. The recognition that the typical steam-engine mechanism was a four-bar kinematic chain with one link fixed has led to some serviceable deductions.

First as to the relative velocity at any point of their unequal travel, of the crank-pin and the piston, or of the cross-head invariably connected to the latter. In

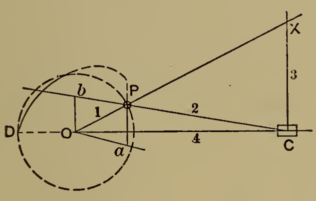


FIG. 365.

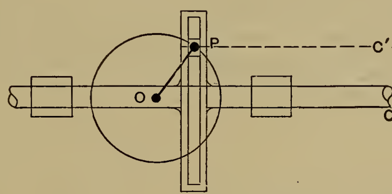


FIG. 366.

Fig. 365 let O be the center of the shaft, OP the crank, PC the connecting-rod, and C the cross-head pin. The fixed link in the chain is No. 4 as in Fig. 348. If the number of revolutions of the engine is known, and OP is known or measured, then the velocity of the crank supposedly uniform is $a \times OP = v$ if a is the angular velocity of the revolving crank $= 2\pi n$. The velocity of the point C is then capable of being found or measured by the principle of the instantaneous or virtual center, which gives the link 2 or PC the same motion as it actually has by the concept of its being revolved around a center located by prolonging lines through the centers of its actual motion until these intersect. In this case, the motion of C is a rotation around a center infinitely distant above or below the line OC and perpendicular to the latter. The motion of P is as it were around the center X found by prolonging OP till it intersects the vertical through C. Then by the principles of rotation,

$$\text{Velocity of } C \times PX = \text{velocity of } P \times CX$$

or

$$\text{Velocity of } C\text{: velocity of } P :: CX : PX.$$

If now from O a line be drawn parallel to PC, and from P a perpendicular Pa be drawn to the latter parallel to CX, then the two triangles PCX and POa will be similar; whence

$$\text{Velocity of cross-head: velocity of pin} :: CX : PX :: Pa : OP.$$

If the velocity be called constant and be represented to scale by the crank-radius OP, then the quantity Pa to the same scale represents the cross-head and piston velocity. An identical construction results when the center line of PC is prolonged and a perpendicular OP is drawn to it, making PbO similar to PCX.

If with the measured Pa or Ob for different positions of P as ordinates a curve be drawn with the diameter of the crank-circle $= 2\ OP$ as base, and a curve be drawn through the ends of these ordinates, the curve will represent the velocity of the cross-head for each position of P when the radius of the crank-circle represents the velocity of the pin. It will resemble the dotted curve in Fig. 365 superposed upon the crank-pin path. If the connecting-rod were of infinite length, or some mechanism were used which would keep the link 2 or the connecting-rod PC always parallel to itself and the line OC, then the velocity diagram of the cross-head becomes a circle, the ordinate in every case being the sine of the crank-angle from a zero in the line OC. (Fig. 366.) The yoke mechanism of Fig. 233 is a device to this end.

This same conception enables the velocity to be computed with which the connecting-rod brasses are rubbing upon the pin at any point of the crank-travel. The two elements at P may be regarded as being the contact of two gears, the radius of one being OP and of the other PX, or the radius to the virtual or instantaneous center. Let the angular velocity of the crank be a as before, and the crank-radius R_c. These are known, and hence the tangential velocity common to the two elements, which must be the same for the virtual rotation with an angular velocity b and a radius R_b. Then

$$R_c a = R_b b \text{ and } b = \frac{R_c a}{R_b} = \frac{V}{R_b}.$$

When the crank and connecting-rod are revolving in opposite directions, the rubbing velocity will be the sum of the crank and connecting-rod rotations. If the crank-pin radius is r_p, then the rubbing velocity will be

$$V_r = r_p\,(a + b)$$
$$= r_p a\left(1 + \frac{R_c}{R_b}\right).$$

This is at its maximum when $\frac{R_c}{R_b}$ is its maximum or when R_b is a minimum, which it is when the point X coincides with C or the engine is on its inner dead-center. When the crank is between the 90- and 180-degree points, the rubbing is in the same direction, and the rubbing velocity

$$V_r = r_p a\left(1 - \frac{R_c}{R_b}\right),$$

a minimum when the crank is at its outer dead-center. If the connecting-rod be assumed to be n times the length of the crank, then for the maximum value above

$$V_r = r_p a\left(1 + \frac{1}{n}\right),$$

in which all quantities are derivable or known. The rubbing never becomes zero.

In the foregoing discussion the term "dead-center" has been used. It is obvious that the length of the instantaneous radius CX becomes zero when the links 1 and 2 are in the same straight line. That is, the cross-head has no motion at these two instants when the line OP coincides with OC and with its prolongation in OD. Power is only exercised or made available when the pressure force makes the piston move through a space. When the piston has no velocity, the steam does no work. Or, stated otherwise, the pressure of the steam P on an area A can only cause the crank to revolve when it has a lever arm or moment to cause such revolution. When the sine of the crank angle in Fig. 366 is zero, there is no leverage or moment to revolve the crank OP. An engine without crank leverage sufficient will not start however powerful it may be compared to its resistance, and an engine in such a relation of its mechanism is said to be "on its dead-center" or "on the center." There are two of these, the inner dead center and the outer dead center, according as the crank-pin is between the cylinder and the shaft, or beyond the latter.

There is also another dead-center on which an engine will not start to move. It is that which is due to the fact that both ports to the cylinder at its two ends are covered by the valve in Fig. 346. The design of the valve should not allow both ends to be open to steam at once: hence there will be a relation of valve to port and to piston position in which no steam nor any turning energy can reach either face of the piston. Hence a dead-center even if the mechanism gave a turning moment. In engines working expansively, later to be discussed, this latter dead point may become a dead period at the end of the traverse in each direction, during which the engine cannot be started from rest by its own steam.

A fourth deduction is the truth that with a connecting-rod PC of finite length the cross-head C has gone further than one-half of its travel when the crank has moved 90° from its inner dead-center. In Fig. 366 with an infinite connecting-rod

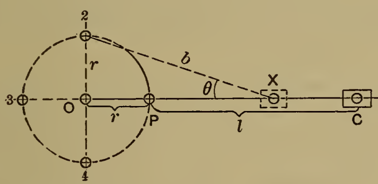


FIG. 367.

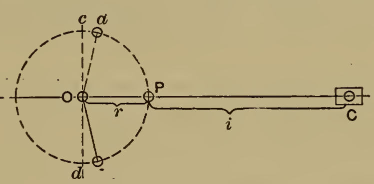


FIG. 368.

this is not the case. This can be shown by two methods. Let the distance from the crank-shaft O to the cross-head C on its inner dead-center be the sum of the crank-length r and the connecting-rod l (Fig. 367). Then when the crank has moved through 90°, the distance OX should be equal to l if the distance XC was equal to r or the half stroke of the piston. But plainly OX is not equal to l, since it is $l \cos \theta$, and $\cos \theta$ is less than unity: hence XC is greater than r, or the piston has moved through more than one-half its traverse in the first 90°. Or the second method of Fig. 368 may be used. At the dead-center the distance $OC = r + l$. With C as center and OC as a radius describe an arc through O. It cannot cut the circumference in the line cd through the 90° and 270° points, but at some points a and b. Let the crank now swing to the point a. The distance along the lines of the mechanism to C will be the line $PX + XC$. The sum of $PX = l$ and XC must be greater than the third side of the triangle $PC = r + l$. But l cannot change in length: hence XC must be greater than r or the half-stroke. This irregularity will become greater as the connecting-rod becomes shorter relatively to the crank, and vanishes when the crank becomes of infinite length as in Fig. 366.

258. Effectiveness of the Crank-connecting-rod Mechanism. The foregoing kinematic discussion enables a most important and serviceable conclusion to be drawn. When a force acts at any point of a mechanism and overcomes a resistance at another point, the work done at the two points must be equal if friction be neglected. Hence in the steam-engine the effort of the steam PA acting through its path $2r$ must be equal to the resistance R reduced to the crank-pin and acting in one stroke through the space πr. If not, the engine must change its velocity until there is a balance between the effort and resistance at this new speed: because it is also true from the principle of work that the efforts in any mechanical combination are inversely as the velocities. If now the sum of all the piston velocities with an infinite connecting-rod be the area inclosed within the semicircle for one traverse or stroke in Fig. 365, then the sum of all crank-velocities for the same period will be the sum of the areas swept over by radius OP equal to the crank-velocity for each crank-angle. But each of these areas is equal to the same thing, $\frac{\pi r^2}{2}$. Hence they are equal to each other, and the value of the piston effort equals that of the crank-resistance in one stroke: or, there is no loss of effort in the crank-connecting-rod combination except that due to friction.* The deduction of paragraf 3 is, therefore, justified, and

* This same result can be obtained by the methods of the calculus. Here it is simpler to get the area for both semicircles or one complete revolution by getting the area of the whole circle from a pole in the circumference. Then $r = a \sin \theta$, in which $\frac{a}{2}$ is the diameter, and $\frac{1}{2}\int r^2 d\theta = \frac{1}{2}\int_0^{\pi} a^2 \sin^2\theta d\theta$. This is equal to $\frac{a^2}{2}\int \sin \theta\delta\theta$

$= \frac{a^2}{4}(1 \cos - 2\,\theta)\,d\theta = \frac{a^2}{4}\,\theta - \frac{a^2}{8} \sin 2\,\theta \int_0^{\pi} = \frac{\pi a^2}{4}$.

Substituting for a its equivalent $2r$ and dividing by 2,

$$\text{Area} = \frac{\pi r^2}{2}.$$

Or, again, let P denote the pressure on the piston-area, and V its velocity at any point of its traverse. Let p denote the tangential effort on the crank-pin revolving in its circular path and v its velocity. These may be uniform or constant, but in any case the time may be taken short enough to have them considered constant without error.

If now a circle be drawn representing the path of the crank, and at any point the pressure P be represented in direction and intensity by a line parallel to the axis of the cylinder acting at that point, it can be decomposed into two components at right angles, one tangential and one normal. The tangential component will be the effort p at that point, and will be perpendicular to the radius. The normal component will coincide with the radius produced. If the tangential component p be projected on the piston effort P, there will be three similar right-angled triangles produced. From these it will appear that the tangential pressure on the pin will equal the piston-pressure into the sine of the crank-angle, becoming equal to it at 90° and 270°, and being zero at 0° and 180°. The velocities of crank-pin and piston will be to each other in such a relation that the pin-velocity will be a mean proportional between the piston-velocity and the projection of the pin-velocity on the difference between the indicated horse-power and the net horse-power is the mechanical friction only. This answers also the recurrent contention of the ill-informed that the crank is an ineffective transformer of motion or transmitter of effort and something else should be substituted for it.

Plainly also the shorter the connecting-rod in terms of the crank-length, the greater the value of the sin θ of Fig. 367 for any crank-angle. This means that the pressure from the cross-head upon the guides is greater with such short rod than with the longer and therefore the friction loss is greater, and the pressure and scraping effects are greater which act to expel the lubricant from between the two rubbing surfaces. The pressure on the guides does not alternate in direction as at the crank-pin, as can be made clear by the four cases of Fig. 369. In case No. 1 the

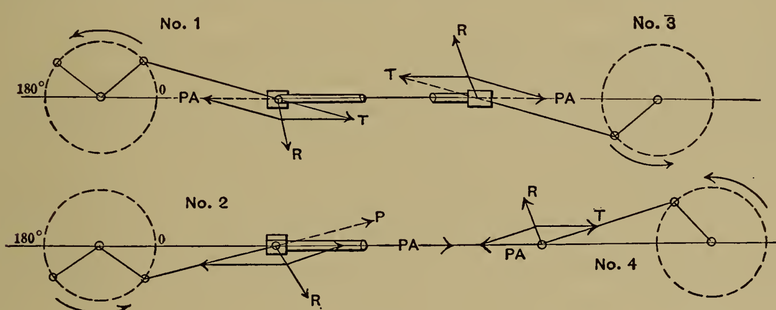


FIG. 369.

crank starts from its inner dead-center and rises from the horizontal. The piston is pushing on the cross-head with an effort PA and the connecting-rod resisted by the crank is thrusting back in the line of its axis T. The resultant of PA and T from zero to 180° is a downward force upon the lower guide. In case No. 2 the crank is returning from the 180° point to zero and the piston is pulling with an effort PA and the connecting-rod is in tension. The resultant is again downward all through this traverse. Such an engine is said to "throw over"; its pressure is upon the lower guides only, with minima at each end of the stroke. In case No. 3 the crank "throws under." The piston is push-

the direction of the piston-velocity. (Legendre, Bk. IV, Prop. XXII.) Hence, from what has preceded,

$$v : V :: p : P,$$

or

$$Pv = pV$$

for an instant of time and for any point. Hence, at any point the work given to the piston equals that received by the crank, less the loss from friction of joints or moving parts. In other words, the crank mechanism is theoretically perfect.

ing with PA and the connecting-rod in compression gives a resultant directed upward; in case No. 4 the crank is returning over the center line, the piston-rod and connecting-rod are in tension, and the resultant is again upward throughout the return stroke. The cross-head can be lubricated by a bath of oil on the lower guide; the lower guide springs less in a horizontal engine when adequately supported by the frame and foundation. Hence "throw-overs" will be usually preferred. The locomotive throws under in going forward. In vertical engines it is indifferent if the two sides of the frame are the same. When one side is open (Fig. 380) then the engine turns so as to thrust against the heavy closed side. Engines to reverse frequently or to do equally heavy work in either direction must be equally resistant in both directions. The resultant R of the four cases must always pass within the contact area of the cross-head and guides, else there is a tendency at work to cock the cross-head diagonally between the guides and jam it so that it will scrape off the oil and run hot.

259. Dynamic Stresses in the Mechanism of the Typical Steam-engine. The stresses which the engine must undergo and resist and which tend to wear it out or injure it are five:

1. Those due to the pressure coming over from the boiler through the pipe. The cylinder receives these in the form of PA upon the piston and the cylinder-cover. The barrel of the cylinder receives them as the boiler does (paragraf 25) in intensity given by PD. The effort PA tends to pull the piston off the rod, or to shear and strip the fastenings: it tends to flex the piston regarded as a beam fixed at the center and uniformly loaded over its area. It tends to buckle or tear the piston-rod apart, regarded as a column fixed at both ends, and as a tension rod alternately. It tends to shear or strip the fastening of the piston-rod to the cross-head, to shear the cross-head pin, and to produce the same effects on the connecting-rod and crank-pin. It tends to push the main bearing away from the cylinder and to draw it inwards alternately. When P is known these values are very definite and easily computed.

2. The forces due to the inertia and the living force of the masses in piston, piston-rod and cross-head which reciprocate, and in crank and crank-pin which revolve, and in the connecting-rod which has a combination of motions which make its center of gravity move in an ovoid curve. The previous paragrafs 256 and 257 showed that the piston came to rest in the cylinder when passing its two dead points, and that it has the velocity of the crank-pin at or before the 90° point of the crank according to the length of the connecting-rod. Hence in the first quadrant all reciprocating masses attached to the piston must be

accelerated or have their inertia overcome. In the second quadrant, the piston and attached masses must be slowed down and their living force absorbed until they come to rest at the outer dead-center again. These forces may exceed the PA values of the previous paragraph, and affect both cylinder and crank-pin, the latter especially if it is the dependence to bring these accelerations and retardations about. The energy absorbed in accelerating is given out in retardation less friction, so that the principal effect is on the structure which steadies the engine and on its bearings and upon the smooth and silent running of the engine. The importance of these inertia effects justifies their further discussion in a subsequent section. The flinging stress in the connecting-rod from the arrest of its mass moving with a lateral component, which occurs when the crank passes its 90° and 270° point, are in this same class.

3. The stresses from water in the cylinder, coming over from the boiler through the pipe or from condensation in cylinder or pipes. This may produce stresses far in excess of the other two. The clearance space between piston and cylinder cover at the dead-centers is intentionally small: as the piston approaches the end of its traverse the steam-valve should close the ports or passages of entry to it and exit from it, and in any case will greatly reduce the openings through which water might escape. If the volume of water entrapped in the cylinder exceeds the volume of clearance and passages up to the valve, such water acts by its practical incompressibility as a solid mass to arrest the piston before its stroke is completed. At the other end of the chain the living force stored in the fly-wheel and the machinery or masses which the shaft is driving are all moving at full rotative speed: this aggregate applied through the crank-pin finds the mechanism of crank and connecting-rod just at that relation of enormous leverage which is given in the elbow-joint press at the moment when the links are coming into line, and transmits this energy to the compression of the water. If the parts receiving this blow are strong enough to resist it, the engine stops. If they are not — and sometimes the energy is so enormous that they cannot be made so — the piston breaks, the cylinder-cover breaks, the cross-head breaks, the cylinder cracks, the bed-plate cracks, or pins at cross-head or crank are bent or sheared, the piston-rod parts, or the bolts fastening the cylinder cover strip their threads, according as one or more of these elements are lower in strength than others. Many able designers make a purposedly weakest point in the engine to concentrate the breakage here where renewal and replacement are also cheapest. The cylinder cover studs are probably the most usual. At sea, where water may be thrown from the boilers by rough weather,

the cylinders have relief-valves to save them. This danger of practically irresistible stress emphasizes the meaning of separators (paragraf 243), drip-pipes and traps (paragraf 250) and non-conducting coverings (paragraf 244). The engine must be started very slowly and cautiously to allow water to get away. The design of Fig. 346 is particularly excellent from this point of view of water relief through its valve.

4. The stresses due to sliding contact, rubbing, and friction. These must be met by adequate areas in the design so as to reduce the unit pressure on them to the least value consistent with meeting other requirements, and by such lubrication as shall keep metals from contact except with a film of oil between. Such bearing areas are those between the piston or its rings and the bore of the cylinder; between the rod and its packing in the stuffing-box; between the cross-head and its guides; between the pins at cross-head and crank and the connecting-rod brasses and between the shaft journal and its bearings in the frame. Adequate bearing area and the reduction of unit pressures on areas so as not to expel the lubricant belong to engine-design and are in a field too large and important to be entered here. Lubrication of such well-designed surfaces will be treated in a later chapter.

5. The stresses due to heating the metal. The engine is bored, planed, and fitted at the shop temperatures; in use it is subjected to temperatures over 300° F. and with superheated steam to a much higher range. These heats cause expansions which deform and warp the fitted surfaces when resisted, and cause a superposition of stress. With very hot gas the castings grow longer or larger, due to a molecular rearrangement in their structure, no doubt; the closely fitted surfaces seize and grip; planes warp and wind. Heat also injures the lubricating quality of badly chosen oils upon such hot surfaces, changing them into gas in part and oxidizing the balance into a gum which prevents easy motion instead of favoring it. Deformations, however, are the worst evil, and the troubles from them should be guarded against in the design so far as possible. Lifting poppet-valves for example are better than sliding ones, and cylindrical piston-valves are better than flat ones (paragrafs 231 and 417).

To this same class belong the stresses in the castings of the engine due to the impeded shrinkage of such castings when cooling from the molten stage in the mold. An initial stress from this cause may be added to the working or heat stresses, and combined they may exceed the resistance of the material. These give special trouble in cast-iron fly-wheels, or when the stress from water comes on.

260. Inertia and Acceleration of the Reciprocating Parts of a Steam-engine. The velocity diagram of the piston and its attached masses (Figs. 365, 366) have made it plain that only at or just before the 90° point of the crank-motion has the piston the same velocity as the uniformly revolving crank-pin. During the first half of the stroke this mass M or weight W is receiving and storing energy imparted to it by the pressure PA and during the last half this energy is being given out or restored to the shaft through the crank-pin. If the engine is rotating slowly this acceleration and retarding of mass may be unnoticed: at high speeds or in the absence of a fly-wheel to mask the process, the smoothness and silence of the engine may be materially affected, and shaking of the engine on its foundation and of the foundation structure itself may result. If the inertia or resistance to acceleration of the reciprocating parts, reduced to a resistance per square inch of piston-area, exceeds the initial pressure P in the cylinder, it is plain that the steam effort did not reach the crank-pin until P had overcome this resistance and there was a balance in favor of the forward effort. This resistance as the speed grows higher may be enough to hold back the engine from attaining any higher speed or running away when the load is suddenly taken off.

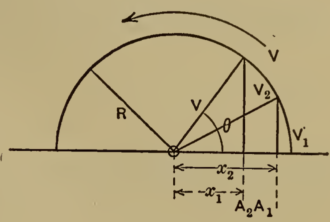


FIG. 370.

It will be remembered that the velocity is the ratio of the space passed over to the time taken to move through that space: the acceleration is the ratio of the velocity at the end of such a time to the time in which that velocity was acquired.* The simplest case will be that of an infinite connecting-rod and a constant crank-pin velocity. Then the semicircle of Fig. 366 becomes the velocity diagram when the uniform crank-velocity V is the radius. The piston velocities for each crank-angle θ become (Fig. 370) $V_1 = V \sin \theta_1$; the living force or kinetic energy for a mass of reciprocating parts $M = \frac{W}{2g}$ at any point of the piston-travel as A_1 is

$$\text{K. E.} = \frac{WV_1^2}{2g} = \frac{WV^2 \sin^2 \theta_1}{2g}$$
$$= \frac{WV^2V_1^2}{2gR^2}$$

since

$$V = \frac{V_1}{\sin \theta} \quad \text{and } V = R.$$

At a subsequent position of the crank and piston A_2,

$$\text{K.'E.'} = \frac{WV^2V_2^2}{2gR^2},$$

so that since the velocity is increasing from A_1 to A_2 the increase will be

$$(\text{K.'E.'} - \text{K.E.}) = \frac{WV^2}{2gR^2}(V_1^2 - V_2^2).$$

---

* With mean velocities, $v = \frac{s}{t}$; with mean accelerations $a = \frac{v_1 - v_2}{2t}$. If the time be made infinitesimal, the statements become true for all conditions; or

$$v = \frac{ds}{dt} \text{ and } a = \frac{dv}{dt} = \frac{d^2s}{dt^2}$$

If the motion of the piston from A_1 to A_2 along the diameter be from a distance x_2 from the center to a distance x_1, this corresponds to a work done under a total pressure P for this purpose of

$$\text{Work} = P\ (x_2 - x_1).$$

But since the radius and sines make a right-angled triangle

$$x^2 + V_2^2 = x_1^2 + V_1^2 = R_2,$$

so that

$$x_2^2 - x_1^2 = V_1^2 - V_2^2.$$

Whence the increase of kinetic energy can be written,

$$\text{Increase of K. E.} = \frac{WV^2}{2gR^2}(x_2^2 - x_1^2),$$

which is equal to $P\ (x_2 - x_1)$.

Whence

$$P = \frac{WV^2}{2\ gR^2}(x_2 + x_1),$$

which can be written

$$P = \frac{WV^2}{gR^2}x$$

if x be the mean distance $x = \frac{x_1 + x_2}{2}$ of the piston from the crank-center during the time in which this acceleration or increase of kinetic energy took place. At the beginning and end of the stroke the change from rest to motion is a maximum; and when $x = R$ under these conditions

$$P = \frac{WV^2}{gR}.$$

This is identical with the expression for the acceleration due to centrifugal force, which implies that the reciprocating parts produce the same effect in the crank-pin in their actual motion as they would do if they were rigidly attached to the crank-pin, and while kept parallel to themselves were rotated around the engine-shaft. Only the components parallel to the cylinder axis are under consideration, however, and this is a maximum as above at the dead-centers and a minimum or zero at the 90° and 270° points. It is positive in the first half stroke and negative in the second half. Since the difference in the squares of the velocities is equal to the difference of the squares of the distances from the crank-center and is a constant equal to R^2, all triangles whose one side is proportional to such change in kinetic energy are similar to each other; so that a vertical line drawn at the dead-center upwards at one end and downwards at the other and with a value to the scale of the crank-pin velocity of $P = \frac{WV^2}{gR}$ will give the relation of P to such velocity.

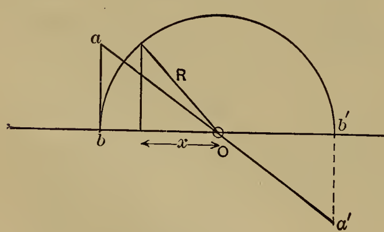


FIG. 371.

Furthermore, a line from the extremities of these verticals to the center of the crank-pin will pass through the point of zero acceleration and will be a straight line. In Fig. 371 the line aa' will therefore give for any point x the value of the acceleration and of the retarding effect for the corresponding crank-angle.

In these, however, the velocity is in feet per second, which can of course be computed from the revolutions per minute of the engine, since the crank-pin velocity V in terms of the revolutions per minute N will be

$$V = \frac{2\pi RN}{60}.$$

Since the pressure P in the horse-power formula is given in pounds per square inch, and is so given all in gauges and problems, it is convenient to reduce both P and W to terms of the square inch of piston area. If P and W be divided by A the area of the piston and the quotient be respectively $\frac{P}{A} = p$ and $\frac{W}{A} = a$, then

$$p = \frac{2^2\pi^2R^2N^2}{60^2gR} = 0.0003\ wRN^2.$$

Since the velocity is in feet per second, R is in feet in the above. The quantity w in massive engines may rise as high as 6 pounds; in high-speed light engines it may fall to 2 pounds. If w be called 3 as an average figure, then

$$p = 0.0001\ RN^2$$

or

$$= \frac{RN^2}{10{,}000}.$$

If the mean piston velocity be used instead of the uniform crank velocity, the latter is greater than the former in the relation

piston velocity : crank velocity :: diameter : semi-circumference :: 2 : π,

whence

crank velocity = piston velocity $\times$ 1.57.

261. Net Impelling Effort upon the Crank-pin. It is now apparent that the full forward pressure of the steam in the cylinder may not reach the crank-pin to produce rotation when it would appear to do so, and that later there may be more effort than that due to the pressure in the cylinder. It is the difference between the steam-pressure and the acceleration pressure which actually presses upon the pin. To show this graphically let the steam pressure effort be constant during the piston stroke. In Fig. 372 let the piston traverse be the base-line SE; the pressure of the steam on the piston be uniform and per square inch of piston area equal to PS. The cylinder work is proportional to the area $SERP$. Let the acceleration pressure per square inch be computed and laid off as a quantity SA acting to reduce the steam pressure effort at the beginning of the stroke and to add to it at the end. Then if $PD = SA$ be laid off downward from PR, the difference will be the net crank-pin effort at each point of the traverse, and the work area will be $SECD$, shown shaded in the diagram. There is lack of full forward effort at the beginning and excess at the end. The fly-wheel will store the excess RCO (less friction) and give it out to help the deficiency POD at the next stroke.

It is an added advantage to the expansive working of steam to be discussed in a latter chapter that it helps to equalize the periodic excess and deficiency due to inertia and acceleration. If the steam effort per square inch be not assumed constant but to have a variation

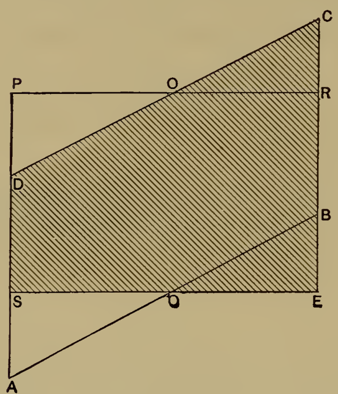


Fig. 372.

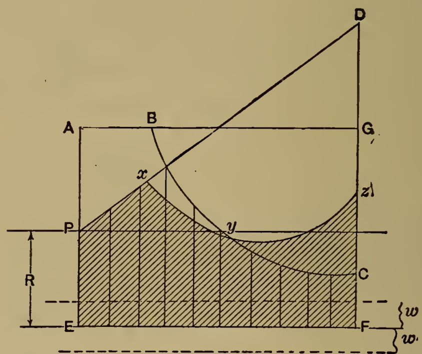


Fig. 373.

such as suggested by the curve ABC in Fig. 373 and the work area below it, and the acceleration pressure be AP laid off downward as before, then the net effort on the crank-pin will be the shaded area below $Pxyz$, much nearer a uniform mean. If the crank-pin resistance considered uniform be laid on upon the base line EF increased above its actual value multiplying it by the factor $\frac{\pi}{2}$ to reduce its longer travel to that of the piston, the excess of net effort over mean resistance R becomes apparent. If the engine mechanism moves vertically instead of horizontally, the weight w of the reciprocating parts becomes a help and a hindrance alternately upon the crank-pin. If w then be reduced to pounds per square inch of piston and laid off above the base-line EF and used additively for one stroke and subtractively for the other, the net crank-pin effort is determined still more exactly.

262. Relief of Crank-pin Stress by Cushioning by Compression of Exhaust or Live Steam. It would appear from Figs. 372 and 373 that the duty of arresting the moving mass of the reciprocating parts places a heavy stress on the crank-pin at the end of the stroke. This is particularly the case where terminal steam-pressures are high and masses or velocities are great. It would be apparent from Figs. 345 and 346 that if a mass of steam-gas be entrapped on the end towards which the piston is moving so as to be gradually compressed in the decreasing volume, such compressed gas could be made to act like an elastic spring, opposing to the piston a force of gradually increasing intensity to arrest its motion and transfer the stress of such arrest to the cylinder cover and take it off the crank-pin. Such entrapping is effected best by closing the outlet or exhaust passage before the end of the stroke of the piston, just enough before so that the rise of pressure due to the diminishing volume shall result in a pressure equal to that of the incoming steam when the inlet from the pipe shall be opened for the next stroke. If the back-pressure when the outlet is to be closed be called p_b and the initial or inlet pressure be called p_i, and the clearance volume be designated by c and the volume to prevail when imprisonment begins be called v, then since the cross section is constant, these volumes are proportional to the fractions of the length of the stroke in each case. By the law of Mariotte for compression of gases,

$$p_b(v + c) = p_i c, \text{ whence } v = \frac{p_i}{p_b} c.$$

The quantity c is a certain percentage of the cylinder volume and therefore of the stroke 2 R; it may be called 2 Rf. The volume actually filled with steam is $v + c$ when the exhaust is closed and compression begins. By substituting the value of $v + c$ will be determined. If the compression be adiabatic the volume will have an exponent greater than unity or will give to the equation the form

$$p_b(v + c)^{1.11} = p_i\,(2\;Rf)^{1.11},$$

which can be solved by logarithms when pi and p_b are known or assumed. Such compression pressures affect the value of the forward effort of the crank-pin in Figs. 372 and 373 and should be incorporated therein as affecting the forward steam-effort ordinates in the part of the curve xyz of Fig. 373 for example.

Another method to ease the crank-pin stress at or near the dead-center is to preopen the inlet valve to the cylinder before the piston reaches the dead-center and thus arrest the reciprocating parts by pressure from the boiler. This is not as economical of steam as the exhaust compression, since the latter makes it unnecessary to fill the clearance with new steam at each stroke, and thus lowers the steam and water rate. Compression also heats the cylinder walls by transmitting thereto some of the mechanical equivalent of absorbing the living forces in the form of heat. Compression is caused by exhaust-lap on the working edge of the valve, to be discussed later (paragraf 425). Preopening of admission is called steam-lead, and will also come up for reference under valve gearing (paragraf 428).

Excessive compression cannot occur with reasonable adjustments of the valve-gear and with steam-lead; the opening of the port releases excess pressure into the pipe. An ideal diagram of the type of Fig. 373 would be one in which the crank-pin pressure was constant during the greater part of the stroke, falling to zero at each end and rising to its full value early in the next stroke. It might resemble the dotted line curve in Fig. 374, in which this ideal is realized. In the broken line curve on the other hand the drop below *SE* to the point *C* shows that the com-

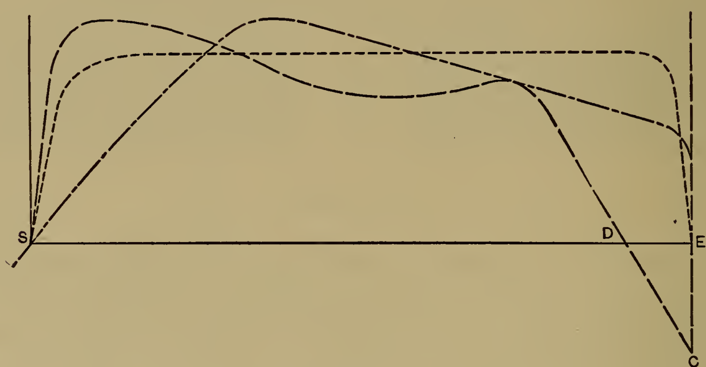


Fig. 374.

pression has been overdone. The net effect is not a forward push but a backward or negative pull in the stroke under consideration from *D* to *E*. There will be a knock in such an engine due to a reversal of the stresses in the mechanism as the piston passes *D*, if there is any play in the connections at the pins, although the reversal of effort at the dead-center would not occur. The engine at a higher speed would not knock, or with a later continuance of the admission or a higher terminal pressure. The dot and dash curve on the other hand is that due to too low initial net effort and too high terminal conditions. The engine should be run at higher boiler pressure for the mass of its parts at that speed, or if the pressure could not be raised, then run it at a lower speed.*

263. The Turning Leverage or Torque of the Crank. In certain cases, as in the locomotive and the motor vehicle, the cylinder must be able

* For further development of these features, consult Arthur Rigg, "Practical Treatise on the Steam Engine."

to start motion by the intensity of PA acting with the crank leverage, independent of the computed horse-power quantities which are created when this force moves through a space and creates a work. In a railroad stop on a curve or a grade, or with a motor vehicle in snow or sand or mud, the engine must be powerful enough in its turning moment to slip its wheels at the lowest speed of piston movement. The foregoing discussion shows that at speed the full steam effort does not reach the crank-pin at once nor tend to turn it uniformly. This at once suggests the two-cylinder engine with two cranks at 90° apart or quartering, so that one shall be delivering to the crank its maximum turning effort when the other is feeblest. This is done in locomotives universally and in many other conditions. This makes it of interest and service to lay out the moment of the turning effort to torque and the moment of the resistance to such effort around the center of the crank as a pole.

The diagram of Fig. 373 or Fig. 374 computed as for a connecting-rod of infinite length when the engine is at speed, gives the horizontal effort of pressure at the crank-pin brass of the connecting-rod for each angle. Then for a finite or actual connecting-rod the turning pressure at the crank will be found by two steps: first, to project that horizontal or parallel effort upon the oblique line of the connecting-rod; second, to decompose that projected effort into two components, one along the center line of the crank (which is without turning effect and only produces journal pressure) and the other perpendicular to the crank axis or normal to the instantaneous radius (Fig. 365) which is the force producing rotation. At any point therefore, as at P in Fig. 375, let the horizontal component be p_h which becomes p_c on the connecting-rod. Its normal component becomes p_n on decomposition, perpendicular to the radius. The turning moment is Rp_n, and varies from R when the force is so small as to be negligible up to a maximum when the component along the crank-center is zero. This kinematic turning moment may therefore be laid out by drawing radial lines from the crank-circle circumference outward, each having the value of p_n computed or determined for that crank position, giving a curve similar to that of Fig. 365. But the dynamic value of this moment is the product of the per square inch value of the piston effort multiplied by the area A of the piston, or p_nAR, in foot-pounds. The uniform (or varying) resistance moment can be similarly laid out. If uniform it will be a circle, and wherever the torque curve falls within it the fly-wheel is giving out energy stored when the torque curve was outside the resistance. If at any point the torque curve falls inside the crank-circle whose radius is R, this means that p_n had a minus value, and the crank and fly-wheel were driving the piston with a negative effort in the cylinder.

If instead of using a diagram for the torque of the crank it is preferred to get the value analytically for any particular crank-angle, as for starting, for example, it will be plain from Fig. 375 that $p_h = p_c \cos \alpha$

$$\text{and } nR \sin \alpha = R \sin \theta,$$

$$\text{whence } \sin \alpha = \frac{\sin \theta}{n}, \text{ in which } \theta \text{ and } n \text{ are known.}$$

Hence, since $p_n = p_c \cos \phi$

and $\phi = 90° - (\theta + a)$

$$p_n = \frac{p_h}{\cos a} \cos [90 - (\theta + a)] = \frac{p_h \sin (\theta + \alpha)}{\cos \alpha}.$$

If now the engine has two cranks at 90°, the second torque curve will have its origin or infinitesimal value for p_n at the points E and D, or its maximal values where the previous curve shows its least. If three cylinders are used and three cranks a still closer uniform value for a mean curve will result, which for any crank-angle will be the sum of the ordinates of all the crank efforts. Four cranks distribute the intensity of the effort on four crank-pins, but if kept at 90° intervals do not improve the turning moment over the results with two. Better balancing of masses, however, may be given, on which this diagram is silent. More than four cranks make multiples of the two and three arrangement, and while introducing no advantages not offered by two, three, or four, they make grave and almost prohibitory difficulties in construction and in cost which are not returned in advantages.

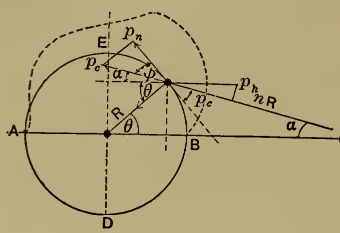


FIG. 375.

264. The Flinging or Slinging Stress in the Connecting-rod. It is evident from the foregoing that the longer the crank with a given rotative speed, the greater the crank-pin velocity and hence the greater the accelerative effort in passing from a velocity zero to a high crank-pin velocity in a given time. This makes it usual to have short crank-radii with high rotative speed and low rotative speeds with the large torque of the longer crank. This will be taken up again in discussing high-speed and low-speed engines, and with short stroke and long stroke (paragraf 280).

It will be obvious, however, that in passing the 90° and 270° points of the crank-pin travel, the mass of the connecting-rod has to be suddenly diverted from its tendency to fly centrifugally outward at that point, and be drawn back to the center line. This tends to shear and flex the rod and to make it vibrate under the slinging effect of its weight applied at its center of gravity. Very long rods are sometimes trussed in the plane of their flinging tendency, as in beam-engines, since to get stiffness by adding metal to the rod is to increase the difficulty to be cured. It would lead too far into the field of engine design to discuss this topic fully, and would therefore be aside from present purposes; but the deduction follows so easily from foregoing determinations that those interested may pursue it as follows:

The centrifugal force developed in a weight w pounds at the end of the crank R feet long is $F = 0.00034\, wRN^2$.

If the rod be supposed to be of steel, then for each inch of rod of cross section A its weight will be $0.28\, A$, since steel weighs 0.28 of a pound per cubic inch. Hence

$$F = 0.0000952\, ARN^2.$$

At the cross-head the centrifugal force is zero, and is a maximum at the crank-pin; it will vary for any small mass of the rod directly as its distance from the cross-head, or the diagram for the distribution of the bending and shearing load will be a triangle as in Fig. 376, and the force at the plane distant from the cross-head C the distance xd will be $\frac{Fx}{l}$ and the moment of the reaction at the cross-head pin $R_c \times l$ will be

$$R_cl = \frac{Fl}{2} \times \frac{l}{3},$$

whence

$$R_c = \frac{Fl}{6}.$$

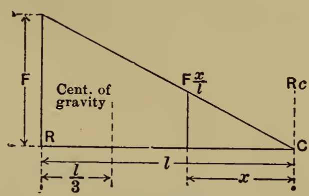


FIG. 376.

The shear at x from the cross-head pin will be

$$S = \frac{Fl}{6} - \left(\frac{Fx}{l} \times \frac{x}{2}\right),$$

which becomes zero when the parenthesis is zero, or when

$$\frac{Fl}{6} = \frac{Fx^2}{2\,l},$$

which is when

$$x = \frac{l}{\sqrt{3}}.$$

The bending moment is at a maximum when the shear is a minimum or zero; so that the bending moment M for a section distant from the cross-head pin C a distance x will be

$$M_x = R_cx - \left(\frac{Fx}{l} \times \frac{x}{2} \times \frac{x}{3}\right),$$

or for its maximum value by substituting for x at R_c

$$M_{\max} = \frac{Fl^2}{9\sqrt{3}} = \frac{Fl^2}{15.6}.$$

If the half-depth of the section be called y in inches and the radius of gyration about a horizontal axis through the center of gravity be called ρ, then the bending stress of f will be related to the bending moment

$$M_{\max} = \frac{Fl^2}{15.6} = \frac{fA\rho^2}{y},$$

whence by substituting for F its value above:

$$f = \frac{0.0000952\,RN^2l^2y}{15.6\,\rho^2} = \frac{0.0000061\,RN^2l^2y}{\rho^2} = \frac{RN^2l^2y}{180330\,\rho^2}.$$

The value for ρ^2 for a rectangular section is $\rho^2 = \frac{h^2}{12}$ when the depth of the rod is h. If the rod be of rectangular section milled on the sides to give it an I section, the area is the exterior enveloping rectangle less the removed part; or

$$\rho^2 = \frac{BH^3 - bh^3}{12\,(BH - bh)}.$$

The advantage of the I section is the getting of the maximum for the radius of gyration by putting the material necessary for strength as far as possible from the neutral axis; and the stress is lessened because ρ is in the denominator. The factor for the area of the cross- section which appeared both in the numerator and denominator has disappeared. Since the bending is a minimum as the cróss-head is reached, the connecting-rod can have the section reduced toward that end.

In the parallel rod which couples the drivers of locomotives and which always remains parallel to itself, the maximum bending moment is at the middle of the rod, and will be

$$M'_{\max} = \frac{Fl^2}{8} = \frac{fA\rho^2}{y},$$

whence

$$f = \frac{0.000012\, RN^2l^2y}{\rho^2} = \frac{RN^2l^2y}{83333\rho^2}.$$

This is about twice that of the connecting-rod which revolves at one end only. Since the bending is a maximum at the middle, it is possible to remove superfluous metal from near the two ends, which explains the appearance of bellying, which was quite usual until the I section supplanted the rectangular section.

265. The Accelerating Effort with Short Connecting-rods. The necessary angularity of the connecting-rod when it is short or when $l = nR$ has a finite or small value, for n alters the acceleration values deduced hitherto for the acceleration effort, or the pressure per square inch on the piston to accelerate the latter and attached weights. The piston is not in mid-stroke when the crank is at 90° (paragraph 357) and the line of piston accelerations is not straight (Fig. 371) but a curve.

Repeating Fig. 365 with necessary modifications in Fig. 377, in which O is the instantaneous center for the finite and infinite rod, but the radius is OC for one and OX for the other, it will be apparent that for the shorter rod

crank-pin velocity : cross-head velocity :: $OP : OC$,

while for the infinite rod, parallel to itself,

crank-pin velocity : cross-head velocity :: $OP : OX$,

whence velocity of CH infinite rod : velocity CH finite rod :: $OX : OC$.

At the inner dead-center

$$OX : OC :: PX : CS :: l : l + R,$$

which may be written

$$\frac{CS}{PX} = 1 + \frac{R}{l}.$$

At the inner dead-center these relations would be

$$\frac{C'S}{PX'} = 1 - \frac{R}{l}.$$

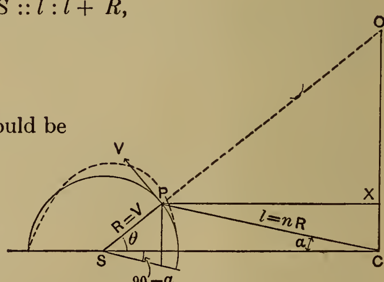


Fig. 377.

The maximum values of the accelerating pressure at the two ends of the stroke instead of being

$$p = 0.00034\, wRN^2$$

and equal to each other in opposite sense, become for the inner end

$$-\,p' = 0.00034\, wRN^2\left(1 + \frac{1}{n}\right) = ce$$

(Fig. 378), or greater than p; and for the outer end,

$$+ p' = 0.00034\, wRN^2 \left(1 - \frac{1}{n}\right) = bd$$

(Fig. 378), or less than p, when the connecting-rod is n cranks in length and n is finite and small. Then intermediate points are found from the curve of velocities in Fig. 365 or 377. The point of zero acceleration will be where the tangent to the velocity curve is zero, and a line from the crank position corresponding and of a length l will cut the center line at the position (f) of the cross-head. Two other points will be those where the acceleration is the same for both the long and short

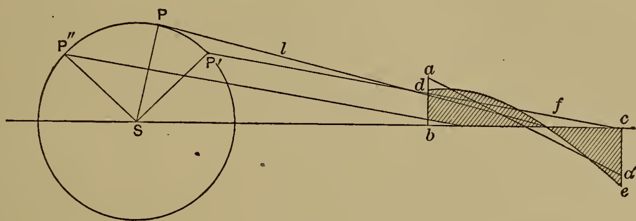


Fig. 378.

rod. With the crank center as a center draw arcs tangent to the short-rod curve, and where they touch erect perpendiculars to reach the crank-pin circle. These will give the 45° and 135° points of the long-rod curve and can thus be located on the line aa' of Fig. 371. The curve will resemble Fig. 378.

To reach the values analytically by computation and without the diagram, let the cross-head velocity be v and the crank-pin velocity V in Fig. 377. Then

$$\frac{v}{V} = \frac{\sin(\theta + \alpha)}{\sin(90 - \alpha)},$$

and

$$v = V\left(\frac{\sin\theta\cos\alpha + \cos\theta\sin\alpha}{\cos\alpha}\right).$$

When α is small as it usually is, its cosine is 1, whence

$$v = V(\sin\theta + \cos\theta\sin\alpha).$$

The diagram shows also that

$$\frac{l}{R} = n = \frac{\sin\theta}{\sin\alpha} \text{ or } \sin\alpha = \frac{\sin\theta}{n},$$

whence

$$v = V\left(\sin\theta + \frac{\cos\theta\sin\theta}{n}\right)$$

$$= V\left(\sin\theta + \frac{\sin 2\theta}{2n}\right).$$

The acceleration of the cross-head is the differential of its velocity in the time dt; or $f_a = \frac{dv}{dt}$ Substituting the crank path $Rd\theta$ for the space, and multiplying the space by dt for the time to get the velocity,

$$f_a = \frac{Vdvdt}{Rd\theta dt} = \frac{Vdv}{Rd\theta},$$

which when substituted for v above gives

$$fa = \left(\frac{V^2}{R} \cos\theta + \frac{\cos 2\theta}{n}\right),$$

whence the value of p for the acceleration pressure with a weight w becomes

$$p = \frac{wV^2}{gR}\left(\cos\theta + \frac{\cos 2\theta}{n}\right)$$

$$= 0.00034\, wRN^2\left(\cos\theta + \frac{\cos 2\theta}{n}\right).$$

When the crank-angle θ is zero or 180° at the two ends, this value is the same as that found by the graphical solution, as it should be. If there are weights of pump plungers or levers to be taken account of, these appear in the value for w, account being taken of the fact that levers may give different strokes and velocities to the parts which they drive from those of the piston.

266. Concluding Comment. It has been the object of the preceding paragrafs to serve as connecting links between the topics of analytical mechanics or theoretical kinematics and the practice of the engine builder, and to open the way to the more advanced study of engine design and the criticism of failures and accidents in the engine-room by the advanced student. The treatment must not be regarded as exhaustive or complete. The dynamics of the fly-wheel will be treated by itself under consideration of this part of the engine construction.

Before passing to an examination of constructive details next to follow, there are certain features of arrangement and certain elements determining the selection of engine-types which should first be examined.

CHAPTER XVI.

THE ENGINE (Continued.)

270. Introductory. The arrangement of engine mechanism suggested by Fig. 345 of the preceding chapter, and those derived from it by inverting the kinematic chain are not the only possible or expedient forms in which the steam-engine may present itself. There are other points of view, some of them matters of personal preference or alleged adaptability for special uses; others again with a valid basis of reason for their selection. This chapter proposes to discuss the following topics:

1. The Horizontal Engine, with cylinder-axis horizontal.
2. The Vertical Engine, with cylinder-axis vertical.
3. The Inclined Engine, with cylinder-axis inclined to the horizon.
4. The Horizontal Vertical Engine.
5. The Direct-acting Engine.
6. The Beam-engine.
7. The High-speed and Low-speed Engine.
8. The Single and Double-acting Engine.
9. Right-hand and Left-hand Engines.
10. Center and Sidecrank Engines.
11. Sundry Special Designs.

271. The Horizontal Engine. The horizontal arrangement of the cylinder-axis is by far the most usual position for factory or mill-engines and for power-plants where room or floor-space is not the governing condition (Fig. 345).

The advantages of the horizontal arrangement are: First, convenience of access from the ground-level to every point of the engine-mechanism. This is a convenience both in operation and in repair. Second, the weight of the engine is distributed over a large area for its support. This is of considerable moment where earth must be depended on to support the foundation, and becomes a critical condition of design for boat-engines, where the light draft imposed by shallow water compels a shallow and therefore flexible or deflecting hull-structure. This is a notable peculiarity of the practice of engine-design for the western rivers of America and for the shallow waters of the British Colonies. Third, the foundation itself does not require to be so massive to hold the engine still and to keep its frame from jar or vibration.

The first of these is usually considered the notable advantage of the horizontal engine.

The disadvantages of the horizontal engine are: First, that the action of gravity on all masses of the mechanism produces a friction which is absent in a vertical cylinder. The spring appliances of the piston which are intended to make it fit the bore steam-tight require to have strength sufficient to support the solid piston in the axis of the cylinder and prevent a wearing of the stuffing-box down on the lower side. The action of these springs increases the friction.

Secondly, due to either of these actions or to both, it is supposed that there is an excess of wear on the bottom elements of the cylinder which causes the bore to wear oval with the long axis vertical.

While the tendency of horizontal cylinders to wear oval is undeniable, it is a fair question whether this may not be caused rather by the springing of the guides and a flexing of the frame than by the action of gravity upon the piston; and in many stationary engines bolted to foundations the change of shape due to expansion by heat not infrequently so deranges the alinement of the engine as to cause the cylinder to wear unequally. The tendency to wear is also diminished by having the area of contact between the piston and its bore large enough so that for a given weight of piston the pressure per unit of area becomes so far reduced as to make wear inappreciable. A great gain is further secured by so selecting the material for the cylinder-casting that it may resist wear by abrasion. The difficulty from wearing is further diminished by the practice quite usual with heavy pistons of prolonging the piston-rod out through the back or head-end through a stuffing-box and with or without a back-end cross-head (Fig. 502). This not only supports the weight of piston, but serves to guide it effectively in the axis of the cylinder.

272. The Vertical Engine. The advantages belonging to the vertical arrangement of the cylinder-axis are the avoidance of cylinder-friction and unequal wear, which are the disadvantages of the horizontal engine. But of more moment than these is the diminished area in ground-plan which is entailed when the length of the engine is up and down. This condition has made the vertical engine practically universal for screw-propelled ships which are not primarily war vessels, and has given to this arrangement its wide distribution in crowded power-plants in cities where ground is costly (Fig. 380).

The objections to the vertical engine are: First, the effort on the crank-pin is greater when the weight of the mechanism is acting downwards with gravity than it would naturally be when the effort of the steam has to lift the same weight against gravity upon the up-stroke.

This must be counteracted, because otherwise the effort upon the pin, and therefore the speed, would be irregular. It can be done either by counterweighting the crank on the side opposite to the pin to which the reciprocating parts are attached, or by means of a steam-cylinder whose area shall be so calculated that the pressure of the steam shall just neutralize the weight to be overcome; or the distribution of steam

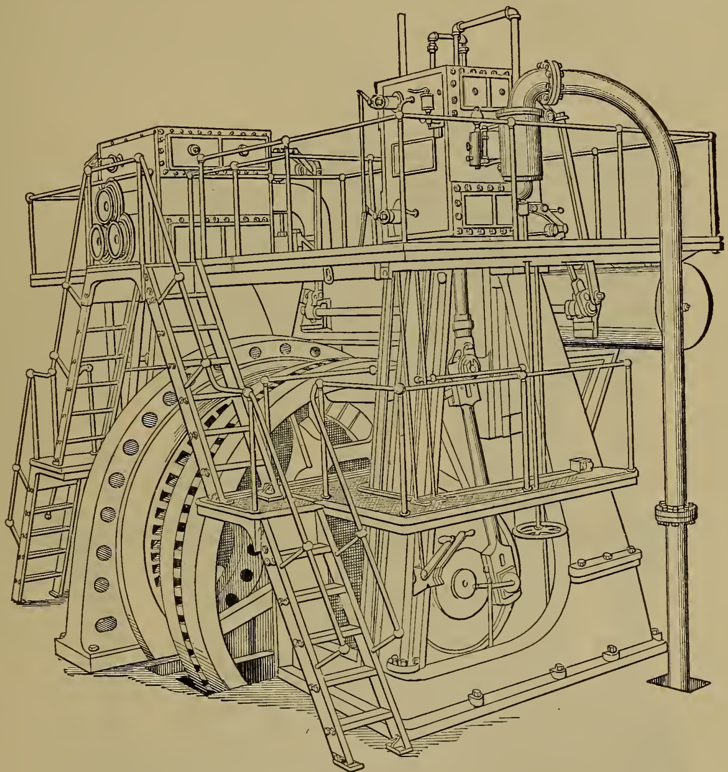


Fig. 380.

to the heavy end of the cylinder can be adjusted so as to develop more effort at that end than at the other. The second difficulty is that in a large engine the different parts of the mechanism will be upon different levels or stories, increasing the number of men required to handle or superintend it, and adding to its cost the price of platforms and railings (Figs. 380, 382). Third, the engine is not so completely and inflexibly secured to its foundation, and a deeper foundation is thereby required, or an unequal settling of such foundation will occur, if the concentrated load is not sufficiently widely distributed. Fourth, when the piston-rod protrudes from the bottom head of such vertical cylinder the combined effects of capillary action and gravity upon the condensation which takes place around the rod in the stuffing-box and upon the cover make it very troublesome to make the stuffing-box tight enough to prevent leakage of water.

273. Direct and Inverted Vertical Engines. The first steam-engines operated for pumping and with beams had the cylinder vertical and with the piston going out of the upper cylinder-head. Hence when the

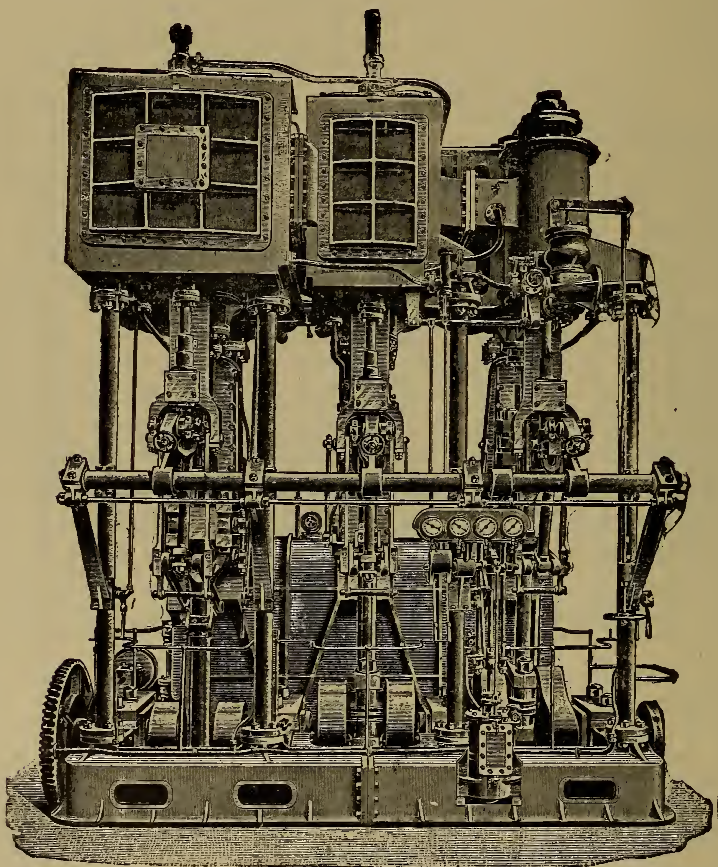


FIG. 381.

cylinder was put over the crank-shaft so that the piston-rod came out of the lower cylinder-head, such an engine was called an *inverted* vertical engine. This arrangement will be universal for engines in which the power is taken off from the crank-shaft as in electric generators, marine engines, rolling mills and the like and where the fly-wheel masses will be large and the reciprocating masses considerable. Scarcely any other arrangement than that of Fig. 380 would suggest itself except for special cases. The general advantage of this arrangement is that the moving reciprocating parts, which are those whose inertia or living force must be taken up by solid connection and for which the crank-pin must provide, are held to the ground through the crank-shaft directly secured to the foundation. The cylinder has nearly the same strain on it as the crank-pin, but these strains are transmitted through the elastic cushioning action of the steam. The cylinder is not a moving part but may be solidly bolted to the vertical members of the frame and anchored to the ground.

For certain uses the inverted vertical arrangement is specially adapted by reason of the location of the engine-shaft near the base. It is this condition which has made this the typical marine engine (Fig. 381); but it is also adapted for any function where the vertical arrangement is either preferred or necessary.

It will again be found the most convenient arrangement for water-works pumping-engines where the level of the water in the well or source from which the pump draws is either some distance below the general surface of the ground or is liable to wide fluctuations. Furthermore, where the pumping organ is a plunger of considerable weight or length it will naturally be arranged to travel vertically and the steam-cylinder which drives it will be inverted vertically above it (Fig. 382).

The support of the massive pump plunger is inconvenient if it traverses horizontally, since it is guided at one end only. Moreover, where floods may prevail drowning out the water-works it is convenient to have the steam-cylinders above the highest flood level. In pumping-engine practice, where the fly-wheel is only a regulator and its shaft transmits no power, the *direct* vertical type has been used (Fig. 383A, Appendix), with the piston-rod passing up from the top of the steam-cylinder to cross-head and connecting-rod above it. This is limited to rather quick-moving engines with fly-wheels of moderate weight, and is little known in America except for pumping, and with one exception has been restricted to factory practice at slow rotative speeds. In European shops such engines are often bolted to the wall, and are then called wall engines. Where a vertical engine has been desired and the inverted type disapproved, either the back-acting design has been chosen (Fig. 361) or use has been made of the advantage offered by some of the beam mechanisms.

274. Inclined Engines. A modification or derivative of the horizontal engine is met in marine practice with side-wheels, where the desired speed of the engine and the limit of speed set by the efficient action of

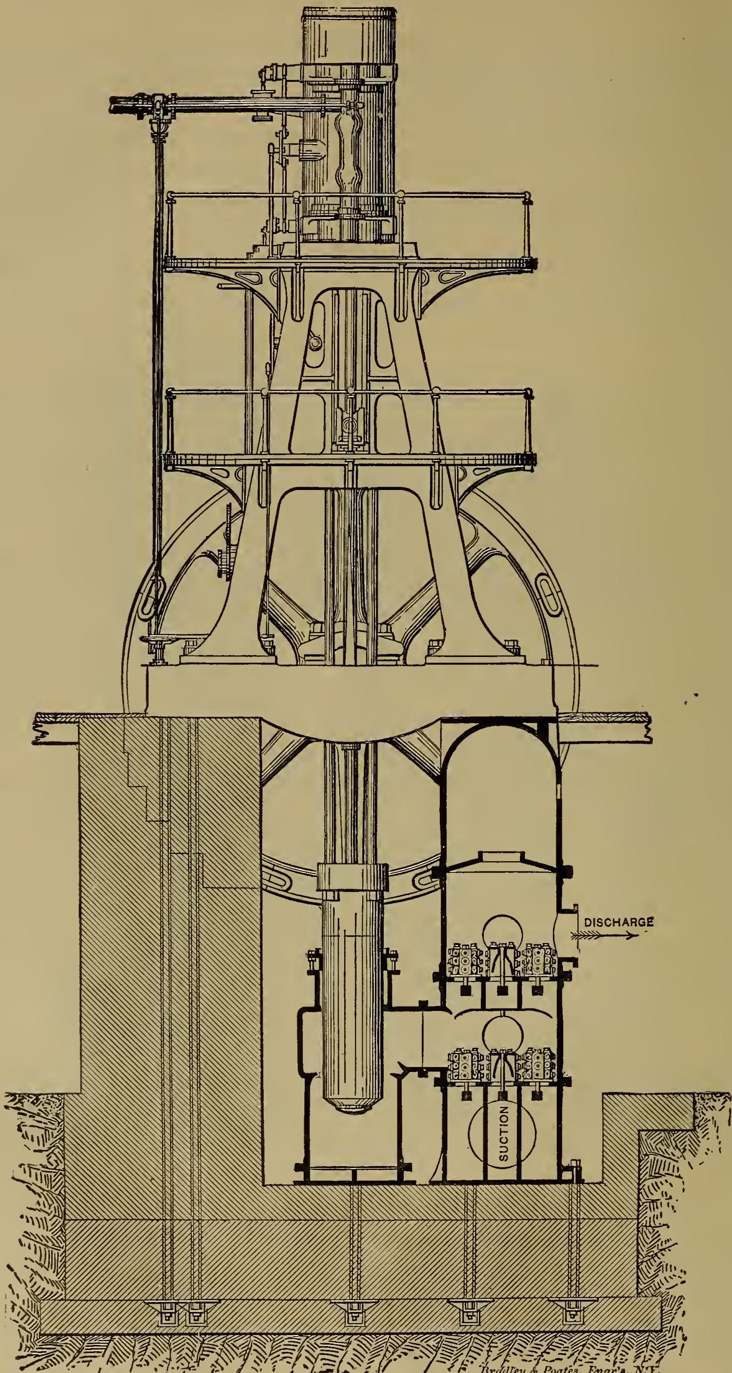


FIG. 382

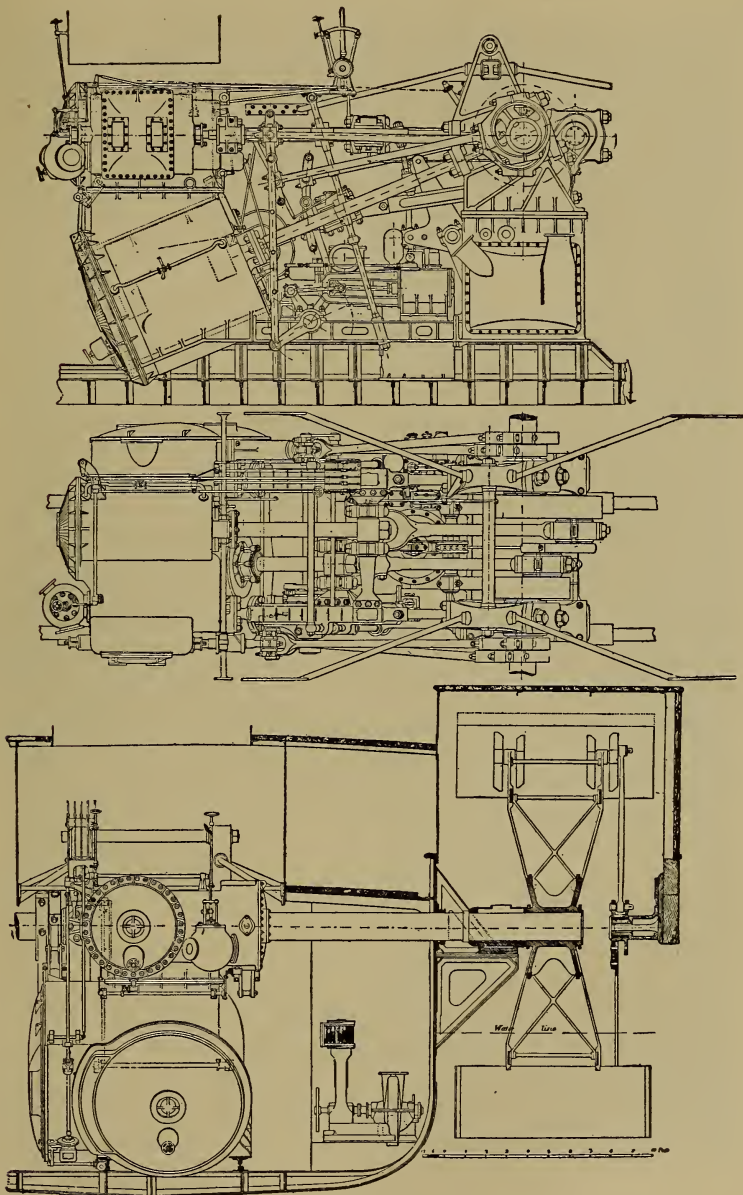


FIG. 384.

the paddle-wheel floats upon the water have fixed the diameter of the wheel, and hence with a desired immersion, the height of the wheel-shaft above the water. To keep the cylinders low down in the hull, and lower the center of gravity to diminish rolling, and to free the main deck from the room they would occupy, the cylinder axis is inclined to the horizon and the piston-rod points obliquely upward. This type has been much used in ferry-boat service, and in the sound and river-boat designs both of Europe and America. Fig. 384 is the engine of a

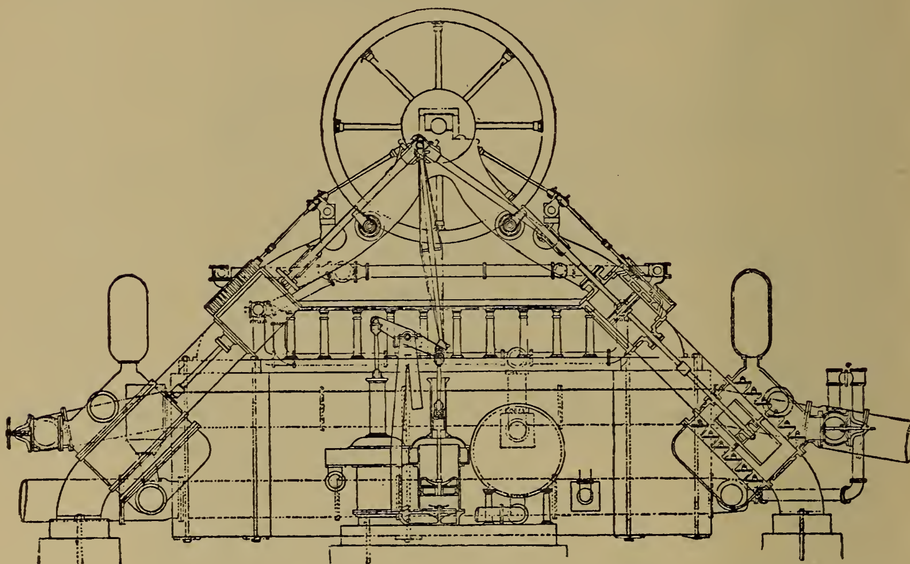


FIG. 385.

channel steamer, similar also to the general arrangement of the Hendrik Hudson engines of the Hudson River. The type is specially advantageous where a light draft and shallow depth of hull are factors. Fig. 385 is a water-works pumping-engine for a town of moderate size where four water cylinders are to have their motion controlled by one crank-shaft and fly-wheel. This gives some of the advantages of the horizontal-vertical design of the next paragraph as respects the quartering cranks and distribution of work, and in general while it does not offer the disadvantages of the two other types to the same degree, neither does it reap their respective advantages. Fig. 386 is the inclined double compound type in use on Long Island Sound. The next type is the more modern stationary solution. A few examples for power uses of self-contained engines of this style were once made in small sizes but are now no longer in the market.

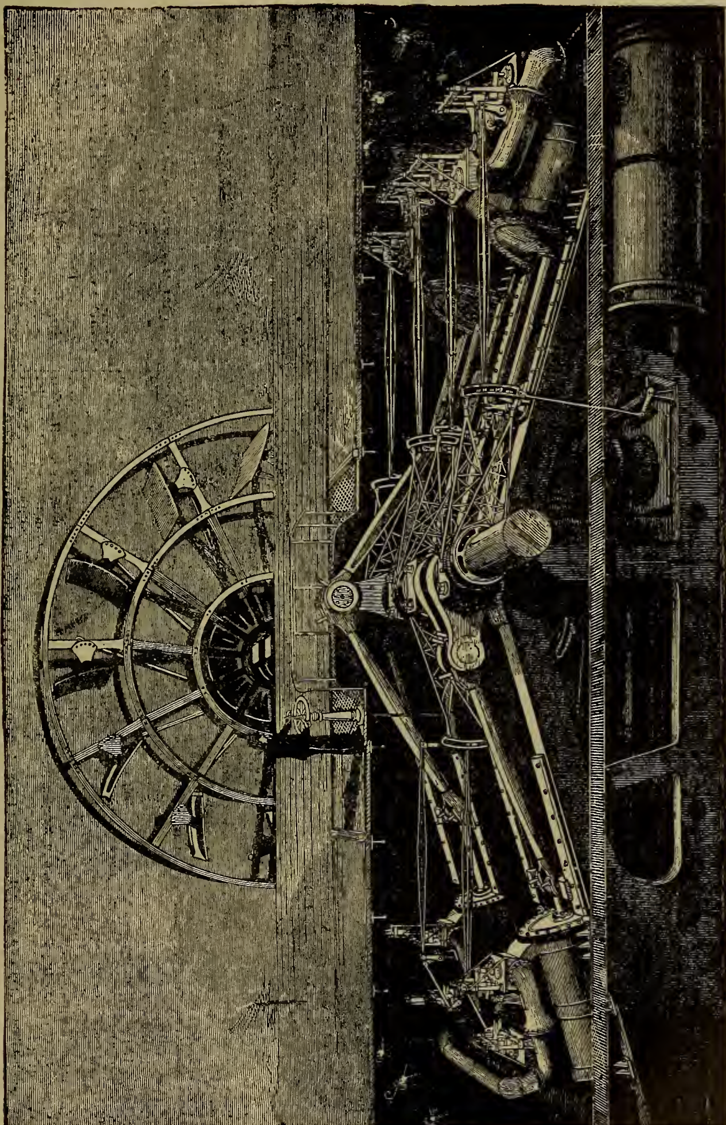


Fig. 386.

275. The Horizontal-vertical Engine. Recent designs of large capacity engines have embodied the advantages of the arrangement in Fig. 386. One cylinder is that of a vertical engine and a second is that of a horizontal engine. The two connecting-rods act upon a common crank-pin on the single crank and drive the shaft. Such an engine is called a horizontal-vertical engine. Its advantages are:

(*a*) Less area occupied in ground plan than if both cylinders were horizontal, and only a very little more than if both were vertical.

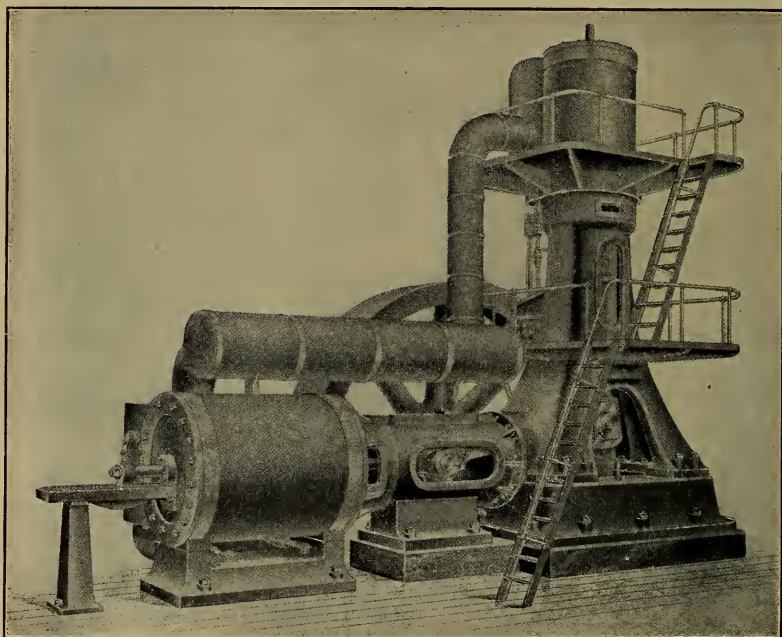


Fig. 387.

(*b*) When the engine is compound and massive, the heavy low-pressure cylinder is steadied close to the ground.

(*c*) The two cylinders act with the turning-moment of one at its maximum when the other is at its minimum, as the mechanisms are quartering, or 90° apart. The inertia of each engine gives out energy to the crank during the quadrant when the other engine masses are giving the least. Hence the shaft turns more steadily; or with equal steadiness the fly-wheel mass may be reduced. (Compare Figs. 371 to 375 and paragrafs 261, 263).

(*d*) This result could have been gotten by any arrangement of cranks at 90° apart, but this gets it without the cost of manufacture entailed by a double crank.

In Fig. 387 of a compound engine, the piping between the two cylinders enables a reheating chamber to be easily introduced to dry or superheat the steam for the low-pressure cylinder.

This arrangement is also well adapted for the compression of volatile fluids, in those designs where a liquid such as oil is used in the clear-

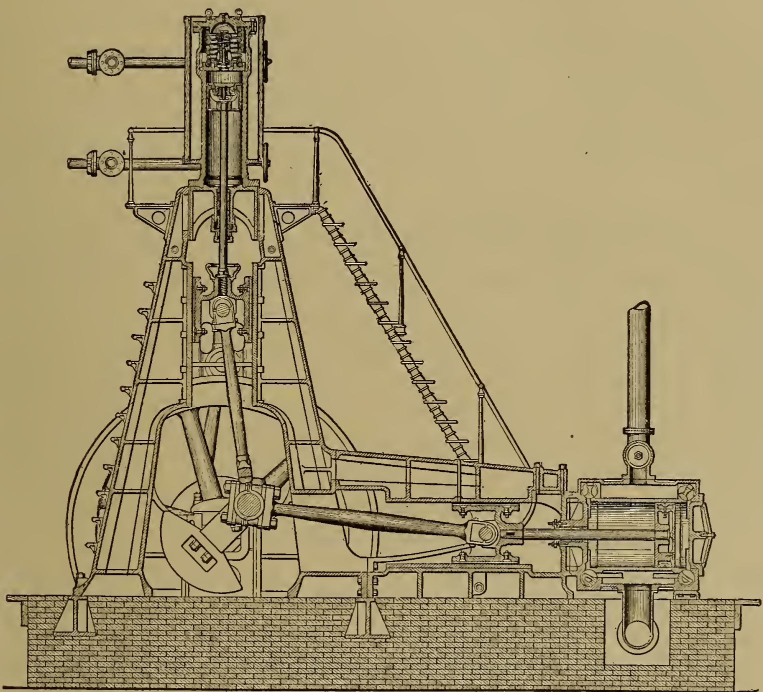


FIG. 388.

ance spaces of the compressor to reduce losses therein to a minimum (Fig. 388). A vertical compressor is therefore most serviceable, and by this type, which quarters the cranks of the driving and driven cylinders, the maximum crank-pin work is received from the driving piston at the time when the maximum compression resistance prevails. The pump-cylinder can be vertical downward or below the axis of the horizontal steam-cylinder to meet the requirements of a water-works pumping-plant with plungers and where a wide fluctuation of supply level is to be faced (paragraf 273); or again a beam may be introduced as in the designs of the next paragraf to change the direction of effort and resistance instead of using the crank and pin for this purpose.

276. The Direct-acting Engine. In the typical engine of Fig. 345, whether horizontal, vertical, or inclined as in the illustrations selected in the previous paragraphs, the power is applied as directly as possible to revolve the crank. Such engines are all called properly direct-acting engines to distinguish them from the back-acting types such as Figs. 360–363.

It is again used in steam-pump practice to denote the type of pump in which there is introduced no rotary mechanism with shaft and fly-wheel weight, the steam-piston acting directly upon the pumping-piston. In this latter sense it is synonymous with the term non-fly-wheel pump, and in the first case it is synonymous with forward-acting engine to distinguish it from back-acting. It is unfortunate that the same term should have so many different meanings, because in the sense in which it is desired to use it here the distinction is to be drawn between direct-acting engines like Fig. 345 and those in which the effort in the cylinder reaches the crank-pin indirectly through a beam.

277. Beam-engines. Since the earliest steam-engines were designed to operate the rods of mine-pumps, it was convenient to locate the cylinder at a little distance back from the mouth of the shaft, and to transmit the motion of the piston to the rod which went down the shaft or pit by means of a pivoted lever or beam. This beam was usually of wood pivoted at the center on convenient bearings, which were in most cases supported upon a masonry wall. When the function of the engine changed from that of pumping to the continuous driving of a revolving shaft the general design was only modified by connecting the outer end of the beam by a suitable connecting-rod to the crank-pin of the revolving shaft, and it seems probable that the term pitman often and properly attached to this organ of a beam-engine mechanism is a survival of the early mining term. The earliest steam-engines in America for marine use were beam-engines, and the preference of many skilled designers of side-wheel vessels for this type of mechanism shows that there are valid reasons for its popularity.

The advantages of the beam-engine mechanism are as follows:

First, the steam-cylinder can be vertical (paragrafs 271 to 275) in cases where the elevation of the cylinder for an inverted arrangement or a back-acting arrangement would be inconvenient. Such advantages as attach to the direct vertical cylinder are obtained.

Second, the cylinder and its weight are kept down low bolted to a bed or sole-plate, with advantages of low center of gravity and steadiness in vessels and elsewhere. The shaft and its bearings are also low down as in a horizontal or inverted vertical engine and with the same advantages.

Third, a long crank arm and a large turning or starting moment, and the consequent long traverse of the piston is possible and yet not too much space in ground-plan consumed. This is a great advantage in side-wheel practice and in pumping. In both these cases the number of revolutions or the number of reciprocations of the piston must be kept low, yet it is desirable that the piston-speed $L N$ (paragrafs 3 and 282) should be made high in order that the engine may be powerful. The beam-engine attains these results in a satisfactory way.

Fourth, the beam-engine secures a flexibility in the alinement of the cylinder-axis in its relation to the axis of the shaft. This is specially desirable for vessels of light draft whose hulls cannot be made absolutely rigid.

Fifth, for engines specifically designed for pumping, and particularly where several steam-cylinders and work-cylinders are features of the design, the beam construction furnishes convenient points of attachment for these various organs.

Sixth, where valid reasons demand that the steam-cylinders be vertical and the work-cylinders horizontal or inclined, while their motion shall be limited and controlled by a revolving crank and fly-wheel, the beam principle lends itself to attainment of this result.

Seventh, the swinging beam rotating round an axis parallel to the crank-pin gives less obliquity to the connecting-rod at many points in its alternate push and pull upon the crank than where the end farthest from the crank is guided in a straight line. The beam engine of Fig. 389 shows the four-bar chain of mechanism in its most apparent form.

It is probable that the union of high piston-speed with slow rotative speed, and the advantages which are secured by the combination of these two features in a flexible alinement, are the cogent reasons for the widespread acceptance of the beam mechanism.

278. Structure of Beam-engines. Fig. 389 illustrates a typical arrangement of an American river-boat beam-engine of the period 1850 to 1875. The type was practically fixed by the late Charles W. Copeland, and the sketch shows the beam supported on a frame of wood which has been variously called the gallows-frame or the A-frame, from its shape. It will be seen to have been well braced by wooden knees. The modern frame is of steel worked up into box-girder forms, securing thereby greater rigidity and less weight than was required in the wooden frames which they have displaced.

The beam itself in early practice was a cast-iron girder with the metal of the flanges so disposed as to secure the greatest strength and stiffness with the least weight. This gave to the beam the form of two semi-parabolas back to back, meeting over the center. The greater lightness attained by using wrought iron for tension elements of the beam caused the open-work or lozenge beam to be early adopted by American designers. Its first use is usually attributed to Stevens.

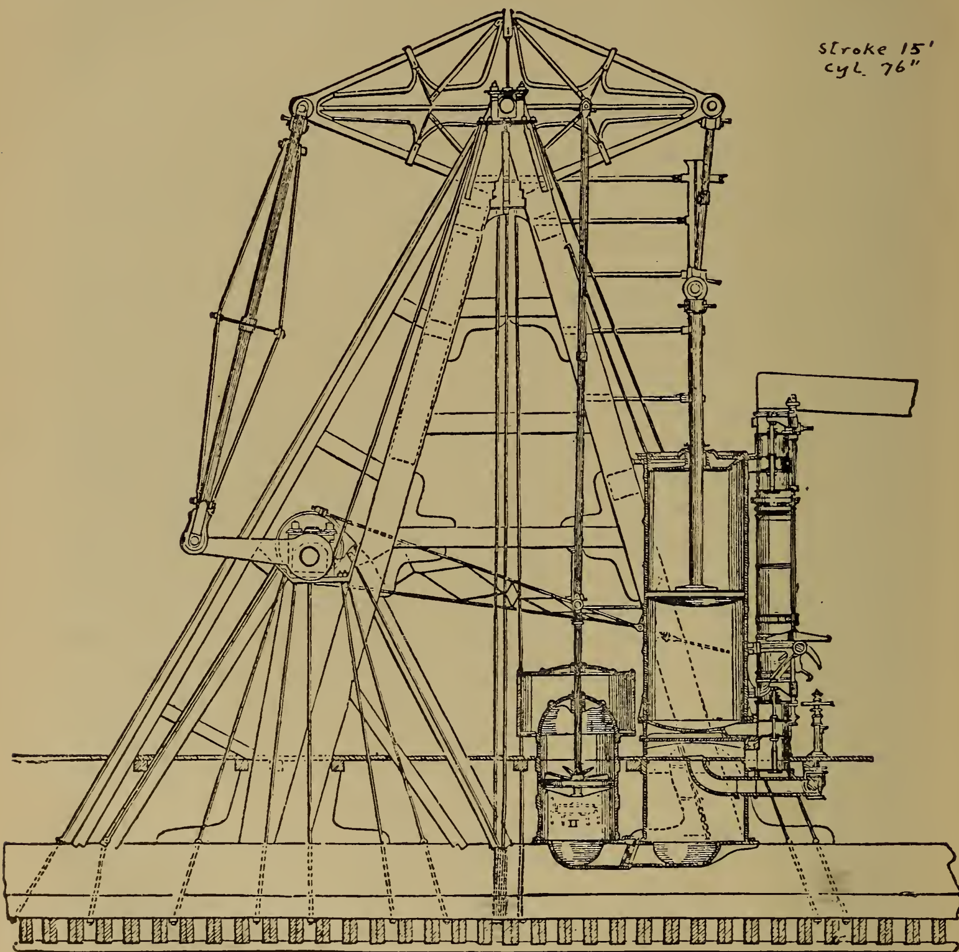


FIG. 389.

The solid wrought-iron forged diamond or lozenge transmits the alternating push and pull of the piston by an alternate tension in its upper and lower halves and the cast-iron center keeps the beam in shape and is exposed to compression only.

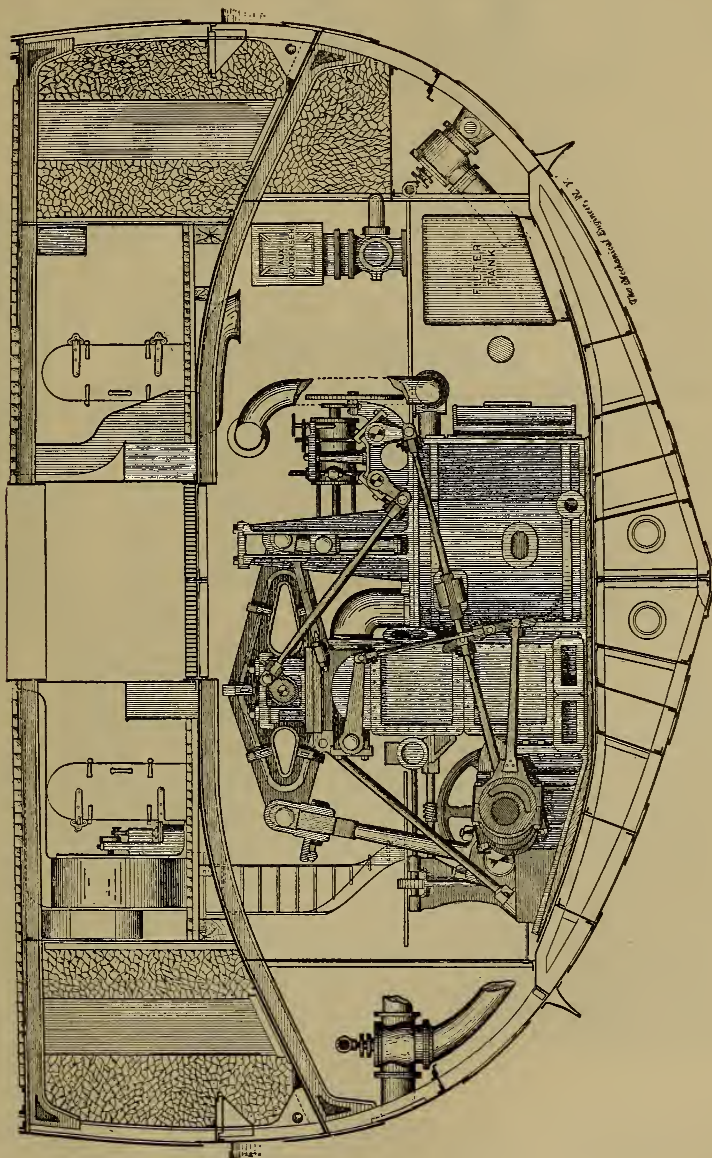


Fig. 390.

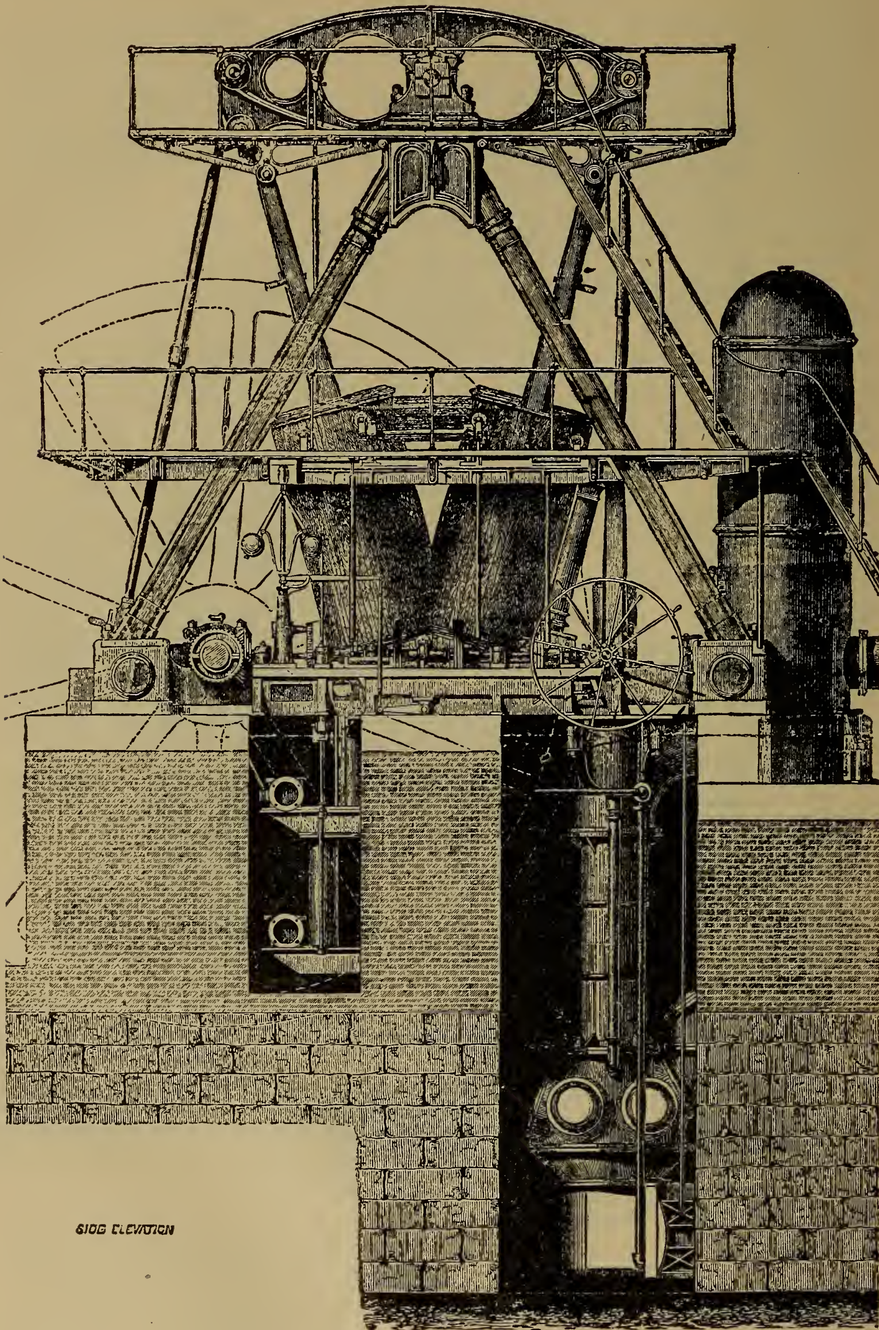


Fig. 391.

The cross-head at the upper end of the piston-rod either is guided in a straight-line path or is steadied by a linkage or parallel motion. The linkage is less usual. Two short connecting-rods connect the cross-head to the beam, one on each side of the latter to cause a symmetrical

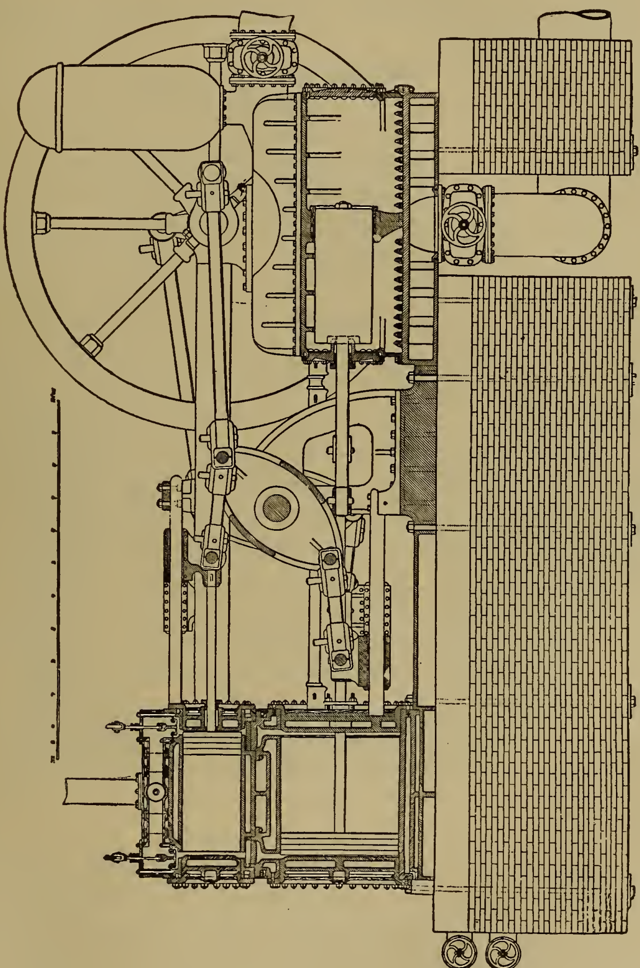


FIG. 392.

application of the force. Great care is necessary in the practical handling of these short connecting-rods as they wear at the bearing surfaces; since if they are permitted to become of unequal length a serious cross-strain is brought upon the cross-head, and a twisting strain upon the beam.

At the outer or crank end of the beam depends the long connecting-rod or pitman. It requires to be long when the crank is long, and it is therefore usual to brace and truss it with light steel tension-rods and king-posts, so that in its amplitude of swing its own mass should not have a tendency to make it fling and flex.

Fig. 390 shows the form which the beam-engine may be made to take for war-ship conditions and where a short stroke and rapid revolution of the screw-shaft running lengthwise of the vessel are the conditions to be met. The vessel is a twin-screw cruiser, and the engines are arranged right and left athwart ship. There is a space between the two shafts for the vertical cylinders and the beam frame, and yet the whole engine mechanism is below the water-line. This latter is the principal object sought in the design, as otherwise the inverted vertical cylinder would have been advisable. Fig. 391 shows the convenience of the beam mechanism when pump-plungers and connecting-rods are to be provided for as well as the steam-cylinder connections. While the inclined-cylinder design permits the straight descent of the plunger-rods, it can readily be seen that a successful design could as easily have been made by having the cylinders vertical and attached to a longer beam, leaving space nearer the center for the attachment of the plunger-rods, and one on each side could have been used. It is an advantage to give a long stroke to the steam-cylinder to secure high-piston-speed, but the pump plungers must move slowly to allow easy flow of water and unaccelerated through the valves.

In Figs. 392 to 395 the use of the beam mechanism for the engine for economical pumping of large masses of water is presented. In Fig. 392 the engine is horizontal and is twinned, with one compound engine complete on each side of the fly-wheel plane, and with a vertical pair of small beams, one for each engine. The beam serves to connect the high and low-pressure cylinders to each other, and to secure for both the regulating effect of the fly-wheel. The pumping work does not go through the beam as in the previous examples.

In Fig. 393 the inverted vertical type of Fig. 383 reappears, with its advantages respecting the pumps, but the fly-wheel shaft and bearings are removed to one side, and the beam takes a tri-angular or three-bearing construction in addition to the main bearing or beam center. In Fig. 394 the same general assembly type is preserved, but the beam loses even a resemblance to any conventional form.

Fig. 395 reverts to the horizontal type for pumping, and introduces the half-beam, or that where the main bearing is not between the points of application of the power and resistance, but the beam is a lever of the third order and not of the first. The fly-wheel being a regulator only, the compression center casting is omitted above the attachment of the piston-rod link and a triangle of tension rods operates the main connecting rod. In Fig. 396A in the Appendix for a single or twin-screw monitor of low-head room and to keep all below the water-line, the beam becomes a rock-shaft, with its main bearings in the sole-plate. The trunk-piston connecting-rod drives the short arm in the line of the

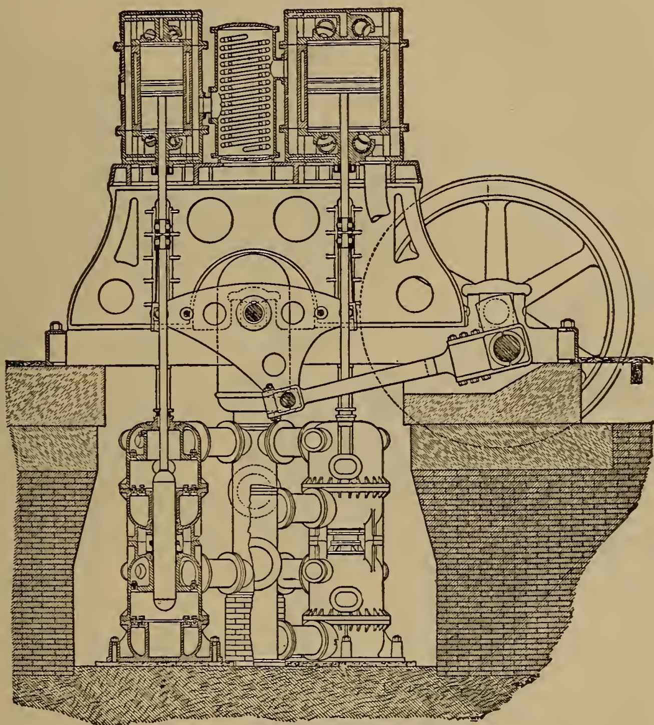


FIG. 393.

axis of the cylinder nearest to the observer; and the longer arm is far enough behind the plane of the cylinder axis to let the connecting-rod to the crank clear the cylinder at the back.

279. Objections to the Beam-engine. The Side-lever Type. Objections to the beam engine are:

(1) The power is applied indirectly to the crank-pin when it goes through the beam. There are frictions at the beam-center and at the multiplied joints.

(2) These latter require special care, since if the beam is kept single to keep it light, then the connecting-rods must be forked at one end to

get hold of pins on each side of it. If the beam is doubled, it is heavy and costly. If the rods are forked, it is difficult to keep the effort going symmetrically through both arms of the fork.

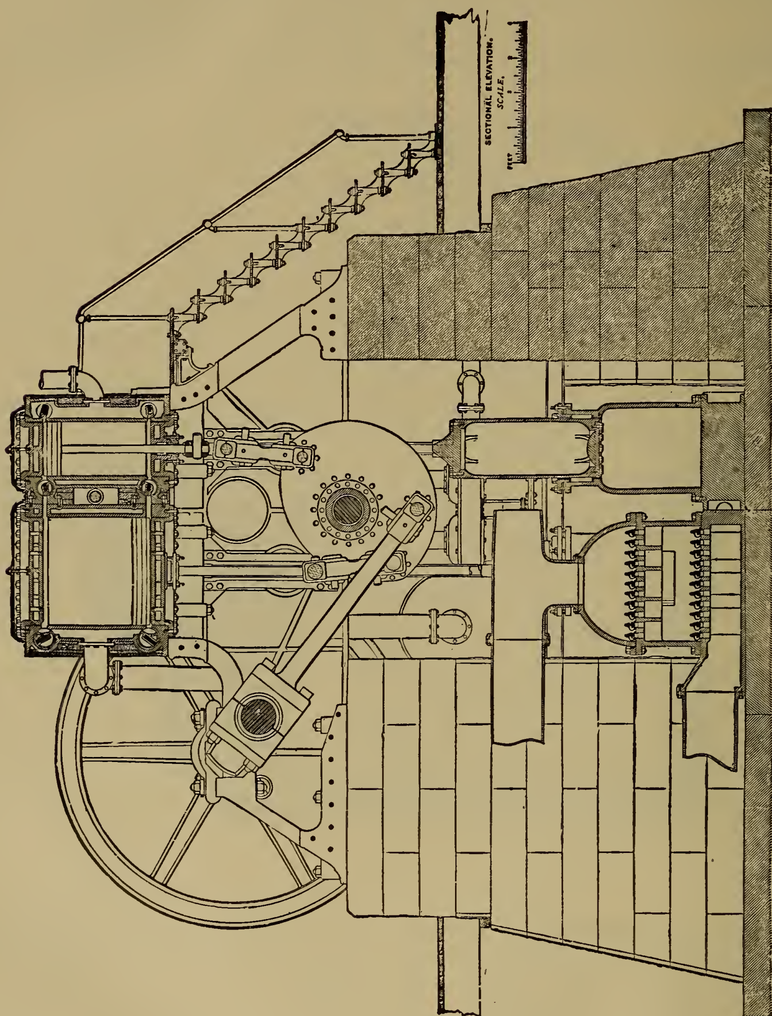


FIG. 394.

(3) The masses of the beam and its extra connecting-rod or rods have to be accelerated and stopped if the engine is to go fast at all.

(4) Extra joints to keep in order and to lubricate.

(5) In marine practice the inconvenient weight of the high frame and the beam at its top, so far above the center of gravity of the hull and its loading.

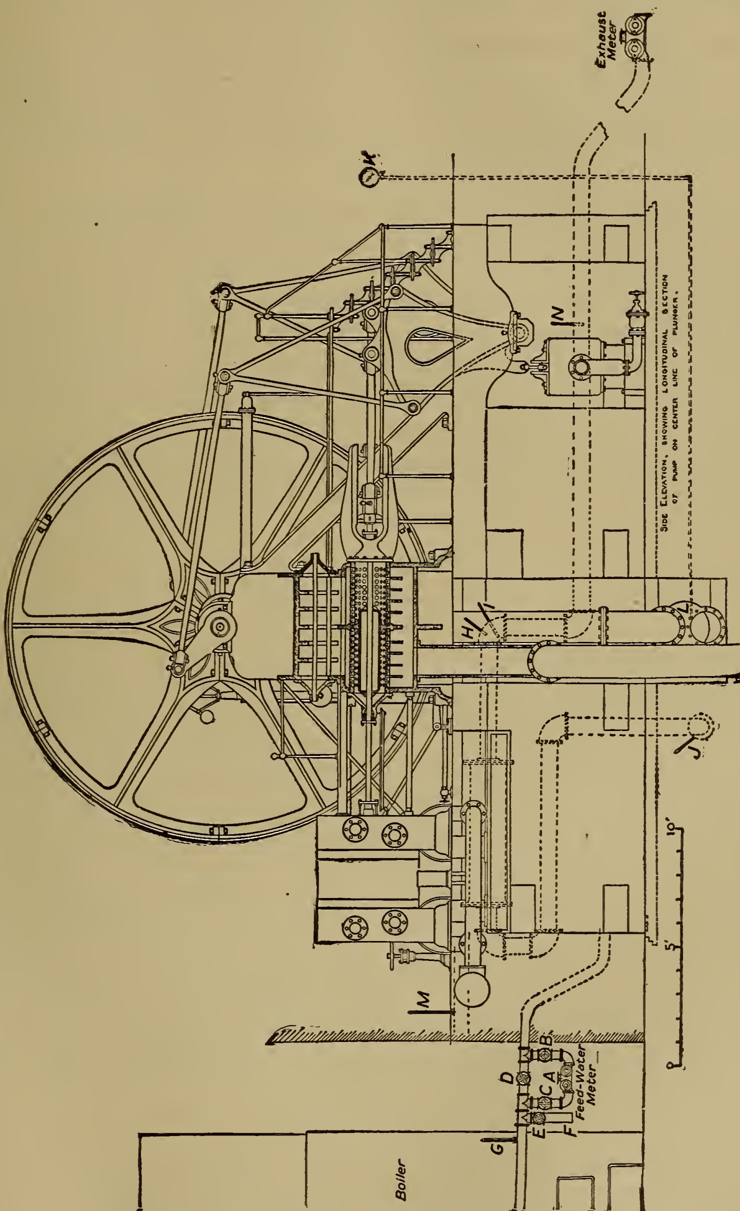

Exhaust Meter
K
N
Side Elevation, showing Longitudinal Section of Pump on Center Line of Plunger.
H
I
L
J
10'
5'
M
D
B
C
A
E
F
G
Feed-Water Meter
Boiler

Fig. 395.

(6) The exposure in warships of a vulnerable part above the armor, whose destruction is a fatal disablement.

The beam-engine has been a favorite for shallow draft conditions where the hull could yet be made stiff enough to hold up the concentrated weight of the engine and its frame. It never has had a vogue for paddle-wheel practice in Europe, and has disappeared from transatlantic and warship practice with the paddle-wheel.

The warship and ocean difficulties gave rise at an early date to the adoption of the back-acting principle to beam-engines with a double beam pivoted below on each side of the frame. From this double beam or side lever the connecting-rod or pitman rose to the level of the main shaft above the beam. This was the type known as the side-lever engine, and was in very general use up to the time when the introduction of the propelling screw displaced the side wheel for ocean service (Fig. 397A in the Appendix).

280. The High-speed and the Low-speed Engine. It will be apparent from the formula discussed in paragraf 3 that the work capacity or power of the piston type of steam engine in the form

$$\text{H.P.} = \frac{PA \times LN}{33{,}000}$$

can be made to vary with the value of the factors in the numerator; but if certain conventions are observed as to usual limits of these values either absolutely or relatively to each other, certain types of engine will result from these assumptions. For example, the piston-traverse is rarely less than the diameter of the piston, and rarely or never more than twice such diameter. The maximum pressure is fixed as a rule by the safe boiler-pressure permitted by the material of the boiler (paragraf 24 to 26) and its shape. Hence the mean working-pressure has also an accepted or usual value. The value of N is apt to be within certain limits by reason of the inconvenient losses in acceleration (paragraf 260), if it is made too large. Hence when N is large for reasons connected with the work which the engine is to do, then the weight of the reciprocating parts should be small; but this weight will be greater with large values of A and with a long crank arm $\left(\frac{L}{2}\right)$ since the connecting-rod length will vary as the crank-arm of which it is a multiple. Two types of engine result, of the same horsepower and the same value of P, according as N or the number of traverses or reciprocations is large or small. The one class with a high value for the reciprocations or revolutions per minute will be called high-speed engines;

the other, with a low value for N, will be called low rotative speed, or low-speed engines. A high-speed engine is one which makes a large number of revolutions per minute. The consequences of this are:

1. The engine has a small cylinder-volume because it fills that volume frequently each minute.

2. The small cylinder-volume both in length and diameter means an engine light in weight.

3. A short length of cylinder means a small crank-arm, a short connecting-rod, and an engine short in length. These three conditions are the same as to say that to increase N diminishes both weight and bulk and acceleration losses with a given power. P also has no weight. When the engine is to move by its own power as in self-propelling motors of any type, this is the important advantage of the system.

4. When the engine makes a high number of revolutions per minute each revolution is made in a fraction of a second, and consequently a variation of either effort or of resistance is more promptly met, and is less noticeable as compared with the mean effort or resistance of any given minute.

5. The regulating mechanism partaking of the rapid rotative motion produces its effect to equalize effort and resistance in a less interval of time than with the slower-moving types.

High-speed engines are those which make over 200 to 250 revolutions per minute with engines of two feet stroke or over, up to the 1000 to 2000 revolutions of small engines for motor vehicles or small boats. They are used in locomotive, motor vehicle, electric light and marine launch practice, in rolling-mills and street railway power houses.

281. The Low-speed Engine. In pumping water or air, in marine paddle-wheel practice, or where gearing is to be used in transmission, the number of revolutions or of traverses should not be high. With P fixed in the formula for horsepower, and N kept small, the values of A and L or the cylinder-volume will grow larger and an engine of greater bulk results, with the disadvantages of this type.

But it will be clear in inspection that if L be made large, or the engine is a long-stroke engine, the product of the two factors $L \times N$ may be the same as in the preceding case, and the engine may have its piston-effort pass through as many feet per minute in this latter case as in the faster revolution type. A distinction must therefore be carefully made between the piston-speed in feet per minute and the rotative speed, since a locomotive of two-foot stroke at 240 revolutions is moving at the same piston-speed as an 8-foot stroke pumping-engine of 60 revolutions.

The high-piston speed-value $L N$ keeps the value for A down, with a given horsepower limit and a value for P given. Such engines will have a long stroke compared to their cylinder-diameter.

The advantages of the long-stroke slow rotative-speed type are the avoiding of the disadvantages belonging to the short-stroke high rotative-speed engines discussed in a previous paragraph. The disadvantages of the high rotative-speed types are:

1. The rapid alternating of admission and suppression of steam through the ports to the cylinder compel large port areas in the design of such engines.

2. The rapid motion of the piston compels a generous allowance at each end between the piston at its dead-centers at the heads of the cylinder.

These two conditions create a clearance-volume of the cylinder at each end which is filled many times a minute with steam which escapes at the exhaust without doing work. The clearance-volume will be the area of the piston multiplied by the allowance length, which is usually a small fraction of an inch. That length and volume will be a greater percentage of a short cylinder than the same length and volume will be in a long cylinder, and it is filled and emptied more frequently in the high-speed engine.

3. Where the stroke is short the surfaces traversed are traversed more frequently and therefore the wear per unit of surface will be greater. This holds true for wear at all rubbing-surfaces.

4. The concentration of friction and pressure upon small areas frequently in action upon each other and the high rubbing-speed at joints (paragrafs 257–261) compel very close attendance upon such engines, because heating and abrasive wear go on with great rapidity when once allowed to begin, from the very circumstances of the case. These two conditions increase the possibility of expense for maintenance and repairs for this class of engines.

5. These conditions of concentration compel lubrication of such engines to be copious to a degree which may be wasteful if safety from heating is to be assured.

6. The foregoing conditions compel a standard of workmanship in the matter of fitting, alinement, and provision for wear which make high-speed engines costly to build and successful only when very well made.

7. The most frequent reciprocations absorb energy in accelerating the masses so many times a minute (paragraf 260).

These objections to the high rotative-speed steam-engine are the considerations which point to either a moderate or low speed of rotation as that to be desired when circumstances permit.

282. Piston Speed Values in Feet per Minute. Since the product LN of the horsepower formula is made up of two factors, a very wide number of combinations is possible. When $L N$ is expressed in feet and their product is less than 200 feet per minute, the engine would be a very low-speed engine; from 200 to 400 feet is low; from 400 to 600 feet is medium speed; from 600 to 800 feet is moderately high; above 800 is high speed, and above 1000 is very high. Many forms of valve-gear preclude the use of high rotative speed, and such engines are best run at speeds not higher than 100 revolutions per minute. In general, when the engine turns less than 75 times in a minute it will be called a very slow-turning engine; 75 to 100 is slow; 150 to 250 is medium; 250 to 300 is fast, and above 350 is very fast for steam-engines. Locomotives usually exceed 300 revolutions per minute. Small motor-car engines and especially internal-combustion engines go much higher than this, 1000, 1200, or even 2000 turns; but attempts to run engines of considerable size faster than 400 revolutions per minute have not been altogether satisfactory.

283. Double and Single-acting Engines. Since the numerator of the fraction in the horsepower formula is made up of pounds, PA, moving through $L N$ feet per minute, it is obvious that N is the number of such traverses made under the power of the effort PA. If only one of such traverses in the two which occur in a complete revolution is made under power, then the engine is called a single-acting engine. If both strokes in and out are made under power the engine is called double-acting. The single-acting engine of a given cylinder-volume is only half as powerful as the double-acting of the same cylinder-volume; or two cylinders of the given volume must be used in the single-acting engine to give it the same power as that of the single-cylinder double-acting type.

Historically the first engines were single-acting, having their cylinders open to the atmosphere on one side. The steam entered at the closed end and displaced the piston, was then condensed, leaving a vacuum behind, and atmospheric pressure forced the piston back to its starting point. They were defensible for pumping purposes, and where poor workmanship and unskilful maintenance were to be expected, because the stresses were never reversed as in the double-acting type and pin-joints need not be so carefully made nor looked after. The Cornish is the only type of large engine single-acting, and this will not be met except for mining, where its slow stroke, governed by the flow of great masses of water and the inertia of massive shaft-rods, would justify the type. It has no fly-wheel nor crank.

284. The Cornish Engine. A form of pumping-engine, single-acting in type, was early applied at mines in Cornwall, England, and has had a considerable popularity for water-works uses and for deep-mine conditions — for the latter by reason of its convenient solution of the problem of massive pump-rods. The Cornish engine appears in two forms. The beam Cornish engine has a vertical cylinder from whose top the piston-rod passes to one end of a beam pivoted at its center (Fig. 402A), to whose other end (which in mine-pumps usually hangs over the mouth of the shaft) are attached the pump-rods. The Bull Cornish, from the name of its first adapter, has the piston-rods coming out of the bottom of the vertical cylinder and directly attached to the pump-rods. This compels the cylinder to be located over the mouth of the shaft or above the plungers. In some French designs of mine-pumps and in some water-works pumps in America, the beam has been below the cylinder. Where the pump-rods and plungers have not the necessary mass, there will be a weight on the pump end of the beam leverage to carry the pumps down and force the water up.

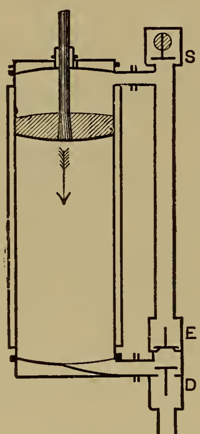


Fig. 400.

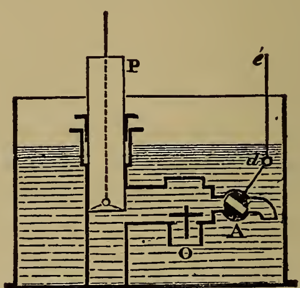


Fig. 401.

The Cornish principle is that of forcing the water up in the rising main or column pipe by the descent of the rods by their weight. The steam-cylinder lifts the rods and overcomes their resistance, but does no pumping, as a rule. The water controls the speed of the descent of the rods; the steam cushioning of the exhaust (paragraf 262), and the absorption of the inertia of the masses of the rods by the resistance (paragraf 260) controls the length of the stroke. There is no crank nor fly-wheel. Each double stroke is complete in itself.

285. Operation of the Cornish-engine Cylinder. The cylinder of the Cornish engine has three valves (Fig. 400):

1. The inlet-valve (S), admitting steam.
2. The exhaust-valve (D), allowing steam to escape, usually to the condenser.

3. The equilibrium-valve (E), opening and closing a pipe or passage between the upper and lower ends of the cylinder above and below the piston when at the upper or lower end.

The steam-valve and the exhaust-valve will be at opposite ends of the cylinder, the steam-valve at the bottom of the Bull engine and the top of a beam-engine. The cycle of operation will be as follows: The massive pump-rods being at the bottom of their motion and the piston at the corresponding end of its cylinder, the steam-valve will be opened and the exhaust-valve opened while the equilibrium-valve remains closed. The pressure of the steam overcoming the weight of the rods, the piston will move and the rods will be lifted. The admission of steam will cease at such a point in the stroke as is indicated by calculation and experiment, in order to impart to the rods sufficient living force to carry them to the end of their stroke. The exhaust-valve will close before the piston completes its stroke, so as to shut in between the piston and the head of the cylinder sufficient steam to form an elastic cushion strong enough to arrest the piston before it strikes the head.

Safety-catches or buffers were usually supplied in old engines to prevent this accident mechanically if the steam should fail to serve.

The massive pump-rods being now at the top of their stroke, the third or equilibrium-valve is opened, permitting the steam to pass through it on its passage to the other side of the piston so as to produce equilibrium of pressure on both sides. The weight of the rods causes them to descend, displacing the water to be pumped with a speed which is controlled by the valves of the pump, and by the extent of the opening of the equilibrium-valve. Both steam and exhaust-valve are closed during this equilibrium stroke, and the equilibrium-valve should itself be closed before the end of the stroke, so as to compress the steam between the piston and the head, cushioning the piston and filling all clearances with steam at inlet pressure. The cycle begins anew by the opening of the inlet and exhaust-valves for the next stroke.

286. Cataract of the Cornish Pumping-engine. It was a feature of the Cornish massive pump that it should be able to make few strokes to the minute with considerable pauses of rest and inactivity between them. The continuous rotation of a fly-wheel was, therefore, excluded, and if the pump-rods actuated the cylinder-valves directly, the pump would stall with all moving parts at rest and there would be no means to open the steam-valve for the next stroke. The working stroke must therefore store some energy in the form of a lifted weight whose subsequent descent, independent of the engine, shall set the valves for the next stroke while the engine proper is at rest. Such weight can be most easily controlled by resisting its descent by water flowing through a controlled orifice. Fig. 401 will show such a device in the form which is known as the cataract. The main-engine stroke raises the heavy plunger *P*. As it goes up water flows freely in through the valve *O*. The plunger released from the main engine starts downward by gravity, but *O* at once closes, and the water under the plunger can only get out slowly through the cock-valve *A*, partly open, if the flow is to be retarded and take a considerable time. The main-engine is at rest during the slow descent of *P*, which at the end of its fall either opens the valves directly or releases detents or catches whereby the valves are permitted to open by another force. The cataract principle, using a spring instead of the weight of the plunger, has been applied to operate the valve mechanism of horizontal direct-acting pumps. As such engines are double-acting, the graduating-valve will be in a connection which joins the two ends of the cataract-cylinder, and the plunger is replaced by the piston which fits that cylinder. It is obvious that in either case work is stored by the working-stroke of the main engine and given out as desired after the main engine is at rest.

Other solutions for the non-fly-wheel type of pumping-engine have been the auxiliary water-motor, having its valves set by the main engine, and whose work was the moving of the valves of the main engine. Such engine could not stall, for the main engine had its valves wide open until the motor reversed them independently of the stroke or motion of the large rods. (See also paragraf 149.)

287. Advantages and Disadvantages of the Cornish Pumping-engine. The primary advantage of the Cornish pump is that the motion of the water through the valves and pipes is made the controlling element. Large masses of water can only be accelerated as demanded by crank-motion at the expense of considerable work which is unprofitably expended. Second, the masses of the pump-rods serve as a reciprocating fly-wheel. Third, the single-acting principle of working enables the Cornish pump to work with much greater economy than less carefully designed pumping-engines belonging to its earlier period. The duty of the best grade of Cornish engine stated in the usual form has been about 100,000,000 pounds of water raised one foot high by the combustion of 100 pounds of coal. Fourth, its ability to work successfully with a very small number of strokes per minute.

The disadvantages of the Cornish pumping-engine are, first, being single-acting it is bulky for a given number of foot-pounds of work. The mean pressure in the cylinder cannot be high, because at the end of the stroke all living force of the reciprocating parts must have been given out. Second, having no crank to limit the stroke of the piston, there is the danger from overstroke either up or down. If from any cause the pump-barrel fails to fill with water, the massive rods descend unchecked and their living force under these circumstances will wreck the engine. Third, the bulk of the cylinder and the masses attached to the piston compel an expensive and massive foundation greatly in excess of that required by an engine of a different type to do the same work. Fourth, the intermittent action of the cylinder compels very careful provision to keep it warm between strokes, and condensation will be considerable in spite of all care.

The Cornish engine is now only in operation where it was early

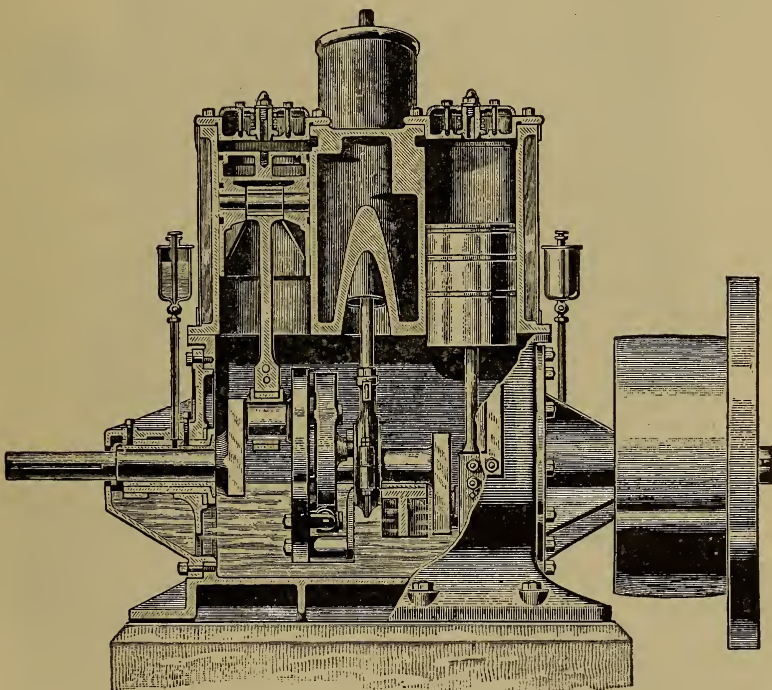


Fig. 403.

installed. In water-works such as in New York, Brooklyn and Philadelphia it will not be replaced when its work is done, nor built for extensions of service. The economy and convenience of new designs have taken away its significance. Fig. 402 in the Appendix is the Brooklyn engine, showing the older type with cast-iron beam, parallel motion, extra plunger-weight and buffers.

288. Single-acting Rotative Engines. The demand for an engine of high rotative speed for motor-car and power service which shall be able to be cheaply built with respect to fitting, alinement and wear has attracted engine-builders to the single-acting principle. In this type, and particularly with an inverted cylinder and trunk-engine mechanism with the energy of the steam acting with gravity downwards upon the upper side of the piston, it is brought about that the effort of the steam through the mechanism is in one direction only. Hence silent running is secured at high rotative speed because the strain on the crank-pin is

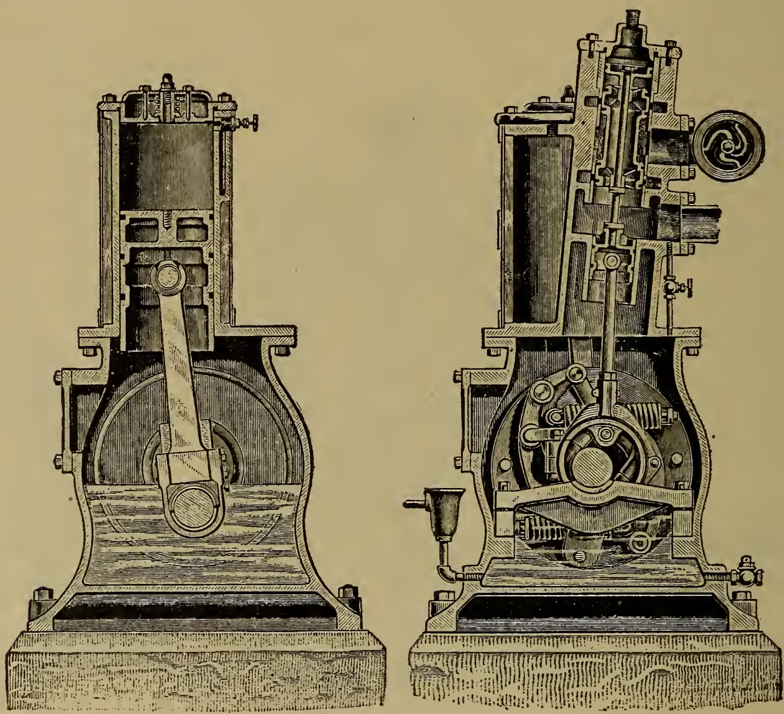


Fig. 404.

never reversed, which will be the occasion for knock or pound in a double-acting engine upon passing the centers, unless the adjustment and fitting are very perfect and the adjustment of the valve-mechanism just right. The danger of overheating bearings is lessened when the adjustments of fit are of less moment, and it furthermore becomes a matter of less risk to make use of high initial steam-pressure in the cylinders. Continuous action is secured by putting two cylinders to act upon the same crank-shaft. The two best-known single-acting engines of the rotative type are the Westinghouse and the Willans. Fig. 403 shows a longitudinal section and Fig. 404 a transverse section of the Westinghouse standard engine, and Fig. 405 a section through the Willans cylinders. The trunk-mechanism is clearly manifest in both designs, and the principle which they both represent of securing self-lubrication

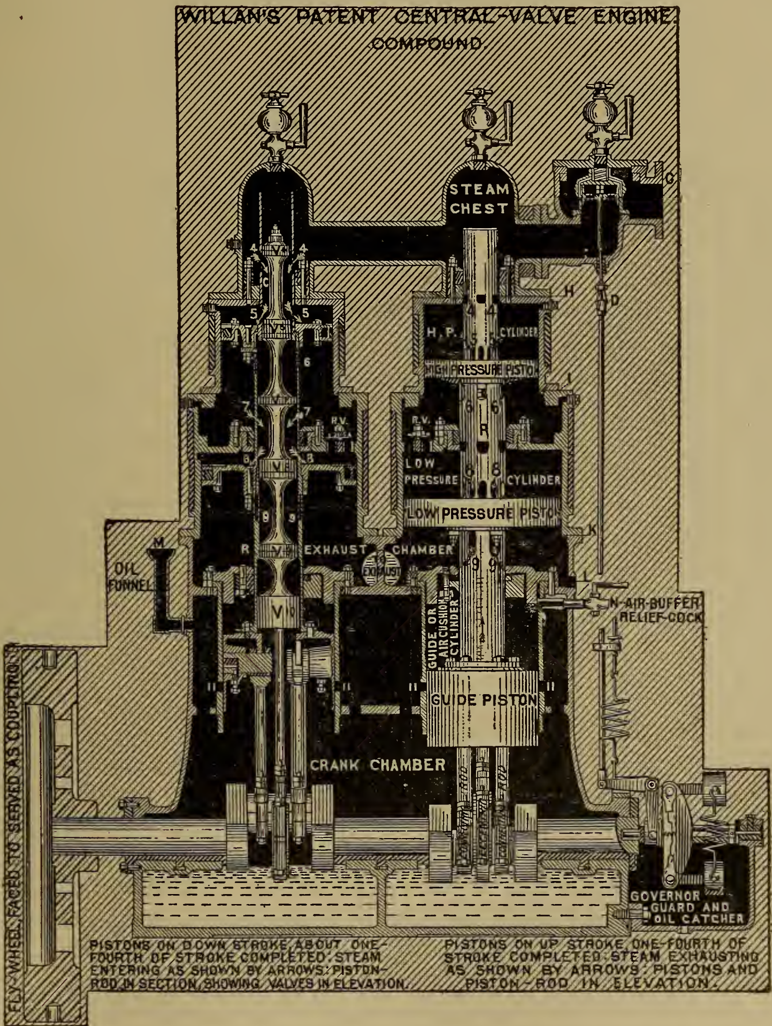


FIG. 405.

by having the crank-shaft revolve in a closed casing which is filled with water on the surface of which floats lubricating oil. The use of pistons of different diameters is very convenient in engines of this type, and will receive discussion in the sequel. These engines may have rotative speeds between 250 and 500 revolutions per minute without difficulty, and have been quite a little used where it was desirable to couple the armature of electric dynamos directly to the engine-shaft. The Willans engine-section shows the characteristic central valve within the hollow piston-rod, and Fig. 404 the possible offset of the piston to one side of the plane of the shaft, since the work is done through 180° only.

289. Right-hand and Left-hand Engines. In the typical engine mechanism of Fig. 345, the cylinder happens to be on the left hand as the observer faces it. It might just as correctly have been at the right hand. Hence two types originate; one will be called left-hand engines when the observer stands in the axis of the engine shaft in front of the crank and finds the cylinder on his left. The other or right-hand engine (Fig. 406) will be that in which from the same position in the line of the shaft axis the cylinder is at his right hand. The selection of one design or the other will be fixed as a rule by the location in the room which the engine is to occupy. The wall will be nearer the back of the engine than the front so as to keep the belt or generator wires as close thereto as possible, and give an obstructed passage from the larger areas of the center of the room to the mechanism. When such considerations are equal then the choice may be of personal preference or whim for the right-hand engine because some one is used to it. The right-hand side of a standard locomotive is always a right-hand engine and the left-hand side a left-hand engine or whenever the cylinder is in front of the engine. When the engine drives by a belt from the shaft the belt should be kept near the wall for safety and convenience. When the shaft to be driven is located behind the cylinder, the engine is said to belt back (Fig. 407). If the other location exists, and the belt from the fly-wheel or shaft pulley is in front of both engine shaft and cylinder the engine belts forward. Since the driving by a belt is always done by the bight of the belt which is in tension, it will be plain that the bottom side of the belt should do the work rather than the top, since any slack on the returning side increases the area of contact between the belt and pulley surface of both wheels in this case. When the top drives and the lower bight is slack, the belt leaves both pulleys sooner than in the other case. Hence when the engine belts back, the crank throws under; when it belts forward, the crank throws over (paragrafs 259, 265).

In the inverted vertical type (paragraf 273) there is no right or left type when the frame is symmetrical and the guides and cross-head permit the engine to turn equally well in either direction. Here the engine should turn clockwise as the observer faces it when the belt goes off to the right, and contra-clockwise when the belt goes to the

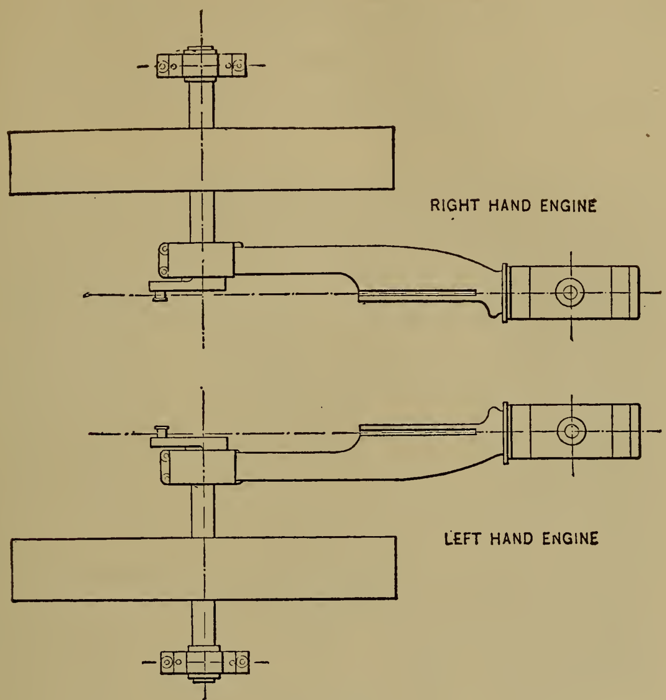


Fig. 406

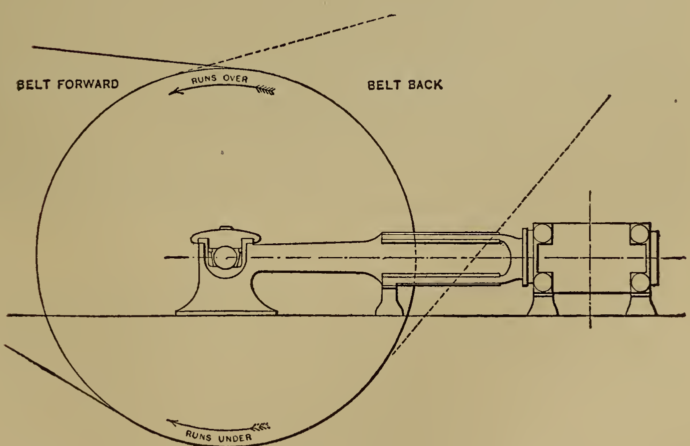


Fig. 407.

left. If, however, the cross-head is guided on one side of the frame only (Fig. 380) then a right-hand engine has the guides on the right side of the observer as he faces the crank and looks along the shaft. Such engines should turn contra-clockwise. A left-hand engine has the cross-head guides to the left of the vertical cylinder axis and in throwing over turns clockwise. In either case if the vertical cylinder were revolved around the crank-shaft until it became horizontal and rested on the ground with the cross-head right side up, it would be necessary to revolve the right hand to the right, when it would still be a right-hand horizontal engine, and the left-hand engine to the left, when it would become a left-hand horizontal engine. Both engines would be throwing over still.

The valve gear of the engine will always be driven from the shaft or the fly-wheel beyond the first or main bearing. This will place the steam-chest for the valve on the cylinder on the right or left of the cylinder according as the engine is right or left handed. The term is often used for a center-crank engine, symmetrical as respects the shaft, to describe the side on which the valve-chest is to an observer standing at the head end of the cylinder and looking towards the shaft.

290. Center and Side-Crank Engines. The diagram of Fig. 406 shows that in both the right-hand and left-hand engine the revolving crank is at the side of the line through the cylinder axis, and so also is the bearing of that shaft on the bed-plate of the engine. Such engines which have crank and bearing on one side only of the axis of the cylinder are called side-crank engines. If now the two cylinders of Fig. 406 be brought nearer until they melted into one, and the two crank-pins became one and in the line of the common cylinder axis, there would be two cranks, or a double crank with each half symmetrical with that axis, and two bearings on the bed-plate, one on each side of the crank in the center. This makes the center-crank engine. The side-crank type requires an outboard bearing beyond the fly-wheel and usually independent of the bed-plate: the center-crank has the fly-wheel usually doubled (Fig. 408) and the engine entirely self-contained. The shaft bearings must be long and ample since the weight of the wheels overhangs. The mechanism is not so accessible in center-crank engines and they will be limited to short-stroke high rotative-speed engines of small or low medium power, and for belt drives. The larger, more massive engine for generators will be of the side-crank type with outboard bearings. The term outboard bearing has been derived from marine practice with side wheels where the bearing for the shaft beyond the wheel or resistance was carried on guards outside of the hull proper.

In Fig. 408* the double cranks are counterweighted and the oiling system for the crank-pin is shown. The crank-case is inclosed. The center-crank type may be called right-hand or left-hand for convenience according as the valve-chest is at the right or left hand of the observer standing at the head end of the cylinder and looking toward the shaft.

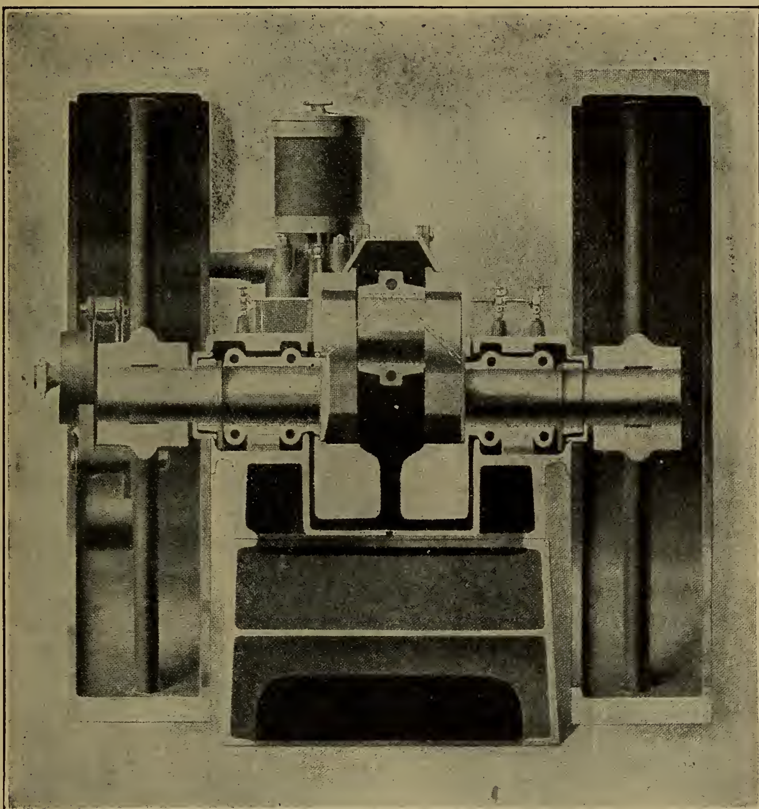


FIG. 408.

291. Sundry Mechanisms and Arrangements. The realization of the satisfactory transmission of motion and force by the crank and connecting-rod has rendered obsolete a number of special designs which have been advanced to improve upon the typical mechanism. James Watt had to avoid a patent in the crank-connecting-rod combination in his early rotative beam-engines and therefore used the sun-and-planet device. A toothed gear was bolted to the side of the connecting-rod

* Ames engines.

and meshed into another of the same or different diameter on the engine-shaft. The gears were kept in mesh by a link connecting their centers. When the connecting-rod was alternately pushed and pulled, the rigid

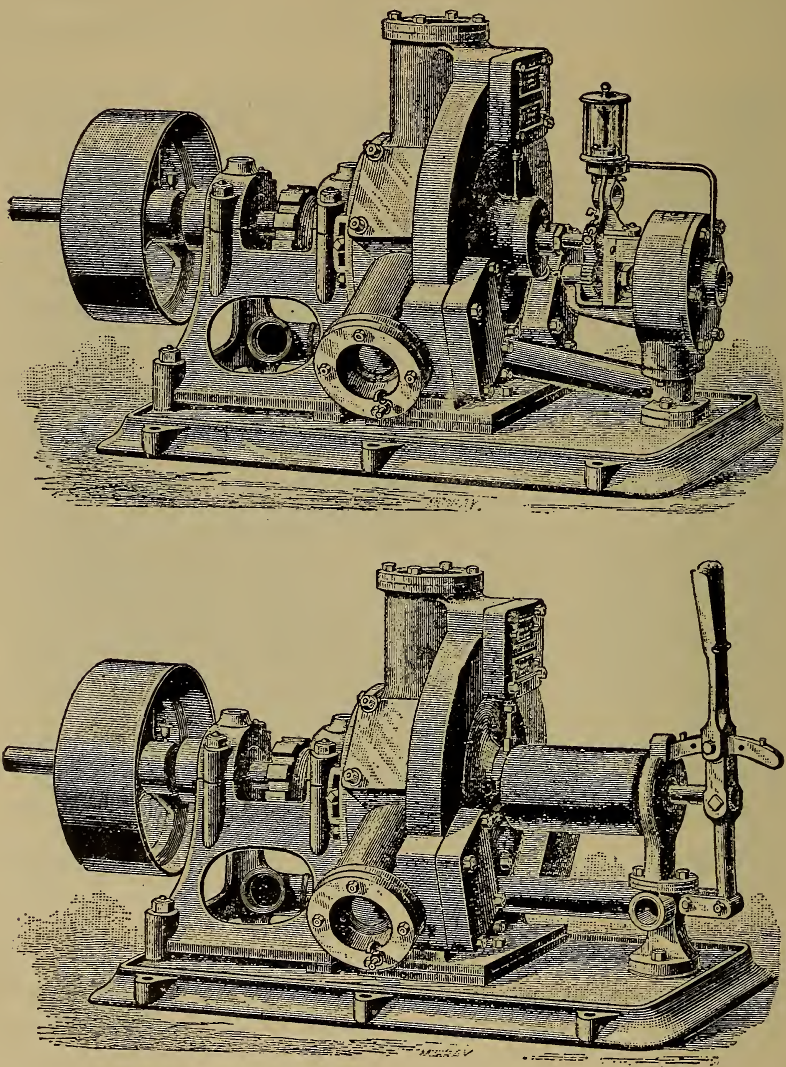


Fig. 409.

gear was compelled to roll around the one upon the shaft, and compelled the latter to revolve, since the other could not. If they were of the same diameter the shaft made two revolutions for a single complete revolution of the rigid gear on the connecting-rod. As soon as the patent expired the crank was at once introduced.

The multi-cylinder engine, designed either to get a required piston area without excessive diameter of cylinder, or to distribute the weights of the engine parts, or to secure more uniform turning moment or continuous expansion of the steam in more than one cylinder, or for other reasons, makes a considerable variety of designs departing from the simple fundamental type. Some of these have been referred to and illustrated and others will be introduced in the later treatment of the compound or continuous-expansion engine. The large gas-engine, which is often single-acting by preference, has presented a number of such types, such as the double-opposed cylinders and the cylinders tandem upon a single piston-rod. The inside-connected locomotive with the two cylinders between the frames compels a cranked axle on one pair of driving-wheels, similar to the multi-cylinder arrangement of the marine engine. In quite small engines or of short crank-arm and intended for special service in capstans and winches, the three-cylinder design of Fig. 409 eliminates the dead-center, since each crank is at 120 degrees from the other two, so that no care is required in stopping, and the rotation will start as soon as pressure is on. They can be handled from a distance therefore. For compactness these are usually single-acting trunk engines, and require little fly-wheel weight, but are limited in size and capacity. They make good engines for steam-steering of large vessels, in the reversing type.

The vibrating piston-engine of Capt. John Ericsson (Fig. 457A) for small launches; the square piston-engine (Fig. 698A); the Colt disk-engine (Fig. 699A) are all designs of interest at present mainly historical and will be briefly referred to in an appendix. Designers in Great Britain and Germany have also proposed mechanisms which have sought to attain special results, such as the Musgrave two-cylinder engine with triangular connecting-rod, and the swinging fly-wheel on a vibrating link for pump-service such as Stannah's. These have never become standard forms or been extensively reproduced.

292. Concluding Comment. The next step after study of these general arrangements of mechanism of engines would be the details of their construction and organs. Before doing this, however, it will be advisable to discuss the behavior of the steam as a motor force in the cylinder and the means to secure economy in its operation, in order that the effect of these influences upon the design of such elements may be made apparent and their significance in relation thereto.

CHAPTER XVII.

EXPANSIVE WORKING OF STEAM.

295. Introductory. The Indicator. From paragraf 3 and the preceding chapters it must have been apparent that the greatest value for the output from an engine in foot-pounds when the size and speed of the cylinder have been fixed is given from the formula

$$\text{H.P.} = \frac{PA \times LN}{33{,}000}$$

when P is the greatest. For when the engine is embodied in metal, A and L and 33,000 are not variable and the speed N has a fixed or determined value which must not be exceeded. How can the value of P be known or measured?

It is plain that it cannot exceed the reading of the pressure gauge upon the boiler (paragraf 171) as a maximum. It may easily be less than this, but how much and at what points in the stroke?

James Watt devised the steam-engine indicator to give to the eye and to record the cylinder-pressure on each face of the piston during its traverse forward and back. It consists of a piston of known area moving with the least possible friction in a little cylinder, whose under side is in communication through as short pipe-connections as possible with the end of the cylinder (Fig. 410). It is usual to make this connection into the clearance, but at such point that the flow of steam through the ports shall not affect the pressure actually prevailing where the indicator is connected. This pressure connected on the under side of the indicator piston would force it upwards in its cylinder, and this tendency is resisted by a spring carefully calibrated with respect to the area of the piston, so that it shall undergo certain definite deformation under certain definite pressure. It will be apparent then that the deformation of this spring will weigh the pressure, and if a tracing point or pencil be attached to the piston, it will draw a curve which will be the ends of ordinates corresponding to the pressure on the indicating piston in terms of the scale of the spring. If a motion of a paper at right angles to the piston motion be provided, a closed curve will be made, which will thus record the pressures in the cylinder at every point of the stroke, if the movement of the paper be produced by a linkage or mechanism driven from the piston by a positive reducing connection. It is usual to reel the paper of the diagram upon a barrel which is rotated through a part of a revolution by a reducing mechanism driven from the engine cross-head.

It will be apparent that with a known scale of spring in the indicator the mean height of the diagram which it traces will be the mean pressure upon the piston. The mean height can either be ascertained by finding the area of a diagram with a planimeter and dividing that area by the

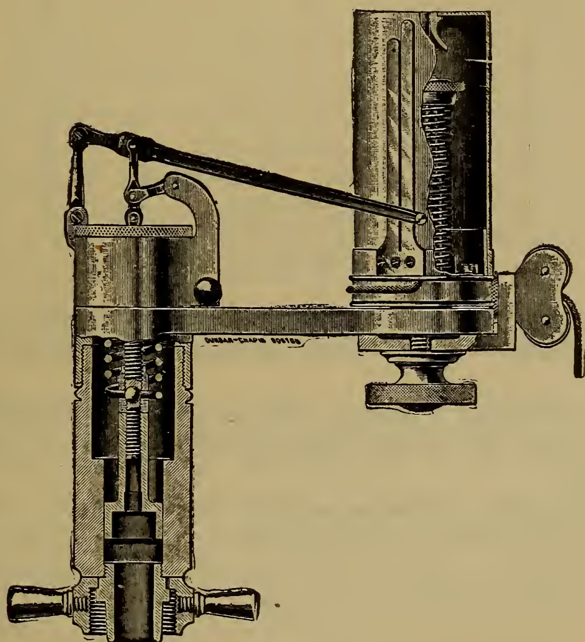


Fig. 410.

length of the diagram, or the mean height of the diagram can be observed by dividing the length of the diagram into equal parts and measuring the height in each segment, adding their aggregate together and dividing by the number of heights measured.

It will be further apparent that the lines of the diagram will indicate the satisfactory working of the distributing valves, or the reverse, by reason of the relation of actual pressures to those which ought to prevail, and furthermore the approximation of the curves of effort to those which theory indicates as desirable. The indicator is thus a check on the setting of the valves, sizes of ports, friction through pipes, resistance to free release of exhaust, excessive condensation, ill-adjusted expansion, and the like.

The errors of the indicator are those due to defective accuracy of springs, inertia in the moving parts, which causes them to move further than simply to balance the pressure, friction which prevents their moving as far as they ought to balance the pressure, and inaccuracy in

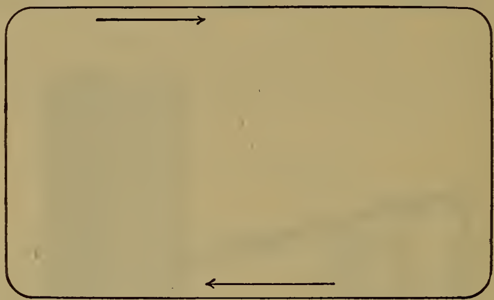


FIG. 411.

the reproduction on the diagram of the motion of the piston in its true relation by reason of defects in the mechanism used to give motion to the paper.

Fig. 410 illustrates a type of such steam-engine indicator. What will be the appearance of the diagram which it traces? It will have two general forms.

296. Non-expansive Working of Steam in the Cylinder. If the valve admitting steam from the boiler and steam-pipe remain open during the full length of the traverse of the piston from one dead-center to the other, the indicator piston should rise at the left hand of the diagram in Fig. 411 to the point represented by the boiler or pipe pressure on the scale of the spring resisting motion of the piston. Then as the piston moves to the right boiler steam flows in behind it and at that full pressure if the passages are adequate until the end of the stroke, at the right-hand end. Then the exhaust-valve should open, with a drop of pressure down to that prevailing in the exhaust-pipe, and under that pressure the indicator pencil should describe the bottom line towards the left, and complete the cycle when the admission-valve opens to let in steam for the next stroke. If the calibrated spring is correct and known, and A and L are known, then the factors of the formula for one stroke are all given

$$\text{H.P.} = \frac{P \times A \times L \times 1}{33{,}000},$$

and the horsepower is computed by direct measurement from the diagram.

Such an engine is said to work non-expansively, or to use its steam non-expansively. It works exactly as it would do if the pipe to the boiler were full of water and entered the boiler below the water line. Exactly the same work is done as would be done by an incompressible non-expanding mass of water forced into the cylinder by the steam pressure. The water would be at the same pressure at the end of the stroke as at the beginning; and to keep up pressure in the boiler a weight of water equal to that withdrawn from the boiler would have to be made into steam in the boiler. The non-expansive process in the cylinder is therefore a purely pressure process and is not a heat process or transfer of heat into work at all. Direct-acting or non-fly-wheel pumps and some steam elevators work non-expansively. The heat process is in the boiler. How efficient is such a process?

297. Efficiency in a Heat-engine. The efficiency of any machine or apparatus is the degree to which it utilizes the energy put into it. Its measure is therefore the ratio of the energy utilized, which is its output in work units to the input of energy; or

$$\text{Efficiency} = \frac{\text{Energy supplied} - \text{energy rejected}}{\text{Energy supplied}} = \frac{\text{Work done}}{\text{Energy supplied}}.$$

Now in the steam-cylinder working its steam non-expansively, a weight W of steam entered the cylinder and filled its volume at a pressure p_1 corresponding in the steam tables to a temperature T_1. If each pound of steam at T_1 can do 778 work units for each heat unit, then for a range from an assumed zero of temperature to T_1 the energy input will be $W \times T_1 \times 778$ foot-pounds. When the exhaust opened, the weight W left the cylinder at the same pressure and temperature that it went in. Hence the

$$\text{Efficiency} = \frac{(WT_1 778) - (WT_1 778)}{WT_1 778} = \frac{0}{WT_1 778},$$

or as a heat utilizer its efficiency is zero.

At the boiler the feed-water came in at T_2 and was raised by the heat of the fire from water at T_2 to steam at T_1. Since weights are the units used, the boiler raised W pounds of water to W pounds of steam, and increased the energy from the beginning of the process to the end by adding the heat aggregate necessary. Hence the fire input was $WT_1 778 - WT_2 778$ and the efficiency

$$E = \frac{T_1 - T_2}{T_1}.$$

But the heat or fuel necessary to heat this water was greater as W is greater per stroke or per minute or per hour. Hence this method of working, while giving the greatest work per stroke burns also the greatest weight of coal possible to utilize. Cannot this be improved upon?

The valve-gear of such an engine can be the simplest and cheapest. It need only open an admission of steam at one end of the cylinder and an exhaust valve at the other and hold these open during the

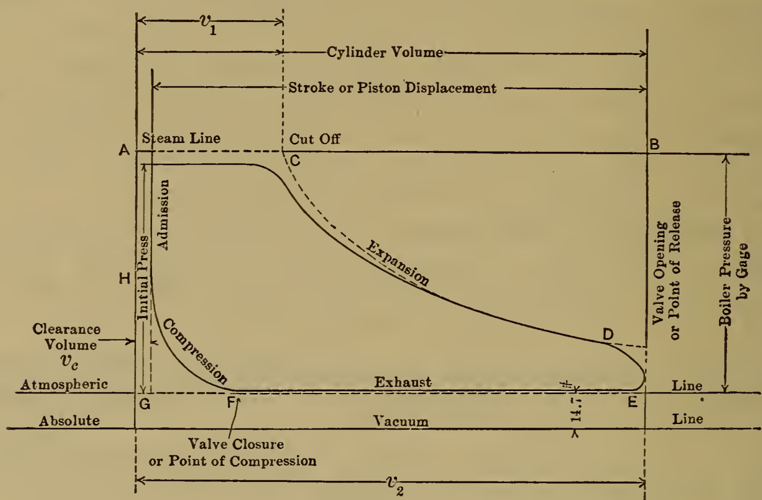


FIG. 412.

piston traverse. For the next stroke, the admission should be at the other end, and the exhaust at the end which has just had the pressure.

The governing of the speed of the engine can be the simplest since a valve to throttle or reduce pressure in the steam-pipe as the load lessens or the speed increases is all that is required or possible.

298. Expansive Working of Steam in the Cylinder. The other form of the indicator or work diagram exhibiting the steam operation in the cylinder has the form shown in Fig. 412. Here the valve admitting steam to the cylinder was not open during the full stroke or piston traverse but during a part only. At some point as C the valve was closed, either by the design of the valve-gear or by its method of operation. Hence while the volume between cylinder head and piston is increasing, there is no fresh weight of steam coming into it: what is now present in the cylinder will follow the universal law of all gases under pressure. It will expand in volume decreasing in pressure according to some law of relation between the two. Such an engine works expansively or works its steam expansively. The point where the admitting valve closed is called the "point of cut-off": such an engine is a cut-off engine. An engine which has the point of cut-off variable for different amounts of work to be done will be said to have a varying point of cut-off, or to be a "variable cut-off" engine. If the engine itself without human interposing can vary automatically the location of the point of cut-off, it is called an "automatic cut-off" engine.

What advantages follow from an adoption of the expansive working of the steam?

299. Advantages of Expansive Working of the Steam.

1. A reduction in the weight of steam furnished to the cylinder per stroke from the boiler. Instead of a volume proportional to AB and the corresponding weight at that pressure, the volume and weight are $\frac{AC}{AB}$ of the full cylinder volume and weight.

2. During the part of the stroke represented by CB the boiler is not furnishing steam to the engine, and during that time therefore heat can go in to the water without any going out.

3. The mean pressure p_m in the cylinder is less than it was in the non-expansive case, since the initial pressure is the same as before and the final pressure less.

4. The stroke is less powerful than it was in the previous system for this reason. If the horsepower is to be kept the same as with non-expansive working the cylinder-diameter or piston-area must be increased; but since the expanding steam is not reduced at once to the low terminal pressure, but drives the piston continually, the increase in cylinder area or diameter will not be proportional to the lowered mean pressure, or compel the same weight of steam to be taken into the volume AC as was formerly taken into AB. If, for example, the mean pressure in expansive working were only 85 per cent of the full pressure of the previous case, so that $p_m = p_1 \frac{85}{100}$, it would only be necessary to introduce the reciprocal of this multiplier into the numerator to keep the horsepower the same, or $\frac{100}{85} A = 1.17\, A'$. But such mean pressure is the approximate result of cutting off at one-half stroke. Hence the increase is 17 per cent in diameter but the steam used is diminished by 50 per cent. Numerical examples of this result will follow.

5. There is less weight of steam rejected at exhaust, for although the

cylinder is full of steam it is at a lower pressure and weighs less per cubic foot.

6. The less weight and at the lower temperature and pressure at exhaust rejects less heat to waste. Hence while external losses from radiation remain the same there must have been more heat transformed into mechanical work than in the former case. That is, the engine is a better heat-engine.

7. The cylinder has now a thermal efficiency as well as the boiler. If the final temperature range above a given zero of temperature at the exhaust or point of release of the steam be called T_3, then the weight of steam W at T_1 entering the cylinder and leaving it at T_3 makes the efficiency

$$E = \frac{(WT_1 778) - (WT_3 778)}{WT_1 778} = \frac{T_1 - T_3}{T_1}.$$

The lower this terminal temperature the greater the thermal efficiency. This carries some interesting conclusions as respects the use of condensing engines.

8. The increased thermal efficiency and the deduction in number 4 above result in getting a greater amount of work per pound of steam than in non-expansive working. As each pound of water is made into a pound of steam by burning a definite weight of coal at a price per ton, the expansive working is more economical in cost than the other, when a given mechanical work is to be done. Since furthermore the efficiencies increase with an increase in T_1 when the limit of T_3 is fixed, the advantages of high initial pressures and the compound engine which make them available are both suggested.

9. Governing the speed by governing the point of cut-off or the ratio of expansion makes the engine make each stroke with an effort proportioned to the resistance. Hence the variation of such engines from the mean speed desired becomes less.

300. Disadvantages of the Expansive Working of the Steam. The offsets to the above advantages are:

10. The larger cylinder diameter (No. 4) makes the engine heavier and more costly.

11. The valve-gear to produce cut-off of the steam, particularly a variable and automatic cut-off, and yet keep the release of the exhaust and its compression constant as is desirable to do make such designs complicated and costly (No. 9). Depreciation and interest charges increase, and possibly also repairs.

12. The temperature range from T_1 to T_3 or to the exhaust temperature T_2 causes the cylinder to cool at the end of the working stroke and

further durin th exhaust. Hence when the new stroke begins the entering steam at T_1 finds the piston and cylinder head cooler than itself, and some steam condenses in heating them up. This condensed steam in drops or a mist or cloudy vapor reëvaporates into steam as the fall of pressure in expanding allows the boiling point of such hot water to be reached. The heat to vaporize the water is withdrawn from the metal of piston or walls or from the expanding steam, cooling them further. All such absorbed heat is swept out with the exhaust

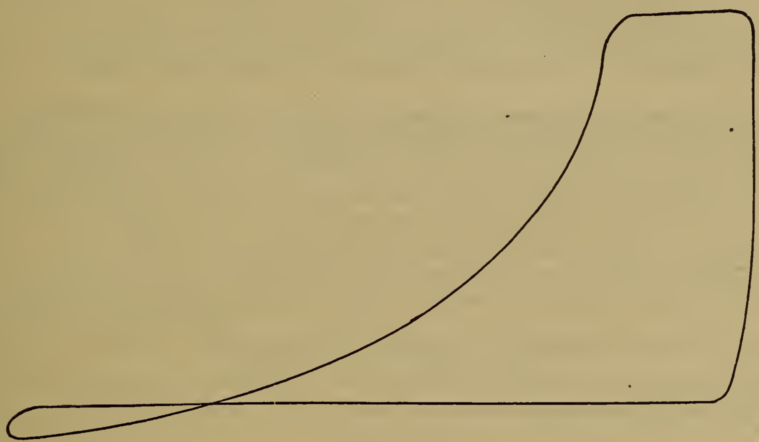


FIG. 413.

steam, and has to be replaced anew by the incoming charge. This process is called that of "internal condensation and reëvaporation" in the cylinder, and it is to diminish it that superheating is intended (paragrafs 228–231).

13. The percentage of the steam per stroke which fills the clearance volume (Fig. 412) is greater than in non-expansive working and consequently the percentage loss is greater. This is more than offset by the gains.

14. A back pressure on the exhaust or negative side of the piston of a given and irreducible value is a greater percentage of the lowered mean pressure in expansion than it was of the greater pressure throughout the stroke in the non-expansive working.

15. The mechanical friction of the engine acting in effect as a negative or back pressure upon the piston diminishes the net or forward effective pressure. When this mean effective pressure is less, the effect of such constant back-pressure effort is proportionately greater.

16. The improper adjustment of the grade or amount of expansion with respect to the initial pressure or the length of the stroke may

produce so low a final pressure that for part of the stroke there is a negative pressure, or one less than the pressure in the exhaust-pipe. The fly-wheel must drive the piston during such period; and when the exhaust opens, the exterior pressure being greater than that in the cylinder, there is a flow backward into the cylinder. In locomotives, exhausting into the smoke-box of the boiler, this causes cinders or grit to rush back into the cylinders, unless relief valves are provided (Fig. 603) opening into fresh air. Fig. 413 shows an indicator diagram with such a negative pressure loop. It may be said in general that the expansion line should never go below the line which represents a pressure just sufficient to overcome the friction of the unloaded engine.

301. Numerical Values of Pressure and Work of Expanding Steam. It will be of interest and service to obtain quantitative expressions for the elements which enter into the behavior of steam in expanding in the engine-cylinder. There are two general laws under which expanding gases like steam may be expected to act when the limits are such that no condensation of steam to water will occur in the process. If the pressure be taken to be that of one atmosphere on a square foot, and be called p_0 and observed: if the volume of a pound at zero C. and under that one atmosphere pressure be called v_0 and the absolute temperature be called T_0, then the quantity $\frac{p_0v_0}{T_0}$ for any gas is a constant and definite quantity and may be called R when computed and observed. For any other temperature a different value for p_1v_1 will exist, but in every case

$$p_1v_1 = RT_1.$$

If now T be kept constant, as occurs essentially in an engine which is steam-jacketed,

$$p_1v_1 = p_2v_2 = \text{Constant}.$$

Under this law, the expansion curve is an equilateral hyperbola, referred to the lines of no pressure and no volume on the diagram (Fig. 412) as its asymptotes: or points of pressure can be computed along the curve by multiplying the initial pressure p_1 by the fraction $\frac{v_1}{v_2}$ in this last equation. If the cut-off be at half stroke for example in an engine of two feet stroke, the pressure at every six inches would be, with initial pressure 100 pounds,

Travel of piston, inches	0	6	12	15	18	21	24
Pressure on piston	100	100	100	80	67	57	50

Similarly if the cut-off be at one-third stroke,

Travel of piston, inches	0	4	8	12	18	24
Pressure on piston	100	100	100	67	44	33

This shows the reduction of pressure at the exhaust (Nos. 4 and 5).

The other law of expansion revealed from observation is that the exponent of v is not necessarily unity, and will not be so when the temperature is not artificially kept constant. The natural way is for the final temperature to be less than the initial, so that

$$p_0v_0^n = p_1v_1^n = p_2v_2^n.$$

This expansion law is followed in a non-conducting and unjacketed cylinder: but as it requires logarithms to use and apply this law to numerical examples the other law will be followed, as the results will be close enough for present practical purposes.

If further the cylinder of the experimental engine be supposed to have a diameter of one foot or an area of 112 square inches or $\frac{7}{9}$ of a square foot, the volume of steam when working non-expansively will be 1.56 cubic feet of steam per stroke; the work done in that stroke will be $100 \times 112 \times 2 = 22{,}400$ foot-pounds, or 14,400 per cubic foot of steam.

In the first cut-off case at one-half stroke, the mean pressure was 85 pounds, but only 0.78 cubic foot of steam was drawn from the boiler. The work done per cubic foot was therefore

$$\frac{85 \times 112 \times 2}{0.78} = \frac{19{,}040}{7.8} = 24{,}400 \text{ foot-pounds per cubic foot.}$$

In the third case, the mean pressure was 70 pounds: the volume $\frac{1.56}{3} = 0.52$ cubic foot. The work done

$$\frac{70 \times 112 \times 2}{0.52} = \frac{15{,}680}{0.52} = 30{,}200$$

foot-pounds per cubic foot. The method for computing the mean pressure will be given in pargaraph 309. It appears therefore that the foot-pounds per pound of steam would be as

$$14{,}400 : 24{,}400 :: 30{,}200 : 100 :: 170 : 210$$

if the non-expansive type be taken as the standard. Earlier cut-offs at $\frac{1}{5}$ and at $\frac{1}{10}$ of the stroke would give mean pressures of 52 pounds and 33 pounds and corresponding values of 37,300 and 47,400 foot-pounds per cubic foot of steam.

But the disadvantages numbered 14 and 15 should be allowed for. An engine in which the expansion curve runs down to the pressure level of the line of back-pressure is said to have "complete expansion." If the back-pressure line was at zero pressure, or that of the complete vacuum, the theoretical values above would be reached. Where the vacuum cannot be perfect, or where the exhaust-pipe is of moderate dimensions and opposes a frictional resistance, these conditions cannot be reached. Hence it will be worth while to assume values for this back-pressure of practice and again tabulate. Table XIV results.

The back-pressure is subtracted from the mean forward pressure to get the mean effective pressure, and the difference multiplied by 2×112 to get the foot-pounds per stroke. This is again divided by the weight of steam admitted at that ratio of expansion.

The back-pressure figures correspond respectively to:

Column No. 4 — Ideal conditions.
Column No. 5 — The condensing engine with no friction allowance.
Column No. 6 — The non-condensing engine with no friction allowance.
Column No. 7 — The non-condensing engine with ten per cent friction allowance.

In comment upon the foregoing computations, it should be noted that the lowered back-pressure of the condensing engine gives it an advantage over the non-condensing engine at all points of cut-off; that to cut-off earlier than one-fifth in the non-condensing engines begins to diminish the work per cubic foot at this pressure, but the difference is not very great when friction is not allowed for, and disappears when friction is taken account of. The work per cubic foot at $\frac{1}{10}$ cut-off in such engines is less than the non-expansive work. It will be seen later again why the cut-offs earlier than one-half in such engines are not advisable for other reasons.

TABLE XIV.

WORK PER CUBIC FOOT OF STEAM EXPANDING WITH BACK-PRESSURE.

Cut-off in Fraction of the Stroke.	Mean Pressure Computed.	Volume of Steam Admitted.	Work Done per Cubic Foot of Steam. Back-Pressure being in Pounds.			
Column No. 1	2	3	4	5	6	7
			0 Pounds.	3	17	27
No cut-off........	100	1.56	14,400	13,900	11,900	10,500
One-half..........	85	.78	24,400	23,600	19,500	16,700
One-third.........	70	.52	30,200	28,900	22,800	18,500
One-fifth.........	52	.31	37,300	35,200	25,100	18,000
One-tenth........	33	.16	47,400	43,100	23,000	8,600

302. Weight of Steam entering the Cylinder at Different Initial Pressures. Steam per Horsepower per Hour. It will be apparent that with a given cylinder-volume filled at each stroke with steam of differing pressures the work of the weight will differ, and their relation to each other becomes of interest. If the same engine of one foot diameter and two feet stroke be taken and the cut-off be assumed at one-third stroke, then the cylinder volume filled from the boiler will be per stroke,

$$\frac{112}{114} \times \frac{2}{3} = \frac{14}{27} = 0.52 \text{ cubic foot.}$$

If the engine turn at 100 revolutions per minute, the volume per hour $= 0.52 \times 100 \times 2 \times 60$. To get the weight per hour, the steam tables must be consulted to find the weight of a cubic foot of steam at the pressure existing at the moment of cut-off. The mean effective pressure multiplied by the area and by 200×2 and divided by 33,000 gives the horsepower. Hence for the assumptions Table XV follows.

TABLE XV.

STEAM PER HOUR AND PER HORSEPOWER AT DIFFERENT PRESSURES.

1	2	3	4	5	6	7	8	9
Initial Pressure p_1	Mean Pressure p_m	Back-Pressure 147+2.3	M.E.P.	I.H.P.	Volume of One Pound in Cubic Feet.	Weight of Steam per Stroke.	Weight of Steam per Hour, W.	$\frac{W}{I}$ Col. 8 ÷ Col. 5.
140	98	17	81	110	3.2	0.162	1960	17.8
120	84	17	67	91	3.7	.140	1690	18.5
100	70	17	53	72	4.4	.118	1400	19.4
80	56	17	39	53	5.5	.095	1150	21.7
60	42	17	25	34	7.0	.075	880	26.0
40	28	17	11	15	10.3	.051	610	40.0

This indicates plainly the economy in the use of higher pressures with a fixed cut-off in non-condensing engines, cushioning and clearance being neglected, and that such an engine is less efficient at low loads than at the higher.

303. Cut-off and Ratio of Expansion. Most Economical Point of Cut-off. It will appear from Fig. 412 and the law of Mariotte that the expansion of steam after the point of cut-off from the volume v_1 to the final volume v_2 will give a ratio of increased volume or of expansion which if denoted by r will give $v_2 = rv_1$; or, $r = \frac{v_2}{v_1}$, a quantity always greater than 1, since v_2 is greater than v_1. The point of cut-off in terms of the piston stroke v_2 is $\frac{v_1}{v_2}$. Hence the ratio of expansion is the reciprocal of the point of cut-off. It is convenient in calculations to use the ratio of expansion and avoid fractional values.

The conclusions of the previous paragraf have shown that it is not desirable to have the terminal pressure less than a given value, fixed by the back-pressure, either actual or assumed to cover friction of engine, condensation water, and other losses. Hence if the initial or boiler pressure be fixed and the terminal pressure also determined, the point of cut-off and ratio of expansion become fixed, since,

$$p_1v_2 = p_2v_1$$

whence

$$v_1 : v_2 :: p_2 : p_1,$$

which gives

$$\frac{v_1}{v_2} = \frac{p_2}{p_1} = \frac{\text{Exhaust pressure + friction + condensation}}{\text{Initial or boiler pressure}}.$$

The ratio of expansion being the reciprocal of this is

$$r = \text{ratio of expansion} = \frac{\text{Initial pressure}}{\text{Total resisting back-pressure}}.$$

For a non-condensing engine, this back-pressure will be made up of 17 pounds from the vacuum for the pipe resistances and 20 pounds for engine friction and other losses, or 37 in all, in smaller engines. In larger condensing engines the value of Willans is 25, made up of 17 + 8, and is representative of experience. If these assumptions be applied to a case using 100 pounds of steam as initial pressure, the corresponding values for r will be as in Table XVI when various total back-pressures are taken.

TABLE XVI.

VALUES OF RATIO OF EXPANSION IN ENGINES WHEN $p_1 = 100$.

Maximum Work in Terms of	Will be obtained per Pound of	By making $\frac{p_1}{p_3}$ equal to	Condition met by —
Indicated work	Indicated steam	100÷17=6	Non-condensing engines
Net or brake work	Indicated steam	100÷27=3.7	
Net brake work	Actual steam used	100÷37=2.7	
Indicated work	Indicated steam	100÷ 3=33	Condensing engines
Net or brake work	Indicated steam	100÷17=6	
Net or brake work	Actual steam used	100÷27=3.7	
Indicated work	Actual steam used	100÷25=4	Willans' rule
Indicated work	Actual steam used	100÷19=5.5	Emery and Loring, U.S.N. tests, 1874–75 $r = 1 + \frac{p_1}{22}$

The historic Loring and Emery tests would give the back-pressure value in terms of the initial pressure $p_3 = 12.5 + \frac{p_1}{16}$. The Gately and Kletch test in 1884 on a Corliss simple engine gave a condensation value of 0.08 p_1, when non-condensing and a value of 0.19 condensing to be added to the fluid and engine friction back-pressure.

John Perry* proposed the rule that the internal condensation as a factor of the indicated steam should be computed by a formula

$$\frac{\text{Steam not accounted for}}{\text{Steam per indicated diagram}} = y = K\frac{1+r}{d\sqrt{n}},$$

in which d is the cylinder diameter in inches, n' the number of strokes per minute, and r the ratio of cut-off. K for non-condensing engines has an average of 15, with a best value of 5 in a well-drained jacketed cylinder with four valves, and rising to

* The Steam Engine, London, 1902, MacMillan and Co.

30 in a cheaper slide-valve engine badly drained and unjacketed. For the condensing engine, he introduces the initial pressure under the radical in the denominator, and the average factor is 120, with corresponding limits of 50 and 300.

An exhaustive test to obtain experimental values of the best point of cut-off from the horse-power formulæ would cover trials for water-rate when

1. The initial pressure is made to vary, speed constant, cut-off constant.
2. Cut-off varying, initial pressure constant, speed constant.
3. Speed varying, initial pressure constant, cut-off constant.
4. Speed constant, initial pressure varied with the cut-off so that the terminal pressure shall be constant, or, $\frac{p_1}{r} = K$, in which K shall have one of the accepted values, say 25. If these results be plotted in curves upon a diagram, it will be found that the varying pressure and varying speed tests plot in a straight line on indicated horse-power abscissæ and weight of steam per hour as ordinates. The other two will be curves. The tangent to such a curve will be the water per horse-power, from the significance of the coördinates, and it will be greatest for the smaller loads. It is only with varying cut-offs that there is a load of maximum efficiency. In multi-cylinder or continuous expansion engines, where the expansion is great in any case, there is not much gain in economy in cutting off early; or there is a considerable range of load with the same economy. In a single-cylinder engine the gain by cut-off regulation is greater than with the multi-cylinder engine, and particularly with the lighter loads.

A rule of some acceptance* is to make the ratio of expansion

$$r = .42\sqrt{p_1},$$

the pressure being counted from vacuum to apply both to condensing and non-condensing engines.

304. Governing in Non-expansive Engines or with Fixed Cut-off throttling governing. It will be apparent from Fig. 411 that in engines taking steam full-stroke the condition of that diagram and with that initial pressure is the full-load value, and cannot be exceeded. The only variation with a fixed boiler pressure is downward. A less load of resistance must be met in the cylinder by lowering the height of the diagram or the value of such initial pressure. This will be done by partly closing the valve between boiler and engine, and thus throttling the flow of steam by changing the pressure energy into velocity through the reduced area of the valve-opening and transforming some heat into work at the constricted point.

If the engine has a fixed cut-off, not variable by hand at will while the engine is running, the same state of affairs exists. The area of the work diagram or the value of the mean pressure must be reduced here also by lowering the height or value of the initial pressure. Governing appliances which act to control the engine speed by throttling or reducing the initial pressure are called "throttling" governors, or

* Buckeye Engine Company, Salem, Ohio,

such engines are throttling engines. Fig. 415 is a type series of superposed work or indicator diagrams to illustrate practically fixed ratio of expansion but a varying initial pressure by throttling action.

The arguments for the fixed cut-off type with throttling variation of initial pressure are:

1. The engine is cheap to build and to buy. The valve-gear for distribution will be simple and therefore inexpensive, and the governor controlling pressure only will be extremely simple, and its valve need not be complicated.

2. The effort of the steam to drive the piston will be exerted through a considerable portion of the stroke. Hence there will be less inequality in the steam effort at the beginning and end of the stroke.

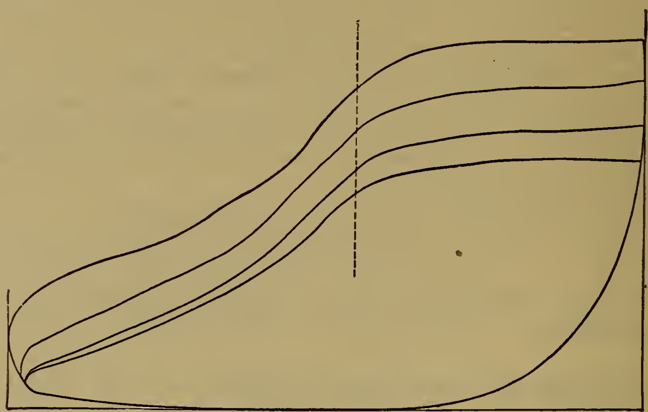


Fig. 415.

3. The effect of driving the steam through the throttling-valve or orifice is to bring about the equivalent of a superheating of the steam. The pressure on the boiler side of the valve is greater than that on the cylinder side. Hence if the steam were saturated at the higher pressure it will have a temperature on the low-pressure side higher than belongs to that pressure, and is therefore in a superheated condition. This has a tendency to dry out moisture in the steam and to diminish condensation in the cylinder. The heat corresponding to friction in the throttle-valve area must also appear in the form of heat, some of which serves to heat and dry the steam.

4. The throttling-engine for these two latter reasons is likely to suffer less from cylinder-condensation. The diminished range of temperature between the two ends and the relatively higher terminal pressure are the reasons for this.

The arguments against the throttling system can only be weighed when such engines are in competition with the class using steam expansively, and the discussion must therefore include the considerations of the preceding paragraphs and be viewed in their light. Independent of these are the following disadvantages:

5. It is not as sensitive as the cut-off engine to instantaneous variation in the resistance. The control by throttling can only take effect in the cylinder at an interval after the governor has acted to throttle the steam or to open the valve wider. The engine meanwhile has had a chance to make at least one stroke under the conditions which prevailed before the change of condition was announced to the governor.

6. The throttling-engine does not regulate as closely to uniform speed as the cut-off engine. The reason for this is partly that explained in the preceding sentences, and partly because the method of controlling by the motor fluid in bulk cannot be expected to be as exact as when the control is exerted immediately upon each reciprocation of the piston.

305. Governing in Expansive Working of Steam. Automatic Cut-off Engines. The principle of expansive working of the steam in the

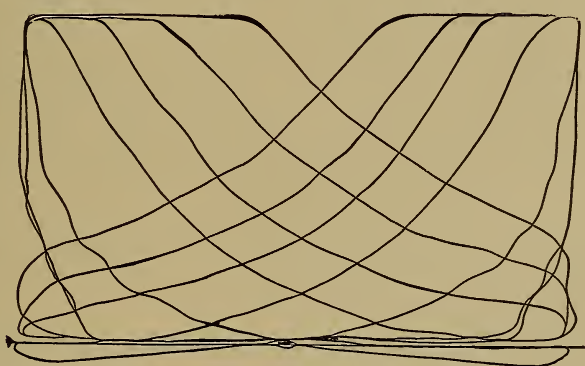


Fig. 416.

cylinder lends itself so easily to solve the problem of varying the effort as the resistance varies, that this becomes a prime factor among the advantages of the system. The valve-gear for expansive working can be so easily made to cause the point of cut-off to vary by action of the governor, that the variation in cut-off by load can become automatic without great entailed expense. Fig. 416 shows the superposed diagrams of a succession of four strokes with varying resistance. The smallest area is a card of practically no load. The cut-off successively grew later as the resistance increased.

Governing by varying the cut-off point without reducing the initial pressure offers the following advantages:

1. The effort is controlled per stroke of the piston. Just enough steam is admitted into the cylinder to do the work of that stroke.

2. For this reason the engine is sensitive immediately to variations of the resistance.

3. It is more certain to be kept by the governing appliance at the uniform or fixed speed, since a variation caused in the governing appliance operates immediately to control the admission for the next stroke.

4. The full energy present in the elastic tension of the steam as it comes from the boiler is exerted upon the piston without undergoing

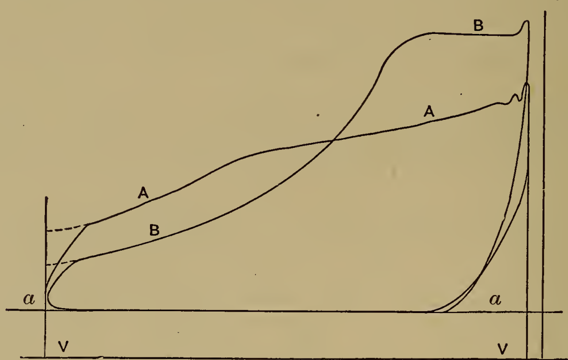


FIG. 417.

the loss from throttling. This may or may not be considered an advantage (see paragraf 304 above).

The disadvantages on this same plane are:

5. The design and complication of valve-gear to provide for properly varying the admission.

6. This complication usually makes the engine costly to build and to buy. If closeness of regulation without the intervention of human agency is not worth paying for, the superior economy of the automatic cut-off engine does not always pay the interest on the difference of first cost.

The automatically governed variable-cut-off engine secures the advantages of expansive working in economy of water rate, but carries with it the disadvantages attaching to that principle as practically carried out. For example let two actual cards be superposed, one of each class as in Fig. 417, assumed to have each the same M.E.P. with the terminal pressure 37 for the throttling card A and 27 for the cut-off card B. The weight of steam rejected at exhaust was that of a given final cylinder volume at 37 pounds pressure in one case and 27 pounds in the other for the same work done. The water rate will be 31 pounds per I.H.P. per hour for the throttling-engine, and 23 pounds for the cut-off, or an apparent loss of nearly 29 per cent over the better performance by the throttling method. The disadvantages will bring this down to 20 per cent or below. The lower terminal pressure causes less heat to be rejected into the exhaust, and reaps the full advantage of having as great a difference between the initial and final pressure and temperatures of the steam in the cylinder as is consistent with doing the foot-pounds of work required for that stroke.

The disadvantages of the cut-off engine are the contradictories of the advantages of the non-expansive engine:

7. There is a wide difference in the pressure and effort at the two ends of the stroke when the engine is working with an early cut-off. This compels weighty reciprocating parts and a massive fly-wheel to take care of these wide variations, and to give out in the latter part of the stroke the excess of work stored in it at the beginning.

8. The lower value for the terminal temperatures and pressures increases the amount of cylinder-condensation by reason of the better opportunity for the evaporation of moisture present in the cylinder either mechanically entrained or as the result of radiation or the doing of work. The evaporation of such moisture under reduced pressure makes the demand for the necessary heat for vaporization from either the working steam or the metal of the cylinder. It is this condition which accounts for the result experimentally found, that in non-condensing engines, such as the locomotive, it does not pay to carry the expansion further than is given with the cut-off at one-fourth of the stroke.

9. It may happen when the engine is very lightly loaded that the cut-off will take place so early that the final volume of the cylinder will be greater than that which the volume of steam admitted would fill at the pressure of the exhaust stroke. The line representing pressures will therefore cross the line representing the return-stroke before the stroke is completed, forming a loop at one end (Fig. 413).

Weight should also be given to the service conditions, because in many classes of work in power-house service the variation of resistance is so wide and so rapid that it would be inconvenient or impracticable to depend on human quickness of perception to provide for it. On the other hand, where the effort is constant, as in pumping, or is progressively varying, as in hoisting from deep mines, and in railway and marine practice, the other method of regulation is close enough to be satisfactory, and particularly where the engine-runner must be in attendance in any case. The automatic cut-off engine is usually the more economical of the two, but it is usually better built in every way, and such excellence of construction would explain some of its economy, by diminishing leakages and losses in friction and back-pressure.

306. The Condensing Engine, with Lowered Back-pressure. The discussion of expansive working in the preceding paragraphs has made it clear that it would be of advantage to diminish the back-pressure upon the piston. When the engine exhausts into the atmosphere, the pressure therein of 14.7 pounds per square inch above a vacuum precludes getting much below 17 pounds for such back-pressure, on account of friction in passages, valves and pipe with its fittings. If, however, the engine could exhaust into a vacuum chamber when the pressure approached zero of pressure, the advantages of such lowered pressure could be secured, less the drawbacks of the mechanical apparatus to create and maintain the vacuum. The properties of steam and of water make it easy and simple to create this lowered pressure in a closed vessel from which the air can be excluded, since by cooling the steam or the vessel below 212° the tables show that the steam cannot remain as steam but must go back to water to such an extent that only the remaining vapor can remain steam at such lowered temperature. The change to water of so large a volume of steam produces a volume filled only with water vapor at the corresponding low tension. This is a vacuum, therefore, of varying degrees of completeness according to the temperature.

To *maintain* this vacuum is a separate process when the vacuum chamber is of limited volume. It tends to fill with the water resulting from the condensation of the steam, and this water has to be gotten out, under the most difficult conditions, since the pressure outside the vacuum chamber is greater than that within it.

The lowering of the temperature in the vacuum chamber will be done with the cheapest available cooling medium. This will be water, with the high specific heat of unity. It must be paid for, either in a tax, or in the interest on the cost of the plant which brings the water to the condensing vessel or " condenser " and its operation. If an infinite or indefinitely great quantity of water can be used or brought, the water will not be sensibly raised in temperature in cooling the condenser and the steam. If the water must be handled by pumps or similar apparatus it is best to reduce the weight of water, and allow the temperature to rise during the cooling process. Hence although the water may be at 60° F. on an average through the year, the condenser temperature will average 110° to 130° F. for these reasons. Air can be used to cool the condenser by causing cooler air to blow upon a sufficient contact surface, but it is much less effective per unit of such cooling surface.

The advantages of the condensing engine are both thermal and mechanical. Among the thermal advantages are:

1. The greater thermal efficiency referred to in paragraf 297 resulting from the lowered value of T_2 in the formula. If it can be possible also to raise the initial pressure as in the dotted lines of the left hand of Fig. 418, the theoretical efficiency is further increased.

2. This is accompanied with more effective realization of the thermal advantages of expansion. The discussion of paragrafs 302 and 303 showed that lowered terminal pressure increasing the ratio of expansion made less weight of steam per horse-power per hour, and hence less weight of coal to make the steam per hour. Fig. 418 illustrates to the eye the gain of utilization of the intrinsic or potential energy in a given

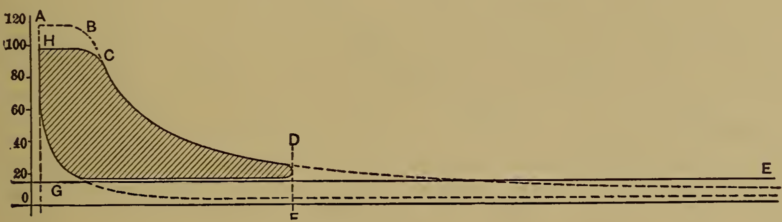


Fig. 418.

weight of steam, if the exhaust need not be opened after a traverse to *DF* only, but the stroke continued by increasing cylinder volume until the atmospheric pressure line *EG* was passed and a terminal pressure reached which was about 10 pounds above zero, and the exhaust return back-pressure be only about 3 pounds. With a given boiler pressure as at *H* the same net forward effect is produced as though the initial pressure had been increased to *A*, so chosen as to give the same increase of area as that secured at the bottom of the card by lowering the back-pressure.

3. This results in a gain in heat consumption or in fuel saving in one of two alternative ways. Either the same work is done in a smaller condensing cylinder which is done in the larger non-condensing cylinder; or in a given cylinder condensing the cut-off can be earlier than in the non-condensing. In either case less indicated steam is used and less coal per I.H.P. Fig. 419 shows two superposed cards of equal area and equal work in a given cylinder for the two cases. The dotted line is the card of work done above atmosphere. The full line shows that the cylinder volume need only be filled $\frac{1}{5}$ of its volume to do the same work

with expansion below atmosphere, as the non-condensing requires a volume of $\frac{1}{3}$ of the whole to do. This gain, if all realizable, would be $\frac{2}{15}$ of the steam weight per hour or 13 per cent under the pressure conditions assumed.

4. A heat saving results from impounding and storing in a hot-well the hot water resulting from condensing the steam. This can be used as boiler feed-water (paragrafs 146 to 151), doing away in part with apparatus to preheat the feed-water (paragrafs 225 to 227) and

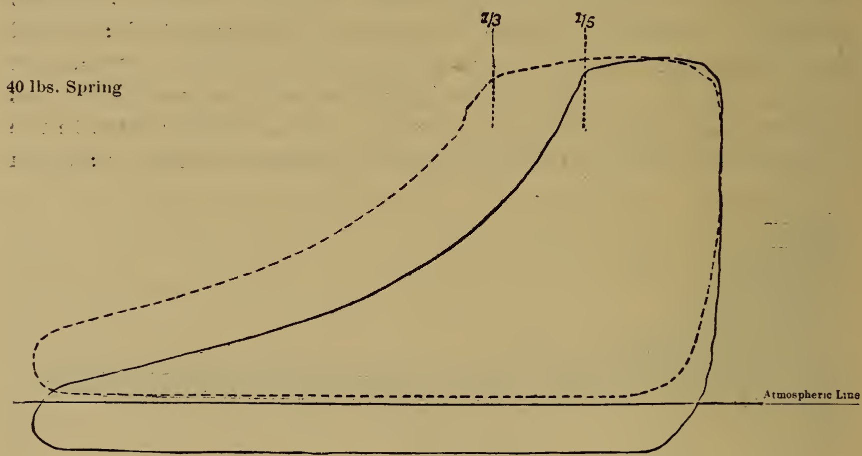


FIG. 419.

lessening the heat-units demanded of the coal to make the necessary steam. In an exhaust to the atmosphere, the heat rejected in the weight and temperature discharged is not returnable to the plant directly.

5. The lowered temperature in the expansion at the end of the stroke causes the cylinder to reject less heat per stroke than in the non-condensing engine.

The mechanical advantages are those of increased power due to lowered back-pressure, or increased area for the indicator card, and the smaller lighter engine cylinder resulting. The heated water produces less strain upon the plates or elements of the boiler, by lessening the stresses from unequal contraction under pressure (paragrafs 193 and 132).

The condensed water is pure distilled water free from chemical compounds. It is of advantage for use in boilers as it deposits no scale (paragrafs 185 to 189), and in marine practice the same pure water can be used over and over again without bringing into the boiler the acids from the sea-water. This is also true for land conditions where good feed-water is expensive or not available.

The physical principle on which the condensation of steam causes the practical vacuum is that one cubic inch of water will form 1658 cubic inches of steam at the pressure of one atmosphere. If these 1658 (often called 1700) cubic inches of steam are cooled back to water, they undergo a reduction of volume in the same proportion less only the volume filled by the tenuous vapor which even cool water gives off in a vacuum. It will only be necessary to draw off the condensed steam as water by proper apparatus to enable the vacuum to be maintained which the condensation has created.

The earliest historic steam-engines of the modern period were all condensing engines. Steam at a comparatively low pressure above the atmosphere was admitted to the cylinder for the working stroke, and upon being condensed the absence of pressure represented by the vacuum upon the working side of the piston was the principal dependence for the power of the stroke. Such engines were called low-pressure engines. When the engine did not condense, so that the back-pressure line in Fig. 418 was at atmospheric pressure or above it, it was necessary that the pressure of the steam in the boiler should be correspondingly raised. Such non-condensing engines were therefore run at relatively high pressure, and were called high-pressure engines. At one time, therefore, high pressure was synonymous with non-condensing, and low pressure synonymous with condensing. This is no longer the case, since nearly all condensing engines of modern construction operate with steam at high pressure.

307. Disadvantages of the Condensing Engine. The disadvantages of the condensing engine are also in two groups, the thermal and the mechanical. In the thermal class are:

(1) The greater temperature range in the cylinder and the lowered final temperature increase the loss by thermal interchange of heat between the working steam and the metal of the cylinder walls. During exhaust the condenser temperature prevails, considerably below 212° Fahr., and greater initial condensation occurs, and re-evaporation loss (paragraf 300).

(2) Condensation increases the heat flow and condensation in the steam-jackets of the cylinder, if these are used, both in the heads and around the barrel of the latter, causing a loss of heat which ultimately flows away with the cooling water.

The mechanical losses are caused by:

(3) The necessity for the cooling water for the condenser. This has to be paid for in taxes or in interest on the plant. In some places

water is hard to get, as in climates where there is a long dry-season, or little rain-fall, or where there are no streams or bodies of water. It may be necessary to cool the injection water and use it over and over again.

(4) The pumps and apparatus for handling this cooling or "injection" water are costly, heavy and bulky, unless a natural flow is available: cooling apparatus if required is also costly and bulky.

(5) The cost and maintenance of pumps and apparatus to maintain the vacuum in the condenser are an added expense to the engine, as well as the condenser, and the power to operate these and the injection pumps are elements of cost to be deducted from the gain from condensing.

(6) If these pumps are operated by the engine itself, they limit its speed, or the engine speed limits theirs, according to circumstances.

(7) The condensed water from the condenser and in the hot-well carries with it the oil used in lubricating the cylinder piston valves and the like. The lubricating material must undergo the cooling of the condensing process, and gradually fouls and stops up the passages through which it passes, or else it goes through to be pumped back with the warmed water into the boiler. This presence of lubricating oil in boilers is a serious annoyance, inasmuch as a coating of such material on heating surfaces prevents intimate contact of water with the metal, and frequently causes the latter to become overheated and so softened as to be easily forced out of shape by the pressure in the boiler. Great care has to be taken to separate the oil from the condensed steam in the condensing engine to prevent this difficulty. See paragraf 246.

The condensing engine can only be used where an available quantity of water for condensation can be procured without excessive cost. This limits the application of the principle to stationary practice on land, but is a reason for the abundant and extensive use of the condensing engine for marine purposes. Condensation by air requires an enormous bulk for the condensing appliances, and where water is costly or scarce special provision must be made for using the same water over and over again. It is practically impossible to operate the locomotive as a condensing engine.

If the heat in the exhaust steam can be used for heating air for buildings and shops or tanks or solutions in manufacturing, so as to release the steam-boilers from furnishing such steam in addition to that required for power, the condensation of exhaust steam is undesirable and unwise. In such cases the engine may be regarded as a form of reducing valve as respects pressure in such heating coils relatively to that in the boilers, so that in effect the manufacturer gets his power for nearly nothing if he can use all the heat in the exhaust steam for heating.

The mechanical apparatus in the way of condensers, pumps and the methods of effecting condensation and maintenance of vacuum pressures and temperatures will form the topics of a later chapter, under engine-room auxiliaries.

308. The Compound Engine. The condensation loss from the great temperature range of the condensing engine suggested the advisability of dividing the expansion process into stages, so as to distribute the heat interchange among two or more cylinders, in each of which the range should be less. This thermal concept brought with it so many mechanical advantages, and so wide a range of mechanisms to embody it, that its discussion will be taken up in the following chapter.

309. The Computation of the Mean Pressure in Expansion. Reference was made in paragraf 301 to the mathematical procedure for computing a mean pressure of a gas expanding in volume. The methods of the calculus are the most convenient.

When a fluid of volume v undergoes an infinitesimal increase in volume dv, while the pressure is assumed constant for that small increase in volume, the work done is pdv. If the fluid at v_1 increases to v_2 and the law of such expansion is $p_1v_1 = p_2v_2 = pv$, the total work done within these limits will be

$$\text{Work} = \int_{v_1}^{v_2} pdv = p_1v_1 \int_{v_1}^{v_2} \frac{1}{v}\, dv = p_1v_1 \log \frac{v_2}{v_1}.$$

But $\frac{v_2}{v_1} = r$, the ratio of expansion (Fig. 420): and if there was work done before cut-off and expansion begin while pressure was coming in p_1 and working through the space v_1, this work was p_1v_1 and should be added to the expansion work. If there is a back-pressure p_3 acting constantly through the space v_2, the total work will be the algebraic sum of the three, or

$$\text{Work} = p_1v_1 + p_1v_1 \log r - p_3v_2.$$

But this work equal to the area under the line bounding the pressure ordinates will be equal to the product of a mean resistance or a mean pressure acting through the same length v_2; hence the value for the mean pressure will be found by dividing the work by v_2; or

$$p_m = \frac{W}{v_2} = p_1 \frac{v_1}{v_2}\left(1 + \log r\right) - p_3 = p_1 \left(\frac{1 + \log r}{r}\right) - p_3 \cdot$$

If there is no back-pressure, p_3 disappears.

If, on the other hand, the expansion is not at constant temperature, or isothermal, then the exponent for v will not be unity. Call it n. Then

$$p_1v_1^n = p_2v_2^n,$$

and

$$\text{Work} = \int_{v_1}^{v_2} pdv = p_1v_1^n \int_{v_1}^{v_2} v^{-n} dv,$$

$$= \frac{p_1v_1^n\,(v_2^{1-n} - v_1^{1-n})}{1-n},$$

which can be transformed into

$$W = \frac{p_1 v_1}{n-1}\left[1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right],$$

and if admission work $p_1 v_1$ be added to it, and back-pressure work subtracted,

$$W = \frac{p_1 v_1}{n-1}\left[n - 1\left(\frac{v_1}{v_2}\right)^{n-1}\right] - p_3 v_2;$$

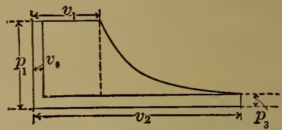


FIG. 420.

so that the mean effective pressure becomes

$$pm = \frac{W}{v_2} = \frac{p_1}{n-1}\left(\frac{n}{r} - \frac{1}{r^n}\right) - p_3.$$

In the steam cylinder of $\frac{A}{144} \times L$ cubic feet, with a cut-off $\frac{L}{r}$ the volume of steam per stroke $= \frac{AL}{144\, r}$. The work done under the hypothesis first presented is $pmAL$ per stroke. Hence the work done per cubic foot of steam is

$$\frac{pmAL \times 144\, r}{AL} = 144\,[p_1\,(1 + \log r) - p_3 r)$$

When p_1 and p_3 are assumed this is a maximum for $r = \frac{p_1}{p_3}$ as discussed in paragraf 303.

The work in terms of pressures instead of volumes above will take the form

$$W = \frac{p_1 v_1}{n-1}\left[n - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right],$$

and in terms of absolute temperatures

$$W = \frac{p_1 v_1}{n-1}\left[n - \frac{T_2}{T_1}\right].$$

CHAPTER XVIII.

THE COMPOUND AND MULTIPLE-EXPANSION ENGINE.

315. Compound Engine defined. A compound engine is one in which the process of expanding the motor or pressure fluid from its initial high pressure to its final low pressure is done in two stages or in two steps in two or more cylinders. The reverse process is done in a compound compressor. The single cylinder of the previous types will be the same in the compound as in the simple, since the terminal pressure determines the final volume of the steam at the moment when the exhaust opens, and this final volume (called v_2 in the computations and in Fig. 420) is the volume corresponding to the product LA in cubic feet in the formula for horsepower. The second cylinder or extra cylinder in the compound is the smaller of the two, introduced between the main or low-pressure cylinder and the high initial pressure from the boiler or reservoir of pressure.

As the expansion line may go down below the line of atmospheric pressure, or may be cut by the line of back-pressure above the atmospheric pressure, there will be two types of compound engine. The condensing compound has the low-pressure cylinder condensing: the non-condensing compound, or the engine which is compound above the atmosphere, has no condensing cylinder, nor condensing apparatus. The small cylinder taking the higher pressures is always non-condensing, and is called the high-pressure cylinder. Its exhaust is the initial or driving pressure for the low-pressure cylinder.

In the normal or type compound, both cylinders deliver their effort to a common engine shaft, either through one crank or through two or more. The compound engine has therefore thermal advantages due to the division of the expansion work on the diagram, and practical or mechanical advantages due to the use of more than one crank-pin and the distribution of effort of the expansion upon several cranks. These will be examined separately.

A triple engine, or triple-expansion engine, is one which divides the work of the expansion process into three stages or steps, and uses three or more cylinders for the process.

A quadruple-expansion engine is one which divides the work of the expansion process into four steps or stages, and uses four or more cylinders for the process. There are no large installations higher than four stages, on account of the inconveniently high initial pressures required in the boiler to make them significant, and the cost of such

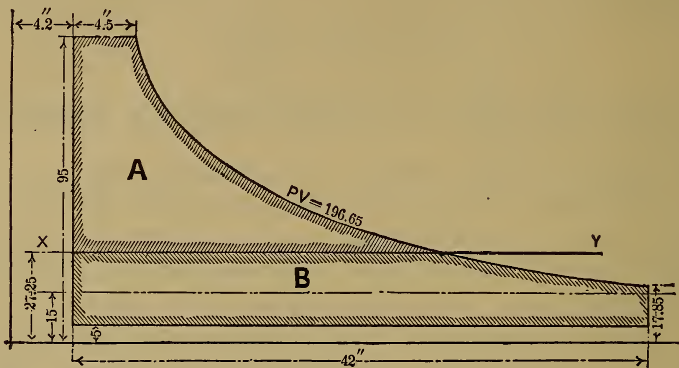


FIG. 425.

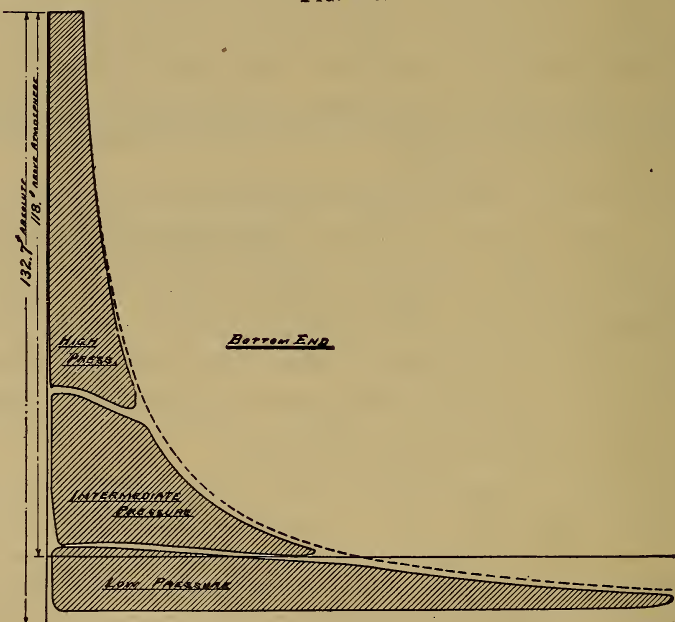


FIG. 426.

large engine. The usual range of selection of type with initial pressure will approximate:

For a pressure below 80 lbs.,	use simple engine;
For pressures between 80 and 120 lbs.,	" compound engine;
" " " 130 and 160 "	" triple "
" " above 170 lbs.,	" quadruple "

If the two pistons of the compound engine are upon the same piston-rod as in the tandem or steeple types to be presently reviewed, the ratio of the work done in the two cylinders is not specially significant. A sound principle of proportioning would be to make the surface exposed to heat transfer in the two cylinders bear a ratio to the heat disposable in each, or to make the heat range equal in the two. Mr. George I. Rockwood has designed engines on this basis with a volume ratio of 7 to 1. Ordinarily, however, the two cylinders have each its own crank; and here the distribution of equal work on each crank makes it a better arrangement to divide the work diagram, Figs. 425, 426, into two parts of equal area for the compound, and into three parts for the triple. This tends to keep the efforts of the cranks in balance. The two cylinders have nearly always the same stroke, so that if the mean pressures are to be assumed equal, the cylinder volumes must be related to these pressures.

Let R denote the ratio between the volume v of the small cylinder and V the volume of the large. Let E be the total number of expansions: then if e be the ratio of the expansion in the small cylinder, $E = eR$. Assume the initial pressure p_1 and the pressure at the beginning of the stroke of the larger cylinder to be p: then

$$p_1 = ep = \frac{E}{R} p.$$

The two mean pressures will be determinable from the formula of paragraf 309, and the work to be equalized will be

$$\frac{v}{e} p_1 (1 + \text{hyp log } e) - vp \text{ for the small cylinder,}$$

and

$$\frac{V}{R} p (1 + \text{hyp log } R) \text{ for the large cylinder,}$$

back pressure in the large being assumed zero for simplicity. Then reducing after equating:

$$\text{hyp. log } E = 1 + 2 \text{ hyp. log } R,$$

or

$$\text{hyp. log } R = \frac{\text{hyp log } E - 1}{2}.$$

If the back-pressure be taken at 17 pounds and the initial pressure at 153 absolute, the ratio of total expansion is 9, and $R = \frac{V}{v} = 1.81$. This explains and justifies the usual ratio of diameter of 1.8 to 2, when both pistons have the same stroke.

Usual cylinder-ratios of practice for usual pressures with triple engines are:

Pressures.	Small.	Intermediate.	Large.
130	1	2.25	5
140	1	2.40	5.85
150	1	2.55	6.90
160	1	2.70	7.25
170	Quadruple engine preferred.		

For quadruple-expansion engines the usual ratios of cylinder-areas and volumes approximate 1 : 2 : 3.78 : 7.70, which may be called 1 : 2 : 4 : 8.

If the principle be adopted that the ratios of areas are to be as the fourth root of the number of expansions, the ratio of the first to the fourth will be as the cube of the fourth root. The ratio will increase as the initial pressure becomes greater; e.g., 1 : 2.2 : 4.8 : 10.6.

The expansion in two stages need not be made in two cylinders only. The high-pressure cylinder having a volume, v, may deliver its steam into two larger cylinders of equal area whose aggregate volume is V. Such engine with three cylinders, however, is only a two-stage expansion or compound engine. Similarly, the triple may have a high, an intermediate, and two low-pressure cylinders, or four in all, and yet be a three-stage expansion engine. The emphasis is upon the number of steps in the expansion of volume and not upon the number of cylinders.

Again, the mechanisms discussed in paragraf 291 where cylinders were twinned, or set up double-opposed, in order to get piston area without excessive diameter for the cylinder, must be distinguished from the compound or triple-expansion engines of several cylinders. The essential point of difference is that twin or multiple-cylindered simple engines receive their supply of motor energy to all cylinders equally from a common source in engines of this class, while in the continued-expansion class, whatever the number of cylinders, the cylinders are in a series, of which some receive their motor energy after others have first withdrawn a part of such energy either in the form of heat or pressure or both.

316. Thermal Advantages of the Compound Engine. The heat saving or thermal effectiveness of the compound and multiple-expansion engine referred to in paragraf 308 appears in four ways.

1. The internal condensation loss in the cylinder is due to the temperature and pressure range with large expansion ratios. Steam cannot exist at T_1 in a cylinder which has cooled to T_2 during the last exhaust stroke. This cooling of the metal seems practically instantaneous, and no engine has ever run fast enough to outrun the condensation process.

The incoming steam at T_1 parts with some heat to warm the metal, and some steam is condensed in that cooling, to be reëvaporated later, and at the lower pressure, getting necessary heat for such vaporization from walls and steam. Now the cooler the walls during the exhaust stroke, the more rapid and complete is the transfer and cooling, since the laboratory and experience show that such transfer is greater, the greater is the *difference* of the temperature of the two masses, one being cooled, the other being heated. If there could be no difference in temperature there would be no transfer and no condensation. The compound, by dividing the pressure range, divides the temperature range, and hence the loss in each cylinder or in both together is less than it would be in either if it were compelled to pass through the entire temperature range of the desired expansion.

2. Any steam condensed in the simple engine cylinder and not evaporated until after the exhaust opens, evaporates without doing work, and both heat and pressure effect are lost. In the compound, such condensation in the high-pressure cylinder, whether due to initial cooling or cooling during an adiabatic expansion, does work in the low, and is sure to be evaporated under the lower tensions of the big cylinder, and not escape as water.

3. The high-pressure cylinder of smaller volume than the low loses less heat from the hot steam by external radiation than if such hot steam were received into the large low-pressure cylinder directly.

4. The mechanical possibilities of the compound enable it to work at higher initial pressures than the simple engine when the back-pressure is fixed, and thus reap the full advantages of expansive working. To increase the pressures in the boilers is to carry more stored energy in a given space; to use higher pressures is to enable each cubic foot or pound of steam to carry more energy into the engine-cylinder, and a given quantity of heat raises the pressure of steam more rapidly after the steam has become a complete gas than it does at lower pressures, when a large part of the heat is absorbed in changing the molecular condition of the water.

317. Mechanical and Practical Advantages of the Compound Engine. The principle of expanding in two or more stages may be applied in many differing arrangements of the necessary mechanism. Mechanical advantages of the type will therefore apply in differing degree to such varying forms, and differing adjustments. Among these advantages are:

1. The high-pressure steam comes upon a piston area at admission from the boiler which is less than in the simple engine of same power. There is less stress on rods and pins and on the piston itself.

2. The clearance volume of the small high-pressure cylinder carries less steam by weight than if steam at that pressure had to fill the same length of the large low-pressure cylinder as in the simple type.

Such clearance steam in the high expands into the low and does work instead of going out with the exhaust.

3. Leakage past valves and pistons is less with the smaller masses of the high-pressure engine. Such leakage again is used to do work in the low. The higher the pressures the more leakage through a given opening in a given time. High heat of steam increases likelihood of such leakage.

4. The equations (paragraf 315) for total expansion $E = Re$, where R is the ratio of the volumes $\frac{V}{v}$, lead to an interesting and valuable feature of the compound. If there is no cut-off whatever in the high-pressure cylinder, and the steam at boiler pressure follows the piston full stroke, so that $e = 1$, the total expansion E is equal to $R = \frac{V}{v}$. That is, by the compound principle expansion is secured while the steam follows full stroke in the small cylinder: or, the advantages of expansive working are secured without the disadvantages of an early cut-off in a single cylinder. If the back-pressure be fixed and the final volume v_2, then the value of the second member of the equation

$$p_1 v_1 = p_2 v_2$$

becomes fixed. The initial pressure p_1 is also usually fixed by the convenient boiler pressure; but for reasons hitherto advanced it should be as great as consistent with safety and convenience. Hence the value of v_1 must be

$$v_1 = \frac{p_2 v_2}{p_1},$$

giving a small value for v_1. If this is a small fraction of the length of the stroke v_2 of a single or simple engine the high pressure p_1 comes on the piston during such a part only of the crank-motion that it is not effective to produce rotation, but the effort goes into the component velocity along the crank itself (paragraf 261) causing bearing friction rather than rotation. In the compound, however, the same expansion is secured and the high pressure exerted through the best crank-angles. This is an argument for the compound system with air and non-condensible gases where the thermal arguments are without significance.

5. This makes the compound waste less applied effort in bearing friction in spite of the increased number of rubbing surfaces. The

heavy pressure is upon smaller piston areas; and conversely, the large areas have only the reduced pressures upon them after some expansion has occurred.

6. The prolonged time for admission of steam through valves and ports to the high-pressure cylinder, as compared to the short period for opening into a simple cylinder to get the same expansion, enables full pressure to be realized in that first cylinder of the compound. The openings would have to be closed so soon in the stroke of a simple engine that the steam would be wire-drawn, or else ports and passages used of such size as to make excessive clearance loss.

Certain compound engine mechanisms derive advantages in addition to the above which are peculiar to them. If the cylinders deliver their effort to the crank-shaft each through its own crank-pin:

7. Such engines avoid the concentration for large engines of great energy on small areas, and enable designers to avoid either excessive lengths or inconvenient diameters for their crank-pins. When the crank-pin becomes of inconvenient diameter with respect to the length of the crank, the angle during which the pressure of steam is available to produce rotation of the crank is diminished.

8. The turning effort is equalized when the compound engine is arranged to have its cranks quartering. The distribution of reciprocating weight over two crank-pins in vertical engines makes balance around the shaft more easy to secure (see Figs. 439, 440). This diminishes the weight of the fly-wheel and the amount or intensity of vibration of the bed-plates.

9. The compound engine gives an opportunity to improve the quality of the steam during the process of expansion when it is possible to use a reheater such as shown in Fig. 393.

318. Disadvantages of the Compound Engine. These are also in two groups. The thermal objections are:

1. The loss of available work or diagram area in free expansion of steam into the passages from one cylinder to another; also friction of the fluid in such passages and condensation therein. These losses show when diagrams of a continuous action of a given mass of steam are superposed to scale as in Fig. 426. and are not incurred in the simple engine.

2. The free expansion or unused energy of the steam expanding in clearances of cylinders after the first.

3. The losses from radiation of heat from surfaces of the additional cylinder and valve-chest.

The objections from the mechanical or practical point of view are:

4. The cost of the cylinders other than the low. In tandem engines

this may mean the cost of piston and cylinder with additional rod, but in cross-compound and fore-and-aft engines it means an additional cost of practically another engine with crank, connecting-rod, cross-head, and the like.

5. The weight and bulk of the additional cylinder adding to foundations and taking up valuable space.

6. The repairs and maintenance and up-keep on such additional parts as are added to the typical simple engine to make it compound.

7. The friction-loss due to the work absorbed by this extra cylinder in operating its mechanism, valve, and the like.

8. The difficulty connected with regulating the power of the engine when the work varies widely, and the first cylinder has measured off a volume of steam adapted to a resistance different from that upon the engine when that volume of steam reaches the later cylinders. This is the difficulty of governing the multiple-expansion engine, except by regulating devices operating upon each cylinder independently.

9. The limitation of power-output because the compulsory expansion makes it impossible to get a larger value for the mean-pressure in an emergency than was intended by the designer. This has been an objection in railway service with two-cylinder compounds, that they did not allow of making up time in storms or delays by making a very late cut-off to get power. The end can be served by fitting a by-pass or small pipe connection from the boiler-pressure pipe to the low-pressure cylinder, and "bleeding" live steam into the low-pressure cylinder through a controlling valve. Since such bled steam to drive the low-pressure piston is necessarily a back-pressure against the high-pressure piston, the engine as a whole benefits only from the fact that the low-pressure piston is larger than the high by a ratio $\frac{V}{v}$. If full boiler pressure is admitted to the low-pressure cylinder the high-pressure piston passes into equilibrium of pressures on both sides of it and goes out of effective action.

10. The high grade of expansion makes it impossible to start the engine from rest if the high-pressure cylinder is on a dead-center or there is no steam getting past that piston to the low-pressure engine. This is met by an "intercepting-valve," a by-pass admitting steam to the low-pressure cylinder directly from the boiler, and by the availability of this large area a powerful starting moment or torque is also secured. The intercepting valve may cause the high-pressure cylinder to exhaust to the open air or to the condenser while it is open, and thus put both cylinders in action for a few revolutions. The intercepting valve may remain under the control of the human intelligence starting the engine,

or the engine may automatically close it and become automatically compound when a certain pressure is reached in the connection between the two cylinders. Such compound engine starts like two simple engines, and the valve is sometimes called a simpling valve, and the operation simpling the engine. It can be used to meet objection No. 9.

11. There has been considerable trouble in compound engines from the accumulation of water in the low-pressure cylinders, particularly when compounding above the atmosphere and using wet steam. The wide range of expansion, the lowered terminal pressure, and the large diameter of the low-pressure cylinder have made this difficulty a very troublesome one in locomotive practice.

It is obvious that the weight to be attached to the above objections is not considered by most designers to be great enough to overbalance the advantages which follow from the principle of compounding.

319. Mechanisms of the Compound Engine. The engine mechanism which is to carry out expansion on the compound principle will embody one of two systems. The one system will be that in which the two pistons will have their motion controlled by a crank or cranks which compel them to be at all times in the same phase of their motion and to pass their dead points and begin and end their stroke and to pass their 90° and 270° points of crank motion together. The pistons at the dead-centers may have completed a traverse in the same or in opposite directions. The other system has the two pistons acting upon cranks which are not in phase, but may be 90° apart or quartering, or 120° apart in the triple engine or three-cylinder compound.

In the first system, the mechanism appears in three general forms:

1. Both pistons on the same rod, tandem, with one piston-rod, one cross-head, one connecting-rod, and one crank.

2. The two pistons each on its own rod, driving a common cross-head or a common beam with one connecting-rod and one crank from that common cross-head or beam.

3. The two pistons each with a complete engine mechanism operating upon its own crank. The cranks may be at 180° apart, or (very rarely) parallel.

The type with the two pistons moving in the same direction together is the simplest form, proposed by Arthur Woolf in 1804 and often called the Woolf type.

The other class compels the use of a receiver between the two cylinders to receive nearly the half contents of the high-pressure cylinder. If the cranks are 90° apart the low-pressure cylinder will be at its dead-center when the high-pressure piston has still a half stroke to make

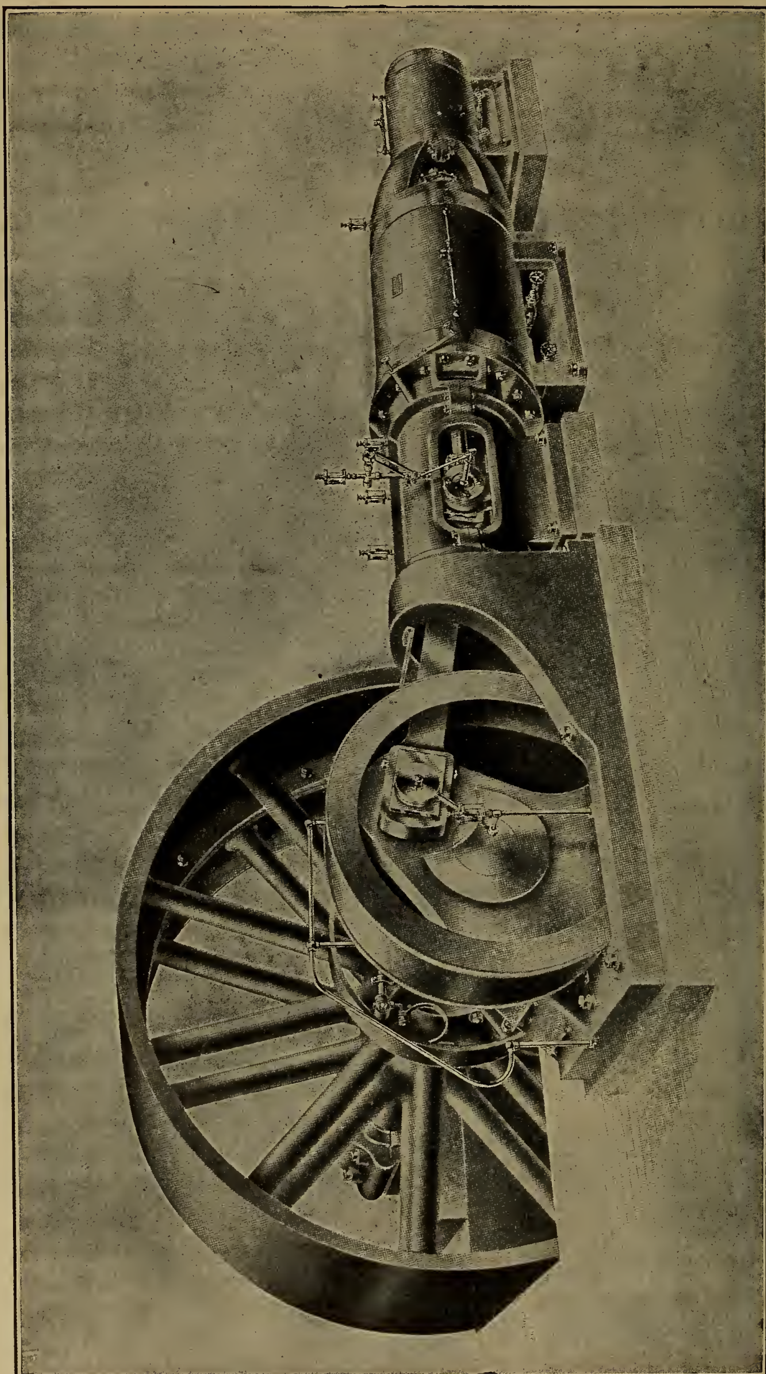


Fig. 430.

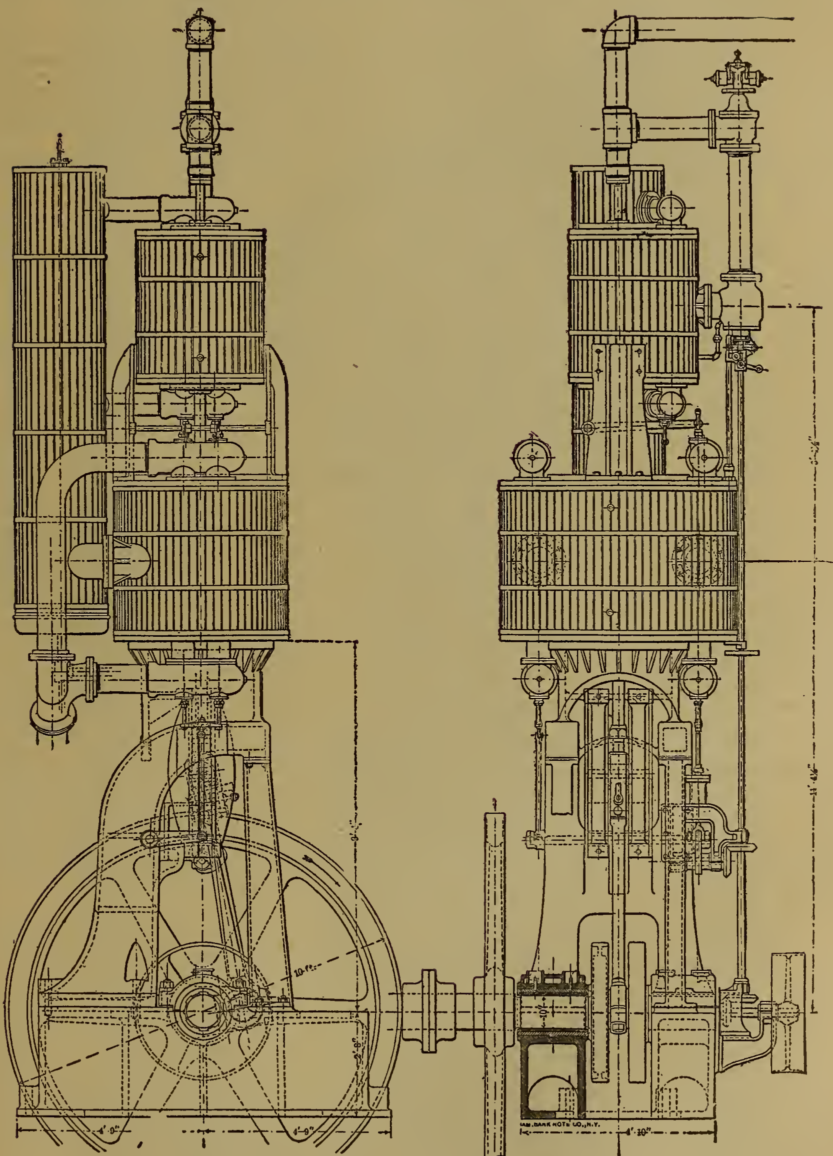


FIG. 431.

before it comes to rest. Hence the exhaust of that high-pressure cylinder must be received in an adequate receiver from which it can pass to the low when the latter is ready for it.

320. The Tandem Engine. The Steeple Engine. Beam Compounds. When the two pistons are on the same rod in a horizontal engine, it will be called a tandem-compound (Figs. 430, 505). When this same engine is preferred with vertical cylinders it is a steeple or cupola engine (Fig. 431).

The feature of the tandem engine is the lowered cost of the type, since the extra cylinder piston and valve with its mechanism are the only parts added to the simple. In the form shown in Fig. 430 a long internal sleeve forms the stuffing-box between cylinders, but it is inaccessible without taking the engine down. In Fig. 430 the cylinders are separated to give access to the two stuffing-boxes now required, and the engine is longer. The steeple compound will rarely be used except in short stroke marine engines because of its inconvenient length. Figs. 430, 431 and 505 show the more usual arrangement with the heavy low-pressure cylinder close to the massive part of the bed plate, and giving the lighter high pressure its support from the low. This is not imperative, and Figs. 433 and 434 show this reversed. The larger diameter of the low permits the valve rod to pass by the high. Fig. 434 is of interest from the presence of a reheater between the steam cylinders, and a device to give out work at the end of expansion instead of a fly-wheel for that purpose. Plungers which swing upon trunnions are forced forward by pressure from the water main at the end of each stroke and help complete the piston traverse under the fall of pressure toward the end of the diagram of work. The objection to the tandem is that the steam from the high-pressure cylinder has to travel the length of the low cylinder outside of its bore to supply the end furthest from the high.

The beam types of mechanism were early utilized by the designers of compounds. The first Hornblower type of 1781 was a beam engine, and his general design of two cylinders side by side acting upon a beam has been widely adapted for water-works pumps. Early simple beam engines compounded by the addition of a high-pressure cylinder were said to be "McNaughted" from the name of the designer who first proposed it (Fig. 435). Or, the two cylinders may take hold on opposite ends of the beam, as in Fig. 391, in which case the two pistons work in phases 180° apart, and very short steam passages are required between the two cylinders. Figs. 393 and 394 show the two pistons connected to opposite ends of the beam, the arrangement used in the great sewage pumping-engines of Boston, Mass., with the beam below

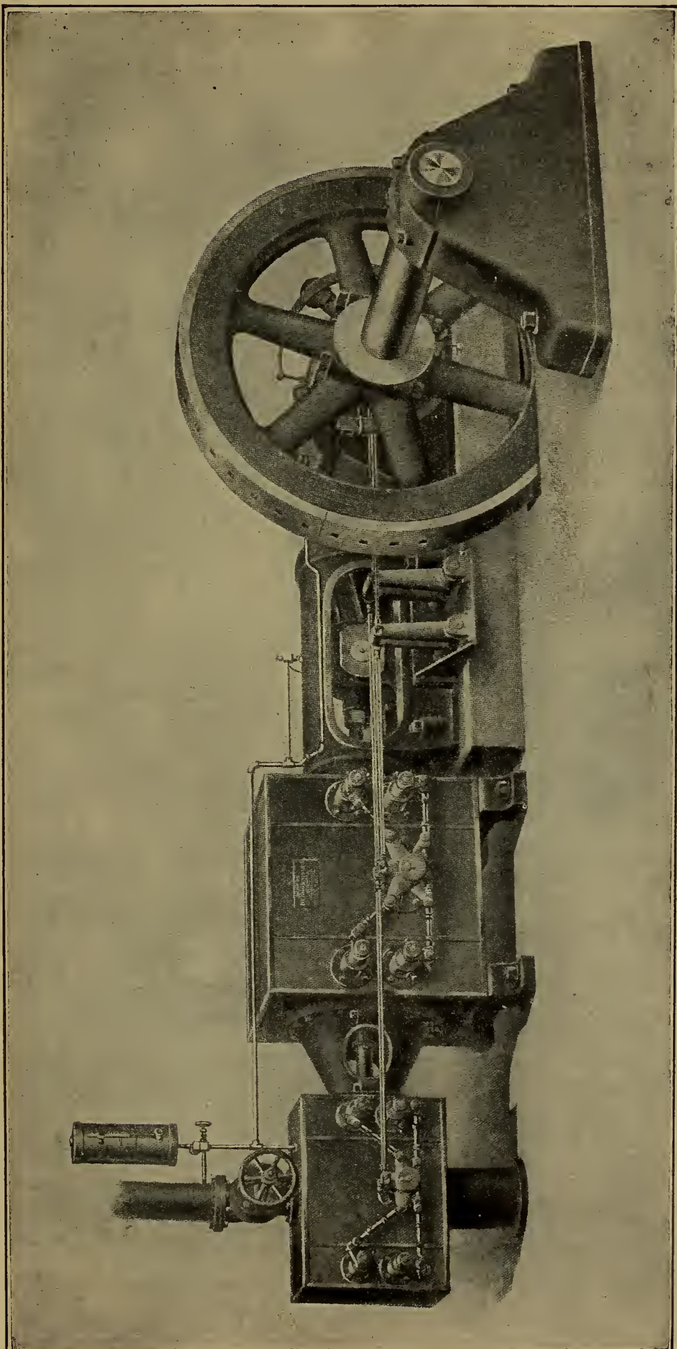


Fig. 432.

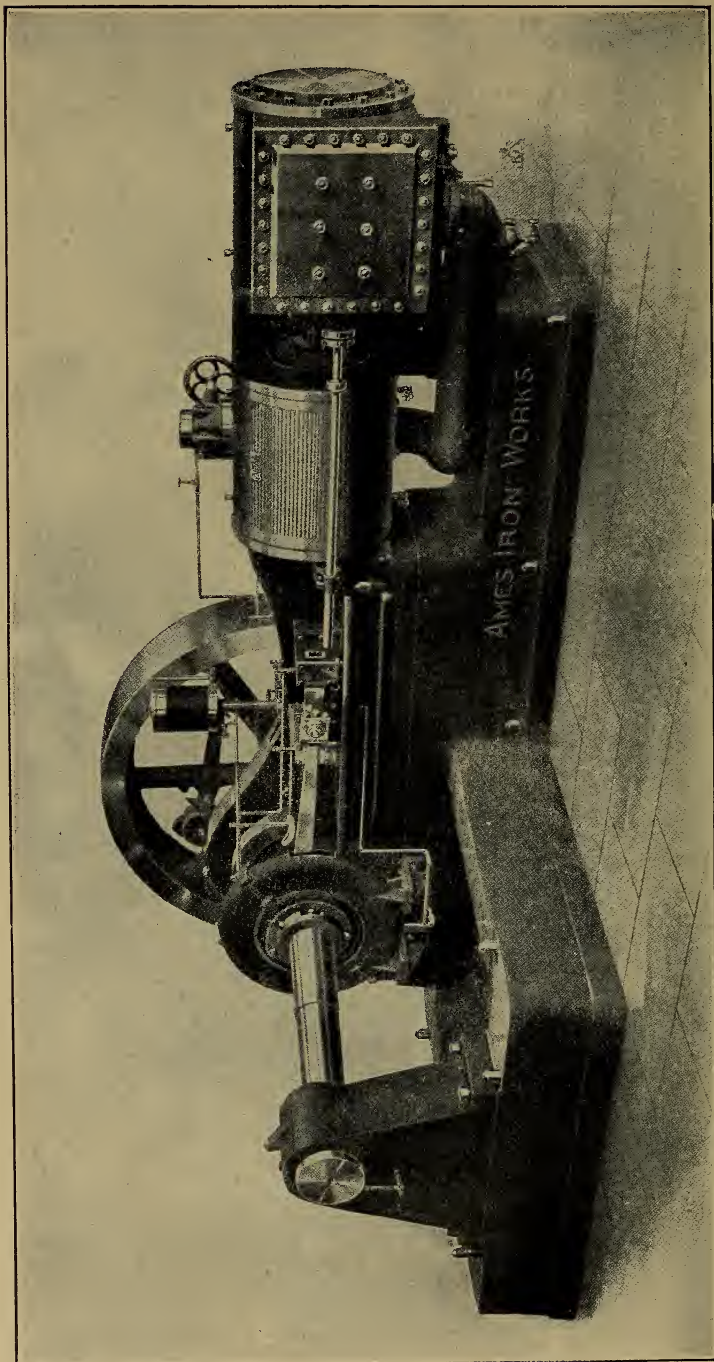


FIG. 433.

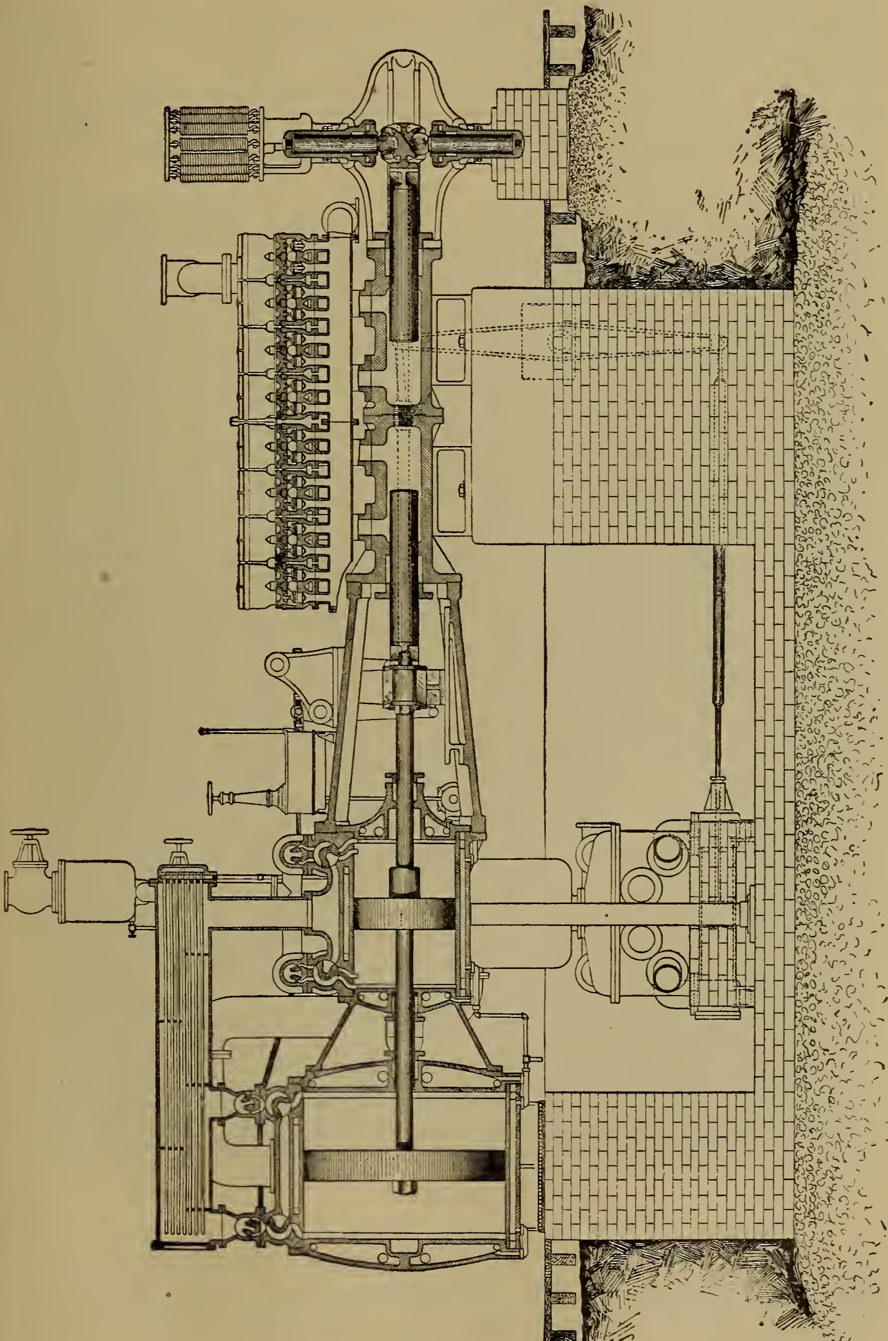


FIG. 434.

the cylinders, and Fig. 392 is a most compact and short-passaged design with the beam length vertical.

The only important example of two parallel cylinders side by side with two separate rods to a common cross-head is in the Vauclain or Baldwin compound locomotive. The difficulty is the varying effort

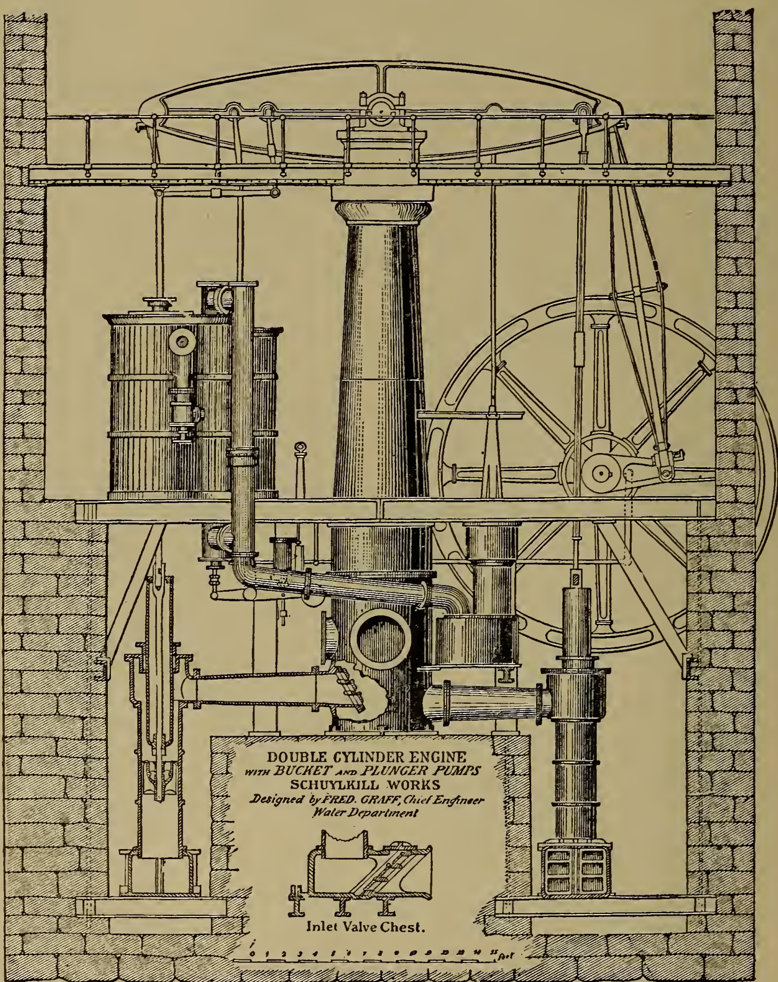


FIG. 435.

upon the two rods except when the conditions of pressure assumed in proportioning the respective areas happen to be met. The valve can be placed between the two cylinders and a simple gear only is required.

The cross-head requires to be very long to prevent "cocking" when starting, and water in the low-pressure cylinder produces a strain out of the axis of the thrust of the connecting-rod and is apt to crack the cross-head.

321. The Fore-and-Aft or Side-by-Side Compounds. The advan-

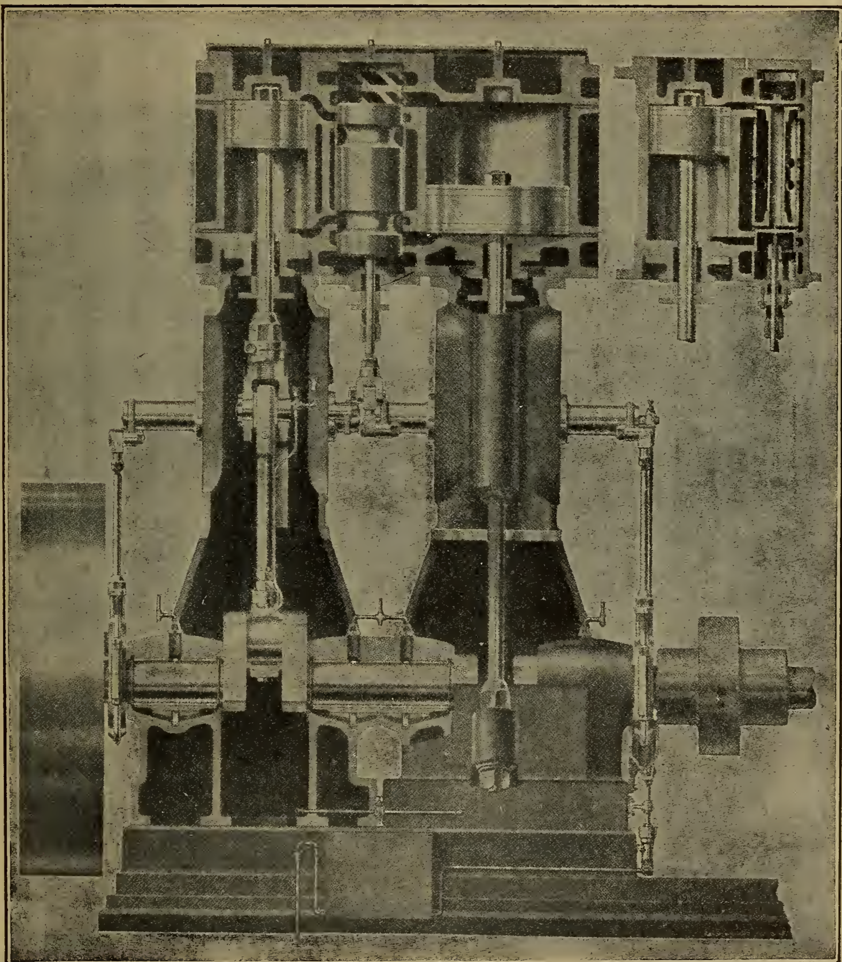


FIG. 436.

tage of getting the work from the pistons to the engine shaft through two crank-pins has given wide acceptance to this type either with the pistons moving together or in opposite phase with cranks at 180°

apart. The latter is the only one in American use, because the weights of the two mechanisms are in balance on opposite sides of the shaft in vertical engines, and this type is practically restricted to such engines. In marine engines the cylinders were one behind the other over the shaft, or fore-and-aft. Fig. 436 shows the side-by-side inverted vertical type and a small detail of the high-pressure valve, and Fig. 437 a developed section of another similar type. These show also what is called the piston-valve. In shallow hulls for light-draft vessels this

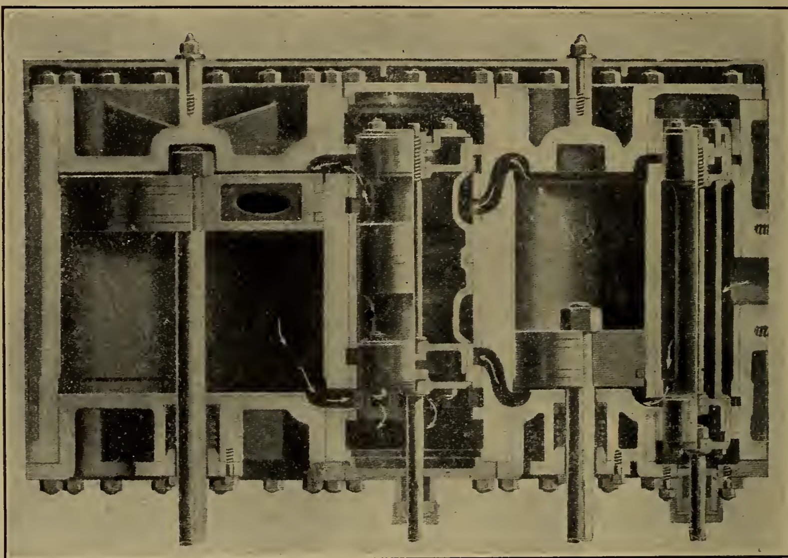


Fig. 437.

arrangement gave less vibration from unbalanced or unequal effort on the up and down strokes and lessened the stresses on the bearing between the cranks. The passages are short as in the beam arrangements for opposite phase of the crank. Figs. 403 to 405 show single-acting compounds of this type.

322. The Cross-Compound Engine. Receiver Engine. The cross-compound engine is the exemplar of the third group when the cranks are quartering or 90 degrees apart. It may be defined as a compound engine with parallel cylinders side by side and cranks at 90° apart: the space between the cylinders being large enough to take in a pipe or chamber to act as a receiver of the exhaust-steam from the high at such times as there is no low-pressure cylinder-volume to receive it. This happens at and near the dead-centers of the low, at which time the high-piston is moving at its most rapid rate, sweeping the exhaust out from the previous stroke. Cross-compounds quite usually have a cut-off valve action in the low-pressure as well as the high, for which there has been no room nor call in the previous types.

Cross-compounds are either horizontal (Figs. 438, 504) or vertical (Fig. 380). It is very usual to put the fly-wheel or generator between the two cranks. It is not clear whether it gets its name from being a cross between the two previous crank arrangements, or because the steam crosses over the space between the two cylinders (Fig. 438). The quartering cranks give more equable turning moment, do not compel all the reciprocating masses to be accelerated at once or in one quadrant, and lessen the fly-wheel weight. The horizontal vertical engine when compound (Fig. 387), as it usually is, reaps these advantages. The inclined engines of Figs. 384 to 386 and 503 are also cross-compound mechanisms.

323. The Triple-Expansion and Quadruple-Expansion Engine. In the triple-expansion engine, the cranks are usually at 120° apart to equalize turning effort and the acceleration influences. These mechanisms are horizontal or vertical, or show the combined horizontal and vertical. The marine use of the triple engine has given a vogue to the inverted vertical type (Fig. 381) and the multiple pistons on opposing cranks enable balance to be aimed at by making the small piston and attached weights more massive than needed for strength. But there will be a period in each revolution when two of the pistons will be descending and only one ascending, and this inequality must be provided for. Moreover, the crank-shaft with multiple cranks is a costly and troublesome proposition, so that even with quadruple expansion, for economy of room and to shorten the crank-shaft only three cranks will be preferred. Figs. 439 and 440 show arrangements which have been used to overcome these difficulties.

324. The Compound Locomotive. The locomotive is a compound above the atmosphere, and its fuel economy diminishes the weight for the tender in a given run. It is at its best in long steady runs with infrequent stoppages and at efforts near the maximum. Frequent stops and starts either in marine or locomotive conditions put the compound under its worst disadvantages, and the heavy draft for power in accelerating trains after stops is best met by the simple engine.

The mechanical arrangement appears in several differing types. First the two-cylinder compound, which is a cross-compound, with a reheating action in the cross-over pipe in the smoke-box. Usually the high pressure is on the left side and the low pressure on the right. The diameter of the low is limited by lack of room laterally. The

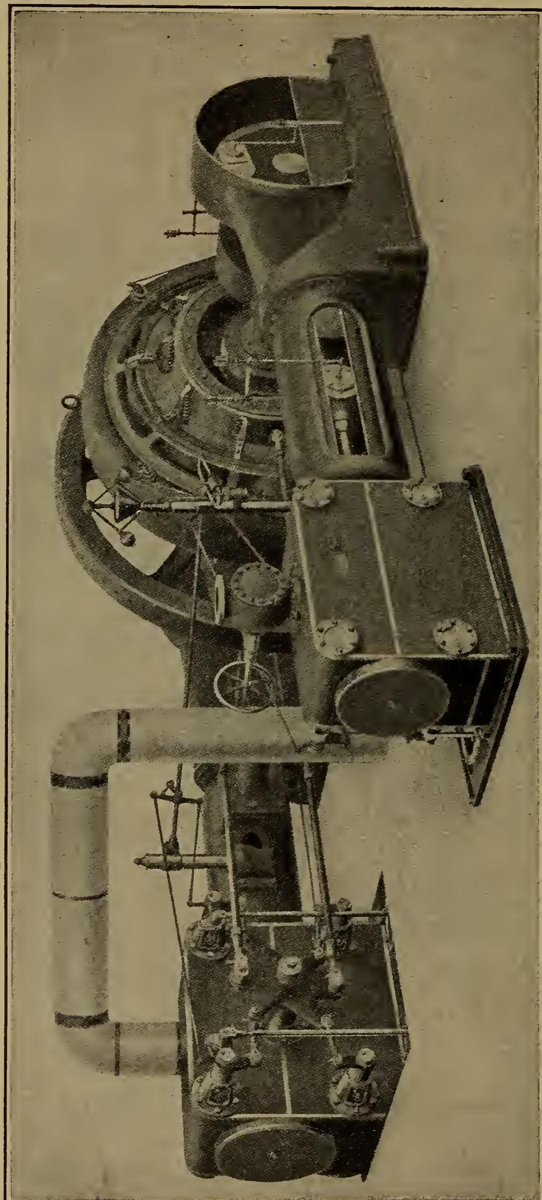
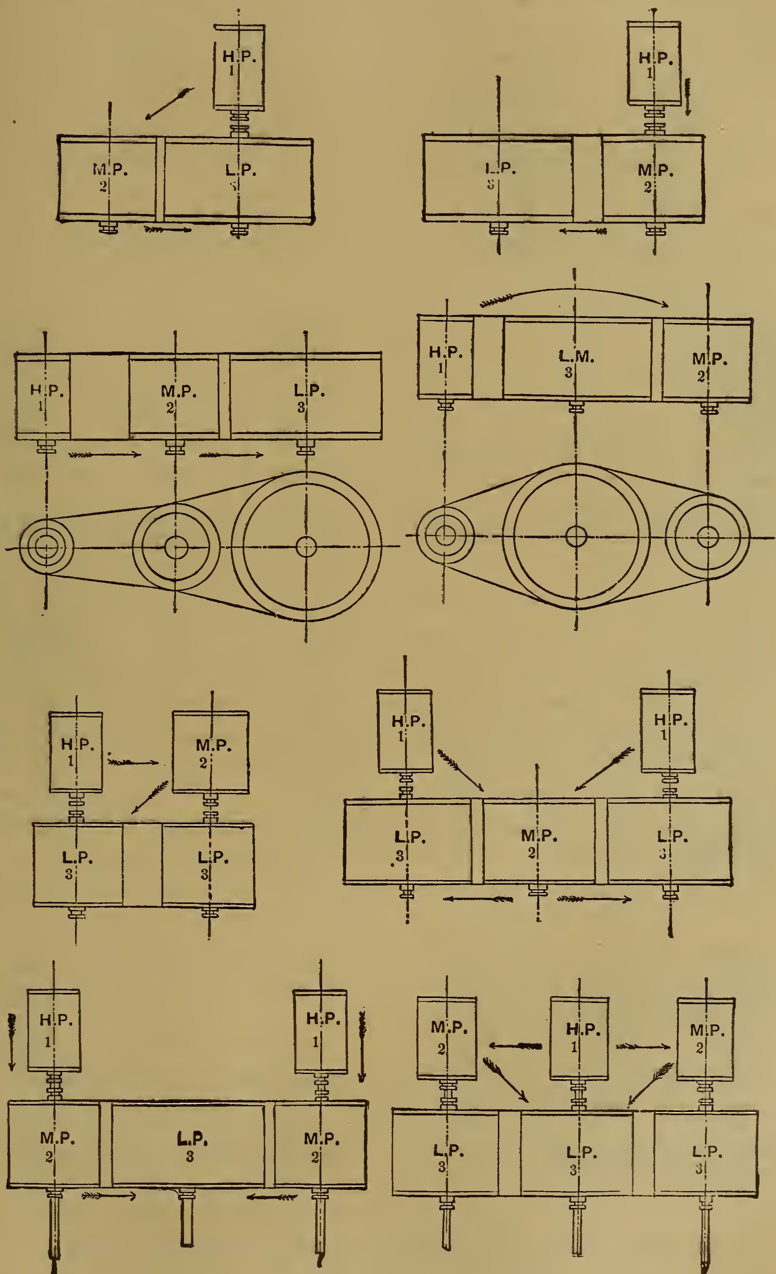


Fig. 438.


H.P.
1
M.P.
2
L.P.
3
H.P.
1
L.P.
3
M.P.
2
H.P.
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M.P.
2
L.P.
3
H.P.
1
L.M.
3
M.P.
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M.P.
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L.P.
3

FIG. 439.

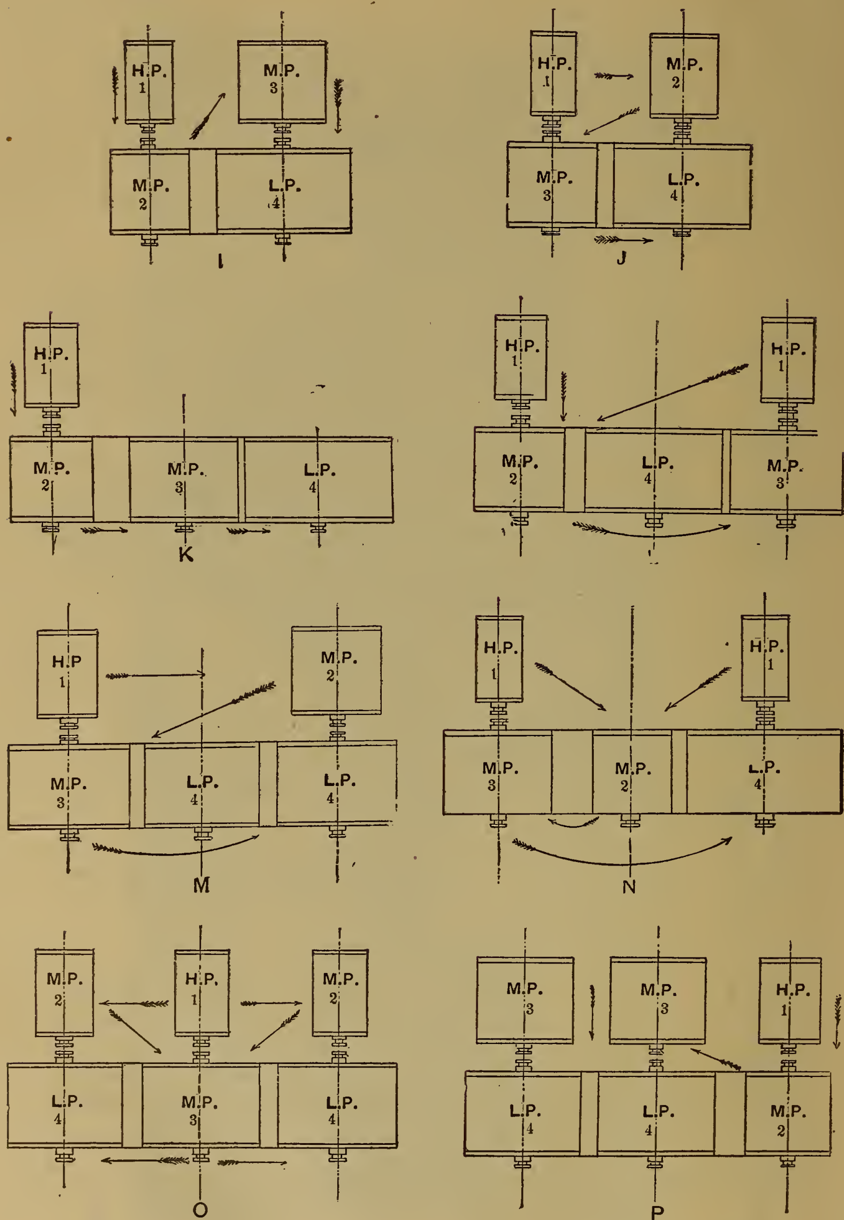


Fig. 440. — Grouping of Cylinders in Quadruple-Expansion Engines.

intercepting or starting valve will be either automatic, starting the engine simple, and changing over as pressure accumulates in the receiver; or it will be under the control of the engine-man; or the operation of the cut-off lever in increasing the rate of expansion after the starting inertia has been overcome will compound the engine. The successful steam motor vehicles are two-cylinder compounds, with the intercepting or starting-valve under the control of the operator.

The second arrangement has three cylinders. This is a more prevalent European design, and is not used in America. Usually the two cylinders which form the low-pressure stage are outside the frames in the position of the usual outside-connected engine, and the high-pressure cylinder is between the frames under the smoke-box. This central cylinder drives the forward driving-wheels by means of a cranked axle, and the outside cylinders drive the rear or trailing wheels by outside crank-pins. This arrangement is also sometimes reversed.

The third arrangement is the four-cylinder compound, in which the high and low-pressure cylinders are attached in pairs on each side of the engine, in the common cylinder location, to a common cross-head, from which the usual driving-rod passes to the crank-pins and wheels. The advantage of this is that the engine works compound just like a simple engine and with the same valve mechanism, except that a simpling valve is required so that boiler steam may be let into the larger cylinder through a by-pass for starting. By using a pipe of small cross-section for such by-pass connection, the pressure on the areas of the two pistons is not allowed to be so different as to cause undue stresses in the cross-head. The high-pressure cylinder is either above or below the larger one, and both move together in the same direction. Hence the construction of the valve must be such as to pass the steam diagonally from one end of the high-pressure cylinder to the other end of the low. This type requires no receiver and permits no reheater.

In the fourth type the two cylinders are tandem on each side, on a common piston-rod on each side. This type avoids cross-strains on the cross-head, and valve-gear and starting devices are the same as in the preceding.

The fifth type has four cylinders side by side, a compound engine on each side of the locomotive, and with the four cranks 90° apart in one of two systems. In both the high-pressure pistons drive a cranked axle on the forward or main drivers, the two cylinders being between the frames, while the low-pressure cylinders are outside the frames, and their pistons drive the rear or trailing drivers by the usual outside-connected mechanism. The pairs of drivers are coupled with side-rods externally. In one system the two cranks of each side of the locomotive are 180° apart, and this pair of cranks 90° from the other pair. In the other system each side has its cranks 90° apart, the two highs and the two lows being each 180° from the other. This latter makes a better balance in running. The advantages of the compound locomotive beyond those which it enjoys in common with any compound engine are the result of the lower terminal pressure at which the exhaust escapes. This is favorable to economy in the fire-box, because the fire is less torn by the pulsation of the exhaust, which causes the draft, and in cities the diminished noise of the escaping exhaust makes the locomotive less of a nuisance and the cinders are less thrown. The increased number of parts, with increased cost and time for repairs to them, and the inaccessibility of parts placed under the boiler, have prevented the rapid or wide introduction of the compound except for continuous through runs of heavy trains. Compounds do not make up time easily after road delays.

325. Compounding above the Atmosphere. The general discussion of Chapter XVII, which showed that the greater the back-pressure the less the advantage from expansion, should have made it obvious that the compound system as a means to secure and favor expansion is not at its best in non-condensing engines. With a given initial pressure or the less the total area for the work diagram the more significant are the losses due to the compound, due to such back-pressure value. The distribution of work effect is present still as an advantage, and in pumping and similar cases the following of the full pressure through the stroke of the small cylinder is a mechanical advantage. Where the exhaust-steam is to be used for heating, and requires heat and pressure to be valuable for this purpose, the divided expansion may still be advisable for this reason, while without significance in heat economy.

With compressed air motors, as for street railway service, where the demand that great store of energy be carried in pressure tanks of small bulk and hence under high tension, and where the exhaust tension must be that of the atmosphere to prevent noise upon the streets, the compounding of the engine is desirable for mechanical reasons. There is no heat gain from the process, except that from distributing an inconvenient temperature drop in the expansion process; but in starting and in equalizing stresses over piston areas the system has proved itself worth while.

326. The Receiver and the Reheater. A rule of very general acceptance is that the pressure in the receiver should not rise to nor exceed one-half that in the boiler or on the forward driving side of the high-pressure cylinder. In the cross-compound, the most unfavorable case would be met when there was no cut-off or expansion in the high-

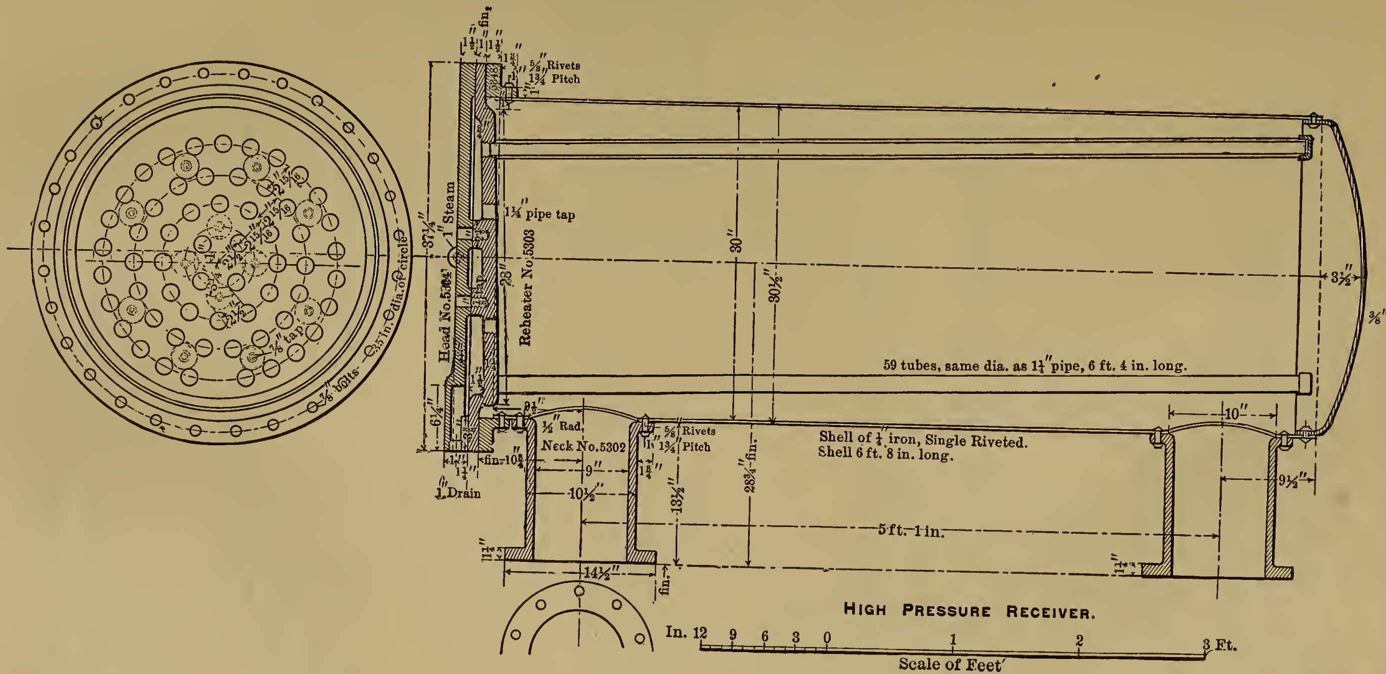

High Pressure Receiver.
59 tubes, same dia. as 1¼" pipe, 6 ft. 4 in. long.
Shell of ¼" iron, Single Riveted.
Shell 6 ft. 8 in. long.
Reheater No. 5303
Head No. 5304
Neck No. 5302
1¼" pipe tap
1" Steam
1" Drain
⅝" Rivets
1¾" Pitch
½" Rad.
⅜" bolts
35 in. dia. of circle
⅜" tap
5 ft. 1 in.
30"
30½"
28¾" fin.
37¼"
28"
14½"
10½"
13½"
10"
9½"
9"
6¼"
3½"
⅜"
Scale of Feet
In. 12 9 6 3 0 1 2 3 Ft.

FIG. 441.

pressure cylinder: one-half the volume of the high-pressure cylinder would therefore have to be provided for at boiler pressure, before the valve to the low pressure opened to receive steam. The limit of pressure would therefore be never reached if the receiver volume was equal to that of the high-pressure cylinder. The ratio varies between 1 and 1.5. In triples it can be less; and in most designs the valve chamber of the low-pressure cylinder, or the pipe connecting the cylinders, gives all necessary volume without inconvenient fluctuations of pressure for the low cylinder, or undue drop or free expansion from the high.

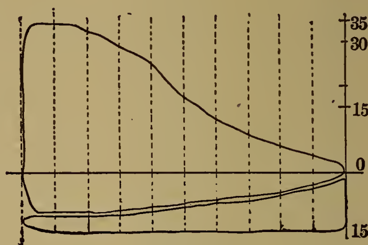


FIG. 442.

Fig. 441 shows a receiver design fitted to take the heating tubes to make such a receiver into a reheater or regenerator to improve the heat quality and dryness of the steam in transit from one cylinder to another. These heating tubes are supplied with hot dry steam from the boiler, always at higher temperature than the used steam which surrounds them. The reheating tends to close up the gaps in the combined indicator diagram. Figs. 387, 393, 434 make obvious the use and construction of such reheaters, which are a means of adding distinct economy. Reheaters require to be drained of condensation, and of course there is also condensation in the heating coils (paragrafs 243, 250).

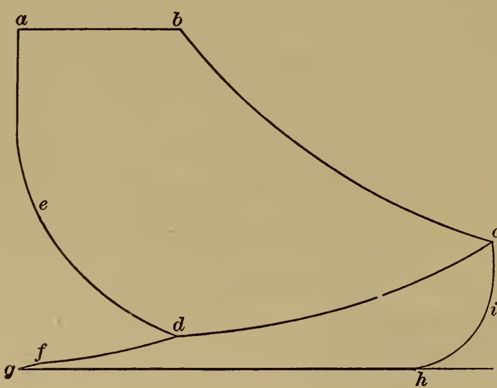


FIG. 443.

327. The Work and Indicator Diagram of a Compound Engine. The discussion of types of compound engine in paragraf 319 and those which follow shows that the expansion process of a given volume of steam will give rise to two differing types of indicator diagram for such combined cylinders. In the one type shown in Fig. 442, taken from a slow-speed low-pressure compound with little or no compression in either cylinder, the diagram of pressures upon the low-pressure cylinder should have its upper line the complement of the lower line of the high-pressure diagram, because the larger piston is being driven by the steam which escapes from the smaller cylinder. Any discrepancy or lack of harmony between these lines in a Woolf engine without receiver indicates losses from friction, condensation, or unnecessary expansion in the

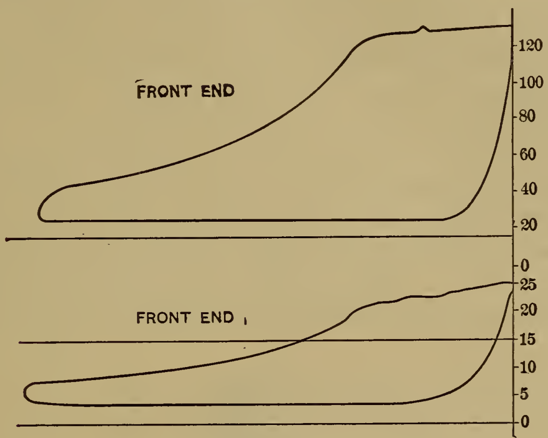


FIG. 444.

clearances or passages between the two cylinders. In the receiver engine the steam in that receiver is to be treated as a steam spring receiving and storing work from the high-pressure cylinder and giving it out unaltered to the succeeding stroke of the low-pressure cylinder.

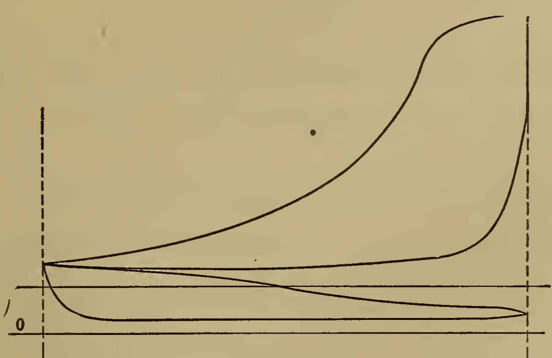


FIG. 445.

Any discrepancy between the lines of the two diagrams for the two cylinders indicates a drop caused by free expansion from the high-pressure cylinder into the receiver without doing work in driving the low-pressure piston, as well as the losses from friction and condensation present in the other type. Such loss by free expansion is not usually recovered, and should be guarded against if the conditions of operating the engine permit it.

For higher speeds, when accelerations are to be provided for by compression, the combined diagram takes the form of Fig. 443. Exhaust from the high ceases at *d* and compression begins, accompanied by a gradual drop of pressure in the low, and loss of indicated work, but not necessarily of crank-pin work, since the masses of the fast moving parts are giving stored energy upon the pins.

When the engine is of the receiver type, on the other hand, the volume of the latter may permit the back-pressure line of the high-pressure card to be practically horizontal, and the cards will be like Fig. 444 when separate, and result in a Fig. 445 when superposed upon the same pressure scale. There are then two procedures possible. One is to get the area of each card and its M.E.P. by usual methods, then

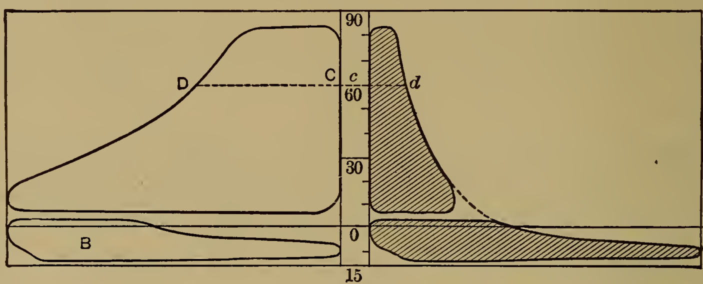


Fig. 446.

before adding them together to multiply the M.E.P. of the high-pressure card by the ratio $\frac{v}{V}$ of the two cylinders to reduce the card of equal length to the same work scale as that of the low-pressure card. Then add the two together for the total mean pressure considered as exerted over the large diameter or last cylinder to get the horsepower. The other plan is to superpose the two cards on a common vertical scale, and related each to a proper allowance for their respective clearances. Then with a proportional dividers or other measuring device, reduce the horizontal measurements of the high-pressure card at several points of pressure by the ratio $\frac{v}{V}$ and draw the resultant expansion curve through the points thus determined. Thus, if the ratio of the cylinders is as 1 is to 4 let *cd* on the right hand of Fig. 446 be laid off one-fourth of the actual measurement of the card, which is *CD* in the original diagram. It will thus appear that the reduced diagram will now represent on the same scale as the low-pressure diagram the work delivered upon its crank-pin, and we therefore have the net indicated work done in the two cylinders presented by the combined diagram shown in Fig. 446 at the right hand. This indicates to the eye the continuity of the expansion in a compound engine, and shows the space for loss between the two cylinders which would not be present if the expansion were in one cylinder only. It is this area which designers of continued-expansion engines are to reduce, and it is in spite of this loss that the compound engine is superior to the simple. Care must be taken in combining the diagrams of effort of steam in the two cylinders to put them in their right relation to each other vertically. This is accomplished by means of properly relating each to the line of zero volume or of no clearance as appears in Figs. 425, 426.

To find the steam in horsepower per hour from such a card, it will be apparent that if

S denotes the weight of steam per horsepower per hour,

w denotes the weight per cubic foot of steam at the mean pressure from the card,

v denotes the per cent of volume of the cylinder or piston displacement filled hourly with steam when the pressure corresponding to w was measured or computed,

Then

$$\text{M.E.P.} \times S = 19{,}800 \times w \times v.$$

The value for v may be taken from either card, and its corresponding pressure. The two values from the two cards will usually differ slightly in an actual case from changes of load or governor action. The mean may be safely taken. The M.E.P., given in pounds per square foot, should be multiplied or divided by $R = \frac{V}{v}$ to correct for the volumes of the two cylinders and reduce to a common standard. This is the steam weight without condensation loss, or allowances for leakage, or for the quality of the steam.*

* The derivation of this formula is:

$$1 \text{ H.P. per hour} = 33{,}000 \times 60 = P_m \times LA \times N \times 60$$
$$= P_m V$$

But only $\frac{vV}{100}$ is filled at P_m, and $S = \frac{w \times v \times V}{100}$

Hence $$P_m S = 19{,}800 \times w \times v$$

P_m must be in pounds per square foot.

328. Concluding Comment. It has been the effort and purpose of the preceding paragrafs and chapters to develop from the simple typical reciprocating steam-engine those forms and arrangements of it which have sought superior economy and efficiency, and particularly in the larger units.

It is now desirable to return to the fundamental classification of paragraf 254 of Chapter XV and examine the engines of the second group. These are engines of the pressure class, but in a continuous rotary group having no reciprocating masses in the mechanism. These are called rotary steam-engines and will form the topic of the next chapter.

CHAPTER XIX.

THE ROTARY STEAM-ENGINE.

330. Rotary Steam-engine defined. A rotary steam-engine is one in which the piston which receives the driving-pressure of the steam does not travel in a straight line along the axis of the cylinder, but revolves in a plane at right angles to such axis. Such piston or pistons — there are usually several in the cylinder — form or are attached to the crank-arm by which the engine shaft is revolved so that their rotary motion around the cylinder axis revolves the crank directly. The rod which was formerly the piston-rod of the reciprocating engine and passed steam-tight through a stuffing-box in the cylinder-head is now the revolving shaft of the engine: the center of pressure of the driving fluid on the piston area is the center of the crank-pin. Piston-rod, cross-head, and connecting-rod, with their mass and inertia, have disappeared, and with them their cost of construction, their joints to be maintained, and their lubrication. The power is applied directly to produce rotation.

The piston will no longer be of circular section but probably rectangular. It must fit steam-tight to the surfaces over which it sweeps, and by reason of unequal expansions it cannot be fitted without packing strips. It must be emphasized that the rotary engine is a pressure-utilizing motor, to distinguish it from the steam-turbine, which it outwardly or superficially resembles. The area to be used in the horsepower formula is the mean or average net effective area of any one of the pistons exposed to the forward action of the steam during that period when pressure from the pipe is acting on it. As soon as a second piston comes into action, the first one is no longer effective, but has the same pressure behind it as it has in front. The piston-speed is the path of the effective center of pressure of such a piston in one revolution multiplied by the number of revolutions per minute. By taking one piston area through a complete revolution, the necessity is avoided of finding out during what fraction of a revolution each piston is effective, and multiplying by the number of pistons. Each of n pistons is assumed to be effective during $\frac{1}{n}$th of the travel of one revolution.

331. Types of Rotary Engine Mechanism. An inspection of Fig. 450 presenting the basal concept of a rotary steam-engine will make clear the difficulty which its apparent simplicity has introduced. Such a mechanism as Fig. 450 would not turn over, because the pressure from the inlet I to produce intended clockwise rotation upon piston a is equated by an equal pressure upon c to rotate contra-clockwise. The steam between pistons a and b is inert, having equal pressures forward and backward over equal areas; while piston b has on one side the pressure prevailing when this space was connected to I and on the other the exhaust-pressure. Hence the net effect would be to drive c contra-clockwise, or on the wrong direction, if the pressure in E is lower than in I.

If, however, the pistons were in the angular relation of Fig. 451 due to a partial rotation and the eccentric ring of that figure were absent, the pistons b and c are in equilibrium of pressures, but worse than

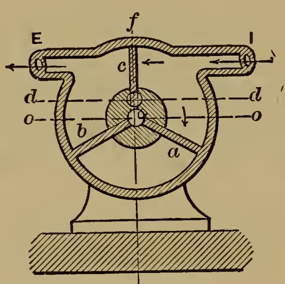


Fig. 450.

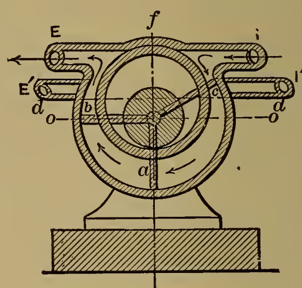


Fig. 451.

this, a passage has been opened directly between inlet I and exhaust E, and steam blows through to waste and does no driving of the engine. Hence a scheme must be found to accomplish the double purpose of separating the exhaust and inlet pressures at all positions of the pistons; and to compel continuous motion of the pistons through the longer arc between the points of inlet and exhaust. This device is called an abutment, and takes many forms. Rotary engines are mainly differentiated from each other by the form of their abutment. In Fig. 451 the abutment is a revolving ring or cylinder, with its center of figure eccentric to that of the path of the pistons. The pistons fit the ring steam-tight with a rolling or rocking joint, and the ring fits both faces or heads of the cylinder closely enough to stop leakage across around its ends. The pistons in passing the point f retreat within the ring surface, but protrude on passing the contact point, and receive the pressure as at c in Fig. 451. As here shown, in the desire to get a considerable path for the piston under pressure, the area exposed, or the value of the turning moment, is small. If the inlet is lowered as in the dotted lines at I', the ineffective arc or length of piston action is decreased but

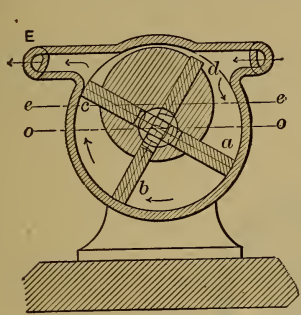


Fig. 452.

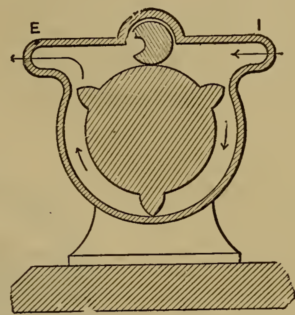


Fig. 453.

the effective area is increased unless, again, in an effort to diminish clearance loss behind *c* in the position shown, a pressure is acting negatively on it to turn the shaft the other way.

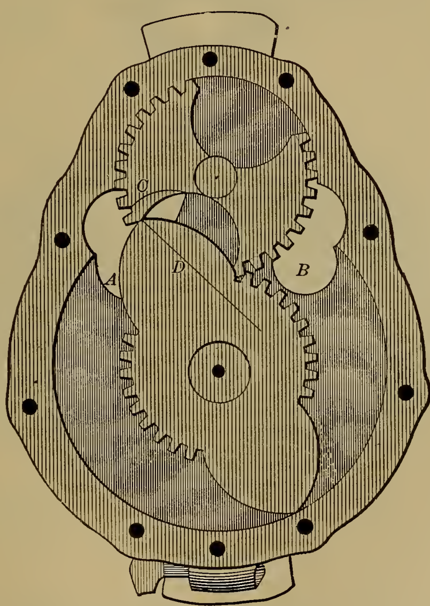


Fig. 454.

Suppose now as in Fig. 452 the abutment ring is made solid, and instead of three pistons, they become four in an effort to lessen the volume of steam filling wastefully the clearance space between pistons *a* and *d*. The pistons can be forced out to their work by springs, doing away with packing strips on the end faces, but the back or negative-pressure trouble appears unless the piston can be kept hidden in the abutment, or such negative pressure on *d* be neutralized as in the sketch, with attendant excessive clearance.

In the forms of Fig. 451 and 452 the center of the abutment is within the travel of the piston tips. It may therefore be called hypocentric.

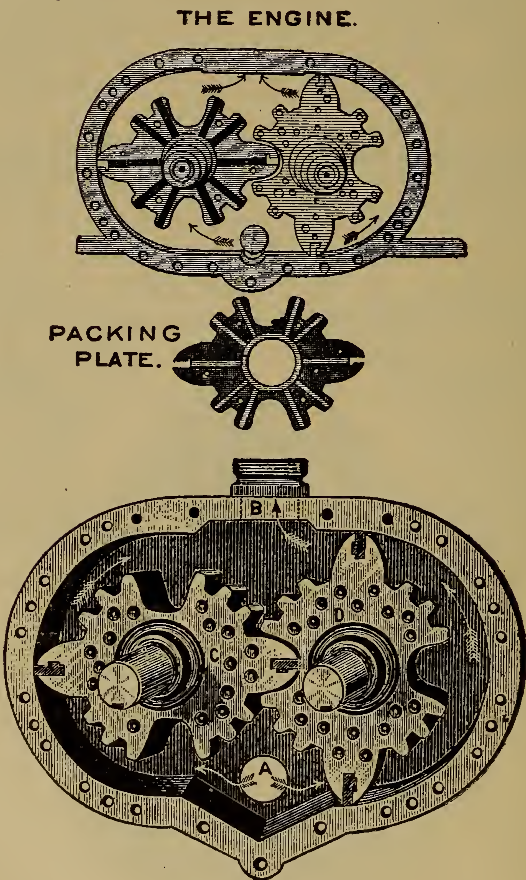


FIG. 455.

In Figs. 453 and 454 are shown two types in which the abutment is outside the piston paths, or its center is epi-centric. The design of Fig. 453 is in effect what is called in kinematics a chamber-wheel gear with a diametral ratio of 1 to 3. If the diametral ratio be 1 to 2 the form becomes that of Fig. 454. The curves of closure are epi- and hypocycloidal with the curves traced by a common describing circle. The other type is kinematically a turning-block slider-crank chain.

If now the abutment organ turning with the crank in the epi-centric class be made so as to have the same diametral ratio, or that of 1 to 1, the form of Fig. 455 results in which the two organs function alternately as piston and abutment. The pressure inlet is at *A* in these two latter figures. The unbalanced pressure upon the piston or crank-organ compels rotation of the shaft, and the two shafts of the piston and abutment element are kept in phase by gears outside the casing.

If the rotary engine is not to be reversed, it requires only a throttling valve upon the pipe connected to the inlet *I*. If it be desired to reverse it on occasions, the form of Fig. 456 shows the simplicity of the required valve-and-seat construction. The sliding valve when thrown to the left by the rack and pinion geared to an external lever, compels a clockwise rotation. To reverse the slide valve is moved to the right; and in mid-gear, with both inlet and outlet covered motion stops. To pack the sliding-contact surfaces, in most cases packing strips are provided which are forced outwards by springs acting radially. The shape of the strips must be such that when the piston leaves the casing towards the middle the strips shall not drop out. For the packing against flat surfaces one design uses radial strips similarly pressed against the flat heads by springs, and the other depends upon counter-bored holes drilled a short distance apart, parallel to the axis within which condensation and lubricant will be caught, which will serve by their capillary action to prevent any considerable escape of live steam.

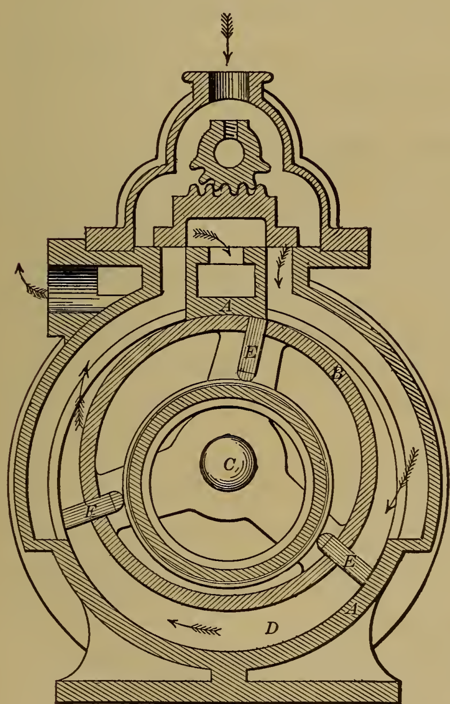


Fig. 456.

It will be apparent that almost any rotary engine can be reversed in principle and become a rotary pump by applying power to the shaft by outside mechanical means and admitting water through the passages with which the steam enters in the rotary engine. This convenient peculiarity has given rise to a formerly popular form of steam fire-engine, in which the steam and water pistons are arranged on parallel shafts properly geared together. Such an engine requires no valves on its water end, and the water used can be full of impurities and solid matter without great inconvenience.

In efforts to compel expansive working of such rotary engines, sliding cut-off valves have been introduced but they have not been significant or satisfactory. Governing will always be by throttling, either on the inlet or on the exhaust. Attempts to use a sliding radial plate as an abutment for pressure or as a partition between the admission and exhaust-pressures have failed from the demand for excessive effort to accelerate the mass of such a plate at the speeds required to get it in place soon enough to do any good; and to introduce any reciprocating mass whatever is to sacrifice the primary excellence of the rotary principle.

332. Advantages of the Rotary Engine. The arguments to be urged in favor of the rotary-engine principle are so many that the skill of innumerable inventors has been continuously directed towards their design. Among other advantages are:

1. These engines are adapted for uses where human power will be required or used to govern them frequently and quickly, as in hoisting and quarry engines; or where variation in resistance is a variation in speed, as in marine work; or where sudden cessations of resistance at speed are unusual or impossible, as in motor-cars and motor-boats.

2. The effort of the steam is applied directly without intervening mechanisms for conversion of the motion. This avoids their attendant friction, their costly fitting, and probable lost motion.

3. There being no reciprocating parts, there is no inertia to be overcome at the beginning of the stroke, with the attendant consumption of energy required to accelerate them.

4. The engine has no dead-center, but will start from rest in any position.

5. Absence of reciprocating parts makes it easy to run the shaft at the highest speed. This has attracted designers of steam-driven dynamos to use this type of engine.

6. The engine becomes very compact from the absence of converting mechanism, so that it occupies little room.

7. The engine has either no valve-gearing, or that which it has is of the simplest character.

8. These features, and the absence of expensive mechanism, make the engine cheap to build and therefore usually cheap to buy.

9. Absence of reciprocating-rods and dead-centers results in a construction in which the presence of condensed steam in the cylinder does no harm. It does not stop the engine from turning, it cannot endanger the cylinder-casting, the engine can be started, even if under water, by simply opening the valve which admits pressure to it; it will start with solid water.

10. Its incased construction and the above peculiarity particularly adapt it for out-door service and exposed places. Weather does it no harm, and its protection from outside injury makes it a serviceable quarry motor.

11. It requires no skill to handle it. If constructed to be reversible as in Fig. 456, it can be reversed from a distance by simple rope and weight.

333. Disadvantages of the Rotary Engine. The objections to the rotary engine are both practical and inherent. The practical objections belong to the difficulty of satisfactorily packing surfaces which do not move through equal spaces in equal times. Those parts farther from the axis move through a longer path in a revolution than those nearer to the axis. The wear from abrasion is therefore greater at one part than another. When the packing-strips have become somewhat worn, leakage ensues, and a noisy rattle from looseness of the fits. A second practical difficulty is the expense connected with proper lubrication of such engines, and the difficulty of taking care of excess of oil rejected by the exhaust. If efficiently lubricated, they consume an excessive amount of oil.

The inherent objections to the rotary engine are:

1. The presence, in the volume to be filled by live steam from the boiler, of an excessive waste space which has to be filled by steam at each revolution, which steam is exhausted without doing all the work there is in it. This corresponds in reciprocating engines to an excessive clearance.

2. The very continuity of the action of the steam upon the rotating pistons precludes the possibility with the single rotary engine of working the steam expansively, so that when the steam leaves the motor it shall have become largely reduced in temperature and pressure by doing work with increase of its initial volume. The expansion is from the boiler and the water in it, and not from the actual volume received by the engine for the work of one stroke. In other words, the rotary engine is a non-expansive engine. These two difficulties make the rotary engine uneconomical.

3. It is difficult to design the rotary engine for large horsepowers: First, because the structure becomes inconvenient the moment that large areas are desired, so as to make a value of PA in the horsepower formula a large factor; second, because it becomes difficult to secure the condition of high piston-speed in feet per minute unless the diameter of the casing be made so large that the difficulties both practical and inherent become nearly insurmountable and the advantages of the rotary principle are sacrificed. Third, because to secure a large starting or turning moment from rest the engine has to be made long parallel to the axis of the shaft; and when this is attempted structural difficulties increase, and the advantage of weight and bulk over the reciprocating engine is lost.

The economy which a single rotary engine cannot secure from its inability to work the steam expansively has been sought and secured in a degree by arranging rotary engines in series upon a shaft, so that the steam rejected from number one engine becomes the driving steam for motor number two of larger volume. By this means the steam when rejected is at more nearly the pressure and temperature of saturated steam at atmospheric pressure than can be attained with the single rotary engine. This does not even yet secure the advantages of expansion in the piston engine, because such increase of volume as is secured by simple enlargement of chambers without the doing of work upon a mechanical resistance is of the nature of free expansion, and results in no economy. The turbine to be discussed in the next chapter has so transcended the possibilities of any rotary pressure engine in the large sizes, that the importance of the rotary is becoming steadily less and less, even for the narrow range of the smaller powers.

A transition design by the late Captain Ericsson for marine purposes will be interesting (Fig. 457A in the Appendix). The piston was a vibrating or rocking element, working pendulum-wise through a limited arc. The piston keyed to the shaft was in effect directly attached to the beam center, from which the connecting-rod drove to the crank in the ordinary beam method. This engine had two valves, admitting and exhausting alternately.

CHAPTER XX.

THE STEAM-TURBINE.

335. Introductory. Historical. The end of the nineteenth century witnessed a return and a renewed recognition of the turbine principle for steam-motors. Hero of Alexander made a steam toy before the Christian era embodying the concept of reaction (Fig. 460), and Branca in 1629 described the turning of a wheel by the impulse effect of a jet of steam from a nozzle (Fig. 461). The modern era began with Gustaf De Laval of Sweden in 1883; with C. A. Parsons of England in 1885; with J. H. Dow of America in 1893; with Rateau of Paris, France, in 1894, and with C. G. Curtis of New York in 1896. These basal patents have been followed by other and later ones in detail. The De Laval nozzle, for instance, dates from 1894, and the Parsons governor from 1895. Other names in an historical list would include: Real and Pichon, 1827; Avery, 1831; Leroy, 1838; Pibbrow, 1842; Wilson, 1848; Delonchant and Tournaire, 1853; Hartman, 1858; Monson, 1862, Perrigault and Farcot, 1863; Moorhouse, 1877; Cutler, 1879; Imray, 1881; Babbitt, 1884; Altham, 1892; Leger, 1894; Ferranti, 1895; Bollman, 1897, and many others. The names identified with manufacture and production are those of Rateau, De Laval, Riedler, Stumpf, Zoelly, Brown, Boveri, Holzworth, Parsons and Curtis. The Westinghouse turbine is the land design of Parsons: the General Electric turbine is the Curtis.

The principle of the steam turbine is the application to a motor utilizing the expansive force of steam, of the methods of making available the kinetic energy of water in the Pelton wheel and the water turbine. The principle of pressure-action upon a slow-moving piston fitting tight in a cylinder bore has been abandoned: instead of pressure the kinetic energy of a mass moving at high velocity because of a difference of pressure, is made to impart such energy to the revolving receiving element or wheel of the motor. Such revolving element is called the "rotor" of the turbine, to include a possible succession of turbines, working in series. The aim has been to utilize to the fullest extent the recognized advantages and economy of the theory of expansion of steam (paragrafs 295 to 303) and secure the same advantages of directness of application of mechanical power which the rotary principle can attain (paragraf 332). The era of high-speed machinery for generation of electric energy has been favorable to the turbine. The era of the slow-speed pumping-engine, mine-hoisting engine, marine and factory engine in the use of power were as distinctly unfavorable to its development. Its use for electric generation and transmission will steadily increase until the develop-

FIG. 460. Hero's Reaction Turbine.

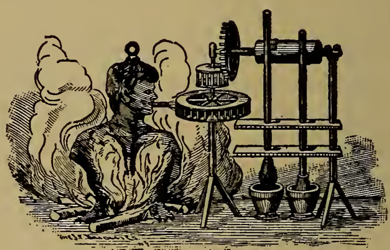


FIG. 461. Branca's Impulse Turbine.

ment of the internal combustion engine and its broadening use narrows the field for the spread of both the turbine and the reciprocating type of steam-engine.

336. The Mechanics of the Steam-turbine. The steam-turbine as a machine to utilize and render available the potential and intrinsic energy stored in a mass of hot high-pressure steam, will apply principles which fall under two groups. The one is the dynamic group, appropriate to the action of the mass and velocity of the steam as a fluid, independent of temperature changes; the other the addition to these effects due to the transformation of temperature energy into mechanical energy in the process of expanding. As a matter of energy due to mass and velocity, the turbine utilizes the impulse of the high-speed jet, and the reaction effect due to changing the direction of the movement of such mass by a force which is used in propelling the rotor. The compounding or stage principle, to use the heat energy in stages of reduction of temperature and pressure, and to eliminate inconvenient high rotor speeds, and the expanding nozzle principle are thermodynamic factors.

337. Impulse Energy against a Turbine Blade. If the mass of the steam coming upon a turbine blade in a second be called M, and the force which accelerates it be called F, and the units of velocity added per second, or the acceleration, be called f, then in a time t the velocity generated or V will be ft: but the space passed over (s) while the velocity was changing from zero to V was the product of the mean velocity $\frac{0+V}{2} = \frac{V}{2}$ and the time t. Hence:

$$s = \frac{ft}{2} \times t = \frac{ft^2}{2} \text{ and since } \frac{V}{f} = t,$$

$$s = \frac{V^2}{2f} \text{ or } V^2 = 2fs.$$

The force F will be measured by the acceleration it can produce in a unit mass M: or $F = Mf$. The unit mass is so much matter as the force of gravity will accelerate in one second and produce the standard acceleration of 32.2 feet per second: hence for mass can be substituted $\frac{W}{32.2}$, and

$$F = Mf = \frac{W}{32.2} f.$$

In a stream or nozzle passing W pounds per second with a uniform velocity V feet per second, it may be considered that a constant force F has acted upon W for a second and then ceased. As in the first part of the deduction, the mean velocity is $\frac{V}{2}$ and the work of such impulsive force is $w_1 = \frac{FV}{2}$. In such a jet acting under the influence of gravity the energy would be Wh, which can be written

$$Wh = \frac{WV^2}{2g},$$

since h corresponds to s in the general case, and g corresponds to f.

But if there are no losses, the energy of the jet should be equal to the work imparted to it, or

$$\frac{FV}{2} = \frac{WV^2}{2}, \text{ whence } F = \frac{WV}{g}.$$

In a flowing stream of area A and weighing D pounds per cubic foot, W pounds flowing per second will correspond to DAV. Substituting in the value for F

$$F = \frac{DAV^2}{g},$$

the impulse effect per second varies as the square of its velocity when D and A are constant. In steam D varies as the pressure, and is taken from a steam table for any assumed case.

If the force F directs the jet delivering W against a surface, and there are no losses, such surface would require an equal force to hold it still; or the full effect of the impulse would be received on that surface. If the jet imparted all its energy to the fixed surface, and the mass M had no velocity in the direction in which it was moving with the velocity V, the impartation of energy was complete. But there is no work done if the surface receiving impact is moving already with a velocity V: the jet did not reach the surface. If the surface receiving the jet was moving with a velocity V_2 the impulse would be $\frac{WV_1}{g}$ before the surface retarded the jet, and $\frac{WV_2}{g}$ after the surface and jet were moving together at their common velocity. The dynamic effect of the jet on such moving vane of a turbine will be the difference between the jet initial and the bucket final velocity; or

$$= \frac{W(V_1 - V_2)}{g}.$$

The energy will be the pressure multiplied by the mean velocity, or $\frac{V_1 + V_2}{2}$. Hence the energy given from jet to bucket will be

$$E = \frac{W(V_1^2 - V_2^2)}{2g},$$

which reduces to horsepower by dividing by 550 foot-pounds per second.

338. Reaction in a Jet and Bucket. If the jet issues from an orifice or nozzle in the side of a vessel, the pressure upon an area opposite to the jet must be the same as that which prevails in a film or section of the orifice just before the jet issues. This reaction pressure is therefore the same as the impulse F, or,

$$R = F = \frac{WV}{g} = \frac{DAV^2}{g}.$$

If that reaction pressure is made to drive that hitherto stationary surface in the direction of the pressure with a velocity having the mean of the sum of the previous two or $\frac{V_1{}^1 + V_2}{2}$, the energy will be

$$E' = \frac{W(V_1^2 - V_2^2)}{2g},$$

as in the previous case.

If the difference $V_1 - V_2$ be called x, the impulse of a mass M is Mx. If the common velocity of wheel and steam when the impulse has been completely absorbed be called u, then the work of Mx is Mxu; and if the steam left the wheel at right angles to the motion of the blade, this would be the work absorbed. If, however, by so curving the blade the mass of steam leaves the wheel not with a velocity u laterally, but with a velocity zero, the force to reduce the u to zero appears as a reaction against the blade, tending to make it revolve. This force must therefore be equal to Mu in intensity, and if exerted through a path u to reduce the velocity of the steam relative to the earth to zero, the work of such reaction will be Mu^2. The total energy is therefore their sum, or

$$E_2 = M(xu + u^2).$$

If the special assumption be made that $x = u$, and $u = \frac{v}{2}$,

$$E_2 = \frac{Mv^2}{2},$$

which is the maximum theoretical value, or gives a theoretical value for the efficiency of 100 per cent. It is impossible to realize a complete negative velocity of u and get the steam out of the buckets. The best which can be done is to get it low but finite and positive.

339. Relation of Jet Velocity to Pressure. In paragraf 337, the kinetic energy of a stream of fluid carrying a weight W through a height h was

$$K = Wh = \frac{WV^2}{2g},$$

if V denote the velocity in feet per second of its flow. In a closed vessel of one square foot of area of cross-section and of a height h, the pressure in pounds per square foot at the zero of height from which h is counted would be Dh when D is the weight of a cubic foot at that pressure. Hence when the pressure is counted from a zero of pressure, the velocity due to a pressure P in pounds per square foot becomes

$$V = \sqrt{2g\frac{P}{D}}.$$

If the flow be into another pressure, then the quantity P in the numerator should be the difference between these two pressures from the zero of pressure. The quantity which will flow needs to be corrected for the conditions of practice imposed by the shape of nozzle, etc.

The velocity having been found due to the pressure, the energy in horsepower per hour per pound will be

$$\text{Energy} = \frac{1 \text{ pound} \times V_1^2 - V_2^2}{2g \times 550 \times 3600}.$$

If V be called 3,000 and the velocity on leaving be called zero,

Energy per pound of steam $= 0.07$ horsepower per pound of steam or the reciprocal of this, 14 pounds of water per horsepower per hour. If the velocity be increased by increasing the pressure, the rates improve.

340. Blade Velocity under the Jets. The assumption of paragraf 338 that the best results are secured by making the blade velocity one-half that of the jets is supported by computation, and holds when there is no angle between the jet and the blade or vein. Fig. 463 shows the De Laval jet angle and the fact that such angle exists makes the wheel velocity a little less than one-half. In the single stage nozzle-form it is usually 47 per cent. When the drop-down pressure is done in one step and the steam has the high velocity thus resulting, the speed of the rotoı becomes inconvenient. For example if $V = 3000$ feet per second, and u for the blade is 1500 and the circumference of the wheel one foot, there must be 1500 turns per second, or 90,000 per minute. Hence the plans of compounding to divide the pressure range into steps and lower the tangential velocity. It is hard to find material to stand up against the centrifugal stresses at these velocities.

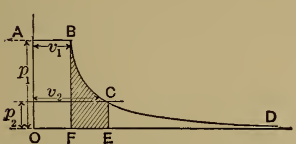


FIG. 462.

341. Thermodynamic Principles underlying the Steam Turbine. The steam turbine must also withdraw energy from the current of steam in the form of heat, as well as of kinetic energy. It is in this that its great advantage lies over the rotary steam engine, in that by utilizing expansion and the heat energy delivered in the process it is a superior heat engine in the more economical exchange of heat into work. Turbines utilize expansion in two ways: by the expanding nozzle, and by the expanding volume secured by compounding.

Repeating the deduction of paragraf 309 for the work in expanding when no heat is added to the steam in the form of heat, and relating it to Fig. 462, it will be remembered that the work in expanding from p_1v_1 to the condition of p_2v_2 in this case was

$$W = \frac{p_1v_1}{n-1}\left(1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right).$$

If the volume v_2 be extended to infinity toward the right the value for its work capacity reaches its limit, or is the total intrinsic energy of such a weight of steam. When v_2 is infinity the work becomes

$$W_i = \frac{p_1 v_1}{n-1},$$

which is an expression therefore for the intrinsic energy. At the condition B it is as above; at the condition C down the line of expansion it is $H_2 = \frac{p_2 v_2}{n-1}$.

If the steam or other gas be confined in a vessel fitted with an outlet nozzle, and the intrinsic energy in the vessel per pound be called I_1, and that in the nozzle be called I_2, external work is done upon the gas inside to get it to the orifice and give it a kinetic energy $\frac{V_1^2}{2g}$; and each such pound does the external work of expanding to p_2 and v_2 in the nozzle, and its state of kinetic energy is then $\frac{V_2^2}{2g}$. Under the law of adiabatic flow assumed for the steam, with the heat or total energy the same at the beginning and end by the principle of the conservation of energy

$$I_1 + p_1 v_1 + \frac{V_1^2}{2g} = I_2 + p_2 v_2 + \frac{V_2^2}{2g}.$$

It is proper to neglect the velocity within the vessel as so small as to be of no consequence when the vessel is of any size, so that the kinetic energy at the nozzle can be written

$$\frac{V_2^2}{2g} = I_1 - I_2 + p_1 v_1 - p_2 v_2.$$

But if it be assumed that there are no nozzle losses nor friction transfers, the second member represents the algebraic sum of the interchanges of internal energy and external work when the pound in question was expanding from a state $p_1 v_1$ to the state $p_2 v_2$. These changes are by the hypothesis due only to the work done by the heat energy in imparting kinetic energy; hence the numerical value of the expression for the kinetic energy is the same as that of the heat given up per pound in such expansion. If for the quantities denoting the internal energy the total heat from steam tables be substituted, and the difference in heat units per pound be multiplied by 778, the product will be the energy in foot pounds per pound; or

$$\text{Energy in foot pounds per second} = \frac{V^2}{2g} = (H_1 - H_2) \times 778.$$

If for the quantities I_1 and I_2 the value for the internal energy be substituted as deduced from Fig. 462,

$$\frac{V^2}{2g} = \frac{p_1 v_1}{n-1} - \frac{n-1}{p_2 v_2} + p_1 v_1 - p_2 v_2 = \frac{n}{n-1}(p_1 v_1 - p_2 v_2).$$

Applying the law for the perfect gases, $p_1 v_1^n = p_2 v_2^n$ and reducing

$$V = \sqrt{2 g p_1 v_1 \left(\frac{n}{n-1}\right)\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

which may be compared with the expression for work in terms of pressures in the piston motor of paragraf 309. If the area of the orifice be A as before, the volume discharged per second is AV. If v_2 be called the volume of a pound of steam at the pressure p_2 taken from the tables, the weight per second will be $\frac{AV}{v_2}$; hence multiplying the value of V by A and reducing,

$$\text{Weight per second } W = A\sqrt{\left(\frac{2\,gp_1}{v_1}\right)\left(\frac{n}{n-1}\right)\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]}$$

This is a maximum when the quantities under the square parenthesis are a maximum. If $\frac{p_2}{p_1}$ be called r, and its maximum found, it becomes

$$r = \left(\frac{2}{n+1}\right)^{\frac{n-1}{n}}.$$

The exponent n is the ratio of the specific heats of steam at constant volume and pressure, and if its usual value of 1.135 be taken the maximum flow will be when $\frac{p_2}{p_1} = 0.577$. This is for straight-sided nozzles. Experiments have shown that as the back pressure is reduced in outflow researches, the flow by weight reaches a maximum for steam and that thereafter further reduction does not increase the flow. If p_1 be increased with p_2 fixed, the weight of flow increases. This confirms the general truth of advantage from high initial pressures. With diverging nozzles the ratio increases up to complete exhaustion. The gain from superheating with steam turbines is both thermal and mechanical. The mechanical gain comes in from eliminating fluid friction on the blades and their mechanical erosion at high speeds of flow from drops of water due to condensation. Turbines work more effectively condensing than non-condensing, by reason of the lowered value for p_2.

It would expand the present available space unduly to carry this treatment further to include the facile and useful deductions from the use of the temperature-entropy diagram. For example, if the entropy-factor have a value of E_1 and E_2 corresponding to the conditions of pressure p_1 and p_2 and temperature absolute T_1 and T_2 the maximum velocity in a nozzle shaped to give an adiabatic expansion would be*

$$V = 158\sqrt{(E_1 + E_2)\,(T_1 - T_2)}.$$

Much experimental work on the flow of steam through nozzles and the best path for the expanding jets has been done by manufacturers and others. Similar tests are the only means of finding the coefficients with which the theoretical computations must be multiplied to conform to actual net output of work.

345. Single-stage Expansion-nozzle Turbine. The De Laval turbine of 1885 and 1894 is the simplest and most direct example of the type having an adiabatic expansion of steam in one or more nozzles from a p_1 of initial pressure to a p_2 at the issue upon the wheel and using only one set of blades and one wheel. The nozzle is so proportioned that the velocity increases as the pressure falls, so that the

* From "Steam Turbines" by Prof. Carl C. Thomas. John Wiley & Sons, p. 70. Consult also Stodola, Treatise on the Steam Turbine.

steam has just enough pressure to overcome friction on the blades and carry itself through the wheel (Fig. 464), and escapes to the low-pressure or condenser side with the least energy in the pressure form. Such energy has been transformed into kinetic energy and the mass of steam

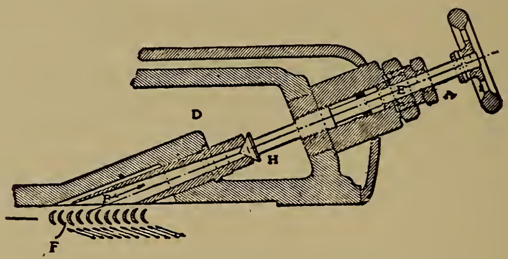


Fig. 463.

at high velocity impinges on the curved blade, and produces its impulse effect. By curving the blades, as in the Pelton impulse wheel, a certain reaction is elicited, and the steam leaves the wheel with little energy either in pressure, heat or velocity. The limit of the number of nozzles (Fig. 465) is only set by the convenience of getting them in, and keeping their jets from interfering with each other. Fig. 463 shows the control of the jet or nozzle by a hand-wheel to vary M, without varying the velocity in the nozzles which are in action. The nozzle, being designed for a particular relation of p_1 to p_2, will not work as efficiently with another ratio. Hence control should not be by throttling the steam pressure, but by varying the weight entering the chamber D at the computed pressure, and passing the nozzles. When the engine is changed from condensing to noncondensing at any time the nozzles should be changed also, but the construction makes this easy to do.

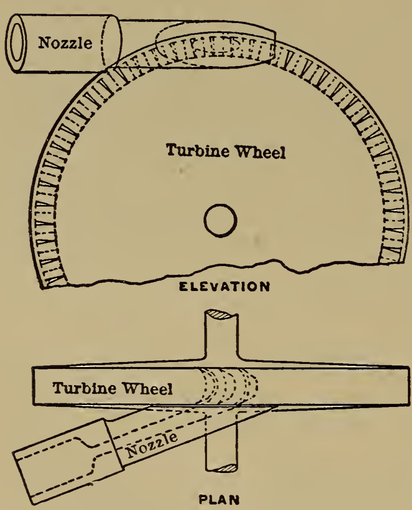


Fig. 464. De Laval Turbine.

The high velocity of the issue of steam, even if that of the blades is one-half of it or less, makes an inconvenient velocity for the periphery of the wheel as respects withstanding centrifugal accelerations: and the number of revolutions of the shaft is too high to use without reducing by gearing. A gearing ratio of 10 to 1 is made a feature of these designs, integral with the machine, so that the shaft from which the power is taken off shall turn at 2000 revolutions or thereabout for a turbine element itself at 20,000. To provide for the running balance of such a high-speed element, and keep it cool and silent, the rotor-shaft is made of very tough steel, but of so small a diameter that it will flex under any lack of homogeneity of the rotor, and let the latter revolve around an axis through its actual center of gravity without undue stress upon the bearings. The rotor itself does not touch the

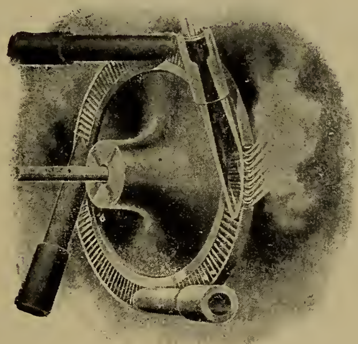


Fig. 465.

casing anywhere, and needs no oil, nor has it any friction. The bearings, however carefully designed, must be very carefully watched to see that they are copiously oiled, since if one gets dry for even a few seconds, its integrity is gone, and perhaps extensive repairs are entailed. The bearings wear rapidly also in careless hands, and noise and inefficiency begin at once. The rotor cannot be of large diameter to reduce the number of revolutions, because centrifugal accelerations and stresses increase with the diameter (paragraf 260) and no material has been found with tensile resistance sufficient to hold against the tearing effect.

346. Single-stage Impulse and Reaction Turbine without Expansion Nozzle. Riedler-Stumpf Early Types. The impulse and reaction water-wheel known as the Pelton wheel served as an early starting point for many designs, having simplicity and cheapness of construction for their advantages. They have been superseded in every successful case by the compound types, to reduce the inconvenient speed. They have been successfully applied in smaller sizes to driving centrifugal pumps — often in two pumping stages — centrifugal fans and blowers, centrifugal drying and cream-separating machinery and electric generators. The high speed gives small size and bulk both to motor and driven machinery. Rateau, Richards, Levin, Zoelly and Kerr have all applied the Pelton double cup-shaped bucket to steam-turbine design.

347. The Pure Reaction Turbine without Impulse. The turbine of J. H. Dow (Fig. 466) is an example of the pure reaction principle without impulse. The superior mechanical efficiency of the types embody-

Fig. 466.

ing both impulse and reaction have put these latter to the front in the larger sizes. For simple high-speed work against low resistances to rotation these machines are compact and cheap. The steam enters in the smaller annulus close to the shaft and expands in volume and in velocity as it works from ring to ring. The modern forms put the successive elements at the side of each other, instead of radially.

348. Compound Turbines. Multicellular Type. The advantages of dividing the expansion process, down pressure and temperature into successive stages commended themselves at an early day. The principle one is the reduced wheel velocity, and the reduced surface velocity of the steam upon the metal of the blades. This latter occasions not only excessive fluid friction, but worse than this, a rapid mechanical

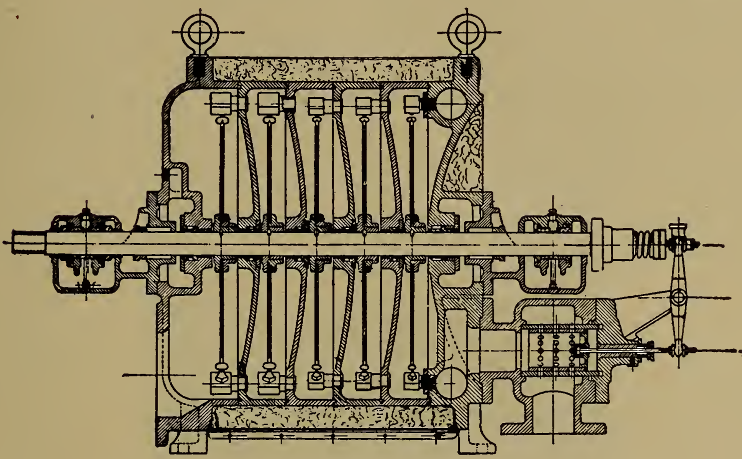


FIG. 467.

erosion. Water-drops in the steam act as the sand particles in a sand-blast at these high speeds. The compound principle has been embodied in two general types. In one, there is a succession of wheels

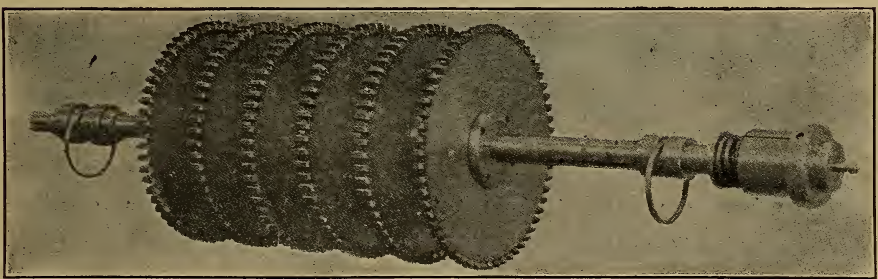


FIG. 468.

of the same diameter in separate compartments, the steam moving in succession with increasing volume and reduced pressure. In the other the wheels are of increasing diameter, and either in the same enclosing casing, or in one divided into a small number of compartments, much less than the number of wheels.

An interesting and successful American type of this class is the Kerr turbine of Figs. 467, 468, 469. The steam enters past the throttling governor into a ring-shaped channel in one head, from which nozzles direct the jets of steam upon the double-cup or Pelton type of buckets on the revolving disk. Between wheel No. 1 and wheel No. 2 is a cast-iron stationary dished diafram (Fig. 467), and the steam is again directed by a second set of nozzles upon the buckets of the second wheel. The nozzles increase in cross-sectional area down-pressure with the increasing volume and velocity, until at the discharge from the last compartment the energy to be rejected has become only just enough to carry the steam away after it leaves the last wheel. The step reduction of pressure lowers the velocity range in each compartment: the number of compartments is adjusted to the initial pressure proposed. The buckets increase in size — not in number — as the

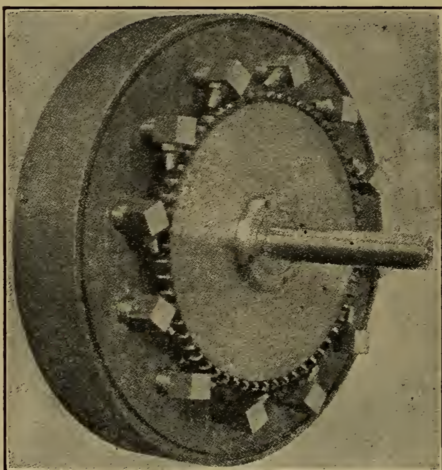


Fig. 469.

volume increases (Fig. 468) and the mechanical or manufacturing detail is simple and direct. There should be no side thrust on the wheels (Fig. 469) because the jet effect is balanced in both directions, and the bearings are so constructed as to keep the wheels in line with the jets and secure effective oiling by the ring-system. In this same class or type are the newer Rateau turbines of Paris and America; the Zoelly of Zurich, Switzerland. With a view to securing divided steps of expansion and cheapness of construction by avoiding the multicellular chambers along the shaft, the same result is sought in the turbine of Fig. 470.* Here the wheel is so enclosed in the construction of the casing that the successive cells or chambers are seg-

* Terry Steam Turbine; Hartford, Conn., U. S. A.

mentally disposed, instead of being side by side. The initial steam, after acting upon the part of the wheel exposed to it, passes into a stationary reversing chamber confronting it, and from this is successively returned four times or more to the wheel in succeeding segments in its periphery, until it escapes into the exhaust chamber from the last. The successive divisions of the expansion into stages enables the speed at the buckets to be kept down to 250 feet per second; the

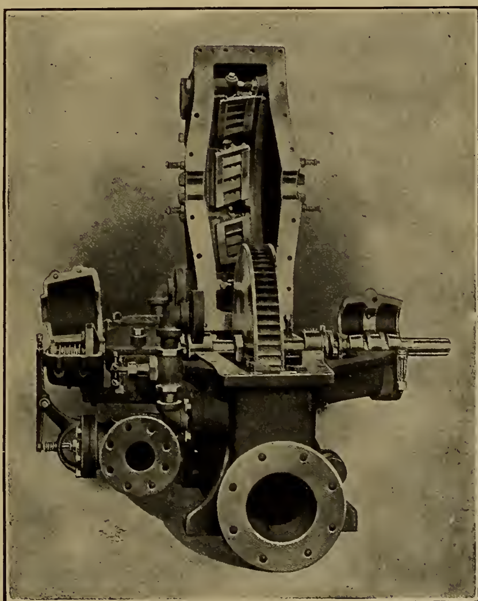


Fig. 470.

machine has its bearings close together and occupies the least area of ground-plan. These types are much used in small electrical, pumping, and fan-blower installations.

In the Zoelly Swiss designs, the successive wheel disks have been grouped into two casings, separated enough to get a bearing between in large sizes. The steam from the higher pressure set crosses over to the head of the lower pressure casing through a pipe. By keeping the drop of pressure in each step less than the limit of 0.577 of the pressure in the compartment previous (as discussed in paragraf 341), straight nozzles and cheap to manufacture can be used without interfering with the expansion process.

349. Reaction in Steam Flow from a Stationary Guide Ring. Stator Rings. The cup-shaped buckets of all the turbines make use of the dynamic principle of reaction whenever the steam leaves the bucket at an angle less than 90° with that at which it entered, so that the deflecting force to change the direction of flow has a component parallel to the plane of the motion of the wheel and in a direction opposite to the rotation (paragrafs 337, 338). In Figs. 463–465 no attempt appears to direct the energy of the issuing jet as it leaves the bucket because it was assumed to have given all its kinetic energy to the wheel in its passage.

In the compound types, however, it will be obvious that a considerable potential energy resides in the steam after leaving the first wheel in a series, and if such energy is allowed to expend itself in causing unutilized motion of the steam in clearance volumes between such rotating power elements, to this extent is this motion wasteful. It is the purpose in the design of what are called the "compound reaction" turbines to direct the flow of steam from the reaction discharge of each wheel into definite channels, and preclude losses by eddies or free expansion. The steam is received from the rotating vane, moving in the wrong direction, and is deflected by the shape of guide channels in such a stationary ring so as to be guided correctly upon the buckets of a second revolving ring less only the loss by skin friction. Using the fixed guide-blade walls as abutments, the pressure and velocity are exerted upon the second wheel, and the steam received into a second fixed ring of guides and so on. The fixed rings or diaframs carrying the guides will form part of the enveloping casing; the revolving rings making up the rotor are fast to the revolving shaft. This idea of securing a reaction for the kinetic energy and utilizing it again with least loss is the basal conception of the Parsons turbine (Westinghouse) and is also embodied in most of the other compounds. It appears in two general forms. In one type there is no impulse and nozzles are not used. In the other the impulse principle is availed of by incorporating the nozzle feature. Where there are no nozzles, the turbine need not be subdivided by partitions into stages. In the nozzle type this will usually be done.

The examples of the compound reaction without nozzles are the Parsons-Westinghouse and the Allis-Chalmers types; and the Holzworth of America, and the Willans and Robinson of England. The impulse or nozzle reaction compound is represented by the Curtis (General Electric), the Crocker-Warren, of America; and the Brown-Boveri, the Sulzer Bros. and the Lindmark of Europe.

The diafram or ring which carries the deflecting guides between the wheels which carry the buckets can be slipped on over the ends of the shaft in small sizes alternately with the wheels and be locked to the casing when the latter is all in one piece or continuous. In larger sizes it is convenient to split the casing along the meridian or greatest diametral section into two halves. Here the guide ring will split with the casing to which it is secured. The guide-blade ring as the element fixed in position has been called with the casing, the "stator" to distinguish it from the revolving element or "rotor."

350. The Westinghouse Compound Turbine. The Westinghouse Company bought manufacturing rights from C. A. Parsons in 1895, and introduced the American product with their modifications in 1899. Fig. 471 is a vertical section. In the form shown the rotor element is made in three diameters, in order that the variation in size of blades may be reduced to convenient steps, and keep the variation of steam velocity and wheel velocity under it within proper ratios. The steam enters at S through a balanced regulating or throttle-valve controlled and operated also by the governor. It enters the turbine passages proper in a complete annular channel between the casing and the solid part of the rotor, and moves to the left past the alternate fixed and revolving blades until the space B is reached, from which it passes to the condenser or to the open air. The depth of the blades is made to increase continuously in each stage or diameter to provide for the expansion.

Since the pressure at A is so much greater than at B and the flow of steam produces reaction components, there is a tendency to end thrust upon the rotor as a whole. This is met by balance disks or pistons PPP connected directly to the chambers of increasing diameter by open passages EEE to reduce this to a minimum. The pistons do not touch the casings, and are kept steam-tight by a succession of rings and ribs which do not touch, but form a labyrinthine or circuitous channel through which only an inconsiderable leakage takes place if any at all. To keep the rotor blades from colliding with the stator guides a small thrust or adjustment bearing T at the right is fitted with collars to keep the standard clearance. This is an eighth of an inch in small sizes, and increases up to one inch in the larger. It will be obvious that great injury can be done to the rotor from a broken blade getting astray in the clearances.

351. Governing and Overload Capacity in the Westinghouse Turbine. It is very desirable that the control of the weight of steam entering the turbine should be direct, and not by means of throttling the pressure and reducing kinetic energy and velocity more than the pressure. Hence the supply to the rotor is made intermittent or pulsating by causing the balanced valve V to rise and fall at intervals, letting full pressure and velocity reach the wheel and then shutting off all supply. The

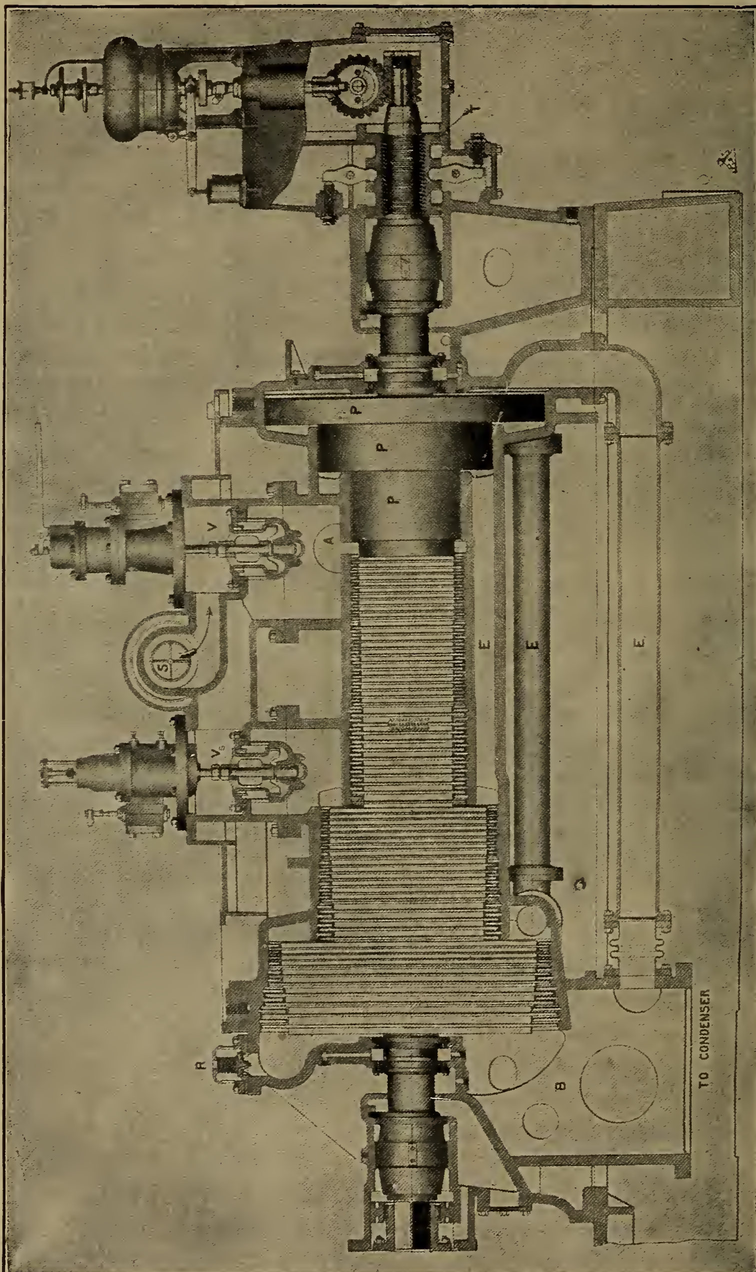


FIG. 471.

valve V is raised by steam pressure admitted below a piston (Fig. 471 at the left) and is quickly closed by the spring. The spindle at the right of Fig. 472 is driven by an eccentric from the worm wheel at the extreme right of Fig. 471. The governor by shifting the position of the fulcrum of the longer lever which drives the pilot valve of the main valve piston makes the lifting steam open pressure sooner and cut it off later per stroke of the eccentric in one set of conditions: and in the other set, the pilot valve is opened continuously and the piston

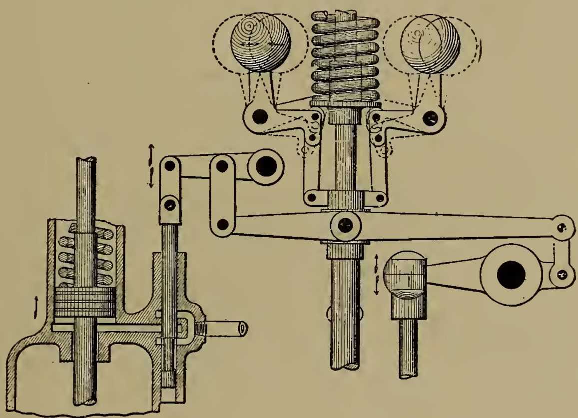


FIG. 472.

never fails to close the steam admission completely. In this case, the pulsation action disappears and the steam enters continuously.

When now the demand for steam exceeds the capacity of the valve V and the annular space through which steam enters the high-pressure end, the capacity of the motor is reached, and there is no means for providing to meet a temporary or continuous overload call. This is met by the use of the supplementary valve V_2 operated also by the governor, and having also a hand control. When the demand comes for more energy, V_2 is opened and allows a bleeding of high-pressure steam into the No. 2 stage of the rotor, where it acts with a larger turning leverage than in the chamber A. This can also be used in starting if the resistance torque is large. The pilot valve mechanism of this by-pass valve is just the reverse of that of the main valve: when the pilot is opened, allowing pressure in the connecting pipe to escape, the pressure of steam upon the valve piston causes it to open against the spring pressure.

Fig. 473 shows the general perspective of a Westinghouse 3000 kilowatt turbine generator. Lubrication is effected by a pressure feed

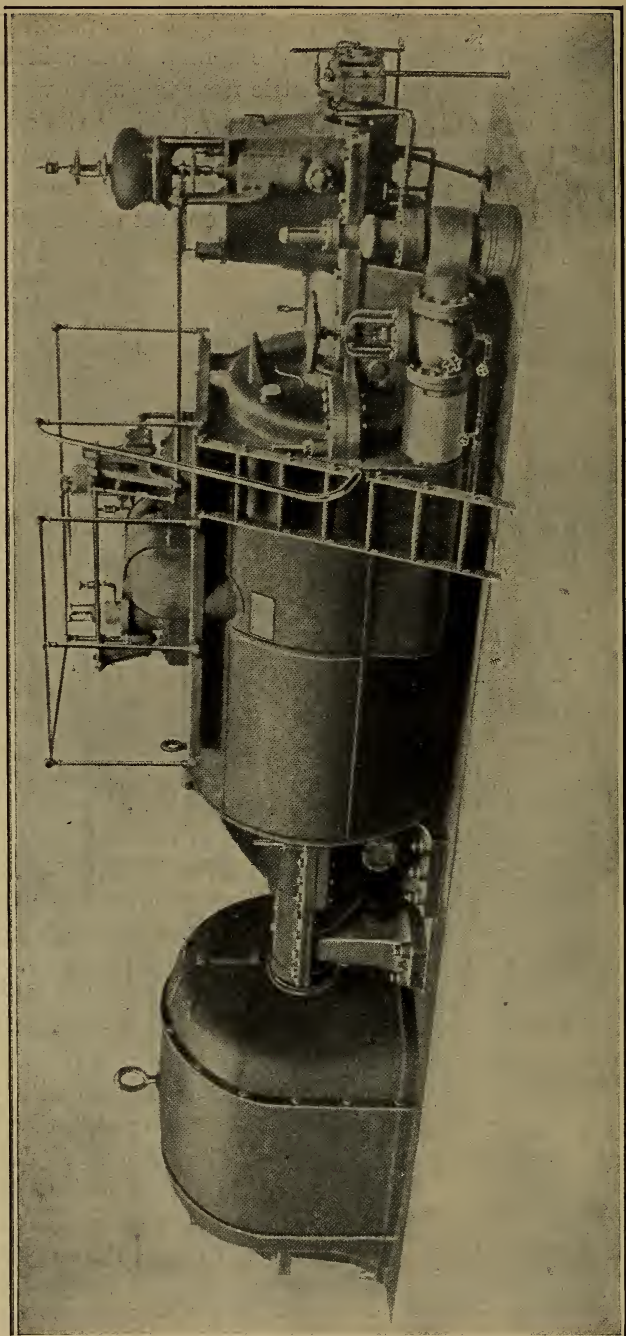


Fig. 473.

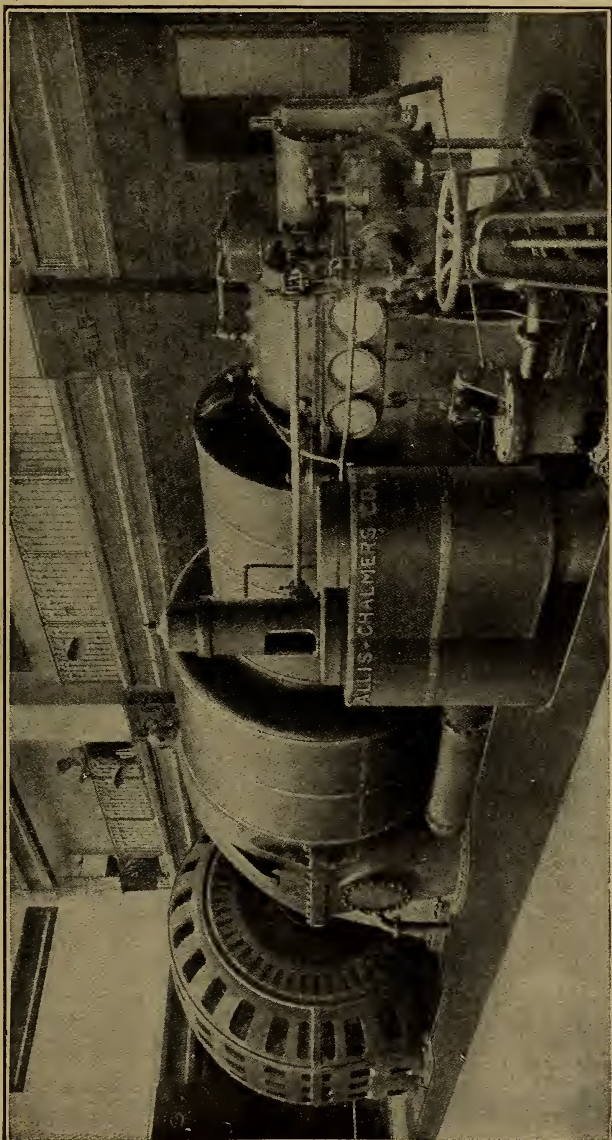


FIG. 474.

through channels in the bearings from a pump driven by worm gear, and fitted with a cooling coil to keep the oil temperature down so that it can be repeatedly used. Packing of the joints where the shaft leaves

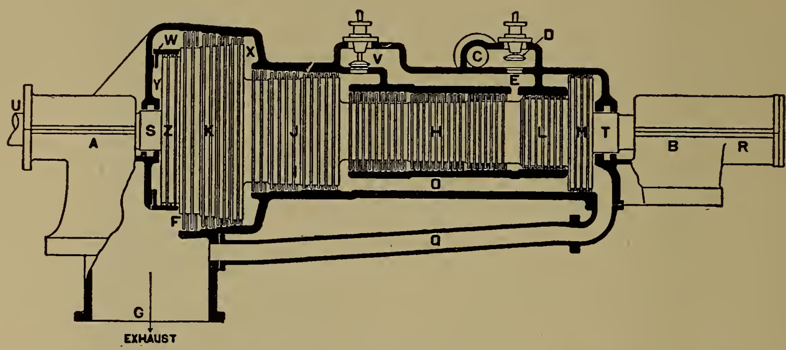


Fig. 475.

the heads of the casing is secured by a water-seal device, where a set of vanes or blades revolve with the shaft in an annular chamber forcing the water radially outward by centrifugal force. The left hand end has to resist indirectly the tendency of air to rush in and break the vacuum: and the right hand end is connected to the same lowered pressure by the balance connection E. The water can be renewed as required, and no parts are in metallic contact to rub on each other.

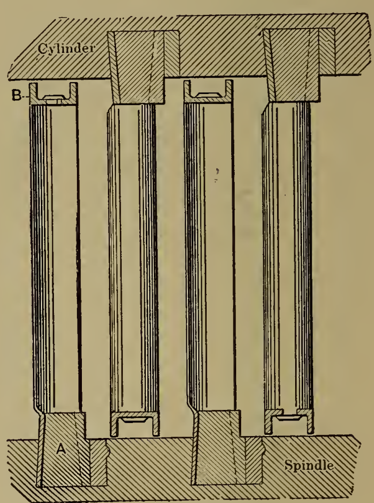


Fig. 476.

352. The Allis-Chalmer's Compound Turbine. This type is made also under the Parsons basal designs for a compound reacticn machine (Fig. 474). It differs from the preceding in constructive and manufacturing detail. In Fig. 475, for example, which is a diagram only, the balance of end-thrust is produced by two areas at the right hand and by one (Z) at the left, the latter working in a supplementary cylinder W. In this latter at the surface Y the same pressure prevails as at X, and the small annular area forcing K to the left is balanced by the larger area forcing it to the right. The blading is also different (Fig. 476), both for manufacturing reasons and to remove a tendency to inconvenient tremor and vibration when the tips of the blades were not confined and strengthened by an enveloping ring. The features of governing, overload capacity, packing, and lubrication are met satisfactorily along accepted lines.

353. The Curtis Nozzle Compound Turbine. The Curtis turbine embodies the two ideas of expanding in a diverging nozzle, securing impulse effect, and the compounding principle of expanding in several

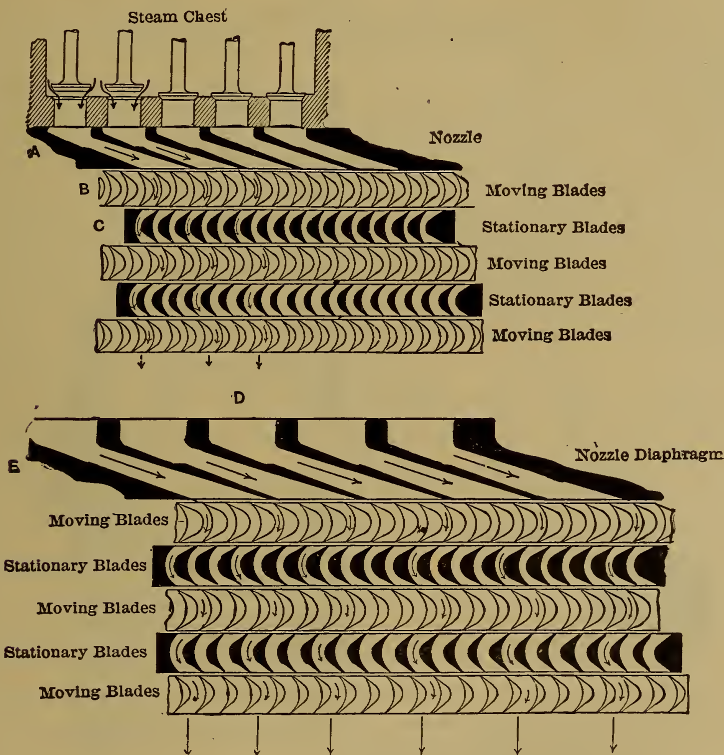


FIG. 477.

stages. For the latter it utilizes the reaction principle of the Parsons type, having alternate fixed and moving elements of guide-blade and bucket.* The turbine is made in two differing forms for land and for marine service. The former rights are held by the General Electric Co.

Fig. 477 shows the type arrangement of the principle. The steam passes through individual nozzles each controlled by its own valve.

* Mr. Charles Gordon Curtis studied engineering at Columbia University, graduating in 1882. In 1907 his creative and practical achievements were recognized by the honorary degree of Master of Science.

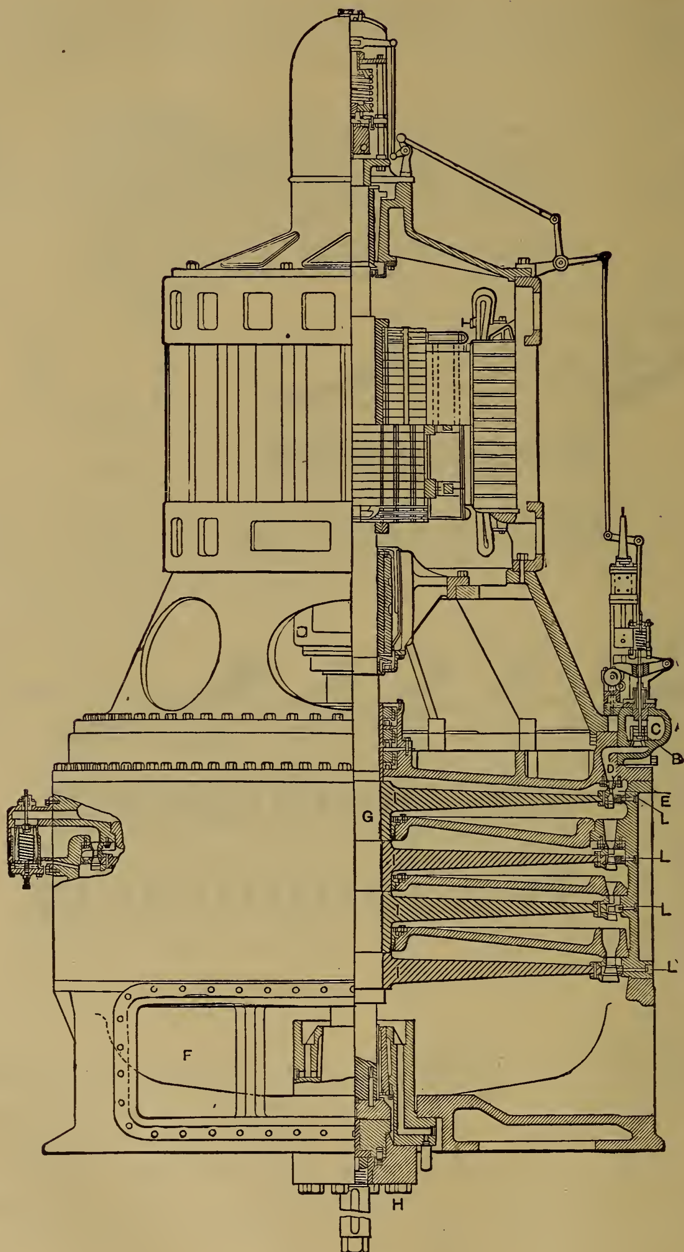


Fig. 478.

These nozzles are as numerous around the circumference as the weight of steam required calls for. The weight is controlled by closing off individual nozzles without throttling effect. It passes through the nozzles undergoing expansion and increase of velocity with a lowering of pressure as in the pure nozzle type, and acts by the reaction system upon the succession of wheels which carry the buckets. The principle of stage expansion is carried out in the larger sizes, each stage having its set of expanding nozzles. For example, if there be four stages in the expansion process, with two revolving wheels in each stage, which is a very usual and successful proportioning with 150 pounds pressure, the drop will be from 150 to 58½ pounds in the first nozzles, calling for a linear velocity of 2000 feet per second. After passing through the first wheels the steam enters the second set of nozzles, in which if expanded to 18½ pounds the terminal velocity will be again 2000 feet per second, due to the increase in volume. A third set of nozzles with their wheels will result in a velocity of 1600 feet per second with a reduction to 3½ pounds pressure; and the final nozzles and the fourth pair of wheels will give a pressure of 1 pound above vacuum, and a velocity of 1400 feet per second, at which the residual energy will be just enough to carry the steam away.

The constructive advantages for electric generators on land which are offered by putting a short turbine such as the Curtis with its shaft vertical, early attracted designers to this plan. The stationary part is symmetrical in shape, easily moulded and cast, and free from irregular disturbance by heat. The structure of the casing forms convenient and adequate support for the generator. Much floor area is saved, and the building will have height in any case. Cost of foundations is reduced in reduction of ground plan, yet with adequacy of support, since alinement is not specially vital. Accessibility all around the machine is secured. The shaft bearings are relieved from the stresses of the weights to deflect them, and their lateral friction is negligible. Hence the shaft does not have to be so massive to prevent distortion: bearings can be placed where convenient. With an adequate step-bearing the relative positions of fixed and revolving parts are definitely fixed, and the short length prevents the inconvenient change of position of parts due to expansion. Fig. 478 shows such a vertical turbine in elevation and section and Fig. 479 enlarged detail of the stages. In the vertical machine, the weight stresses are transferred from horizontal bearings of the other types to the foot-step bearing: but the friction herein can be made definitely and positively a fluid friction and the shaft be compelled to float upon an oil film. Fig. 480 is a type of turbine footstep. The block A on the end of the shaft is of cast iron and turns with the shaft. The similar cast-iron block B is kept from revolving and is adjustable vertically by the heavy screw S. Both blocks are recessed on their faces at C for about one-half their diameter to form an oil cavity supplied by pressure from below through the pipe D. The blocks separate when the pressure is great enough to balance the weight, and oil flows outward and then upward between

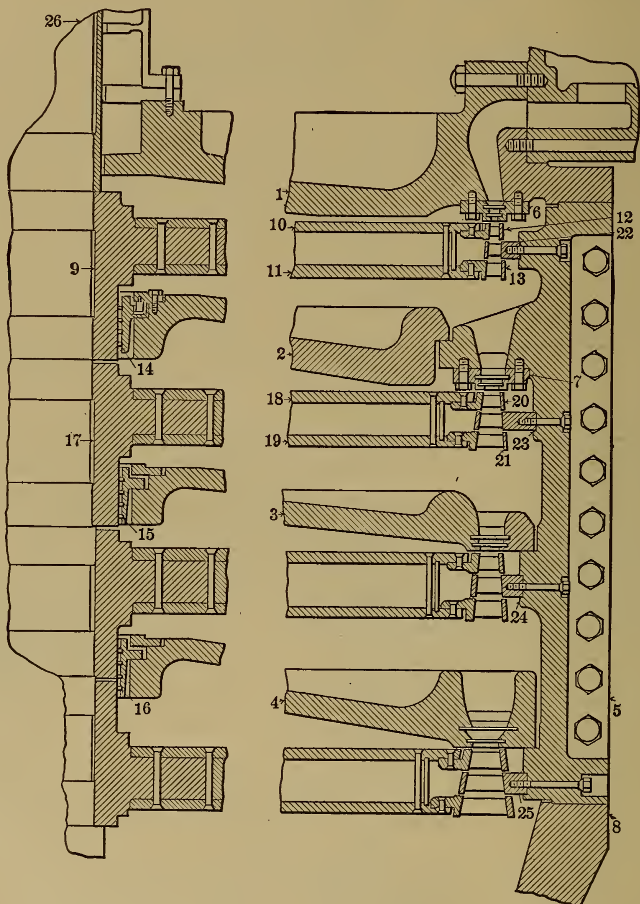


Fig. 479.

the shaft and the steadying bearing to an overflow from which it escapes. The oil pressure is kept constant by an accumulator, so that flow should not cease if the pump stopped. The pressure varies with size of the generator: with a 5000-kilowatt machine it is made about

1000 pounds per square inch. Water may be used instead of oil. The pump is usually electrically driven.

The Curtis turbine is also made for horizontal shafts, as is required for marine practice. This will be referred to in a later paragraf.

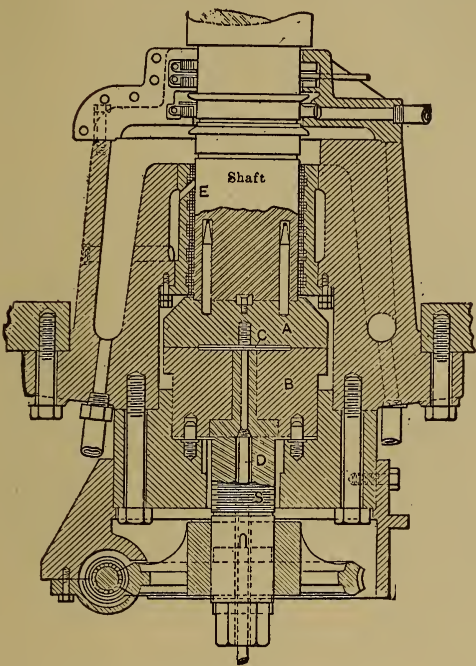


Fig. 480.

354. Governing and Overload in the Curtis Turbine. The smaller types of Curtis turbine are governed by throttling. In the larger sizes each nozzle or group of nozzles is controlled by its own poppet-valve, opened and closed by the governor. The actuation of the groups of poppet-valves may be effected by a hydraulic piston, whose valve is controlled by the governor, or the transmission may be mechanical by pawls or dogs; or an electrical system can be used, the valve spindles being surrounded by a solenoid coil whose action balances a spring. As more or less current passes in the solenoid, the spring is strained more or less, and the valve shut or opened correspondingly.

Overload capacity is secured either by making the number of nozzles for the first stage larger than is necessary for the normal load, and having the extra steam capacity available to meet the call for greater weight of steam, or by-pass valves may be used as in the Parsons, to bring an extra pressure and velocity further down the series. The valve at the left in Fig. 478 is a by-pass valve to give the operator control of the pressure after the first stage.

355. Low-pressure or Exhaust Steam-turbines. The high velocity with which steam, even at low-pressure, rushes into a vacuum has been utilized in making a sort of compound engine or continuous expansion system in which the exhaust from a large reciprocating engine non-condensing, running intermittently, is passed to a condensing turbine running continuously. Such engines are the hoisting or winding engines in mines, and the reversing or continuous rolling mill engine in steel works. Heat which would otherwise be wasted in the exhaust from such engines is saved and used. The range of temperature corresponding to a pressure range from atmospheric pressure down to that corresponding to 27 inches of mercury vacuum pressure (1.6 pounds per square inch absolute) should give an electrical horsepower for each 31 pounds of steam so used in driving a generator. For each 1000 pounds of exhaust steam rejected, therefore, there could be gotten over 30 horsepower if thus utilized.

This system requires an accumulator between the two engines to receive any excess from the one engine, and to keep the turbine going during intermissions of exhaust supply. Such accumulator is a heat storage reservoir, and either water alone, or cast iron with water is the cheapest available material. In the former case the exhaust steam passes directly into water, which it heats and makes into steam without loss except that due to radiation: in the other, the water is in shallow cast-iron trays, and the steam surrounds all as in an open feed-water heater (paragraf 227). The capacity for such accumulator will be computed by assuming the weight W of steam per hour to be supplied to the turbines, and a time t during which no heat reaches the chamber from the exhaust. Then since each pound of water at atmospheric pressure can absorb 966 units of heat, the heat capacity in B. T. U. for each permitted degree of temperature rise or fall will be:

$$\text{Heat capacity} = \frac{W \times t \times 966}{60} \text{ B.T.U. per minute} = H.$$

If the permissible temperature range be a number of degrees $r = 10$ or more, the weight of water will be

$$\text{Weight of water} = \frac{H}{r}.$$

The output from such compounding of reciprocating engine already installed and a low-stage turbine is much greater than the increase due to attaching a condenser to the former engine. The results of tests seem to show such increased output to be over twice what the application of condenser alone would give (paragrafs 306 and 501).

356. The Steam-turbine for Marine Use. The convenience of operating condensing apparatus at sea, and the great advantage to the turbine when it operates condensing have added to other advantages of the turbine for vessels. The absence of vibration at speed, the lowered center of gravity, the diminished attendance, the lessened oil

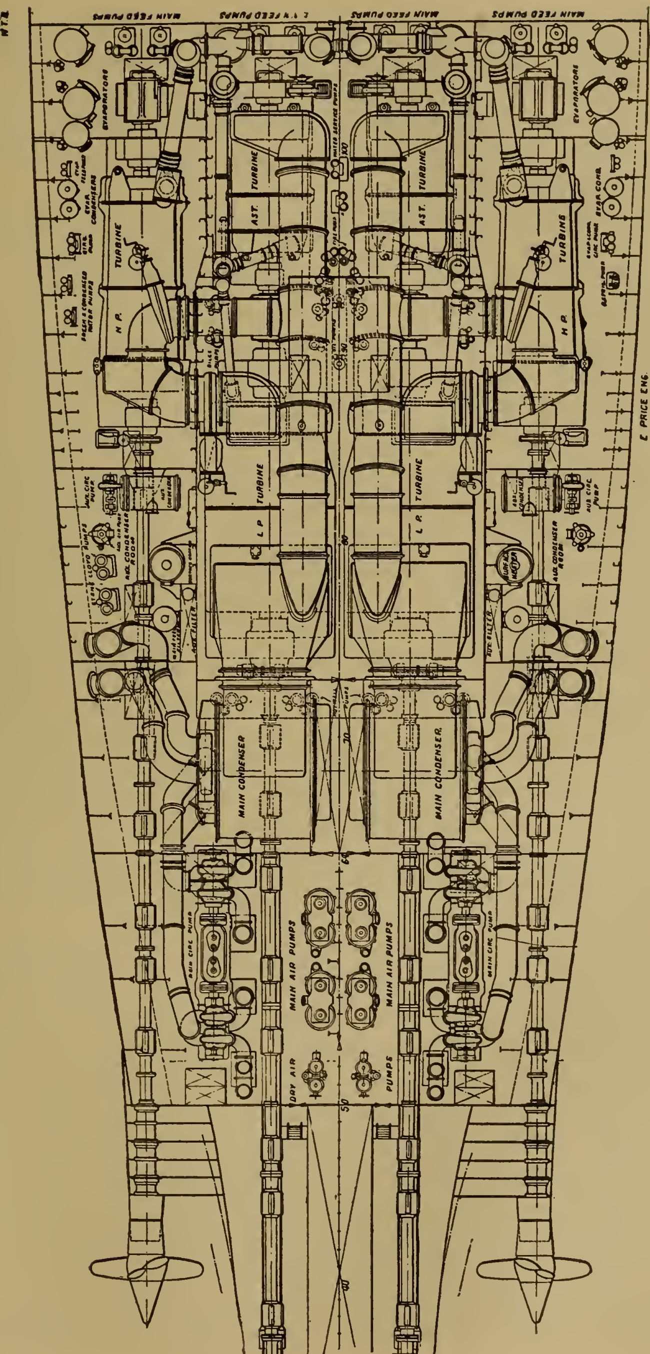

MAIN FEED PUMPS
MAIN FEED PUMPS
MAIN FEED PUMPS
EVAPORATORS
EVAPORATORS
TURBINE
TURBINE
H. P.
H. P.
AST. TURBINE
AST. TURBINE
L. P. TURBINE
L. P. TURBINE
MAIN CONDENSER
MAIN CONDENSER
MAIN AIR PUMPS
MAIN AIR PUMPS
DRY AIR
PUMPS
E. PRICE ENG.

FIG. 481.

consumption, bulk and weight, the less danger from racing when the screw rises toward the surface should all be listed as advantages, and the fact in particular that probably the great increase in propelling power demanded to meet increased speeds in transatlantic passenger service could not have been met by reciprocating steam-engines within the available limits of space.

As against these must be placed the high speed of the screws and the fact that the steam-turbine does not reverse. The high speed of the screws compels a small diameter and small surface for the screw blades, and introduces a source of inefficiency (called cavitation) because at excessive speeds the water is forced backward faster than it will flow around the circle described by the tips of the blades, leaving an area of lowered density behind the wheel, instead of solid water for the reaction to be exerted on. This may be met by enlarging the turbine diameter, when the economy of room and weight vanishes. The gain in weight seems to be about 5 per cent. The diminished area of the blades of the high-speed wheel diminishes the promptness of the response of the ship to manœuvering needs in harbors, and lowers the holding power as compared to big screws of large surface. The non-reversibility requires additional turbines for backward motion and manœuvering, and these are a drag in forward motion if separate and not driven forward in normal use. If many turbines are used as is the usual solution, those close to the hull at the sides are usually the reversing ones, and in reversing these are not efficient, detracting from manœuvring power. The turbine is not efficient at speeds below the normal or maximum: hence for cruising in naval work the practice in certain forms has been to have special cruising turbines, with their added weight and room. The turbine battle-ship gives a very steady deck from which to range and fire the big guns.

Fig. 481 shows plan of the six turbines required to operate the four screws of the Mauretania on the transatlantic service. The two outer shafts are the high-pressure ahead turbines, and the two inner ones are the low-pressure. On these latter are two independent turbines taking steam directly for going astern. In backing the high-pressure turbines and wheels are not in use, or the arrangement of Fig. 482 may be used, with eight turbines on the four shafts, and two additional turbines to give intermediate pressure ranges for economy at slow speeds while cruising as in war-ship conditions. Fig. 483 shows a Curtis horizontal two-shaft four-turbine design of same capacity and guaranteed superior economy. The marine turbine cannot operate any of its auxiliaries itself, but these must all be independent (paragrafs 145 to 151 and 500 to 530). Fig. 481 shows the number and multiplicity of

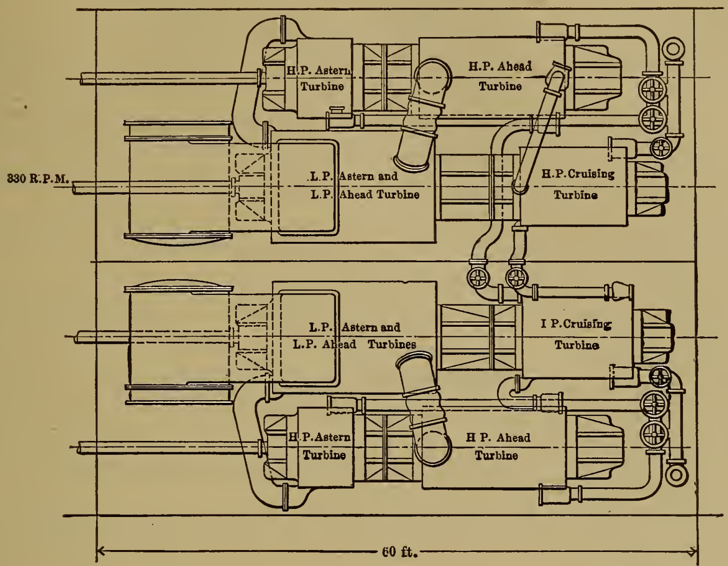


FIG. 482.

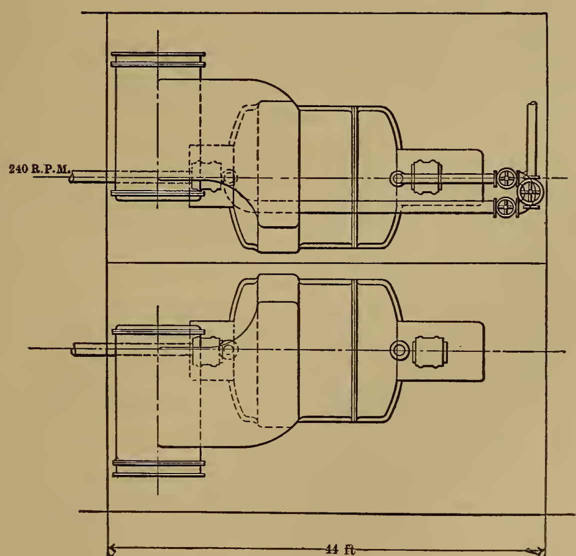


FIG. 483.

such auxiliaries in a marine power plant. The condensers for turbines will be discussed in a later chapter.

357. Advantages of the Steam Turbine. The steam turbine has advantages over the rotary steam-engine, and also those which both offer as compared with the reciprocating engine. In the first class are:

1. It secures the advantages of expansive working. These appear in economy of fuel and low water rate, or in general higher thermal efficiency.

2. There is no piston friction, no need for cylinder lubrication, no piston packing, no leakage past pistons, no wear.

3. It has no limits of size or effective capacity, but the larger the unit the more effective.

As compared with a reciprocating type of motor:

4. There are no reciprocating masses to be stopped, started, and accelerated and retarded.

5. The power is applied directly to produce rotation, without the friction of joints in a mechanism.

6. Continuous rotary motion eliminates danger from entrapped water in the cylinder.

7. Foundation weight and mass are reduced, as there are no shaking forces resulting from changes of direction of motion of masses, or steam effort within the casing. The foundation work is simpler and less costly, as there are no fly-wheel pits, etc.

8. Valve gearing disappears, or what remains is of the simplest type.

9. Superheating and its thermal economy is easier to apply, since no trouble is caused as respects lubrication within the cylinder and deformations of castings due to a high temperature.

10. Maintenance and up-keep are diminished where there is no metallic contact of pistons and bore, valves and seats. There are no pin joints needing adjustment for wear, with knocking and pounding after wear begins.

11. There is no kinematic dead-center, nor need for a mechanism to "bar" the engine past its points when the valves are closed.

12. The angular motion of the shaft is uniform and regular per revolution, without the disturbing effect of variable effort on the pin with high expansion.

13. This favors close regulation to speed.

14. The avoiding of cylinder lubrication makes the condensed water in condensing types free from oil without expensive appliances to make it so.

15. Wear or indifference as to adjustment does not seriously affect the steam economy; settling of foundations does not affect alinement.

16. The erection is therefore cheap and rapid.

17. The time to get the machine going from rest and in a cold state is less than with the reciprocating engine of same capacity; the operation of starting is simple.

18. The rotor is its own fly-wheel, and the weight and space for the latter are saved, while angular regulation is secured and retained.

19. The cubic space occupied in ground plan and elevation is less for the turbine than for the reciprocating engine, and the weight less. The values to be given to this advantage will be determined by the two sets of conditions chosen, and whether the shaft may be revolved at high speed or not. As respects marine conditions of older type as compared with modern turbines it would appear that with steamer engines of the ordinary marine type the weight will be from 300 to 500 pounds per horsepower, and in high-speed naval practice about 150 to 165, with a present limit of 42 pounds. A turbine of the Parsons type weighs in the neighborhood of 30 pounds per horsepower. The reduced weight of the moving parts helps to keep the friction low.

As compared with gas engines running down to much less than this per horsepower — say 10 to 20 pounds and less when speed is high — this would prove only a provisional argument. When the turbine must reverse, as in marine practice, the gain in weight diminishes for the plant as a whole, or where the number of shafts multiplies. The more elaborate installation of auxiliaries also tells against the turbine, while still leaving the margin in its favor. Transatlantic steamers have a margin of 10 per cent gain in weight.

But the gain in space occupied is more apparent. Fig. 484 is based upon a 1000-kilowatt horizontal turbine as compared with an engine of cross-compound Corliss type, with the generator between the cylinders, the latter being 28 and 56 inches by 48-inch stroke, turning at 95 revolutions. As the unit increases in size, the comparison becomes more favorable to the turbine, and particularly if the compact vertical type be chosen. Fig. 485 is the equation of a slow-speed massive engine of horizontal vertical type giving 5000 kilowatts with the equivalent Curtis. The best case for the reciprocating engine is a high-speed vertical cross-compound, compared with a horizontal turbine. The worst case for the turbine is a slow speed of shaft with the necessity for reversing imposed. The boilers and much auxiliary machinery will be the same.

20. The attempt to reverse at speed does not endanger the structure of the engine.

21. In marine and warship conditions, the low head-room required

lowers the center of gravity, increases stability, makes the engine less vulnerable behind incomplete armoring, and increases a cargo capacity.

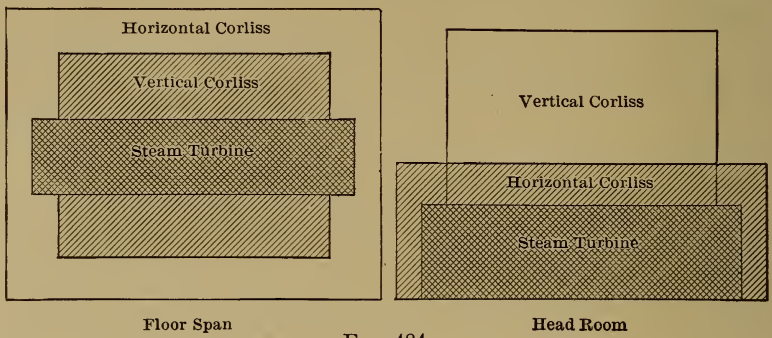


FIG. 484.

The smaller diameter of the wheel at high speed lessens the draft of the vessel and enables high speed to be secured if the motor is powerful enough. The weights and diameters of wheels and shafts are diminished.

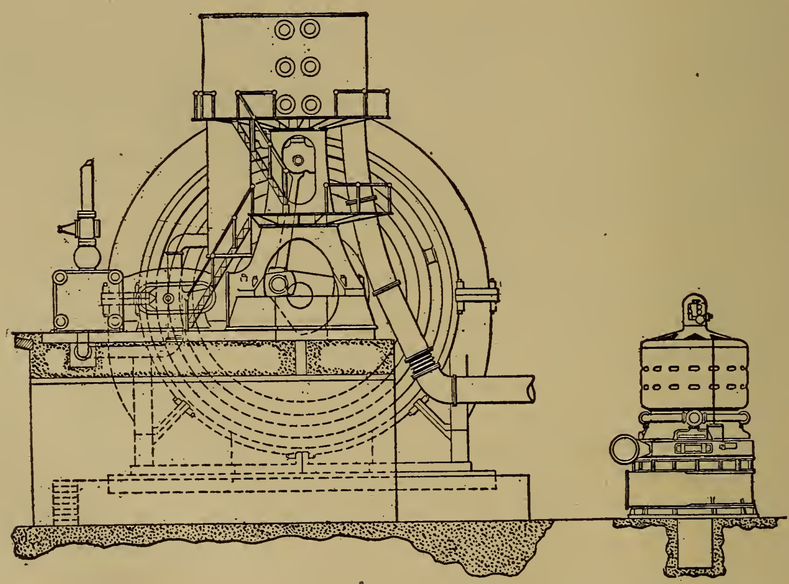


FIG. 485.

It would seem that shaft limits had been reached with reciprocating engines, and that to get required speed and power through two shafts and wheels would have been scarcely possible, and there would not have been room for four reciprocating engines below decks in the big ocean liners. The absence of vibration is a great feature for a passenger vessel.

22. The avoiding of contact surfaces, and the keeping of unbalanced pressures off from the moving surfaces, make the manufacturing process a simpler one and the equipment for it less varied.

23. The economy of coal and the purchase and operating costs will be discussed in a later paragraph.

24. The turbine runs economically over a wide range of loading, from less than its normal load to a considerable overload. The reciprocating engine has a most economical load, and does not do its best on either side of this (see Fig. 486 and paragraf 303).

358. Disadvantages of the Steam Turbine. The disadvantages of the turbine have differing significance according to the uses to which it is to be put and the standard used for comparison:

1. It does not normally reverse. When reversing is a necessary feature it must be gotten by a separate rotor with guides and buckets set for such reverse motion, either on the same shaft or capable of being geared to it. Separate piping and valves must be provided.

2. It has not a large starting torque or turning moment at low speeds to overcome heavy resistances in getting under way. These are the conditions in starting railway trains or commercial motor vehicles and in working the latter in deep snow, mud, or sand.

3. The high rotative speed when the resistance to be overcome is unfavorable thereto. Turbine pumps in successive stages have arisen to remove this difficulty, but air and gas compressing under high pressures, the rolling mill, the mine-hoist, and the factory transmitting power by mechanical means must be operated by the slower moving reciprocating engine or objectionable reducing gears must be introduced. In marine conditions, the small screw at high speed lacks holding power by reason of its reduced area.

4. The stored energy in the mass of the rotor revolving at high speed with low friction. This makes the stopping of the main shaft a slow process requiring many minutes, and in case of accident or emergency a quick stop often becomes imperative. In electric drives, the opening of the switch may accomplish all that is required; in mechanical drives, a clutch may be interposed between motor and transmission and be thrown out by power. But a brake or the use of a reverse turbine as such will be required if the rotor itself is to be stopped. In marine practice this slow arrest of the screw invades promptness of response to manœuvering orders.

5. The economy drops with lowered speed below that for which the machine is designed.

6. The conditions of running cannot be changed respecting speed or initial or final pressures without affecting economy and efficiency.

7. Fluid friction of water in the steam retards speed.

8. High-speed turbines have a tendency to hum. This is partly nozzle and blade noise, having a likelihood of reaching a musical note or tone which the air will transmit. Or the noise is from the generator. This air vibration is different from the vibration of foundations or solid masses, and can be checked by use of double doors and windows. Generator hum is lessened by inclosing this also and sending a forced circulation of air within the casing.

9. The wear of the blades by erosion, due probably to water and its mechanical action in condensation during expansion, replaces the wear of pistons and cylinders in the other type. Water backing up from jet condensers into the lower stage has also made trouble from brake-action and breakage of blades which were not strong enough to withstand impact against solid water in a mass.

10. Breakage of a blade usually destroys many others before it becomes comminuted and gets out of the way.

11. The turbine shares the peculiarity of all high-speed machinery that when anything goes wrong in its lubrication the harm comes very quickly. While there is less to oversee and do about the turbine than in the reciprocating engine, yet the greatest care is required to see that steps and bearings are in good condition and to keep all apparatus in prime condition.

359. Costs of the Steam Turbine. The question of the cost of the turbine appears in the usual two forms. The first is the purchase cost and the second is the cost of operating. As the question always appears as a competitive one as respects a reciprocating engine of the same capacity, a type must be chosen for each alternative. The first cost of the cheaper to build will not come down much below the standard of the more expensive to make so long as competition exists.

To assume a set of conditions, let a 1000-horsepower unit be selected, with a 750-kilowatt generator. The turbine has no indicated horsepower properly so called, so that the electrical output must be its standard. If the steam pressure be called 150 pounds, a cross-compound of 24 and 50 inch cylinders and 42-inch stroke at 100 revolutions should give 1200 horsepower to allow for loss and possible overload. Assume the generator to be of 2300-volt capacity, alternating, 3-phase, 60-cycle.

The turbine and generator together at $30 per kilowatt would cost $22,500. The engine alone at $20 per kilowatt will cost $15,000 or a little less; the generator a little less than $9000. Each will require an exciter at $1500 with its own engine.

The turbine condenser will probably be made of surface type for safety. If 4 square feet be allowed per kilowatt, there will be 3000 square feet of tube area, and the cooling water will be 60 to 70 pounds per pound of steam. Independent steam-driven centrifugal circulating pump will probably be chosen with 12-inch area driven by an 8 × 8 inch engine, and an independent air-pump steam driven. Condenser and pumps amount to $5200. For the reciprocating engine the condenser can be smaller, with 35 pounds of water per square foot of cooling surface and 10 pounds of steam condensed per hour per square foot. A direct-acting circulating and air-pump steam driven can be used with such a surface condenser, making the total cost $3000. If a siphon or barometric condenser is selected, requiring from 40 to 50 pounds of water per pound of steam, the cost falls to $2500 for the reciprocating engine and to $4000 for the turbine. The latter will need 60 times the weight of the steam it condenses. A simple jet condenser for the engine would cost $2000. An allowance of $200 should be made for erecting the turbine condenser.

The foundation cost is greatly in favor of the turbine. If made of the same depth, say 12 feet, and at a cost of $6 per cubic yard, the turbine foundation of 40 cubic yards of concrete will be $250 and the engine foundation of 276 cubic yards will be $1656.

The turbine will have a superheater, say for 100 degrees. The cost of the superheater is at $2 per boiler horsepower for 50 degrees and $2.60 for 100 degrees. If the turbine uses less than 16 pounds of water per horsepower per hour, 500 boiler horsepower should be enough (paragraf 10). Hence the superheater will cost $1300. An excess of freight and erection charges for the engine and generator should come to $2500. Summarizing:

TABLE XVII.

COMPARATIVE INSTALLATION COSTS OF TURBINE AND ENGINE.

Item.	Reciprocating Engine.	Turbine.
Engine	$15,000	$22,500
Generator	9,000	
Condenser surface	3,000	5,200
Foundations	1,656	250
Superheater	....	1,300
Erections	2,300	....
	$30,956	$29,250

That is, the engine plant costs a little over $40 per kilowatt and the turbine plant a little over $38. In larger turbine units, and by omitting the superheater and modifying other details, the price per kilowatt in 3000 units is about $20.

The saving in cost of land and buildings is very considerable for the turbine. The turbine-room assumed at one-half the floor area of the engine-room will give a material reduction of cost when land is dear. The engine-room would have an area of 7600 square feet in the above plant, and the turbine of 3600 square feet floor space saves 4000 square feet.

With respect to operating cost, the standard report of recent research is from the plant of the subway electric power station in New York City.* The data of the first column are from a year's record of costs in the plant. The others are from costs obtained elsewhere, reduced to the conditions to which the first column applies.

TABLE XVIII.

CHARGES PER KILOWATT-HOUR FOR POWER-PLANT MAINTENANCE AND OPERATION.

Item.	Reciprocating Engines.	Steam Turbines.	Reciprocating Engines and Steam Turbines.
1	2	3	4
MAINTENANCE.			
1. Engine-room	2.57	0.51	1.54
2. Boiler-room	4.61	4.30	3.52
3. Coal and ash handling	0.58	0.54	0.44
4. Electrical machinery	1.12	1.12	1.12
OPERATION.			
5. Labor, coal and ash	2.26	2.11	1.74
6. Removal of ashes	1.06	0.94	0.80
7. Rental of dock	0.74	0.74	0.74
8. Labor, boiler-room	7.15	6.68	5.46
9. Supplies, boiler-room	0.17	0.17	0.17
10. Coal, boiler-room	61.30	57.30	46.87
11. Water	7.14	0.71	5.46
12. Labor, engine-room	6.71	1.35	4.03
13. Lubrication	1.77	0.35	1.01
14. Supplies, engine-room	0.30	0.30	0.30
15. Labor, electrical	2.52	2.52	2.52
16. Relative cost of operation and maintenance	100.00	79.64	75.72
17. Relative investment cost in percentage	100.00	82.50	77.00

* Henry G. Stott, "Power Plant Economics." Trans. Amer Inst Elec. Engrs., January, 1906.

360. Performance of the Steam Turbine. Great care should be exercised in generalizing from comparative tests of turbines with reciprocating engines. The reciprocating engine has a most economical load at which it may be run; the turbine runs with essentially the same economy over a wide range from below its normal load to a considerable overload — even as high as 150 per cent. The turbine is also usually helped by its more recent installation and with higher pressures, superheat, and better vacuum conditions to favor it than the older reciprocating engines.

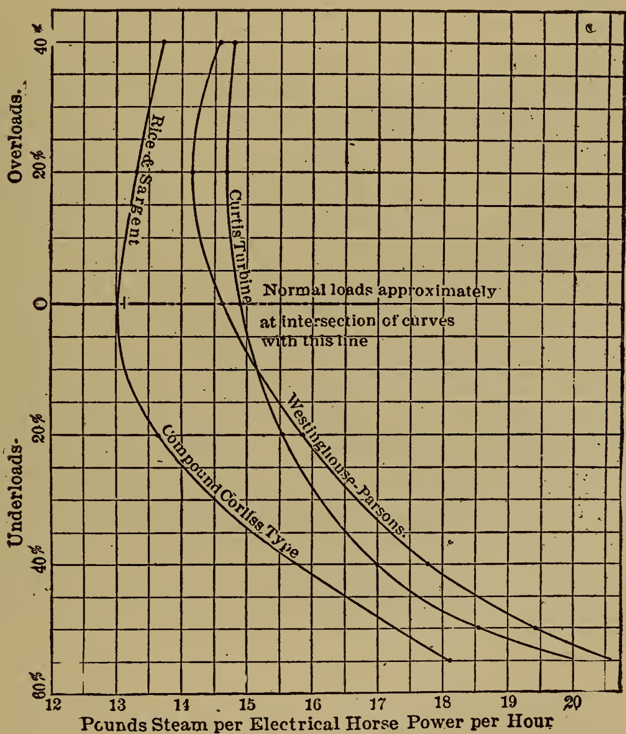


Fig. 486.

Under steady loads, the reciprocating engine at its best speeds will show a little better economy than the turbine. Under average conditions, the average high-grade reciprocating engine has the same economy as the average turbine. Fig. 486 shows plotted comparisons showing the water rate with underload, best load, and overload of three types. When builders are allowed to fix their own conditions and test in their own shops, the water rate will run from 14.5 down to 12.25 in sizes from 1500 kilowatts up to 3000 kilowatts.

Fig. 487 shows results of tests under varying conditions. The Curtis marine guarantee is for 14 pounds at 21 knots, and 23.1 pounds at 10 knots speed.

From the absence of alternations of temperature the turbine suffers less from internal condensation than the reciprocating engine; consequently the meaning of high superheat is not so great. The gain from it is rather a mechanical action in diminishing fluid friction among the blades than the thermal advantages it offers in the other type. Care must be taken not to let any excess of temperature get past the

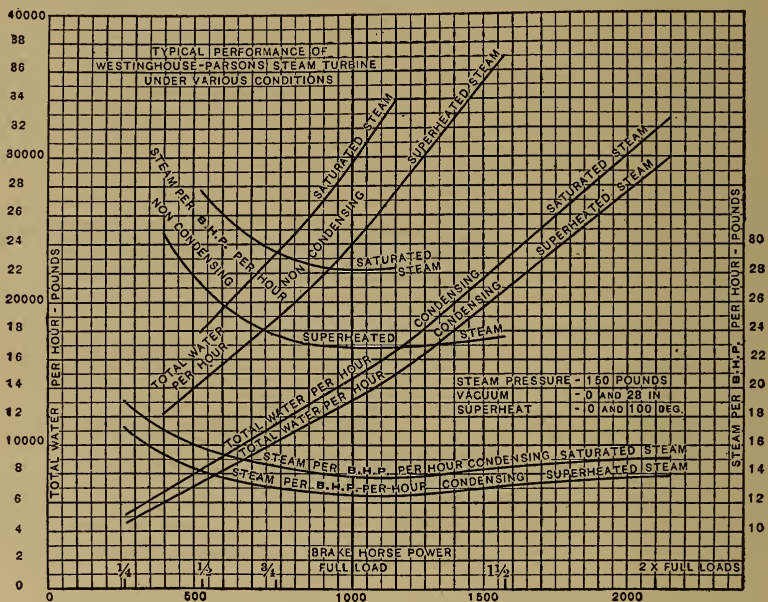


Fig. 487.

lower wheel unreduced by doing work, since its effect will be to increase the weight of condensing water or else to impair the vacuum. The turbine is exceedingly sensitive to the excellence of the vacuum.

361. Concluding Comment. The demand for the steam turbine and its success in meeting it are the outcome of the growth and development of the electrical transmission of energy in the form of light and power. The electrical generator has called for high speed; the turbine is at its best under these conditions. The economy of the steam turbine and its conveniences (paragraf 357) have called for its application to other functions than driving generators, and high-speed rotary machinery has been developed to meet the attributes of the turbine. It is safe to say, therefore, that more and more the turbine will take on itself the work heretofore done by the reciprocating engine, or as a supplement thereto (paragraf 355). This means that the turbine marks a stage in the development of the heat-power plant of importance analogous to that of the separate condenser, the principle of continuous expansion, the automatically controlled cut-off, and the economy of superheat. Where the speed is constant and high; where rotary machinery is applicable; where the resistance varies with the speed from zero at starting to the full load, the turbine has the field of the future, and will be rivalled in it only by the internal combustion motor. At first such motor will be reciprocating; if later knowledge of the laws of expansion should reveal some way of applying such expansion of hot gases to driving a gas turbine, this latter will again supplant the reciprocating or piston internal combustion motor. Hence the steam turbine has the electric power and lighting central station, electric lighting of trains and vessels and isolated plants; it has the high-speed passenger and naval cruiser service, the yacht and launch, the rotary blowing and pumping motor service. Most factories of the future will use electrical transmission of their power, with all the conveniences of that system.

The field least open to the turbine or closed to it is that where the maximum resistance has to be overcome in starting from rest, or at very low speeds of the motor machine. That is, the requirement of maximum starting moment or torque with diminishing intensity of effort per unit of time as the speed increases is that which the method of utilizing expansion in the reciprocating engine is best adapted to meet and the turbine least adapted for. These appear in the locomotive attached to a heavy train and the commercial motor-vehicle; in high-pressure pumping of large masses of liquid or against the resistance of long pipes; in compression of gases or air; in the side-wheel or paddle-wheel engine, the metal-working rolling mill, the hydraulic transmission of power or high-pressure fire-service for cities, in the freighter or towing service at sea or on inland waters. The reciprocating engine of good construction and high economy already installed will not be displaced until worn out many years hence. The reciprocating engine therefore retains and will retain sufficient importance to justify the further and exhaustive study of its constructive details, valve gearing and governing. The first step in logical sequence will be the foundation and bed-plate of a typical reciprocating engine, to whose study the treatment now returns.

CHAPTER XXI.

ENGINE FOUNDATION AND BED-PLATE.

365. Introductory. It must be obvious that when a piston is moved back and forth in the cylinder of a reciprocating engine under a pressure from expanding steam, the pressure upon the cylinder head is equal to that upon the moving piston. When the latter tends to move forward the cylinder has an equal tendency to slide backward. If the crank is to be moved by the piston, the cylinder must therefore be held stationary. It is important that the shaft of the engine be prevented from any other motion than its rotation in the bearings. The push and pull of the connecting-rod must be constrained so as to be in the desired directions only, to prevent wear, noise, heating, and lost work. Kinematically the engine being a four-link chain, one of these links must be anchored to something fixed, in order to render definite the motions of the other three. These conditions require that the cylinder guides and shaft bearings be so bound together that the forces at play shall not produce any undesired motions of the parts, nor cause shocks, jar, or vibrations to the machinery or building of which the motor engine is an element. The fixed link of the mechanism, or the connecting and unifying part of an engine to which its elements will be attached to keep them together and in line, will be called the bed-plate or bed, in stationary horizontal engine practice. In locomotive and vertical engines it is usually called the frame; in beam-engines for marine work, the lower element of the bed or frame is often called the sole-plate.

The bed-plate or frame should make the engine self-contained and capable of being erected anywhere where its weight can be adequately supported. This is measurably true of small and medium-sized engines where some attention has been paid to balancing. In the general case, however, it is not convenient to give to the bed-plate the mass necessary to withstand completely the shaking forces released in the engine when at work; in large engines the bed cannot be made rigid enough of itself to keep perfect alinement if the support below it is inadequate. Hence for both of these reasons it is customary to put the engine bed upon a masonry or concrete foundation, so that mass and support and alinement shall be furnished by it. Masonry and concrete give a mass absorbent of vibration and easily constructed in place. The foundation can go down below the frost-line, and is not liable to destruction below ground. Cast-iron surfaces, or sub-bed-plates, are designed with certain light and high-speed engines to lift the engine above floor-levels and enable small fly-wheels to turn clear of the top of the foundation. The bed-plate will be securely bolted to the foundation to make them one and compel them to act together in resisting shaking and vibration. How shall the desired mass of foundation be computed or arrived at? The foundation must resist motion laterally and up and down from the engine forces within it, or the pull of belts or transmission apparatus; it must support the static weight without tendency to unequal settling in the soil or ground on which it rests.

366. Mechanics of the Engine Foundation. Shaking Forces. The treatment of Chapter XV and paragrafs 259 to 263 requires to be extended when the engine is to be conceived as suspended from above by flexible cords with springs to carry its weight, and therefore free to move vertically or horizontally under the action of the forces at work within its own construction. The foundation is to take the place of the free suspension and resist any stresses which cannot be otherwise provided for by a balance of forces in the engine itself. The foundation discussion therefore opens up the dynamics of the engine masses in both the plane of the cylinder axis and at right angles thereto. In the former discussion only the inertia effects parallel to the cylinder axis were computed, and with respect to their effects upon the turning moment at the crank-pin: it may be that the effort to balance effects of mass and inertia will result unfavorably to the turning moment. The designer must decide which set of conditions is to be paramount. In the locomotive, the motor-vehicle, and the marine engine, the balance is so much the most important that it must receive first place whatever the effect on the turning effort at the crank-pin; in high and medium-speed engines of stationary practice there is usually a compromise with the balance partial but not complete, and in the horizontal plane where such balance is the most important. In very slow engines it can be disregarded, although the engine wears longer when attention is paid to exact design.

When the steam enters the cylinder behind the piston, it exerts a pressure PA upon the piston and the cylinder head. It has been made clear from paragrafs 260 to 263 that this effort does not reach the crank-pin and hence the bearing of the shaft, after motion of the piston begins, until the masses attached to the pin have been accelerated at or near half-stroke. Hence the piston head is forced away from the crank-shaft by the force PA with the mass of the reciprocating parts as its abutment during the acceleration; and during the retardation in the second half of the stroke the crank-shaft is being forced in the other direction by such stored living force given out upon the pin and shaft. The mass of the piston cross-sections and the cross-sections of rods and pins are all fixed primarily by the metal necessary to resist the intensity of PA; the mass due to this computation adds the value of the inertia stresses which these same materials and elements must resist. Hence it would appear that the stresses would fall into four groups:

1. Those imposed from without from the intensity of PA.
2. Those resulting within from PA and resulting centrifugal accelerations due to the speed. These two give the masses in action.
3. The turning moment of the crank-pin where these impressed forces pass effectively to their external work.
4. Unbalanced or balanced forces which do not get away to do industrial work at crank-pin or crank-shaft, but must be cared for in the foundation.

These last are the shaking forces. They arise from two origins:

1. Unbalanced forces and moments due to the reciprocating parts.
2. Unbalanced moments and forces due to the rotating parts. Inequality of the turning effort due both to the varying cylinder pressure and angular velocity of the fly-wheel and crank will be factors in this, or inequalities due to defective adjustment of the governing function to the resistance, or wide or sudden variations in such resistance as related to the stored energy of the fly-wheel. It would appear therefore that reciprocating balance can only be partly attained by a rotating mass, as is usually the practice, and that under variation of load and speed the balancing masses will disturb a previously existing rotating balance.

The theoretical condition for an equilibrium or balance around an axis of revolution is twofold. First, the algebraic sum of all moments revolving in the same plane around the axis of revolution shall be zero. If not, there will be another axis in space where they will be zero, and the system will press bodily or jump the shaft and weights to get into this axis. Mathematically, if W be the weight of each mass and R its distance from the center of the shaft, then $\Sigma\ WR = 0$. Secondly, if, as is the usual case, these weights or masses are not in one plane, the shaft will have a tendency to flex or bend, until the condition is reached that the algebraic sum of the moments $\Sigma\ WR$, when multiplied by a lever-arm y measuring the distance of the plane of rotation of each weight from a plane at right angles to the axis of rotation, shall also have an algebraic sum of zero, or $\Sigma\ WRy = 0$. The first condition gives a standing balance; the second gives a running balance.

Four differing sets of conditions will present themselves — the horizontal single and multi-cylinder engine: the vertical single and multi-cylinder engine.

In the single horizontal engine, the pressure PA at the dead-center is resisted by the main bearing and the cylinder head, which are in equilibrium. The intensity of this may usually be made secondary to that of the inertia effect of the reciprocating parts. This was shown in paragraf 260 to be for each square inch of piston area to be

$$p = 0.00034\, wRN^2,$$

or for the entire weight of the reciprocating masses

$$P = 0.00034\, WRN^2$$

when the obliquity of the connecting-rod is neglected or it is regarded as of infinite length.

Hence a balance weight W_1 in the plane of the crank-pin, opposite to the application of the reciprocating weight treated as a revolving weight and with a radius r_1 from the center of the shaft, would be

$$0.00034\, WRN^2 = 0.00034\, W_1 r_1 N_2,$$

whence

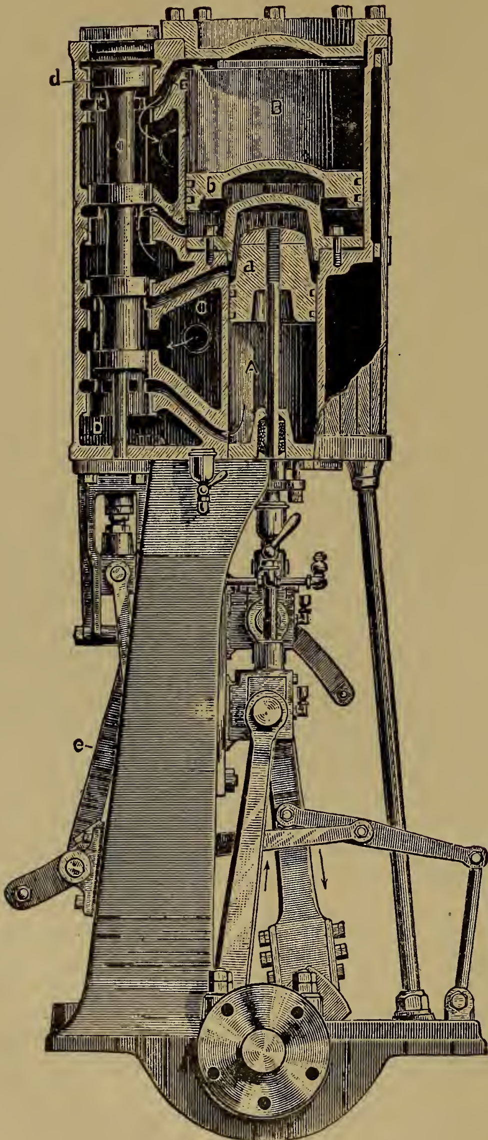
$$W_1 = \frac{WR}{r_1}.$$

It is not quite fair to consider the whole weight of the connecting-rod as reciprocating; the conventional allotment is to take one-half as rotating and one-half reciprocating. Then the value for $\Sigma\, W$ being found, their centrifugal moment about a plane parallel to the rotation of the crank-pin is made zero, to prevent a bending of the crank and pin or shaft. If this last were not done, there would be a tendency for the engine to shake sidewise and move laterally on its foundation at the end away from the crank — a revolution around a vertical axis at right angles to the shaft. Two horizontal cylinders side by side as in the cross-compound could be balanced with cranks at 180° except for the obliquity of the connecting-rod. With cranks at 90° the tendency to rotate horizontally around the vertical axis through the horizontal shaft reappears. The four-cylinder engine can be perfectly balanced, except for the small error of the connecting-rods. The only connecting-rod engine perfectly balanced is the type of Fig. 490, known in America as the Wells engine and in England as Barker's, in which the upper cylinder has two piston-rods and connecting-rods and the lower one has one. These act on crank-pins at 180° apart, and by proportioning their weights and masses both the pressure values and inertia forces are in equilibrium at all times less only variations in the expansion in the high-pressure cylinder. Three-cylinder engines with equal reciprocating masses and rotating moments are also in balance except for the twisting tendency to rock around an axis at right angles to the shaft.

It is a great advantage of the center-crank engine that the counterweight to balance the reciprocating mass when divided between the two cranks becomes symmetrical respecting the vertical axis through the shaft, and steadies the cylinder end from its tendency to sling.

But it is evident that by introducing the revolving counterweight to steady the engine from horizontal shaking the unbalanced revolving weight is going to produce shaking vertically. To attempt to balance the counterweight is to unbalance the reciprocating mass again. Hence the foundation becomes inevitable if the horizontal engine is to be balanced horizontally, so that foundation bolts can hold the bed down. In vertical engines, the balance in a horizontal plane or at right angles to the cylinder axis is the important one, so that equality of turning moment is usually sacrificed in practice.

In vertical engines with two cylinders side by side, the tendency with quartering cranks is to rock or swing about the edge of the bed-plate at right angles to the



DESCRIPTION.

A—The high pressure cylinder.
B--The low pressure cylinder.
C—The steam chest.
D—The receiver.
E—The exhaust passage.
a—The high-pressure piston.
b—The low pressure piston.
c—The piston valve.
d–The valve casing.
e—The reversing lever.
F—Crank-shaft.

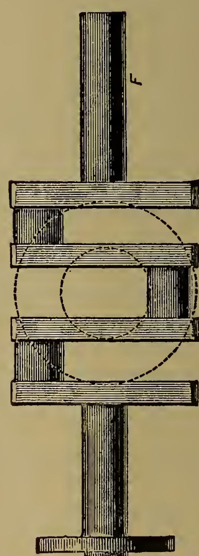


FIG. 490.

shaft. The four-cylinder side-by-side balances again, and the three-cylinder type except for the tendency of the cylinders to oscillate. The initial pressure also is effective to shake in vertical engines.

In the locomotive and the motor car where there can be no foundation, the practice has been to balance about two-thirds of the reciprocating or centrifugal weight and let the counterweight produce its effect downward upon the track and upward upon the mass of the engine or motor with the springs as absorbers between. The side swinging of the locomotive, called wig-wagging or elbowing, is much the most objectionable, as it causes jerks upon the traction or draw bars and makes the engine liable to derailment. Any tendency of the cylinders to lift is also dangerous, particularly on curves. Locomotive balancing also includes a balance for the effect of the reciprocating parts on the other side of the engine, at 90° from the one under observation, and the further factor that the counterweight being in the driving-wheel itself must move through space at twice the velocity of the engine itself or of the reciprocating parts when the counterweight is at the top of the wheel. The parallel or coupling rods also have to be allowed for in the computations for balance.

In computing the shaking forces, which can be done either analytically or graphically, it will be plain that the interest of the designer is upon the axial components parallel to the axis of the cylinder and those transverse or at right angles to them. The reciprocating masses act axially, and will be equal to a force $0.00034\ WN^2R$. The centrifugal components will be decomposable into two at right angles for each crank angle θ, giving the axial component $0.00034\ WN^2R \cos \theta$ and the transverse $0.00034\ WN^2R \sin \theta$. These should be added together when both are in action, giving a resultant in a curve on a straight line base.

A rough practical rule has been to give to the foundation a weight at least ten times the value of the impinging force which it is to resist. This is a practice based upon giving to an anvil ten times the mass of the hammer head delivering blows upon it. In applying it the plan has been:

First, to have the foundation deep enough to raise the engine cylinder axis to the desired level above the engine-room floor.

Second, to have it go deep enough into the ground or far enough below the surface to be beyond the effects which cause unequal settling either from frost, vibrations, or the influence of loads borne by adjacent ground. The depth below the surface desirable for an engine-foundation will vary, but it is rarely safe to permit less than three feet of foundation below the general level. In excessive cold and in exposed situations the effect of frost will be felt down to six feet below the general level.

Third, the engine-foundation must furnish sufficient mass to absorb vibration if the bed-plate is not massive enough to do it alone. The foundation being made up of masonry is easily and conveniently built up in place, while to make a massive casting would not only be more costly per cubic foot, but would make weights of such magnitude that handling would be troublesome.

When the foundation rests on rock and is not sufficiently massive it has been found that the vibrations caused by reactions in the engine are transmitted almost perfectly to the adjoining foundation upon the same rock. Great care has to be observed to attain success.

Fourth, the foundation must have area enough to support the concentrated weight of the engine upon the ground by distributing that weight over a sufficient area to prevent settling. Accepted figures for the supporting power of different soils are given in the following table:

Alluvial soil	from	.5	to	1	ton per square foot.	
Clay, soft	"	1	"	1.5	"	"
" dry	"	2	"	4	"	"
" thick	"	4	"	6	"	"
Sand, clean dry	"	2	"	4	"	"
" compact	"	4	"	8	"	"

Gravel and coarses and, from 4 to 8 tons per square foot if protected from water.

Hard rock, up to 200 tons per square foot in thick strata.

If the soil is so unreliable as to require piling, crib work, and other artificial underpinning, the student is referred to text-books which make a specialty of foundations.

Then if when these requirements have been met the weight or mass is less than one-tenth of the force per stroke due to the skaking effort, the necessary additional mass is added in depth and the other dimensions to make it so.

367. Construction of Engine-foundations. The foundation for very large engines will be of cut or dressed masonry according to the usual specifications for first-class masonry. Where the importance of the structure warrants it, tunnels or thoroughfares will be made or left through the mass of the foundation by which access may be had to the lower ends of the bolts by which the bed-plate is bolted to the masonry (Fig. 491).

For small engines, footings of rough masonry or ashlar may be used to distribute the pressure, and on this footing the foundation proper of brick will be built. The third plan is to make the foundation a monolith of concrete. Upon a proper footing to distribute the weight, a box of rough boards without top or bottom is laid, and within it successive layers of cement concrete are thrown in and well rammed until the desired height is reached. When brick is used it should be of first quality, hard-burned, and laid in cement-mortar. Common lime-mortar is liable to crumble and disintegrate under vibration, and the whole principle of the foundation is to have it act as a solid mass. When appearance is to be considered, the face of the brick foundation may be made of face or

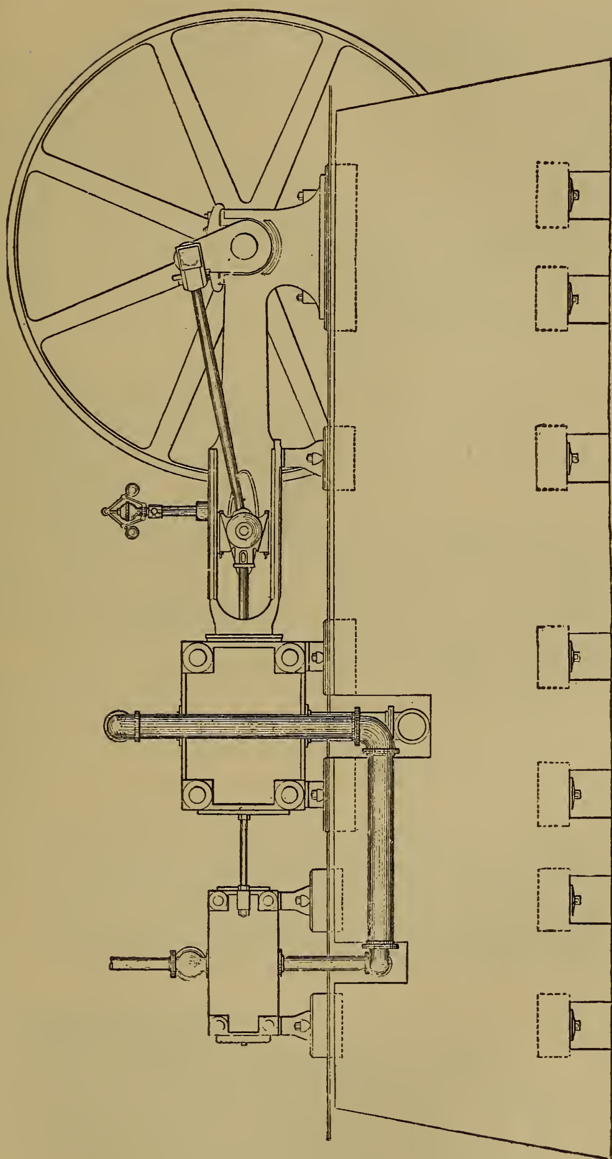


Fig. 491.

pressed brick, while the interior is of ordinary grades. Since the bed-plate is to be bolted to the foundation, the greatest care must be observed in locating the necessary bolts in their proper places.

368. Footings to Prevent Vibration. The mass which it is convenient to get in a vertical engine bed-plate is often not enough to provide for the absorption of all vibration. The vertical engine, when chosen because floor-space is to be saved, does not call for extended area in the foundation, so that sufficient mass can only be gotten by going deep. Where this is inconvenient, or where rock is struck, engineers have had to provide special footings to arrest vibration. It has been tried by some to underlay the foundation proper with timber or rubber, but a springing material of a class to which these belong is often the occasion which causes the very difficulties they are designed to prevent.

Vibration of machinery or any solid substance is of two sorts: the material either swings crosswise as in the vibrating string of a musical instrument or in a flapping belt, or the motion of the particles is lengthwise or parallel to their long axis. If the oscillation or vibrating period of the material used as an absorber of vibration happens to coincide with the vibration period caused in the engine-frame by the speed of reciprocation or by the belt-flap, the deadener partakes, and multiplies the objectionable vibration. What is to be sought to deaden vibration and arrest its transmission is some material to underlie the foundation which shall be without any resilience whatever.

Probably no better material is to be found for the purpose of stopping vibration than sand, if it can be kept dry and all motion prevented, and the foundation-block itself is of sufficient mass. The foundation-pit is dug two or three feet deeper and two or three feet wider on all sides than the foundation proper is to be. This pit is surrounded with proper sheathing to prevent the displacement of the sand, which is filled in two or three feet below the bottom of the foundation, and then around it on the sides as it is built up. Hair-felt or mineral-wool layers have been used underneath the footing-course. If the foundation-block is not massive enough, these methods or expedients only aggravate the difficulty which they are intended to cure. Very satisfactory results have been obtained abroad from the use of asphaltic concrete for massive foundations. It possesses a certain sort of elasticity with its massive character, and its period of vibration is so definite and so much shorter than the period of the engine's vibrations that the latter are broken up and neutralized before they reach the transmitting rock or hard-pan.

If any part of the engine bed or frame be flexible from its construction or from the slacking off of bolts, a possible vibration of the frame itself may take place, due to its elasticity and the modulus of the material.

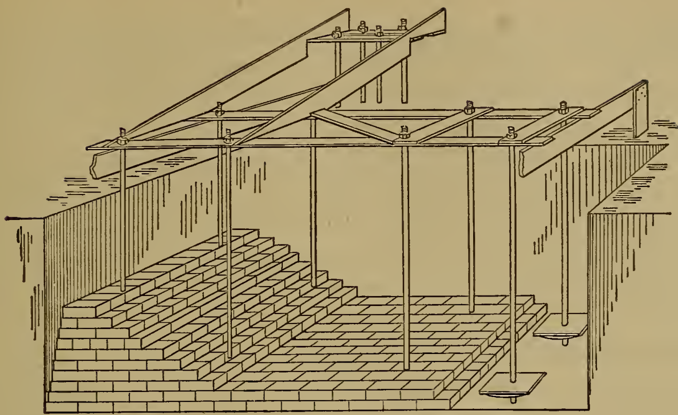


FIG. 492.

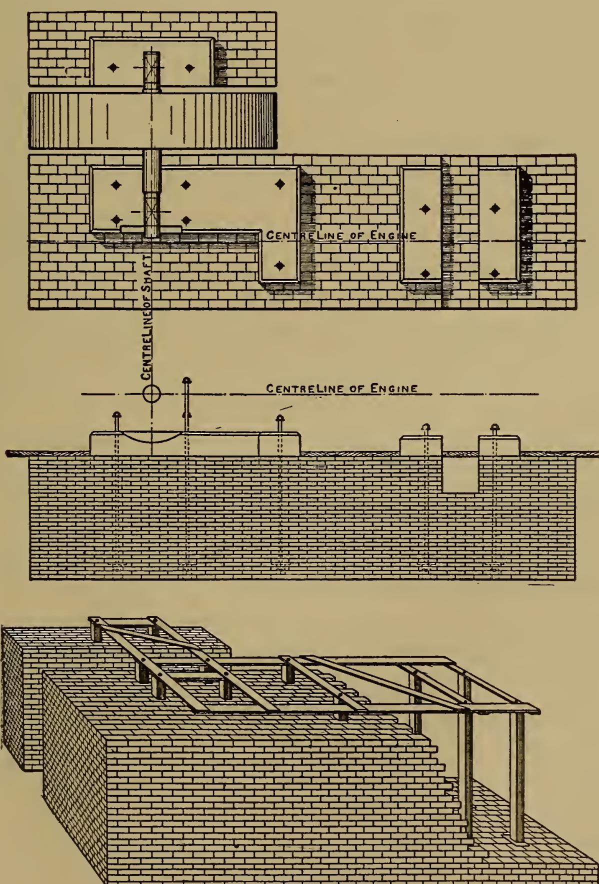


FIG. 493.

Stiff frames with a very short natural vibration period are better than simpler ones of a longer period, since the possible synchronizing of the reciprocation period with that of the vibrating bed would make dangerous and impossible stresses.

Most annoying vibrations are caused in high-speed engines by the impact of steam in an exhaust-pipe with elbows. The difficulty is intensified when there are water or oil drops in the exhaust current. Their impact against the elbow which deflects them will set lengths of pipe atremble, and their motion will be transmitted over a very extensive area.

369. Foundation-bolts. The bed-plate requires to be strongly and stiffly secured to the foundation in order that the latter may act with the bed-plate as one mass, and to prevent the bed-plate from moving upon the foundation. These bolts will vary in size with the size of the engine, but it is very undesirable to use bolts of such small diameter that it can be possible to twist them off with any ordinary wrench. Common diameters of bolts for engines of medium size would be from $1\frac{1}{8}$ to $1\frac{1}{2}$ inches diameter. The largest engines will require 2-inch bolts, but the smallest would use $\frac{3}{4}$-inch. The length of these bolts will be determined by convenience. It is desirable to have them go a good way down into the foundation, if not all the way to the bottom, in order that the upward strain upon them may be widely distributed in the foundation.

The location of these bolts in the foundation must be determined by the holes in the bed-plate through which they have to pass. It will be seen by examining typical bed-plates that as a general rule there are bolts at the cylinder end, or in the feet, and bolts at the crank-shaft end (Figs. 491 and 493). The bolts, furthermore, have to be built into the foundation, and at such a height that when the foundation is completed, and the bed-plate placed upon it, the upper end of the bolts shall protrude through the holes in the bed-plate enough to take the nut which these upper ends are to carry.

The method used to secure this object is shown in Figs. 492 and 493, which present a typical arrangement for this purpose. The wooden frame is called a template. It has holes made through it at points which correspond to the holes in the bed-plate, and when the nuts are in place on the upper end of the bolts the template is adjusted to the proper height above the datum plane, or plane of reference, and the foundation is built around the hanging bolts. The lower ends of the bolts are fitted with thread and nuts, on top of which rest the bearing or distributing plates or washers of cast or wrought iron. The distributing-plate is to enable the effort of the bolt to be borne by a number of bricks without danger of pulling through, and the nut and thread permit a vertical adjustment of the bearing-plate so that it shall come at the under surface of a joint in the coursed masonry. To permit of a certain limited horizontal adjustment of these foundation-bolts, several builders have surrounded the bolts with a length of pipe or a hollow wooden box making a round or square tube reaching from the bearing-plate to the top of the masonry. The diameter of this pipe is so chosen that the bolt can be deflected within the hole which the pipe makes, and, after the bolt is in place and the alinement completed, the space between the bolt and the pipe is filled with cement and the position of the bolt is fixed. The template in Figs. 492 and 493 shows the bolts required for the outer bearing of the engine-shaft attached to the principal template. This is usual when drawings of the template are furnished by the engine-builder and it is desired to make the foundation all in one piece. Where the length of the engine-shaft makes it desirable to have a separate foundation for this outer bearing it is usually more convenient to work with an independent template.

370. Alinement of Foundation-template. The foundation-bolts of the bed-plate will bear a certain relation to the axis of the cylinder. The axis of the cylinder should be in a plane truly at right angles with the axis of the engine-shaft. If the engine-shaft is to drive a line-shaft by belting or gearing, these two shafts should be truly parallel. Hence it is of prime importance to have the cylinder-axis perpendicular to the line of shafting, and the template which carries the bolts must be very carefully placed or squared with respect to these determining lines. The drawing furnished by the builder of the engine from which the template is to be made usually has on it the center-line of the cylinder, so that it can be laid out upon the boards of the template.

For the obtaining of the vertical plane through the cylinder-axis a line stretched over the foundation-pit and carried to suspended plumb-bobs is the usual device. For laying off the center-line of shafting or wall-lines the expedient of snapping a chalk-line upon the floor is the most convenient. The centers of the shaft are transferred to the floor by plumb-lines, or offsets may be taken from permanent walls. Such center-lines having been established, the plane at right angles to each is established by points and lines, using either a transit with graduated horizontal limb and making repeated readings, or by the ordinary geometric methods, or by the use of a massive T square whose head and blade exceed six feet in length and whose squareness has been carefully verified.

If a pulley or belt-wheel has been placed upon the line-shaft to which it is desired to draw a perpendicular line, a most convenient method is to stretch a twine or fine wire across the diameter of the pulley as nearly as the shaft will permit. With pulleys which have been turned, the edges of the face determine a plane perpendicular to the axis, so that the tense string touching the face at one point will only touch the face at a point on the other side of the shaft when the further end of the string lies in a plane which is perpendicular to the axis. This same method is a very convenient one to extend for the purpose of bringing two shafts parallel to each other where both carry pulleys, but is only applicable for either use where both pulleys are so fitted as to run perfectly true when the shafts revolve. It is one of the prime advantages of the system of electrical transmission of power that such care in alining the engine is of no importance.

371. Locating the Bed-plate on the Foundation. The foundation being completed, the bed-plate is to be lifted upon it and lowered into place with the bolts passing up through the holes in the bed-plate.

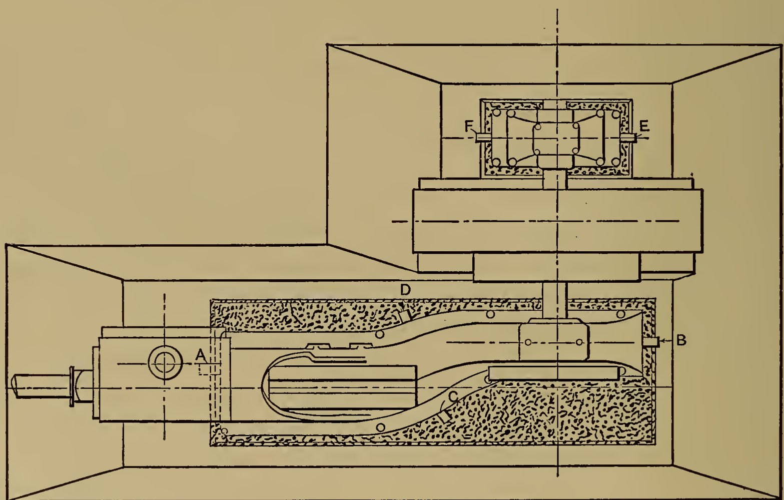


FIG 494.

Where cranes or similar lifting appliances are a feature of the power-house equipment this process becomes simple. In their absence the bed-plate must be lifted by jacks and blocking high enough to clear the bolts. It must then be rolled on skids into place, and then lowered by the successive withdrawing of the blocking.

The masons or bricklayers who have built the foundation do not usually have appliances for working to as close dimensions or as accurate levels as the setting of the engine requires. Furthermore, the top of the foundation is rarely a true plane, while the bottom of the bed-plate is very nearly a plane as a rule. It is necessary, therefore, to make a joint between the bed-plate and the masonry-work which shall support the bed-plate all over and in a plane as nearly level as it can be made. This process is so much easier when the brickwork or jointed masonry is covered by a single flat cap-stone, that where the dimensions of the foundation permit its use it will be preferred. It is usually a sawed or planed slab of bluestone or flagstone from four to six inches thick, and a little larger than the foundation-pier to which it serves as a finish or coping. The holes for the foundation-bolts have to be drilled in it, and it is lowered to its place upon a good bedding of cement. In the absence of such a cap or coping the bearing of the bed-plate comes upon a surface which is full of joints. The bed-plate is lowered over the foundation-bolts, and rests upon thin flat shims, or wedges of metal, which are placed on each side of the bolts between the bed-plate and the foundation. *A, B, C,* and *D* in Fig. 494 show such adjusting wedges. The nuts of the bolts are then screwed home, compressing the shims, while the bed-plate is carefully leveled as the strain is taken at each bolt. By driving in or loosening the shims any distortion or warping of the bed-plate by the bolts is prevented, and the bolts are tightened home until they refuse to go further.

The bed-plate is now rigidly bolted to the foundation and rests upon a number of points in a plane. Between the bed-plate and the foundation is a place between the shims equal to their thickness, and this joint requires to be filled. The materials used for this purpose in setting a bed-plate and making the joint are five. They are methods applicable to the setting of any machinery.

1. Shredded oakum may be driven into the joint with a chisel, as the seams of wooden vessels used to be calked. This makes an elastic sort of joint, but it lacks permanency.

2. Felted hair is used in the same way and has the same properties.

3. A rust-joint, as it is called, may be used. This is made by taking a thin cement-grout into which cast-iron borings or chips are introduced with a little powdered sal ammoniac and flour of sulphur. A dam of putty or clay is made around the outside of the bed-plate, and this mixture run into the joint and well worked in with a trowel. The rusting metal unites the mixture to the iron, and the cement to the stone.

4. The sulphur-joint. This is one of the most widely used methods for bedding the engine. A clay or putty dam is made around the bed-plate, and the ordinary roll sulphur melted in an old kettle and poured into the joint between the bed and the masonry. It expands on solidification somewhat like ice to fill every interstice and give full support

to the bed-plate. It undergoes no deterioration from oil or vibration. If care is not taken in melting the sulphur, it will become too hot and begin to oxidize, giving off an irrespirable gas. Sulphur in melting becomes fluid at a comparatively low temperature, becomes more viscid as the temperature rises, and passes to a second fluidity just before it is ready to burn.

5. The type-metal joint. Advantage is taken of the property possessed by certain antimony alloys (such as Babbitt, type-metal, etc.) of expanding at the moment of solidification, to use them for bedding or jointing bed-plates. The method of using them is the same as that practiced with sulphur, and they are preferred by many engineers particularly for bedding the narrow feet used with Corliss bed-plates.

372. Alinement of Outer Pillow-block or Shaft-bearing. In engines of the center-crank type (paragraf 290, Figs. 406, 408) the unit is self-contained, and all bearings are on the bed-plate itself. In the side-crank type (paragraf 289) only the bearing next to the crank is carried by the bed-plate and the bearing which is outside the generator or belt or fly-wheel and near the outer end of such shaft on the separated element of the foundations must be carefully adjusted. This care is necessary for the following reasons:

If the outer bearing is too high or too low, it will force the crank to revolve in a plane making an angle with the true vertical plane, and twist the connecting-rod in each stroke. If out of place in a horizontal plane while correctly located in a vertical plane, it will force the crank to revolve in a plane which makes an angle with the axis of the cylinder, in which case it will bend the connecting-rod in each stroke; or it may be out of place in both planes, so that the connecting-rod will be both twisted and bent. The effect of either or both errors of alinement of this outer bearing is to wear the crank-pin out of its cylindrical shape, and to cause a knock or pound, and heating at the joint, which no adjustment of these bearings will cure. The proper method of alining the outer bearing involves, first, the establishment of the true axis of the cylinder after the bed-plate is in place and the foundation-joint complete. This is best done by stretching a fine piano-wire through the empty cylinder, carefully adjusting it to the center of the bore and fastening it tightly stretched to walls or fixed objects. To get the wire central is a matter of painstaking care and trial with gauges of wood or metal whose length is the cylinder-radius. The axis of the cylinder being established, the shaft and crank are put in place in the bearing on the bed while the outer bearing is provisionally supported and located. The shaft is then turned until the crank coming towards its inner dead-center touches the wire which marks the prolongation of the cylinder-axis. It will touch it at a certain distance from the end of the crank-pin and from one of its collars (Fig. 406). The shaft is then turned over until the crank-pin approaching its outer dead-center touches the wire. It will only touch it at an equal distance from its end or some fixed collar if the shaft is revolving around an axis truly at right angles to the wire. The outer bearing should be adjusted horizontally until the wire cuts the crank-pin at the same point and in the same plane on its outer and inner centers.

The adjustment of the horizontal plane may be effected by a sensitive level if the engine has also been leveled in the plane at right angles to the cylinder-axis in setting upon the foundation. A more sensitive and satisfactory vertical adjustment of the outer bearing is made by putting the crank-pin at 90° from its dead-center, and holding a plumb-line so as to touch the wire at the pin, noting the distance of the vertical plane thus established from the end of the pin or a fixed collar. If the plumb-line touching the wire also touches the crank-pin at the same distance from the reference-mark when the pin is at half-stroke below the wire, then the pin is turning in a vertical plane through the wire, and the outer bearing requires no vertical adjustment.

Where adjustment is required the usual procedure is followed of correcting half the error and testing the alinement again. The outer

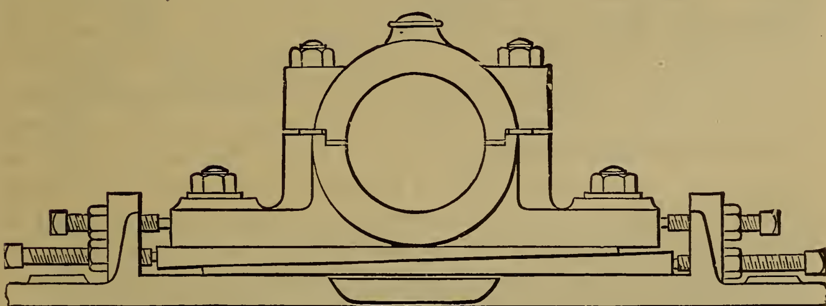


FIG. 495.

pillow-block is often made to rest upon a special foundation-plate which has provision for the adjustment upon it of the bearing proper in both the horizontal and vertical planes (Fig. 495). The vertical adjustment otherwise is made either by shimming or filling in with sulphur or type-metal below the plate, and the last and finest adjustment can be made by liners underneath the bearing-brasses. The alinement of vertical engines is usually simpler than that of horizontal engines, because the bearings are always on the bed-plate and have been made right as to alinement by the builders in their shop-handling. The alinement in the erection of beam-engines is a simple and obvious extension of the principles laid down above. The vertical cylinder-axis and the vertical through the center of the crank-pin when the latter is at the top and at the bottom of its travel determine the vertical plane in which the beam must play, and the crank-pin at its 90° and 270° point must remain in that same plane. The alinement of engines afloat is so complicated by the motion of the hull that little use can be made of perpendiculars and horizontals, and the center-lines must be depended on entirely.

373. Forms of Engine Bed-plate. Horizontal Types. The bed-plate of a horizontal reciprocating engine appears in a comparatively few forms. The strains being the same in kind, differing only in intensity, the material to resist them will naturally be disposed in much the same way by different designers. Historically an early type is known as a tank or box bed-plate. It consists essentially of a box very much longer than it is wide, without top and often without bottom. The sides are made up of a combination of moldings, and the top of the sides is formed into wide flanges upon whose upper surface are bolted the cylinder and guides and the crank-bearing of the shaft. The space between the sides gives room for the motion of crank and connecting-rod. It doubtless received its name from the practice with condensing engines of utilizing the area below the cylinder and mechanism to accommodate the tanks used for the hot or the cold well (paragrafs 508 and 515). Fig. 507A in the Appendix shows a tank bed-plate of the ordinary type. It may also derive its name from its resemblance to a cast-iron trough. It usually has cross-ribs or girts to give it stiffness.

Derived logically from the box or tank-bed is the one-piece bed or two-piece bed and sub-base type of Fig. 496, which shows three forms by different builders. The cylinder casting is placed upon the end of the bed-plate instead of on the top, and is either cast in one piece with it, or bolted up. The one-piece method is open to the objection that a failure of any cylinder detail which necessitates a new cylinder casting necessarily compels a new bed-plate. The guides are either formed in the proper recess of the casting (middle and lower parts of Fig. 496) or are separately mounted as in the upper figure. The cylinder and guides in the middle and lower system are bored by one bar so as to compel alinement. The cylinder is free to expand from the inner head without flexing itself or the bed, and the exhaust pipe leads away very directly. This form of bed is very usual for small and medium sized center-crank engines and lends itself easily to the side-crank type also by molding an addition or bolting it to the sub-base to carry the bearing for the outer pillow-block (lower detail of Fig. 496). This

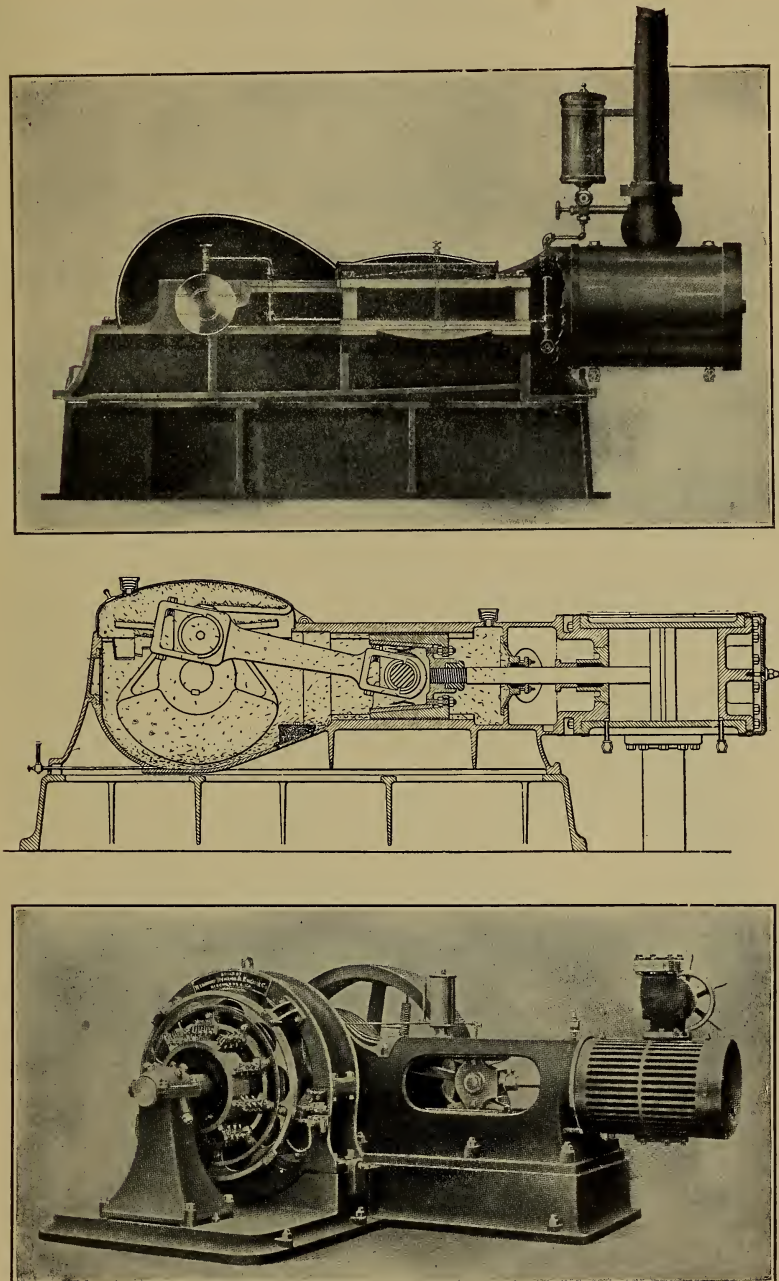


FIG. 496.

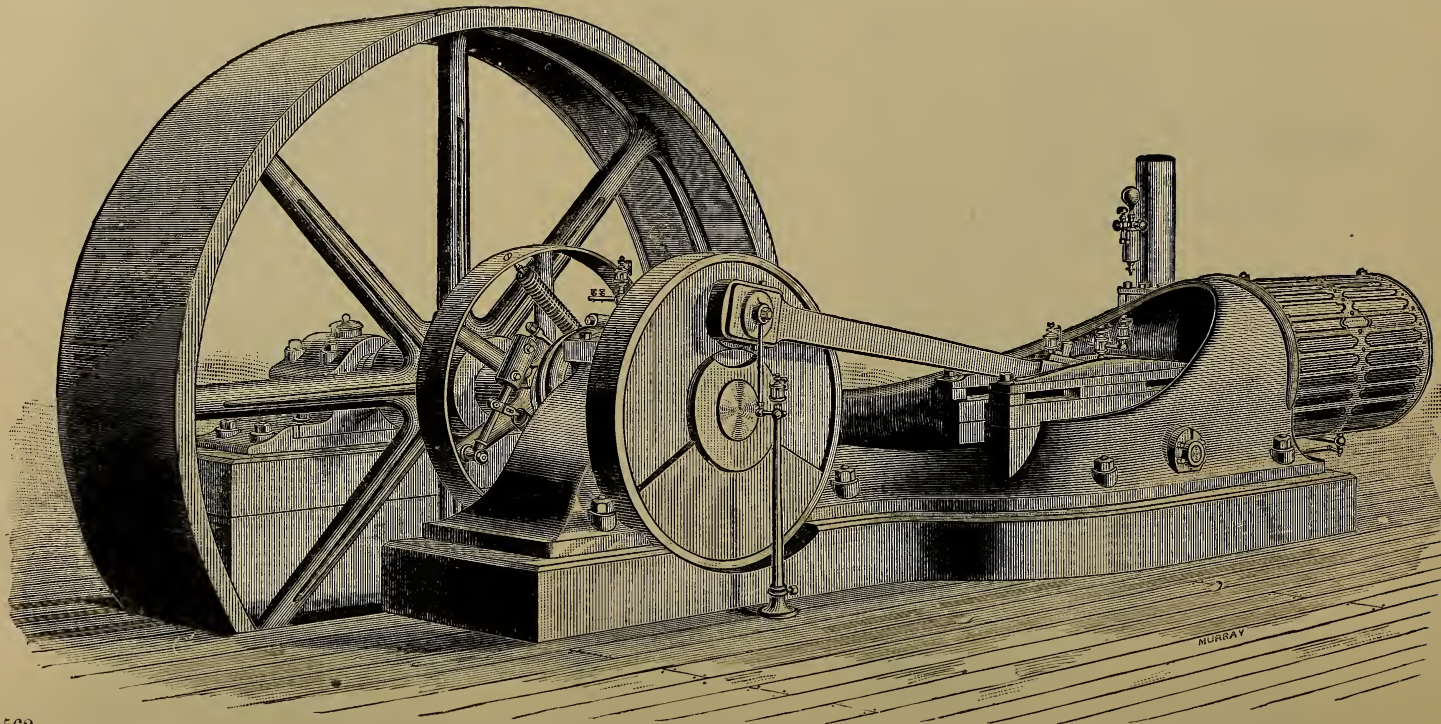


Fig. 497.

appendage can carry also the mountings for the pole pieces of the generator. The fundamental conception of this type of bed-plate was due to Mr. Charles T. Porter of America and was first largely used in England by Tangye. It is known by either name in the standard form shown in Fig. 497, with the guides separately attached. The recent American modifications and developments of splash lubrication of the mechanism have developed the enclosed unit types of Fig. 496 and the semi-enclosed of Fig. 498.

For slower speed engines with longer crank-arm, longer cylinder, and longer connecting-rod allowance, this form becomes too flexible without

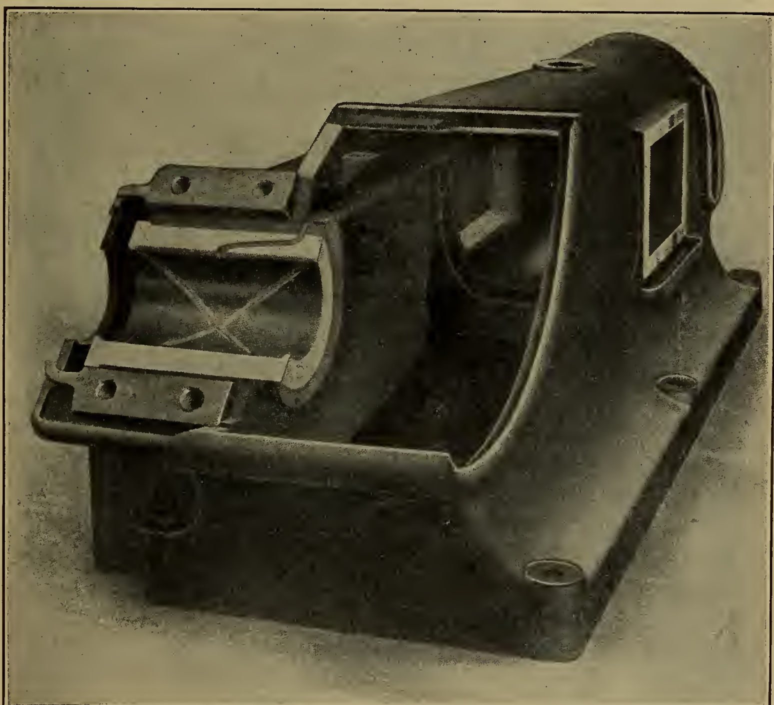


Fig. 498.

a foot or support for the cylinder. Fig. 499 shows the type which will result for a belt drive with independent outboard bearing, masonry cap-stone, and an independent foot for the cylinder. This latter should enable the cylinder to go and come laterally by expansion by oval holes for the bolts which are vertical.

A fifth type of bed-plate adapted only for long-stroke slow rotative

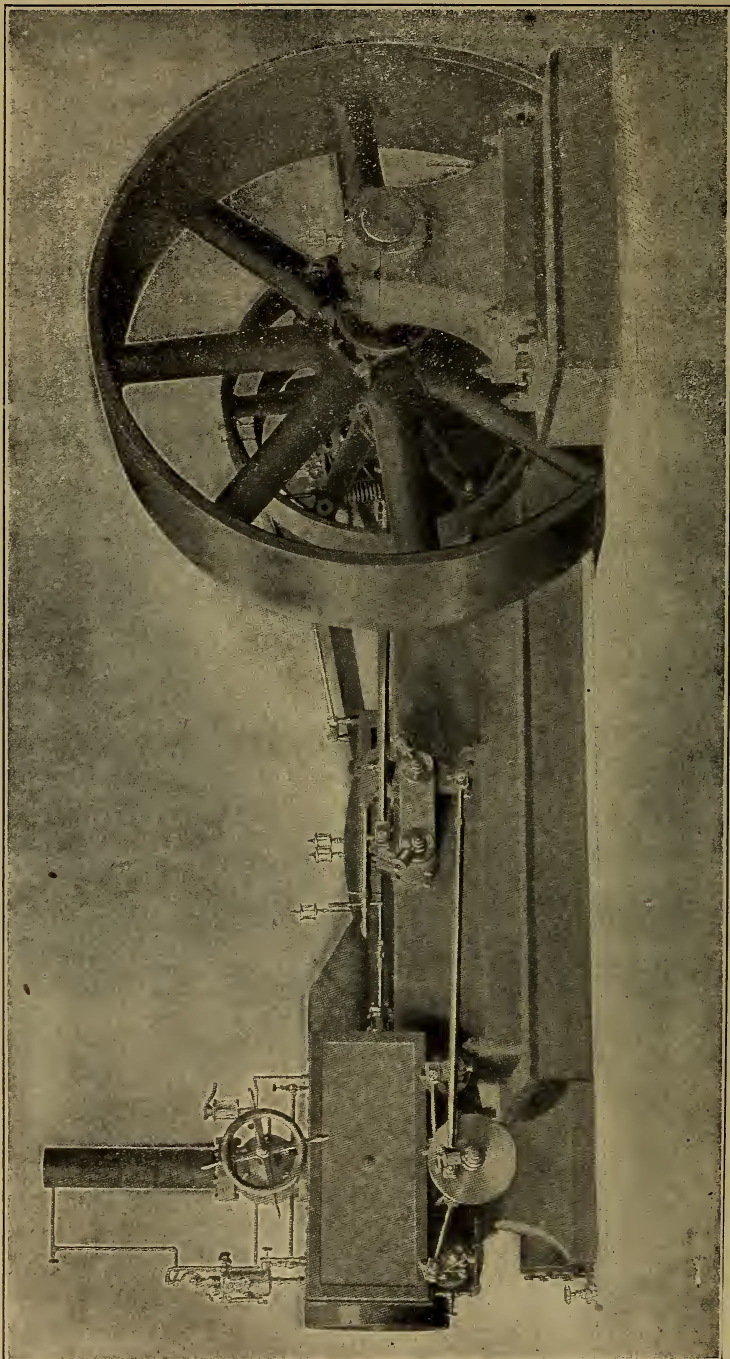


Fig. 499.

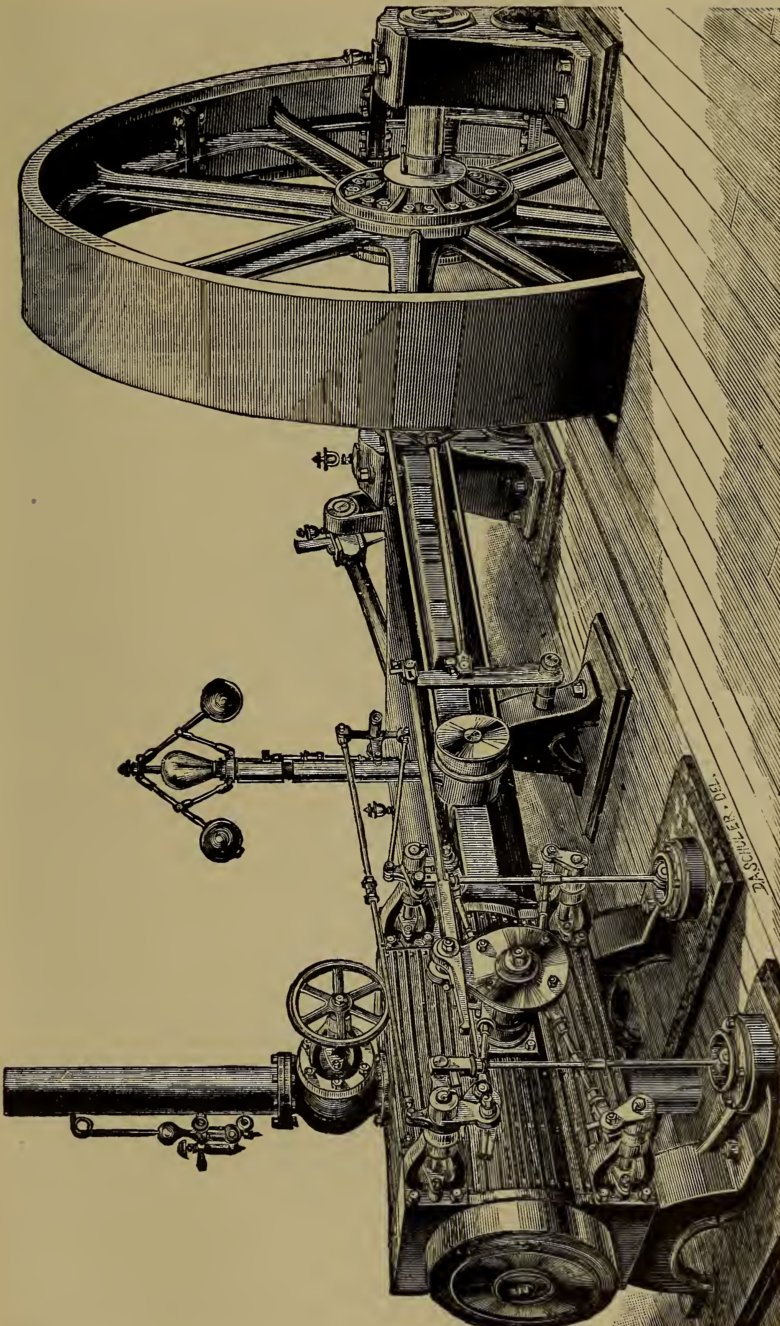


Fig. 500.

speed engines is designed so as to dispose of the metal required in a bed-plate more economically in the line of the stresses. It appears in many forms identified with the names of a number of various builders. That which is usually identified with the name of Corliss in America transforms the bed-plate into a brace between the two independent castings of the crank-bearing and cylinder. Each of these has its own supporting foot or pedestal, and the bed is a casting bolted to each, and either not supported by any contact with the foundation or by a central foot only. This form of bed-plate is sometimes called the girder bed-plate because the shape of the brace, in order to resist the strains upon it, becomes that of an I in both the vertical and in the horizontal plane. Fig. 500 represents a standard bed-plate of this type, and Fig. 501 shows a section through the girder at the guides (see also other Corliss designs, Figs. 438, 504).

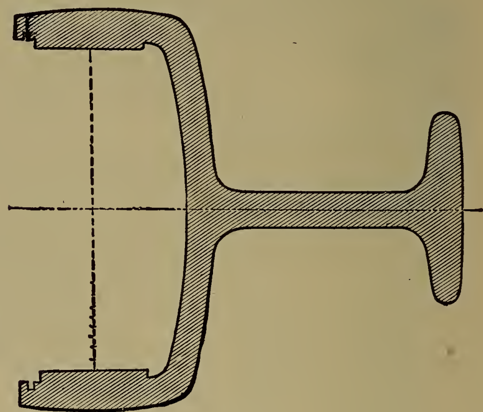


Fig. 501.

The foundation plan of Fig. 493 is intended for an engine of this type of bed-plate, and shows the separate cap-stones. It has become recognized, however, more and more that economy of metal in the bed-plate offers no advantages except in marine, locomotive, or motor-vehicle practice. Hence the newer designs for heavy duty particularly have shown a return in even slow-speed types to the greater masses of the Porter and Tangye types, and make the guide-section of cylindrical or box type, with or without its own foot (Fig. 502) for the medium sizes or carry the sweeping curves of solidity and grace pleasing to the eye as suggesting strength back to the cylinder head, with the guides supported as in Fig. 503. In the heaviest of all demands, for compound tandem rolling mill and street railway service the bed-plate will be carried solid at least to the first cylinder, Fig. 504, and the second one will have its own additional support, with a tie-piece to the first cylinder (Fig. 505). There are of course numberless modifications of the types which have been selected, each of which will be deserving of study, both as to the disposition of the weight of metal and the pleasure which they give to the eye as respects line and proportion. A most interesting departure

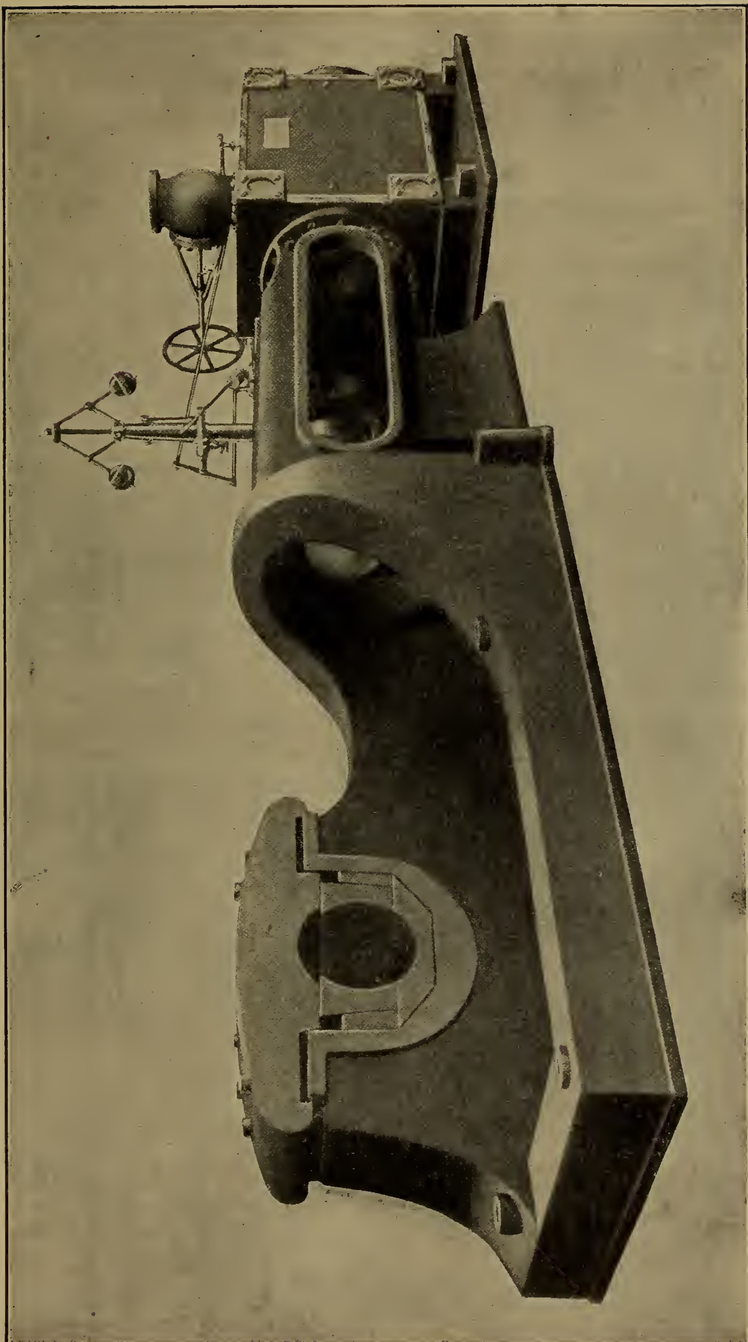


Fig. 502.

from conventional lines for a high-speed engine of light and medium power was the bed of the "straight-line" engine of Fig. 506. This design carries the principle of permitted expansion to its logical end by having the crank-bearings tied to the cylinder by straight-line braces bolted to both, but the cylinder is not fastened to the foot which supports it, but simply rests upon a bearing-surface. The engine can be designed to have all components downwards upon the pedestal in the absence of rigid connections, which removes the tendency to distort in expanding. The straight-line principle appears in many of the high-power heavy-duty types.

374. The Bed or Frame of a Vertical Engine. It has been seen (paragrafs 273, 274) that the vertical engine has the cylinder almost always over the shaft. Hence the frame becomes a proper casting to carry the crank-shaft from which a suitable columnar structure shall arise to carry the weight of the cylinder and serve also to guide the cross-head. The general appearance of these columnar castings in the usual marine engine has given them the name of A frame (Figs. 389, 391). In recent designs, to secure greater accessibility for the mechanism one side of such frames is made of hollow steel columns or rods, such as are fitted on the engine shown in Figs. 380, 381. Such engines are called open-front or open-side engines. Accessibility is a prime necessity of good design for vertical engines of this class, and is much better secured with such open frames.

In beam-engines the bearing for the beam requires to be so designed as to keep satisfactory alinement. In early designs it will be found to resemble a massive column or pillar (Fig. 402A); in later engines a nearer approach has been made to the A frame or gallows-frame usual in river-boat practice. The gallows-frame in recent large engines is made of steel plate molded into box-girder shape and strongly braced.

In the development of the construction of the typical engine the obvious detail to follow those of foundation and bed-plate should be the cylinder and its attachments.

CHAPTER XXII.

ENGINE CYLINDER, PISTON AND PISTON-ROD.

375. The Cylinder Casting. The discussion in paragraf 259 of the stresses upon the elements of a typical engine should have made clear that overlying the computation of thickness for the walls to resist the stress PD by a balance of resistance $2\,tf$ are such other stresses from water and from heat that the thickness of metal to be used in the cylinder is fixed rather on the basis of experience, by the condition of stiffness against deformation, and to be thick enough to permit of reboring when worn.

The metal to be used for a cast-iron cylinder should be a uniform close-grained iron having a certain hardness or ability to resist abrasion. Experience in mixing irons in the foundry is of great use in this respect, and excellent results have been obtained from the use of an iron containing manganese. This metal seems to give a smooth or slippery surface to resist abrasion, while working easily under the pointed cutting-tools. The design of the cylinder will be widely modified by the design of the valve-gear to be used and the type of valve chosen. In the horizontal engine the valve-chest for sliding flat valves or piston valves will usually be at the side of the cylinder, or at the side and bottom if more than one valve is used. For rocking or oscillating valves of the Corliss type, the seats and chambers will be made in the heads. The valve-chest is rarely put on the top of the cylinder except in locomotives. The reason for the preference for the side is the directness of the connection from engine-shaft to valve for driving it, while when the valve is on top or out of the plane through the center of the shaft, the motion to the valve has to be indirect or by means of levers and rock-shaft. In vertical engines the valve can always be directly driven. Inspection is invited of the cylinder sections shown in Figs. 436 and 437 for variations from the typical design selected as Fig. 510 to show certain features of construction.

The casting of the cylinder is either made all in one piece with the massive bed-plate or it is bolted to it. When made in one piece, as is usual in engines of the Tangye bed-plate pattern, a joint is avoided at the crank end of the cylinder, and no difficulty is to be experienced from the cylinder shifting its alinement with the bed. On the other hand, the finishing of that end of the cylinder is made more difficult. In Fig. 510, which represents a bolted cylinder, there will be observed radial set-screws attached to the flange on the crank-head, whose func-

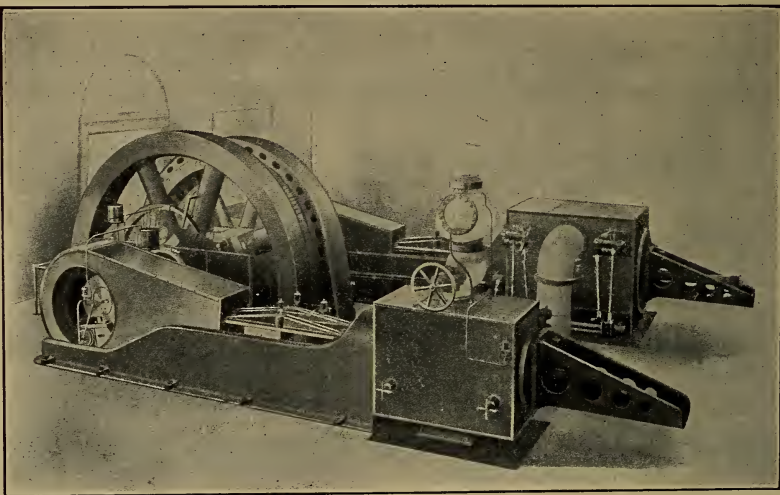


Fig. 503.

tion it is to secure and adjust the alinement of the cylinder and the bed-plate.

The cover of the cylinder is bolted to the cylinder proper by means of

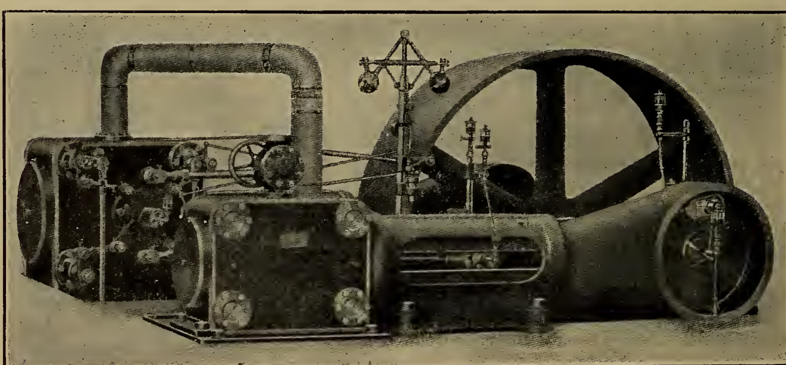


Fig. 504.

a series of studs, whose inner end is tapped into the flange or solid metal of the cylinder, and whose outer ends carry nuts by which the cover is held steam-tight to its place. The joint between the cylinder and the

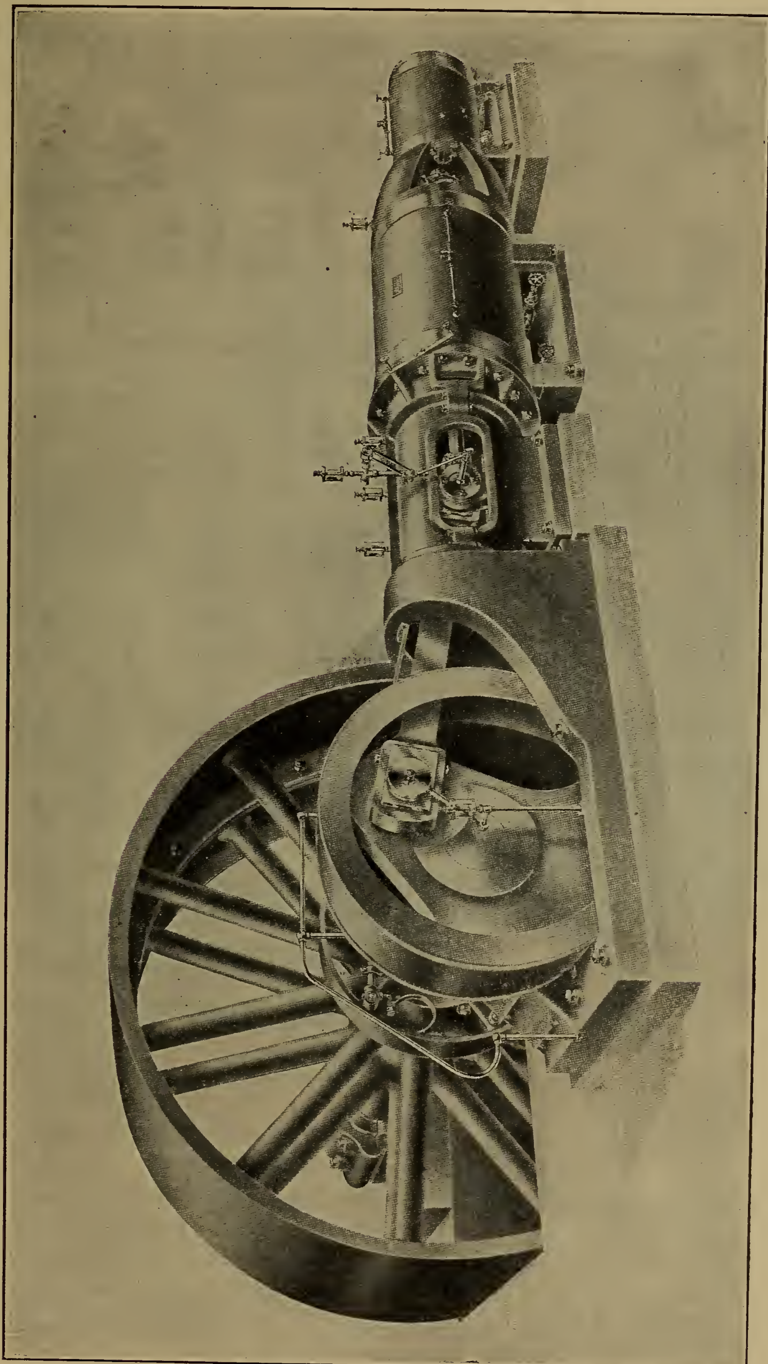


Fig. 505.

cover is a ground or metal and metal joint and requires no packing, or at most a gasket of oiled paper. Many designers use a cross-section of the studs so that in case of entrapped water the stretch of these bolts within their elastic limit shall open the joint enough to release the water.

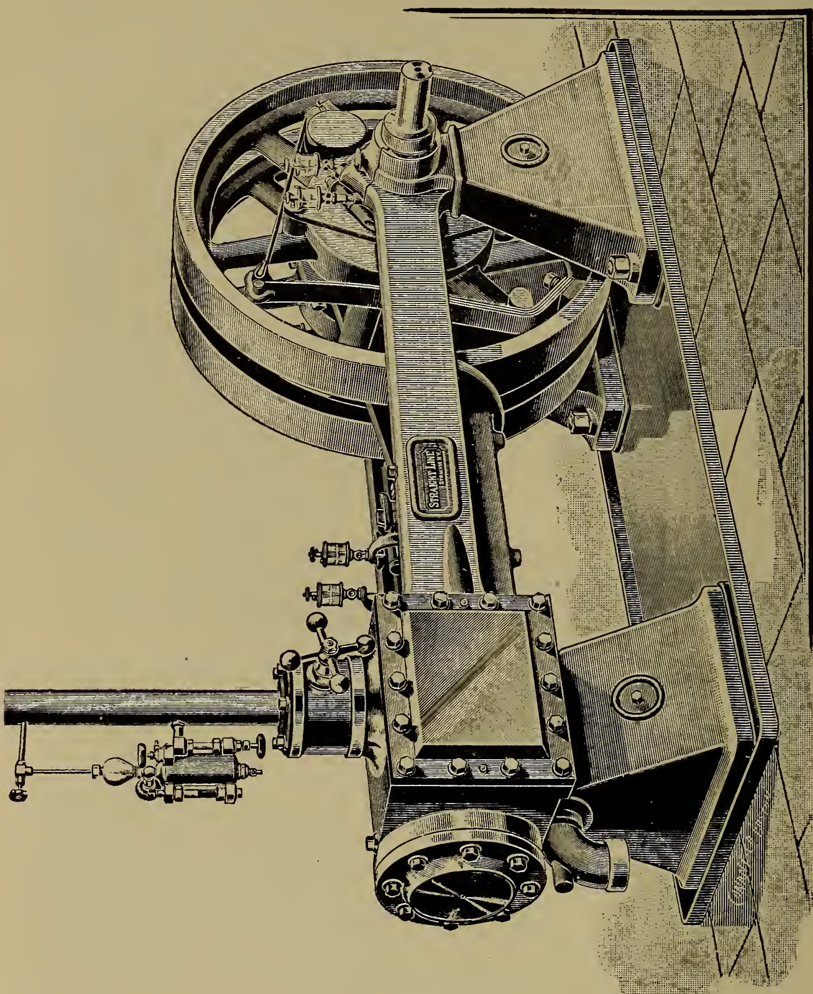


Fig. 506.

These cylinder-covers are often cracked across by water, and precautions must be taken to prevent such accidents. The cover is usually so modeled on the inside as to enter the bore of the cylinder and help to reduce the waste room or clearance. While Fig. 510 shows the head covered with a false plate, it is easily seen that the head might be cast with hollow recesses in it in which steam can be circulated to prevent heat-losses and form a part of the steam-jacket to reduce condensation of the first steam on entering. The objection to the jacketed head is its continual supply of heat to the exhaust steam which is leaving the cylinder (see paragrafs 228 and 316). The jacketed head is the largest area exposed to the entering steam, and is the most effective

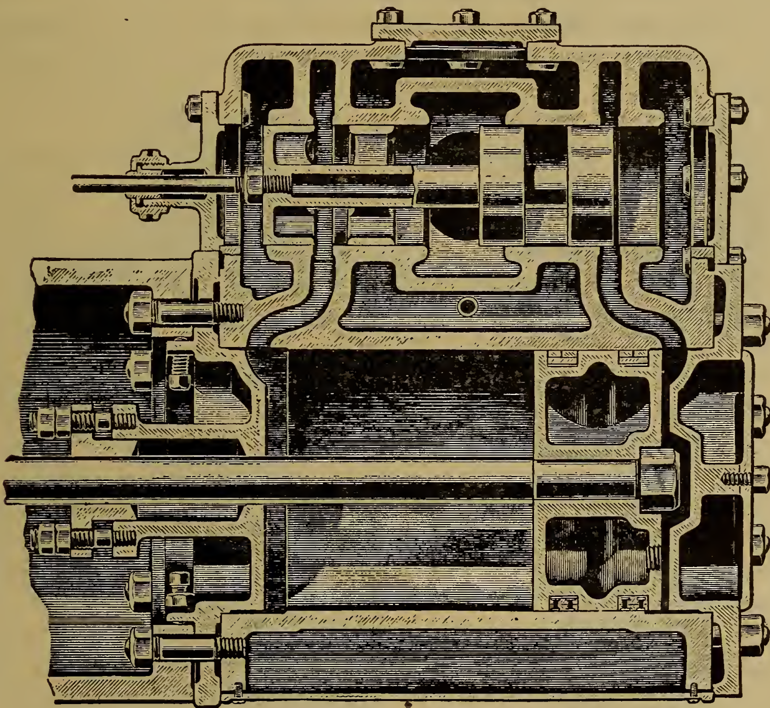


FIG. 510.

in reducing initial condensation. Jacketed and other forms of head will be noted in Figs. 436, 496.

The cylinder should be bored in the shop in the position in which it is to work. That is, a vertical cylinder should be bored on end, and a horizontal cylinder on its side. The reason for this is that the weight of the metal in the cylinder will distort it while the boring-tool develops a true cylinder. The cylinder which was bored vertically will sag and shorten the vertical axis when laid on its side, while the cylinder bored horizontally under strain of its own weight will go out of round when stood up on end so that the weight is taken off.

The valve-chest is usually cast on the cylinder and in one piece with it so as to avoid joints. It will be constructed with a convenient lid or bonnet so that access can be easily had to valves and seats for examination or repairs. The nuts on all studs of covers, lids, and bonnets will be carefully case-hardened to prevent injury from wrenches. And it is best to use only fixed or box spanners accurately fitted to such nuts in order to avoid mutilating the corners. Such wrenches and spanners accompany every well-made engine.

376. The Counterbore. By reference to Fig. 510 it will be observed that the bore of the cylinder at its two ends is slightly larger than the standard diameter through the rest of its length. This enlargement of the bore is called the counterbore, and its object is threefold.

1. The piston in its motion should slide up to and beyond the end of that part of the cylinder on which the piston bears. In other words, it must traverse the entire length of the cylinder-bore proper. Without this precaution the pressure of the piston or its rings, wearing the bore up to a certain point only, will develop a shoulder at that point, and any change in the length of the connection between the piston and crank-pin caused by wear will make the piston bring up against this shoulder at one end or the other and cause a knock or pound. If the piston laps over into the counterbore at each stroke, it wears the whole length equally and no shoulders should occur.

2. The slight enlargement simplifies the operation of getting in elastic rings such as are fitted to most pistons to make them steam-tight.

3. The counterbore, undergoing no wear in use, serves as a truly cylindrical surface to re-establish the axis of the cylinder for reboring in case of wear.

The counterbore and the steam-passages into the cylinder should be so related to each other and to the bore proper of the cylinder that the pressure of steam entering the cylinder should not come upon the piston sidewise, but from the end. If this detail is disregarded, the steam-pressure will at admission drive the piston against the opposite side of the bore and cause a disagreeable knock or pound. This will be worse at the head end, because the rod is more flexible. It is mitigated by prolonging the piston-rod out through the head.

377. Cylinder-Cocks and Relief or Snifting-Valves. To drain the cylinder and to get rid of excessive water of condensation, a hole is drilled into each counterbore at the lowest point of the cylinder, into which a pipe-connection is tapped. These drain-pipes are controlled by valves, and discharge either into a closed tank, or into the condenser or a drain, or simply into the open air, as may be convenient. The valves are called cylinder-cocks, and will be opened when the cylinder is to be warmed at starting, or when it gives indications of excessive water by the noise of snapping or cracking, like a hammer-blow, which is the indication of its presence. In large cylinders, which will be weak to resist the action of water, and in marine engines, where the pitching and tossing of the boilers may cause abnormal quantities of water to come over with the steam, automatic relief-valves are provided to open of themselves in such an emergency. These snifting-valves are usually plain conical valves opening outwards, and held in place by a coiled or flat spring. The tension of such a spring is made greater than the usual steam-pressure, so that in normal conditions they remain on their

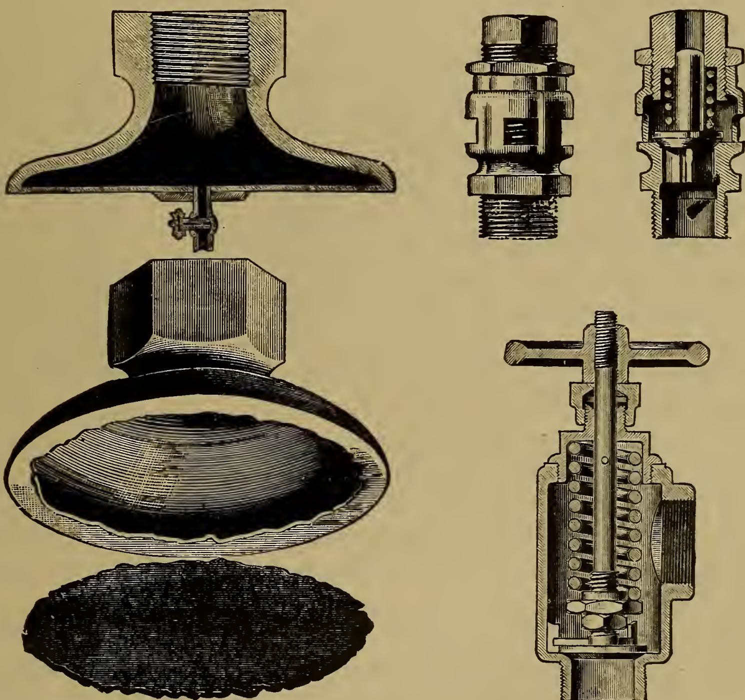


FIG. 511.

seats. Excessive pressure from water lifts them off their seats against the spring and relieves the cylinder and its cover. Fig. 511 shows two types of relief-valves and a form of breaking cap. A special brass fitting screws into the cylinder, and has a thin plate soldered over the large opening, but not too strongly. The plate is easily renewed if forced out by excess of water.

378. The Cylinder-Jacket or Lagging. The radiation of heat from the cylinder must be reduced as far as possible. This is desirable, first, to diminish condensation of steam which ought to do work in the cylinder, and, second, to keep the engine-room cool. Furthermore, the doing of work in the cylinder by expansion condenses a certain weight of steam, and it becomes desirable to diminish internal waste in the cylinder from re-evaporation of such condensed steam as far as possible. For this purpose the walls of the cylinder are often cast hollow so that live steam from the boiler can circulate through these hollow passages and keep the working bore hot. This hot steam surrounds the working bore, and the appliance to keep it there is called a steam-jacket. Fig. 615 shows the steam-jacket and cylinder, and Fig. 510 shows the valve-chest thus jacketed. The constructive difficulty of the hollow bore comes from the unequal expansion of the outer and the inner wall in cooling. This makes the inner wall very liable to crack in service in large engines. The difficulty has been met in two ways. First, by making the bore of the cylinder an inner lining which fits in properly prepared shoulders or flanges in the outer casing which forms the jacket. The joint between the lining and the rest of the casting is made by copper rings. The cylinder-cover closes down upon this lining to prevent displacement. The other plan is not to make the jacket a continuous casting, but to have its two halves united by an expansion-ring of some flexible metal which will make the joint steam-tight, but will yield to changes of length (Fig. 512).

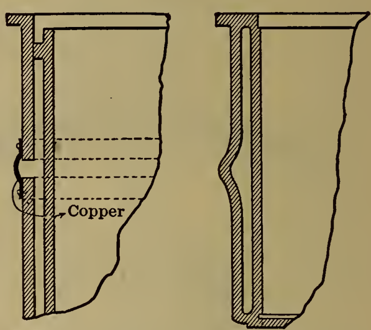


Fig. 512.

Outside of the jacket, or protecting the cylinder-casting proper if there is no jacket, is a provision for some non-conducting material. This may be hair-felt, mineral wool, or wood, or combinations of these with asbestos board. This non-conducting material may be held in place either by narrow strips of wood, or by thin staves of cast-iron, or by a sheathing of Russia sheet iron. This is called a lagging. The choice of method will be fixed by the taste of the designer, and it may be embellished by the use of polished rings. Its object is to prevent radiation and at the same time to produce a pleasing effect to the eye (Fig. 604).

379. The Structure of the Piston. The piston is to fit the bore steam-tight. It must therefore have sufficient area of contact with the bore to bear efficiently and to accommodate the packing devices. It is therefore not calculated as a rule, but receives a length which is the result of experience in the main. By reason of its size it would have unnecessary weight in large engines if made solid, and for the sake of lightness it is usually to be met in one of three forms:

1. The solid piston, which is usual in small engines only.

2. The box piston. In this the two faces of the piston are of solid metal, but the spaces between them are made hollow by the use of cores, in casting, having the shape of a sector of a cylinder. Such cores form the piston into a series of internal chambers separated from each other by partitions which form stiffening ribs to prevent the piston from being forced out of shape. These cores, which form the chambers, are supported upon feet of their own material which will leave holes in one or the other face out through which the material of the core is withdrawn. These holes in the face are then tapped, a plug is screwed in to refusal, and the metal of the plug cut off. The hollow where the core has been is at first filled with air only, but water or oil is apt to work through the pores of the iron into the cavity more or less. Some ugly accidents have happened from the heating of old pistons without a previous venting of these cavities. An accumulated pressure from air or gas heated to a high tension has rent the piston in pieces.

3. The spider-and-follower piston. In this the piston is made in two pieces or more. The solid part, called the spider, consists of one face and the side or contact surface. This cup or dish-shaped part contains the center hub to which the rod is attached, and from it to the sides radiate ribs which give stiffness and strength. It is these radiating ribs from the central body or hub which give it the name of spider. The other face of the piston is a separate plate which bolts to the ribs or hollow of the spider and forms the cover. It is called the follower. When the follower, instead of forming the entire face, is merely a ring rather than a plate, it sometimes retains its older name of junk-ring. It received this name when the packing material was hemp or junk and access was had to the grooves in which this junk was packed by the removal of the ring. In most cases the follower-plate comes off the piston or spider on the side opposite the piston-rod. An exception is met in beam-engines, where the piston-rod goes out through the top of a vertical cylinder. Convenience of access from the top induces the follower plate or ring to be on the piston-rod side in this case. The follower plate or ring is fastened to the spider by bolts which are themselves made of bronze, or their nuts are. The object of this practice is to prevent the nuts rusting fast to the thread and refusing to come off. The piston in most cases is made of cast iron. This is because of the convenience of shaping and fitting, but furthermore because it is desir-

able that the piston and cylinder-bore should be of the same metal or of equal hardness. Recently some locomotive pistons have been made of steel disks and of aluminium or other bronzes, for the sake of lightness; but when steel is used a cast-iron outer shell has often been fitted which forms the contact-surface with the cylinder and carries the packing appliances. It is likely that steel and strong metal-plate pistons will come more and more into use.

In vertical engines it is common to round the upper face of the piston or to make it convex, while the lower cylinder-head is made similarly

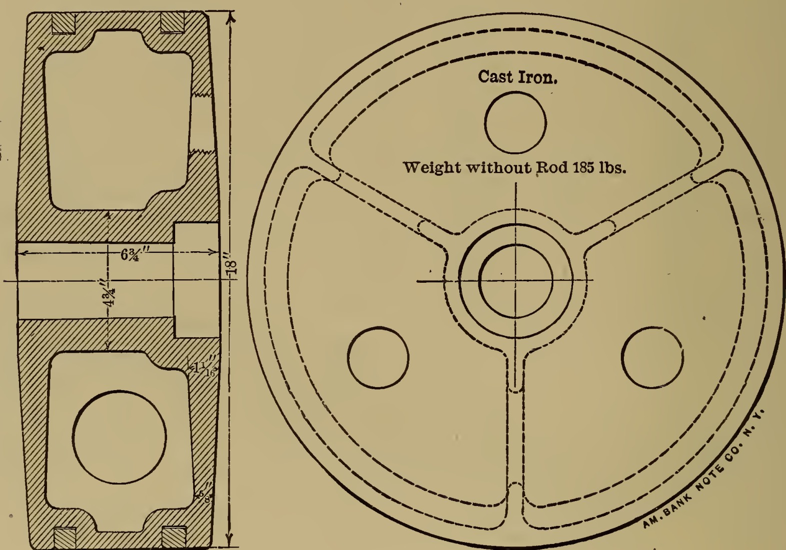


FIG. 513.

convex upwards and the lower face of the piston correspondingly concave. The object of thus doming these surfaces is to cause them to shed water outward from the center to the bore so that it will pass into the exhaust-passages and the drip.

Figs. 590 and 601 show the typical solid piston; Figs. 510, 513, and many others, typical box or hollow pistons; and Fig. 514 the usual form of follower piston used in locomotives. Fig. 515 shows the new type of steel-plate piston.

380. The Piston-Packing. The piston cannot ordinarily be fitted to its bore so as to be steam-tight. This is, first, because the piston and the bore are fitted cold and will expand unequally when heated. If the

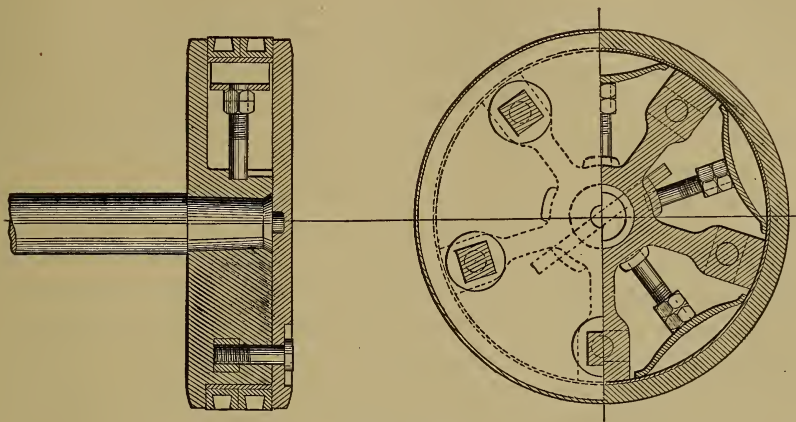


FIG. 514.

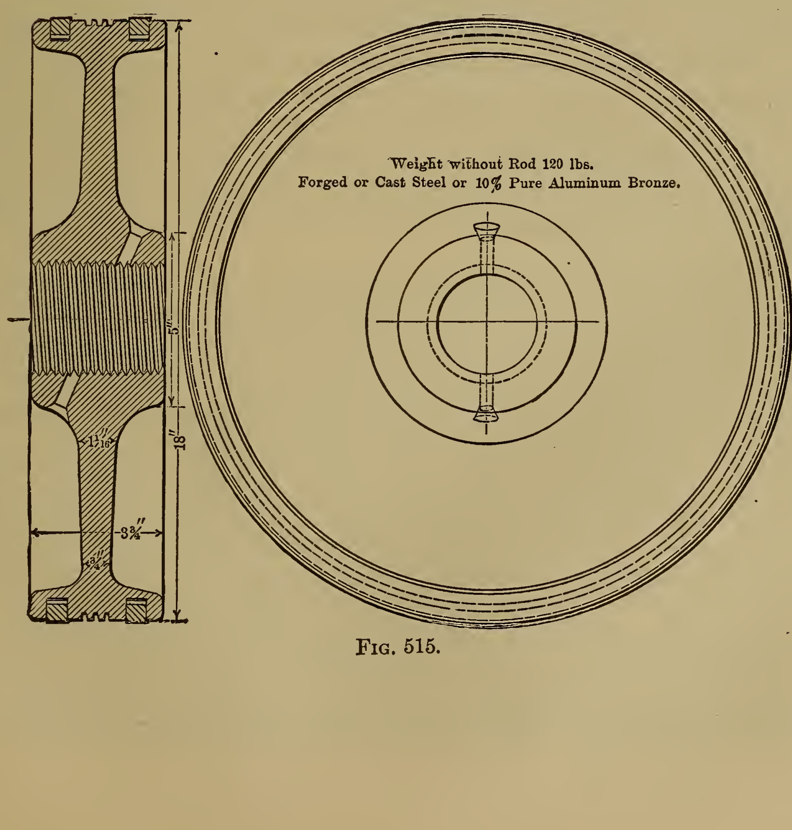


FIG. 515.

bore expands more than the piston, it leaks. If the piston expands more than the bore, it is seized by the latter too tightly to be moved if it was a close fit when cold. Furthermore, wear of the contact-surfaces would make a solid piston fit loosely in the bore after a certain time and permit leakage. If the piston leaks, steam passes directly from the inlet to the exhaust-pipe, and so to waste without doing work. This increases the consumption of steam per horsepower and the consumption of coal. For such reasons some form of packing appliance to make a steam-tight joint and allow for expansion and wear has been used from the beginning.

In the first steam-engines made, before the machine tool known as the boring-machine had been invented, the cylinder was cast as nearly cylindrical as possible and smoothed by hand. A joint between the piston and the cylinder was made by coiling a plaited square gasket of hemp-fiber or junk into a wide groove formed in the piston. This gasket was made of an eight-strand braid, and was held in place and forced outwards by screwing down the follower-plate or junk-ring (hence the name). These elastic or fibrous packings were adequate for low pressures and low temperatures, such as prevailed in the early days. They can still be used for water-packings, and combinations of canvas and rubber may still be used under conditions of this sort. What is known as the cup leather packing can also be used with cold fluids. An annular ring of leather, having an exterior diameter greater than that of the bore, is pressed into the bore when wet so as to turn cup-shape, and is drawn up against the piston and held in place by a ring acting just like a junk-ring. The cup of the leather ring or disk which lies against the bore is pressed outwards by the pressure of the fluid, and leakage is prevented.

The only way in which pistons can be made tight without packing-devices is by the use of what is called leakage-grooves. These are a series of shallow grooves turned in the sides or bearing-areas of the piston and so numerous that the pressure leaking from one groove to the next shall not have time to establish itself in all of the grooves and pass from the last into the exhaust side during a period occupied by one stroke. The principle is that pressure must be fully established in the first groove before steam will leak from the first groove through the narrow space between the piston and bore into the second groove, and so on. Such pistons would not be tight if they stood still or moved at low velocity. At high speeds they serve their purpose if there are enough grooves, but their presence makes the piston of unusual length in the direction of its motion. The grooves become filled also with the lubricating material, and with water of condensation, which helps to make the joint tight. They have less friction than elastic packing.

381. Piston-Rings. By far the most usual method of making a piston steam-tight is by means of rings which fit in grooves turned in the bearing-surface of the piston. It is intended that these rings shall fit their grooves on their sides closely enough to prevent leakage around them, and that they shall be forced radially outwards with sufficient force to prevent steam leakage between them and the bore. Such rings are called piston packing-rings, and they will differ with different designs according to their material, according to their number, and according to the method used to keep them tight against the bore.

The materials used for piston-rings are cast iron, steel, and composite metals. The advantages of cast iron are, first, its cheapness; second, that it has the same hardness as the bore and so does not wear it unduly; and third, its convenient elasticity.

The advantages of steel are its elasticity and that it is not as fragile as cast-iron rings. Cast iron has been known to break from shock or vibration while in service and cause unpleasant consequences in the cylinder. The use of steel rings for pistons is attributed to Ramsbottom of England.

The composite rings are brass or bronze rings, or rings of such metal in which recesses are cast and in which recesses some soft bearing metal like babbitt is cast to form the contact with the cylinder-bore. The object of these composite rings is to obtain a bearing metal softer than the bore, so that the wear shall be concentrated upon the rings, which are easy to renew. The objection to the steel rings is that they are likely to abrade the cylinder by their superior hardness or density, and to rebore the cylinder is more troublesome and expensive than to renew a worn-out ring.

The ring in order to be elastic must be a non-continuous ring, or with a break at some point in order that its length may vary. This joint between the two ends of the ring must be prevented from allowing a leak. This is done either by simply making the joint a scarf-joint, or by fitting a tongue-piece which shall slip in the ring at one end while fastened in the other and thus close the joint (Fig. 516). It is very usual to have two rings, so that the joints in the two rings may be on opposite sides of the piston. Fig. 513 shows the rings in separate grooves, while Fig. 510 shows two rings in the same groove.

To press or force packing-rings radially outward against the bore, five methods are usual.

1. To depend on the elasticity of the ring itself. This is applicable to pistons up to 16 or 20 inches in diameter, but is not desirable for larger sizes. It is used both with steel and cast-iron rings. The ring is turned as a solid ring to fit a diameter larger than the bore.

Usually the proportion is a quarter of an inch larger for each foot of diameter. The finished ring is then sawed apart and sufficient metal taken out at the joint to permit the ring to be squeezed together so as to enter the cylinder. It will tend to expand to its original size against the restraining bore, and this pressure makes a steam-tight joint. Such rings are called snap-rings. They do not guide the piston at all, as they are loose in the grooves sufficiently to move freely, but not enough to leak. To keep the radial pressure of the ring against the

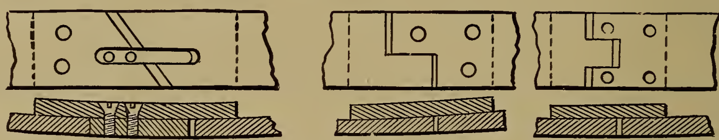


Fig. 516.

bore the same at every point so as not to wear the cylinder unequally, the thickness of the ring should be graduated and should be different at different distances from the joint.

2. The packing-ring proper of cast iron and steel is forced outwards by an inner or spring ring. This is a common plan in large vertical engines where the weight of the piston does not come upon the rings or springs. It can also be used in horizontal engines of medium size (Fig. 510).

3. Flat springs, pushing the rings radially outwards at several points of the circumference. This is a favorite locomotive design and for larger horizontal engines (Fig. 514). The flat springs can be adjusted by nuts or screws to give greater or less tension, and in horizontal engines with heavy pistons the tension on the lower springs may properly be made greater than on the upper. This type is applicable only to pistons of the follower type, and the adjusting of the springs requires that the follower be removable. In vertical engines these springs should all be set out equally, and a clever design by which a taper pin exerts radially a pressure on the studs which carry the springs is shown in Fig. 517, whereby the cover does not have to be removed.

4. The packing-ring may be forced outwards by positive means, such as screws or wedges or combinations of them. The idea is that with a true bore there is no occasion for elastic pressure upon the packing-ring, but that it causes unnecessary friction. If the ring is set out just enough not to leak, and the bearing-contact of the ring and bore is large enough, there is no occasion for give or take in the ring. The wedge or screw is variously applied, either to enlarge the diameter of a split ring by separating its ends or by pressure exerted radially upon the

packing-ring or the inner bull or junk-ring. This type of packing is applicable, of course, to follower-pistons only.

5. Steam packing. Fig. 518 shows a plate piston packed with two rings. The groove behind each ring is connected at several points

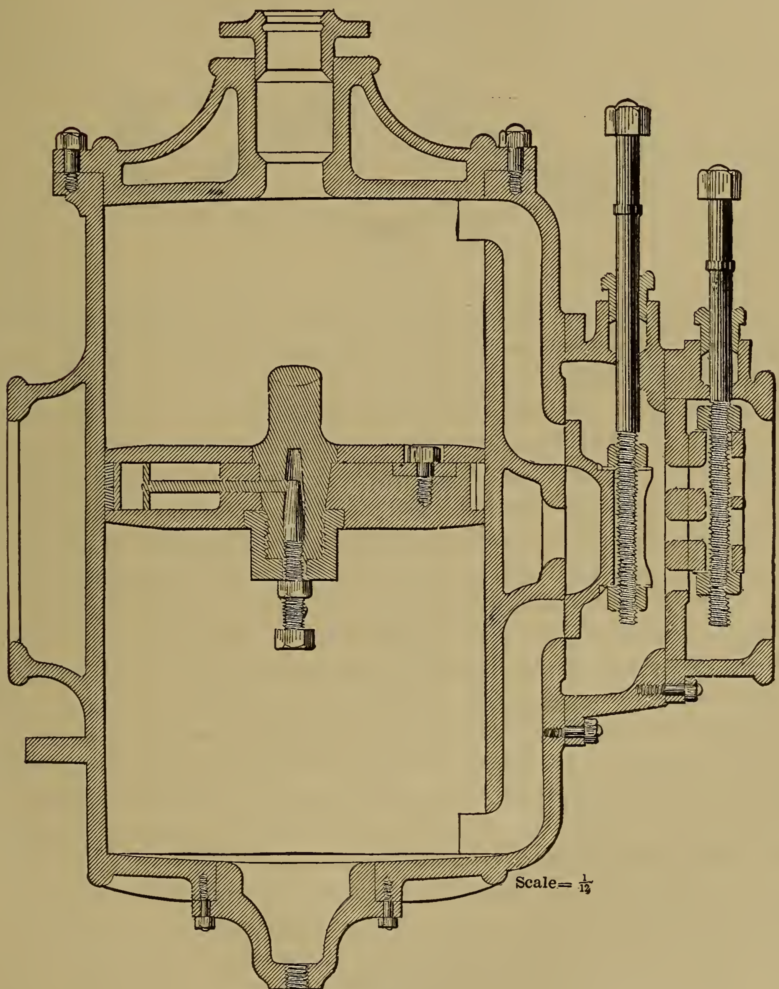


FIG. 517.

through small holes with the steam-pressure acting on the piston, so that the packing-ring is forced outwards by an elastic pressure of steam behind it. It is usual but not necessary to make these packing-rings in segments which overlap each other so as to prevent leakage at the joints, whereby the steam-pressure does not have to overcome any resistance in the metal of the ring in forcing it out. When steam is shut off, the steam-spring ceases its action and lessens the friction in the cylinder. This form of packing was first associated in America with the name of Dunbar and has been much used.

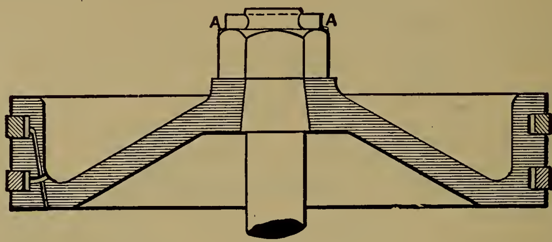


Fig. 518.

A modification of the principle of steam piston-packing has been ingeniously applied in some large horizontal engines with a view to diminish the friction of the piston and its tendency to wear the bottom of the cylinder. Steam is admitted through a hollow piston-rod to a place on the bottom of the piston, extending like a groove part way around its bottom surface. The area of this groove is calculated so that with the usual steam-pressure the upward reaction of the steam in it which comes from the hollow rod shall just balance the weight of the piston. Rings prevent the steam from leaking out of the groove, and in normal conditions the piston should slide upon a layer of steam and without metallic contact with the bore, so as to be nearly frictionless.

382. The Piston-Rod. The piston-rod has to transmit the motion of the piston to the mechanism outside of the cylinder. It has to withstand both push and pull, and the former without bending. It is rarely massive enough to have no tendency to bend with the weight of the piston when the latter is at the head end of horizontal engines. If calculated as a pillar for compression, it will be abundantly strong to resist tension provided that it be properly secured in the piston. The piston-rod has also to withstand the tendency to abrasion or to wear out of round where it passes through the cylinder-head and its stuffing-box. For these reasons a great many piston-rods are made of high-carbon steel which has been treated by the process known as cold-rolling, which gives it a particularly dense, hard, and close texture on the outside, and so increases the modulus of elasticity as to increase its resistance to bending from the weight of the piston. The usual methods for fastening the piston-rod to the piston are five:

1. The piston-rod is threaded and the piston screwed on it with a thin jam-nut or set-screw, to prevent unscrewing (Fig. 515).

2. The piston-rod is formed with a shoulder, and between the shoulder and the end a straight or tapering surface which ends in a screw-thread is turned. The piston is bored to fit the straight or tapering end of the rod, and when the rod is in place the thread on the rod protrudes enough to take a strong nut. The collar and the taper surface take the push of the piston, and the nut takes the pull. These methods have the advantages of being cheap, and the joints between the piston and rod are easily broken (Figs. 510, 518, 615).

The objection to the second plan is that the projecting nut requires that a clearance be made for it (see Figs. 510, 517), and there is always a possibility that the screw-joint exposed to push and pull will in time work the nut downward along the threads so that the joint becomes loose. When this happens it makes a knock or pound which is hard to locate. The nut is liable to corrosion in the cylinder and to rust to its threads. Where it may be expected or desired that the joint between piston and rod is to be frequently broken, the nut may be made of a bronze alloy.

3. The taper is drawn in by a key of metal (Fig. 514). The end of the rod is formed into a tapering or conical surface which fits a corresponding hole in the piston. A rectangular slot is cut at right angles to the axis of the rod, and a similar one across the hole in the piston. These slots are so related to each other lengthwise that a rectangular key driven through the slot when the rod is in place shall bear in the piston upon the end nearest the large base of the cone, and in the rod upon the end nearest to the small base. The driving in of the key draws in the male cone of the rod into the female cone of the piston with a very strong pressure until the key refuses to be driven farther.

This method is an elegant one, but is applicable to follower-pistons only. The key is within the hollow part of this piston, it entails no clearance, it is very strong, and the joint between piston and rod can be easily loosed if necessary. This is done by the use of a special offset key driven after the original key has been removed, and which reverses the pressure by which the piston was drawn on the rod, by having its bearing upon the opposite ends upon the slot in each. The objection to it is its cost and the possibility of the joint working loose from a slacking off of the key. There is not much weight in these objections. The taper of the rod may be either 1 in 32 or 1 in 64, according to the amount of force with which it is desirable to draw the one cone over the other.

4. Riveted rods. The end of the rod with collar or taper surface fits the piston and projects slightly through it. The projecting end is then upset and turned back upon itself as a rivet is headed. Such riveting of the rod may be done hot or cold. If done hot, the rod in

shrinking as it cools draws the piston more tightly against the shoulder or the taper. The heat may injure or scale the surface of the rod. The advantages of this method are that it is cheap and tight and takes no room. The joint cannot be broken without destroying the rod. It is a favorite joint in small cheap engines, where the value of the rod is so slight as not to warrant the cost of an expensive joint. The cold-riveting of the rod does not injure the rod by scaling, and can easily be made tight against the least motion. The head of the riveted rod is often formed in a cup-shaped depression or countersink.

5. Shrinkage-joints. This is a very elegant joint for pistons of medium size. The hole in the piston which is to take the rod is made straight and cylindrical, but is smaller than the diameter of the rod in the proportion of .0025 of an inch for each inch of such diameter. This makes a hundredth of an inch for a four-inch rod. The piston is then heated to low redness, whereby the hole is expanded sufficiently to permit the rod to enter it. As it cools it contracts upon the rod, and seizes it with a pressure so great and firm that the rod will part somewhere in its length before the piston will slip off. The advantages of this joint are its tightness; it can be broken by heating the piston while the rod is kept cool; it involves no clearance. The objections to it are its demand for exact working to dimensions if it is to succeed, and the strain on the piston and the effect of heat upon it. This method of making joints by shrinkage is often used about the crank for its shaft and pin with the same advantages.

In follower-pistons the joint with the rod is often designed so that the follower-plate shall cover over it and remove any necessity for clearance in the cover.

The front end of the rod is to be secured to the cross-head. This must be a joint easily to be taken apart, since the cross-head must be put on after the piston and rod are in place in the cylinder. It will therefore be found that much the most usual plans are to thread this outer end of the rod and screw it into the cross-head with a jam-nut to prevent unscrewing; or to taper the end of the rod and the hole in the cross-head, and draw them together with a transverse key; or to split the metal of the cross-head and close the joint by bolts. The screw plan will be used on small and medium-sized engines, and the key on medium sized and large. Figs. 525 to 531 will serve to illustrate typical methods of securing the rod to the cross-head.

383. The Stuffing-Box. The hole through which the piston-rod must pass steam-tight through the head requires to be fitted with special devices to prevent leakage. As in the case of the piston, the rod must be surrounded by an elastic and adjustable material which shall permit the rod to pass in and out with the least friction, and which yet shall seize it tightly enough to prevent leakage of steam when the pressure is on and prevent the entraining of water with the outward motion of the rod on the exhaust-stroke by a sort of a capillary action. The combination which is used for this purpose is called a stuffing-box. It consists of a sort of annular box or cavity, the packing proper which goes into that box, and the gland by which the packing is compressed and held in place. There must also be a method for tightening and holding the gland.

The typical stuffing-box is exhibited in Figs. 510 and 598A. It is quite usual where the rod enters the bottom of the stuffing-box to force a bronze annular bushing into the hole in the cylinder-head so as to make the rod fit this bushing quite closely. The advantage of the bushing is that it can be easily forced out and replaced when it becomes inconveniently worn. It is preferable to have the softer bushing worn by the rod rather than to have the more costly rod worn by the harder metal of the cylinder-head. The bottom of the stuffing-box cavity tapers inwards towards the rod, and the inner end of the gland likewise. The effect of this is to produce a component inwards against the rod when the gland brings pressure parallel to the rod, and thus to compress the contents of the stuffing-box inwards upon the rod. Fig. 510 shows the gland drawn inwards by two stud-bolts. This is a most usual plan with rods of medium size. For small rods such as valve-stems the arrangement shown in the same figure and in Fig. 598A is more usual because of the room which is required for bolts of practical size. For such small rods the outside of the stuffing-box, instead of being formed into a flange, is threaded, and a hollow nut fitting over the gland will draw the latter inwards when screwed upon this stuffing-box thread. This is the usual method for valve-stems and similar small rods. For large rods above four or five inches in diameter two bolts are not enough to draw the gland symmetrically inwards and prevent it from cocking or binding sidewise, which would cause great friction and wear. Care must be taken to prevent this in any case, but with very large rods requiring four or six bolts in the stuffing-box, as in marine practice, the nuts are often made into small pinions or gears which work into one large gear so that the turning of one turns all the bolts at once, as in the self-centering chuck. This difficulty is avoided when the gland-nut is used.

For the packing material to be used in the stuffing-box the qualities to be sought are elasticity and low coefficient of friction, absence of abrasive effect upon the rod, and capacity to prevent and absorb leakage. Early packing materials were hemp and cotton fiber plaited into gaskets and laid in loosely. More recently combinations of cotton in the form of canvas with rubber have been much used. The rubber gives elasticity, the canvas the quality of absorbing and holding the lubricant. The lubricant not only diminishes friction but opposes the passage of water. Paper-fiber also has been popular, and hemp with graphite. Packings of this class are laid in the stuffing-box in a spiral coil, the thickness of the packing material being standardized to standard dimensions of the space in the stuffing-box which the packing is to fill. Packings of one-half, five-eighths, or three-quarter inch thickness will be usual in engines of medium size.

The objections to these fibrous and rubber packings are first encountered with high pressures of steam, and secondly with high heats. Oxidation and abrasion of the fiber under pressure and heat and a hardening of the rubber under heat make it necessary to renew the packings frequently, and they have but a relatively short life of entire tightness. This trouble is particularly present in vertical engines with the piston coming out of the bottom of the cylinder. Unless the packing be excessively compressed so as to cause undue friction, the rod will draw water out with it past the packing by a sort of capillary action. Combinations of asbestos-fiber, which is not affected by heat, have given great satisfaction in stuffing-boxes, but exceeding care must be used, both in manufacture and in use, that there be no hard or gritty particles of the mineral. Where care is not taken the rod becomes fluted or scored lengthwise from the abrasive action of such hard spots.

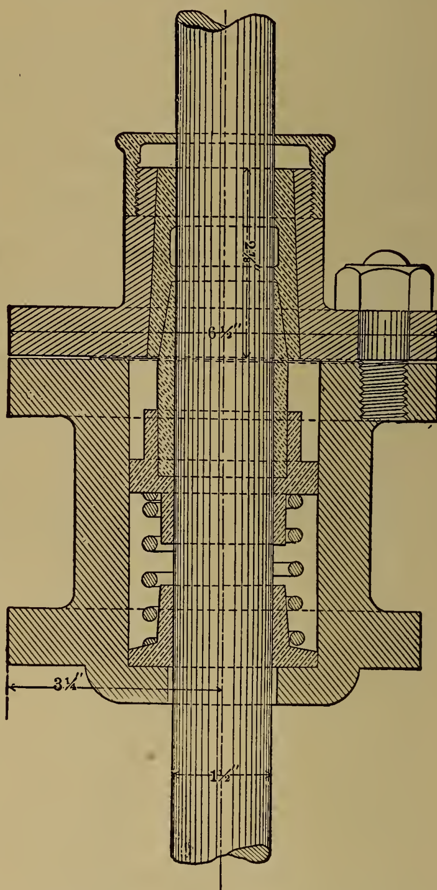


FIG. 519.

To make a more mechanical method of packing which should last longer and resist both heat and pressure, a wide variety of metallic packings has been made. The principle of such packings is to have a series of split rings whose exterior surfaces slope alternately from and

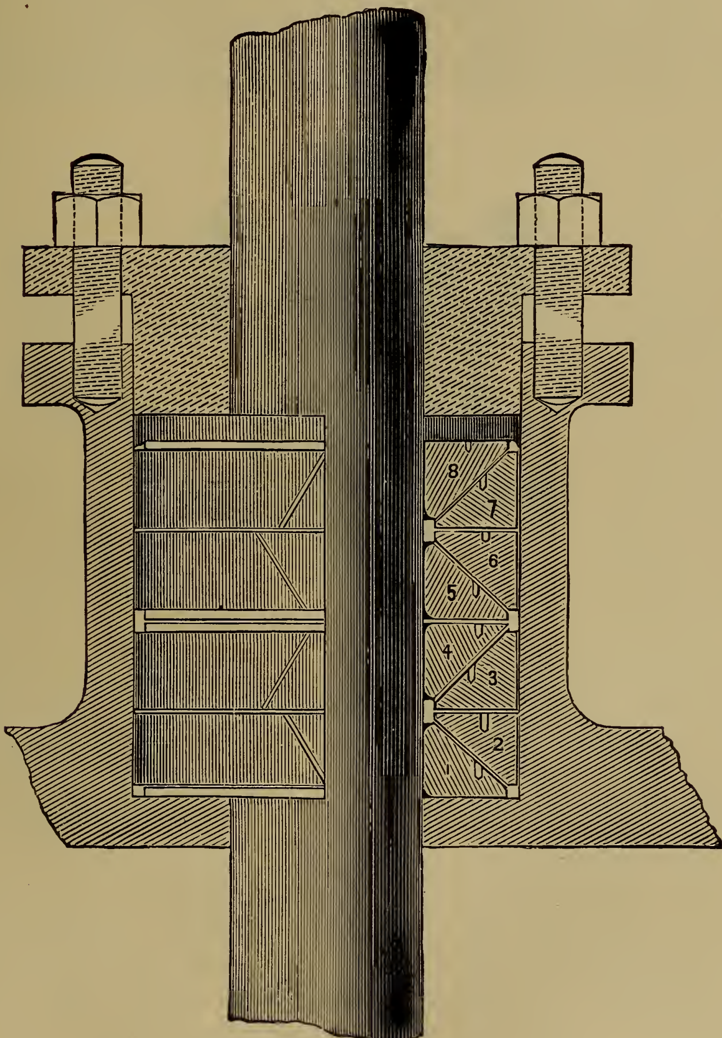


Fig. 520.

towards the rod, so that when endwise compression is exerted by the gland they close inward upon it. Sometimes a coiled spring is introduced behind the gland, so that the compression of the split rings may be an elastic force instead of a positive and unyielding compression. Furthermore, such rings are often arranged so as not to fill the stuffing-box space sidewise, but to admit a certain give-and-take if the rod and the axis of the cylinder should not happen to coincide perfectly. The most striking illustration of this will be found in the method of construction in the Straight Line engine, Fig. 526. Here the packing is really a long cylinder which has a motion around a spherical joint in

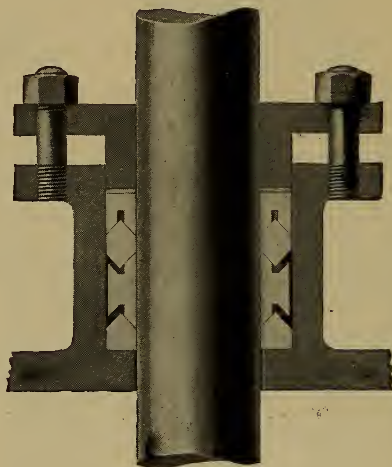


FIG. 521.

the end of the cylinder to permit of adjusting its own alinement. The design of Fig. 521 aims to avoid a difficulty with such metallic rings, which form shoulders upon each other in the slight motion upon each other one to the rod and expansion by heat and high pressure. Each section covers the one next it up to the groove, so that the moving parts slip entirely past each other, and no ridge of an unworn area can be formed. This has been used with highest pressures.

Certain forms of metallic packing are shown in Figs. 519, 520, and 521. If the piston-rod is to project through the back head, a stuffing-box is also required there; but it is of less importance if the path traversed by that projecting rod is inclosed in a steam-tight cylinder which it fits nearly tight. Provision must be made, however, in this case to get rid of water which may accumulate there from leakage.

384. Air-Valves. In engines of the locomotive class where the mechanism of the engine may be expected to run on for considerable periods after steam is shut off, provision must be made to guard against the pumping action of the piston in the cylinders. The continual exhausting of the contents of the cylinder makes an inward pressure, and dirt, cinders, and other foreign matter would thus be drawn in. This difficulty is met by having a valve opening inward attached to the steam-chest which will be shut upon its seat when pressure is on the valve; but will open by atmospheric pressure and let clean air enter when the pressure falls below atmosphere. Fig. 603 shows the principle of these air-valves upon a locomotive valve-chest.

CHAPTER XXIII.

CROSS-HEAD GUIDES AND CONNECTING-ROD.

385. The Guides and Slides. The cross-head gets its name from the fact that it is the head of the piston-rod, and as ordinarily constructed it forms a T or cross-shaped head to such rod. The cross-head and the guides which control its motion are counterparts or complements of each other, and the form, number, and arrangement of guides will be dependent on the preferred arrangement of the cross-head.

The British term for the cross-head is the motion-block.

The condition which the guides must fulfill is that of keeping the end of the piston-rod from bending out of the axis of the cylinder when the strain on the connecting-rod produces such a tendency (Fig 369). The plane or planes of the guides must therefore be truly parallel to the prolonged axis of the cylinder, and it is the convenience of securing such parallelism by means of the level which makes it so desirable that the engine bed-plate and the cylinder-axis should be truly horizontal upon the foundation. In many forms of bed-plates the guides are formed and finished in the bed-plate casting and at the same setting of the tool at which the cylinder is bored. This insures a common axis for cylinder and guides. Where the guides are loose and need to be set up on the bed-plate great care must be exercised in their alinement.

When the fine-wire axis is established (paragraf 372), this is best done by means of special fixed gauges or trammels. In the absence of such appliances the ordinary surface-gauge, or better the micrometer surface-gauge, may be used. When one guide or one pair has been made parallel to the axis, the other guide or pair should be made absolutely parallel to the first. To have the alinement of the guides defective is to invite wearing at the stuffing-box and wearing of the bore of the cylinder out of round, and to cause unnecessary friction and often a knock or pound in the engine which is hard to locate; or even a breaking of the piston-rod.

The guides may control the cross-head by action in a vertical plane or in a horizontal plane. They may be in number, one, two, or four. Their surfaces are exposed to abrasive wear, and they should be massive enough or so shaped or supported as to resist the tendency to deflect. To resist abrasion they are often case-hardened, and to resist deflection they are often made thicker as the distance from the supporting ends increases.

Where but one guide is used it will appear in one of two forms. In the first form the cross-head will be arranged to embrace the rectangular guide on all four sides with the piston-rod and cross-head pin in the

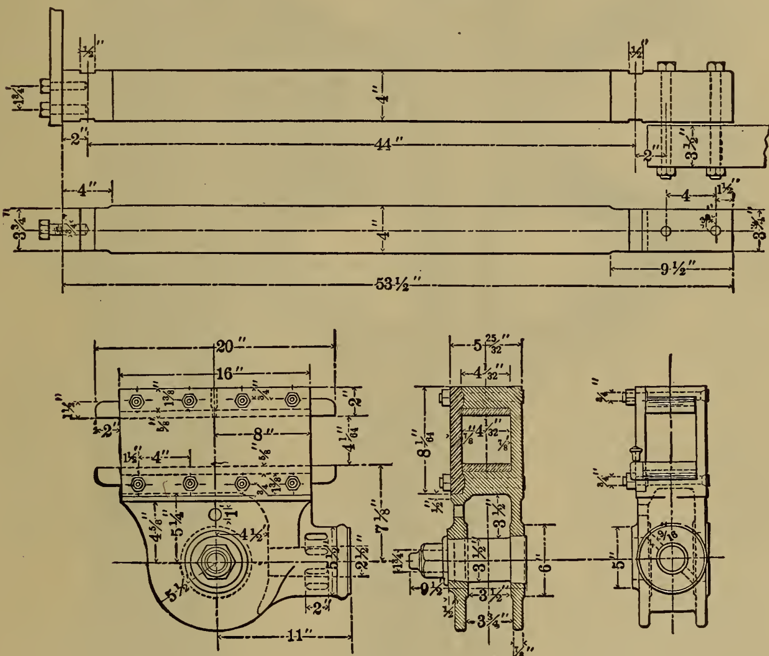


FIG. 525.

plane of the guide and below it. The cross-head pin must be far enough below the guide (Fig. 525) so that the swing of the connecting-rod at its widest amplitude shall clear it. This form of cross-head is used quite a little in locomotive practice, but care must be taken that it should be long enough not to cock or bind upon its guide. This is likely to occur with short cross-heads of any form if the line of the resultants due to the reaction of the connecting-rod on the pin passes at any time outside of the center of the pin, or even near the edge of the bearing surface. The other form of single guide is sometimes known as the guide for the slipper cross-head, and is shown in Fig. 526. The engine in this case usually turns in one direction only, and the guide is a flat plane surface with suitable edges to prevent sidewise motion of the cross-head. Fig. 526 shows also a convenient method for adjusting the plane of the guide

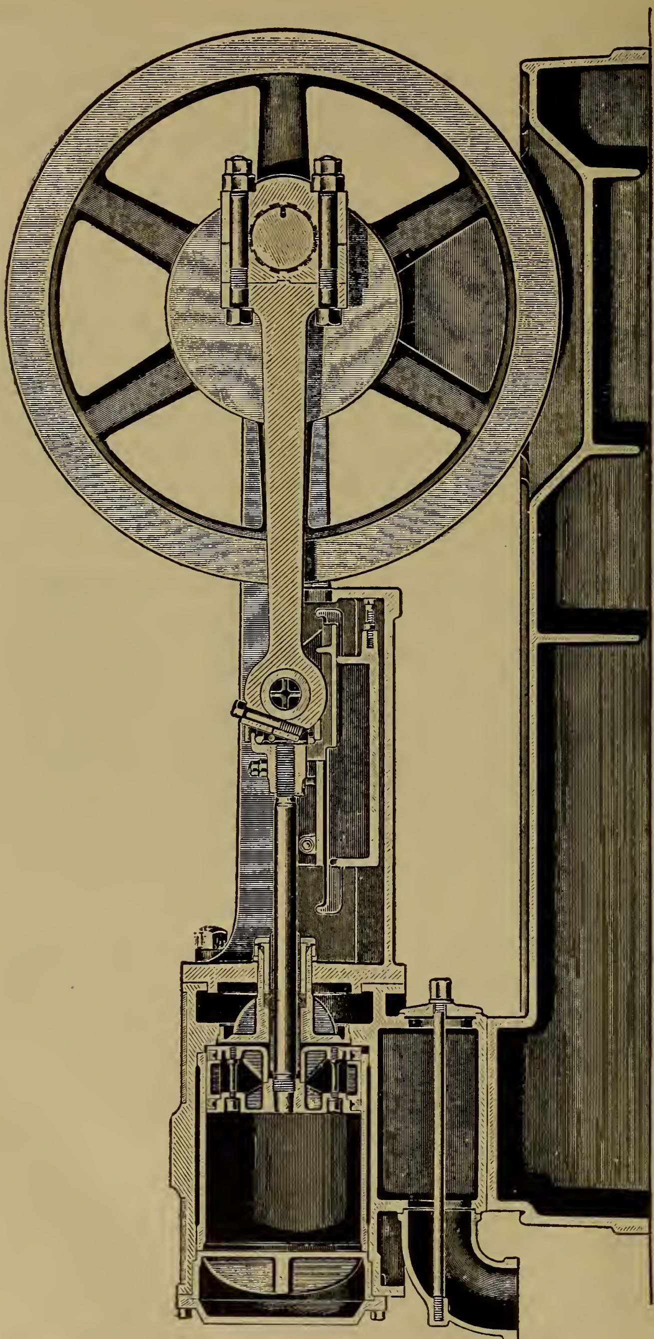


Fig. 526.

in case of wear of the rubbing surfaces. This form is very easy to lubricate.

When two guides are used they may either embrace the cross-head if the guides are in the plane in which the connecting-rod oscillates, or the cross-head must embrace them if they are in the plane at right angles to that in which the connecting-rod oscillates. If there are four guides, they will embrace the cross-head in either arrangement. This must lead to the discussion of the cross-head.

386. The Cross-Head. The cross-head for a single guide has been already discussed. With two guides it is much more usual to arrange them to guide a vertical cross-head, which is one guided in the plane in

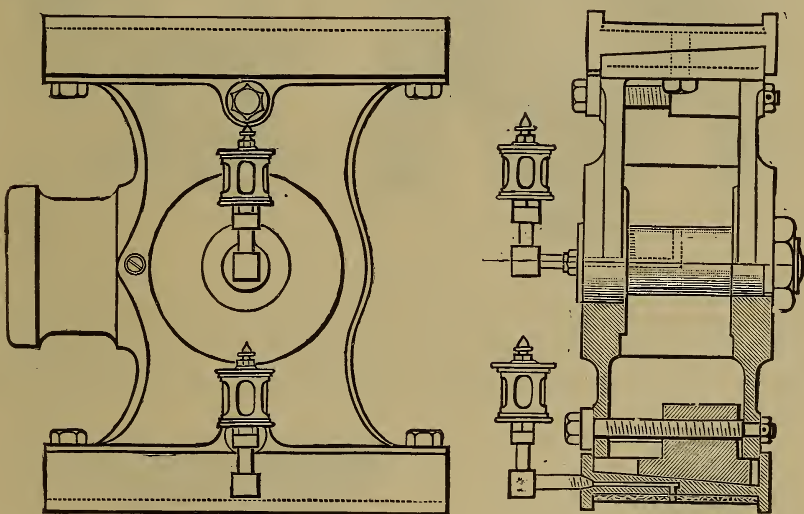


Fig. 527.

which the connecting-rod oscillates. With this arrangement the guides must be far enough apart to clear the connecting-rod in the angle just before half-stroke, when it departs furthest from the cylinder-axis. This makes the cross-head of sufficient extent laterally to meet the contact-surface. With such vertical cross-heads the guides may be flat and plane (Fig. 527), they may be cylindrical (Fig. 528), or they may be each in two planes inclined to each other (Fig. 529). The great advantage of the cylindrical guiding surface (Fig. 528) is that the cylinder and guides are so conveniently bored at one mounting with a boring-bar having two cutting heads. This secures coincidence of the axis of cylinder and guides. The objection to it is that there is nothing to prevent a twisting action except the attachment of the connecting-rod to the crank-pin. It is a very usual method in relatively small engines. With any of these cylindrical cross-heads the guide-surfaces usually are molded and finished in the solid metal of the bed, and the

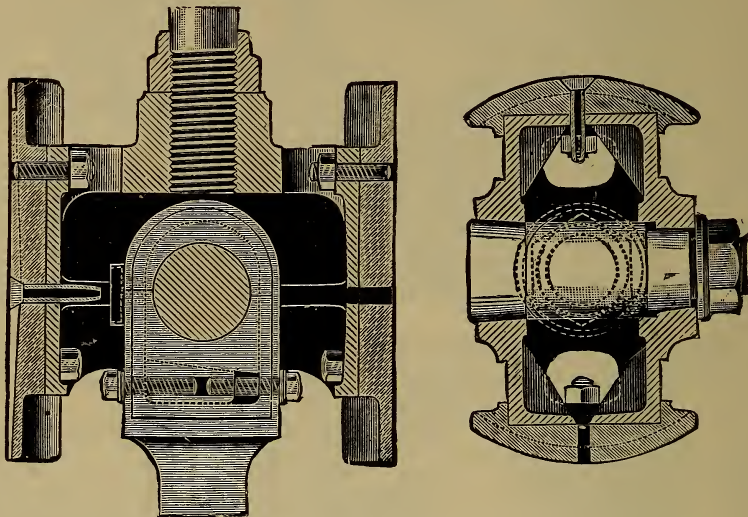


FIG. 528.

adjustment for wear and for symmetry with the cylinder-axis under wear is affected by adjustments in the cross-head itself. The contact-surface of the cross-head is usually made by special metal pieces which

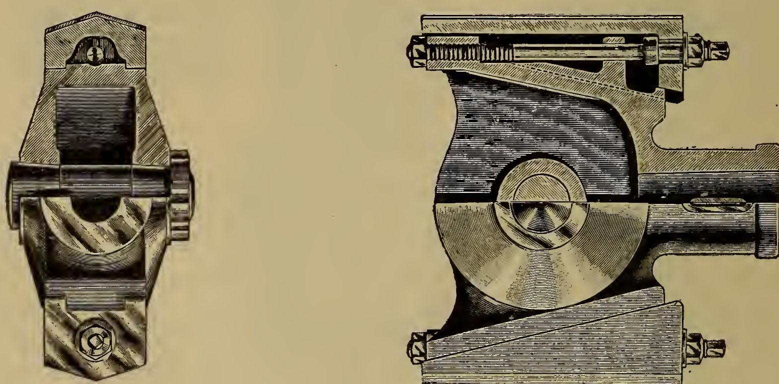


FIG. 529.

are called gibs. These gibs may be simply cast-iron shoes, cast-iron shoes with recesses for Babbitt or other bearing-metal, bronze shoes, or shoes of some wood well calculated to resist abrasion, such as lignum vitæ. The principle of these gibs is that they shall concentrate upon themselves the wear and shall be cheaply renewable. They should furthermore have a low coefficient of friction. These gibs being detached from the solid metal of the cross-head can easily be made to be adjustable in the plane at right angles to the cylinder-axis, by means of screws or wedges or bolts. Fig. 527 shows the adjustment by means of lateral wedges, and Fig. 529 the adjustment by longitudinal inclined planes. In early Corliss cross-heads the central part was attached to the shoes or gibs by bolts of some diameter which were separately adjustable and held by jam-nuts when the adjustment was complete. A simple type for small engines is often met in which an occasional variation can be made by having the adjustment-bolt fixed in position, while washers or liners of thin metal or even of paper are taken out of the space between the collar and the gib as wear or adjustment may require.

The cross-head using two guides in the plane at right angles to the oscillation of the connecting-rod has the cross-head embrace the guide on three sides with gib adjustment. This is a usual adjustment in beam-engines such as Fig. 389. The gibs are like those shown in Fig. 525; but as they will be on the outside of the guides, their adjustment becomes very simple by the use of screws passing through the solid metal of the cross-head and embedding slightly in the gib. This can also be used on the vertical cross-head, but is not considered so satisfactory and mechanical an arrangement. Nearly all inverted vertical engines are guided in the plane of the connecting-rod when they have an A frame, or else make use of the slipper one-guide cross-head when they have open frames as in Figs. 380 and 381.

The four-guide cross-head has been a favorite form for locomotive practice and in much stationary practice. It makes a comparatively light element, and yet the contact-surface is abundant and generous. The two guides on each side of the connecting-rod can come as close together as convenient instead of having to be at a determinate distance apart. Fig. 530 will show the general appearance of a cross-head of this type, which has the further advantage that by generous bearing-areas the pressure per square inch may be so far reduced that no appreciable wear is to be expected during the lifetime of the engine, thus simplifying the construction of the cross-head, doing away with gibs and their appurtenances. The slipper cross-head has this same advantage. Where gibs are thought desirable they can be easily introduced, or wear may be taken up by the introduction or removal of liners of thin paper under the blocks which separate the guides at their ends. If the contact-pressure be kept below 40 pounds per square inch of area, and a proper lubricant kept continuously supplied, a thin film of oil will be always separating the surfaces, and if they never touch they never wear. Care must be taken that the design of the cross-head prevents the resultant of pressures ever passing outside of the contact-surface. If it does, there will be a tendency for the cross-head to cock or press a corner down upon the guides, scraping off the oil and setting up abrasive wear. It is best to have the pin on which the connecting-rod swings in the center of the length of the cross-head for this reason. The gibs, furthermore, should have grooves cut diagonally or zigzag fashion in their contact-surface to hold the oil and distribute it sidewise over every element of the guide. It will be apparent that the lower guide needs to be lubricated in a horizontal engine which throws over, and the upper guide or upper gib in an engine which throws under (Fig. 369). The resultant of alternate push and pull is always in one direction for the engine which turns in the same direction.

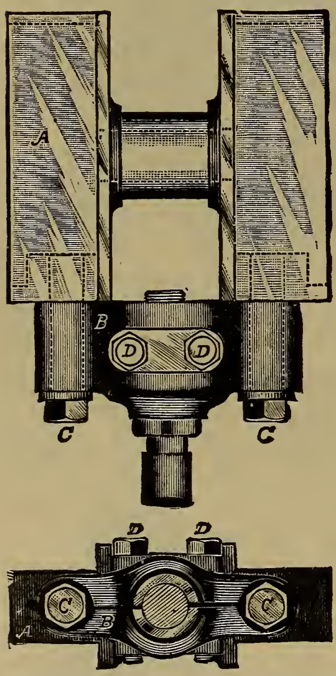


Fig. 530.

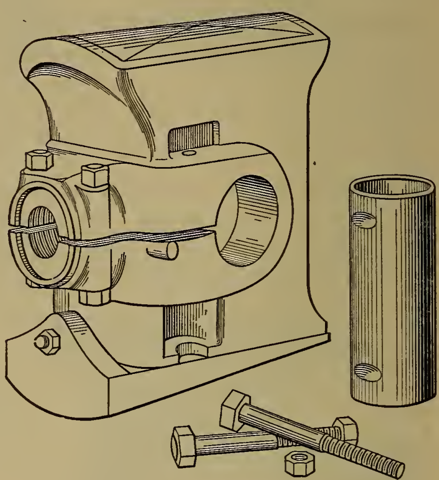


Fig. 531.

Figs. 530 and 531 show the split metal construction for seizing the piston-rod; Fig. 531 the shell type of pin.

387. The Cross-head Pin or Wrist-Pin. The connecting-rod requires a pin on which to oscillate while transmitting its motion to the crank. It is usual to make this pin fast in the cross-head and have the connecting-rod swing on it. This, however, can be reversed if necessary.

The cross-head must transmit the effort to the connecting-rod through the axis of the piston-rod and the connecting-rod. Hence the wrist-pin must either be borne in a hollow in the cross-head or, if the cross-head is solid, the connecting-rod must have a forked end and take hold of the pin on each side of the cross-head. There are objections to this latter plan, to be discussed hereafter, so that it is most usual to support the pin so as to have it in double shear. In vertical cross-heads the pin is apt to be made a tapering fit in its hole so as to be drawn to a tight bearing by its nut. A small key also is used in addition to prevent turning (Fig. 529). In horizontal cross-heads the pin is usually inserted from above into a proper slot. The guides and a steel bolt through the guides keep it from displacement. It is usual to make the wrist-pin hollow in order that oil may be introduced through the center, and so out by a radial hole to the contact-surface (Fig. 531). In the Porter wrist-pin the surfaces outside of the sector of steam effort are flattened away so as to form an oil-cellar from which the surface of the connecting-rod will continually draw oil upon working surfaces.

388. Parallel Motions. In beam-engines, where the guides for the cross-head can only be secured by braces to the frame, which makes their alinement troublesome and uncertain, it has been quite usual to dispense with guides, and to control the cross-head by means of jointed linkages. These linkages are so designed and proportioned that the motion of the cross-head is compelled to be in a straight line, either exactly or so very nearly that the error is inappreciable. Such linkages are called parallel motions. The best known are Watt's, Evans', Russell's, and the Peaucellier cell. (See Fig. 402A.) Their use is restricted in modern practice to a very narrow scope, and the student is referred to treatises on kinematics for a discussion of their properties.

389. The Connecting-Rod. The connecting-rod in the typical engine mechanism must transmit the alternate push and pull of the steam effort to the revolving crank-pin. It must furthermore withstand the tendency to bend transversely due to the flinging effect caused by its own weight or mass as it passes the half-stroke point and has its transverse motion suddenly changed (paragraf 274). Furthermore, its bearings at the two ends are exposed to friction and wear, since the entire pressure on the piston must be borne upon the relatively small areas of the pins, and the crank-pin rubs its contact-surface in the connecting-rod through a space equal to its own circumference in one revolution and under the pressure due to the steam. It will be apparent that the flinging strain will be greatest with a long rod and at high rotative speeds. The rubbing difficulty will be greatest with high pressures and large diameters, and the wear greatest with high rotative speeds.

The cross-section of the connecting-rod to meet these requirements is in most cases an elliptical or oval, or even an elongated rectangle with rounded top and bottom having the longer axis in the plane of motion. Lengthwise the greatest section is either put at the middle or in more modern practice it is gradually tapered from the cross-head to the crank (Figs. 526, 537). Flinging effect is zero at the cross-head pin and is greatest at a point just behind the crank-pin, or more properly at the radius of gyration of the rod. In very long connecting-rods, such as are used in river-boat practice East and West, the connecting-rod (here often called a pitman) is braced by a king-post trussing of wrought-iron rods whereby strength to push and pull is fully retained and yet a much lighter rod results than would be the case if stiffness were sought by a solid deep rod (Fig. 389). A section of steel rod which has become much used in locomotive practice and elsewhere where the conditions for the connecting-rods are very severe is the I-shape section, in which the two flanges give strength against deflection and all unnecessary metal and weight are withdrawn which would bend the rod (Fig. 376, and paragraf 264).

The effort of the connecting-rod tends to deflect the end of the piston-rod in a vertical plane. The shorter it is the worse this difficulty. With a connecting-rod of infinite length there is no tendency to bend the cross-head and piston-rod. Ordinarily for practical reasons the connecting-rod will be two and a half to three times the length of the crank. It will be apparent that a connecting-rod of finite length introduces an irregularity into the motion of the piston (paragraf 260). The piston has moved through more than half-stroke outgoing when the crank is at 90 degrees from its dead-center, and on the return from the outer dead-center it has not moved through half-stroke at the 270-degree point. These irregularities affect the accelerating of the reciprocating parts, but in ordinary cases are masked by the fly-wheel and by the steam-distribution.

390. The Stub End. In order to provide for the concentrated strain on the crank-pins and cross-head pins of engines an especial appliance has become nearly universal. The pins are usually of steel, carefully hardened in best practice, and it is desirable that they should not wear by abrasion, but that if wear must occur it should be concentrated upon the surfaces which bear upon these pins, rather than on the pins themselves. Furthermore, the construction of these bearing-surfaces should be such that wear may be easily taken up to prevent lost motion or pounding, and that when worn they may be easily and cheaply refitted or replaced. These conditions have brought about the combination of brasses, strap, gib and key, or cotter and wedge which is known as the stub end of the connecting-rod.

The brasses are two half-cylinders which embrace the pin and form the bearing. They are called brasses when made of bronze (copper-tin alloys) as is usual, and even if made of cast iron. They may either be true bronze bearings, or they may be made with recesses into which Babbitt or other bearing metal is cast to form the actual contact-surface. The special purposes served by the brass or bronze bearing are, first, that it is easily cast and tooled; second, it is softer than the steel pin, and the wear will be concentrated upon it; third, it has a low coefficient of friction in case lubrication should become defective; fourth, it has a high conductivity for heat, and so draws heat of friction from the pins.

In marine practice, and elsewhere where it would be inconvenient or impossible to stop for any time, spare brasses can be kept on hand to replace those which must be allowed to wear themselves out; and the replacing of such worn brasses is not a matter of shop repairs, but can be made by simply taking down the joint.

The brasses should touch each other at the point which divides the bearing in two halves. As the bearing wears and lost motion begins, the brasses should be filed or scraped down until the wear or lost motion

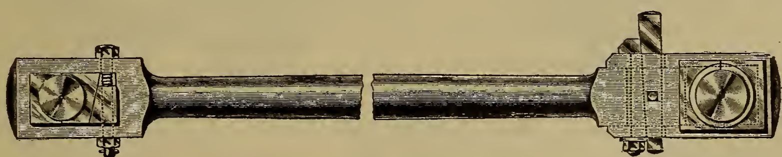


Fig. 532.

is taken up. Another plan is to have the joint open a little when the two half-bearings are in place and fill the gap with liners of thin sheet metal so that the bearing can be made solid. As the bearing-surfaces wear, these liners are successively taken out until the joint comes brass and brass, when refitting is necessary. Not to fill the opening between the brasses is to invite a cramping of the bearing upon the pin with friction heating and all attendant difficulties. In some locomotive practice in the past the brasses have been capped so as to incase the crank-pin completely and keep dust out.

The end of the connecting-rod proper bears against the outside of one brass while the other is drawn against the first half by a U-shaped forging called the strap (right-hand end of Fig. 532). The strap is carefully adjusted to the brasses and the connecting-rod end, and is held in place and to its work by the combination which is known as the gib and key, or cotter. From Fig. 532 it will be seen that the gib and key in this form of stub are counterparts and form compensating inclined planes. As the key slides along the gib, the width at any section is increased. If then the gib and key be fitted in slots in the connecting-rod body and in the strap so that the key rests against the outer edge of the slot in the connecting-rod, the effect of driving down the key will be to draw the strap back, since the gib bears upon the strap, but is free from the inner end of the connecting-rod slot. By drawing back the strap, the joint in the brasses closes together and the key refuses to drive. A set-screw keeps the key from sliding out, and a solid construction results which is nevertheless easily removable and adjustable.

It will be apparent that as the brasses wear in the form of stub shown, and the key is driven down, the effective length of the connecting-rod shortens. In time also the slots in the strap and rod end will come to match, and the key will drive no farther. This difficulty will be met either by renewing the brasses altogether or by fitting in between the rod end and the inner brass liners or shims of sheet metal which will move the center of the bearing outwards as much as the wear has shifted it inwards. The form of stub shown at the right end of Fig. 532 is called an "open stub." If open stubs are used at both ends of a connecting-rod, its effective length is shortened at the two ends by driving in the keys.

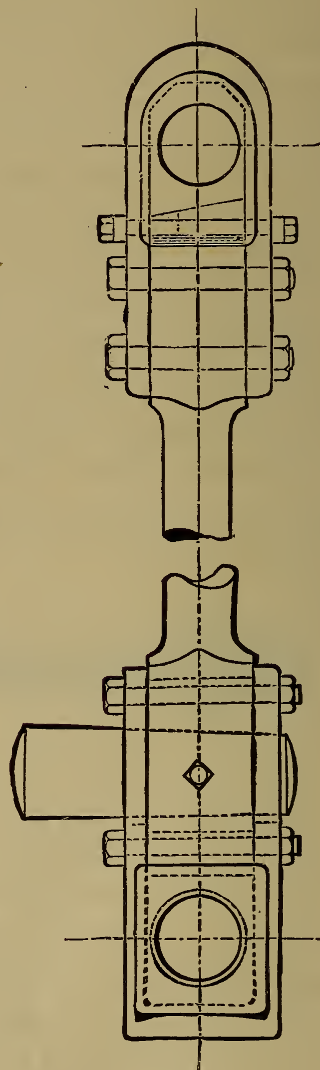


FIG. 533.

It is called a closed stub when the gib or key bears against the inner brass directly, with the end of the rod as its abutment bearing-surface. The left end of Fig. 532 and the upper end of Fig. 533 show this construction using a wedge instead of a gib and key. As the wedge is adjusted inwards the inner brass moves towards the outer and away from the center. The closed stub thus lengthens the rod to take up wear. If a closed stub is used at one end and an open stub at the other the distance between crank-pin and piston is varied by the difference in wear at the two joints; and if there is no difference in this wear, the length of the mechanism remains constant. This plan is much the most usual and to be preferred. The closed stub may be applied either to a

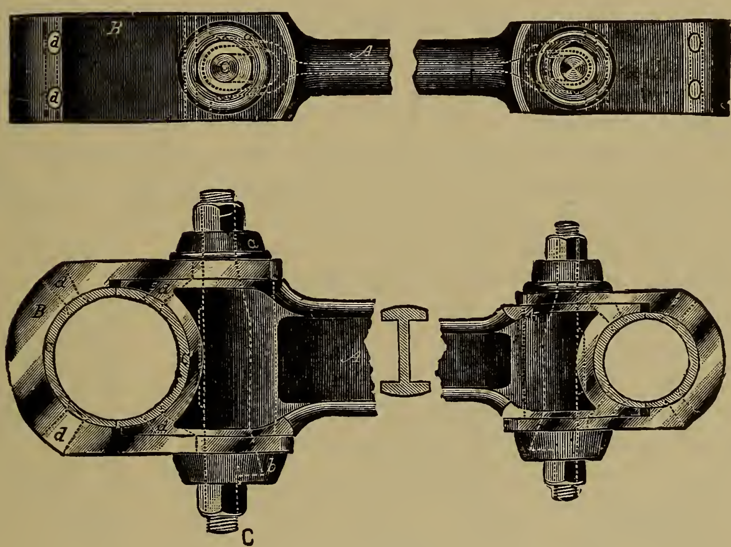


FIG. 534.

rod whose end is forged solid (Fig. 532), or the strap may be strongly bolted or keyed and bolted as shown in Fig. 533. The wedges which are much used in modern engines for setting up brasses are operated by screws and are fully shown in the illustration. To prevent the loosening of keys in large engines and at high speeds, where the set-screw would not be enough, the end of the key is sometimes drawn down to a rod and threaded. A nut on this rod bears upon a Z-shaped bracket bolted to the rod and holds the key in place. It is the trouble arising from slinging of the key which has caused the wedge with its bolt to receive preference in modern usage. Fig. 534 shows a stub of this type, and illustrates also the I section of the rod. A form of stub first introduced in marine practice is shown in Fig. 535. The gib-and-key construction is abandoned, and the half-brasses are held together in a jaw by bolts parallel to the length of the rod. These bolts have to withstand the push and pull of the rod, but they make a very stiff and strong stub particularly well adapted for crank-pins of considerable length. They are also the foundation for very deep connecting-rods made of hollow tubes for compression, while through-bolts resist the tension as they hold down the outer halves of the brasses (Fig. 363).

Another form of stub is known as the round-end stub. The end of the connecting-rod has a tapering hole within which is inserted a bronze bushing which fits the taper on its outside and the cylindrical pin on its inner side. As wear takes place, the split in the bushing is filed out, and the bushing forced a little farther into the taper hole whereby it is closed together. This is particularly adapted for parallel rods and side rods of locomotives, where it is necessary that the length between the centers should always remain the same. With the gib-and-key plan one end of such side rods had to have double keys, one outside the brasses and one inside. Sometimes this cylindrical bushing plan is used without a split and provision for adjustment. When wear has become enough to be annoying, the worn bushing is thrown out and a new one put in place. In light rods with small pins the solid eye is sometimes split, a little metal sawed out, and the split held from opening by a bolt. As the bearing wears, tightening of the bolt closes up the slit and takes up lost motion.

An interesting provision for taking up wear with ordinary brasses is shown in Fig. 536A in the Appendix. A cavity behind the inner brass is filled with steel balls, and into that cavity a set-screw projects. The balls displaced by the screw press the brasses outward, and yet are practically immovable from an outer force. They act like the particles of a fluid to exert equal pressure upon the brass. This device is the invention of Mr. C. W. Hunt, but is open to the objection that the pres-

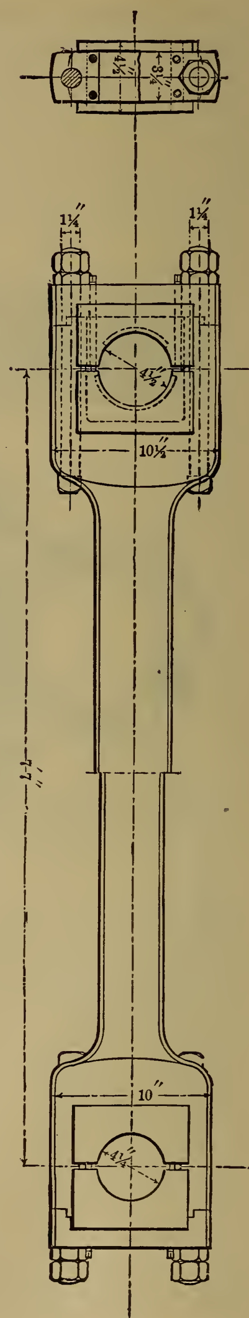


Fig 535.

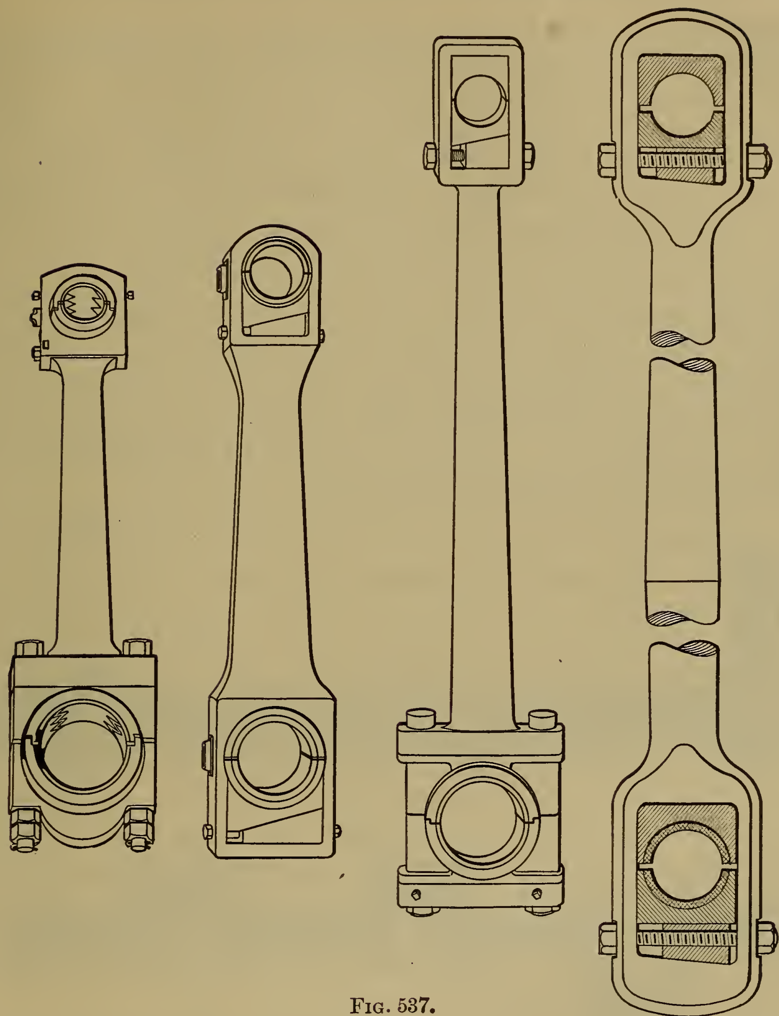


FIG. 537.

sure from the adjusting bolt through the balls to the brass becomes excessive in heavy hands, and deforms the contact surfaces.

Fig. 537 shows strapless types and combinations of design.

391. Forked-end Connecting-Rod. Double Rods. When it is convenient or necessary to have the cross-head pin supported at its middle and to have the motion taken off symmetrically on each side of the axis of the piston-rod, the connecting-rod end may be formed into a sort of rounded Y with a bearing on each arm. This bearing will be of the usual stub construction, with provision for taking up wear (Fig. 380). The difficulties and objection to this forked-end construction are those which result when the two bearings wear unequally. The consequence of unequal wear is that one side or the other draws the farther end of the rod to one side and against the collars of the crank-pin when it is keyed up after refitting. The strength of the connection may be enough to keep the rod continually out of straight, causing friction, heating, and wear. The only proper way to treat such a forked rod after refitting is to take off the crank-pin strap and brasses and, with the brasses of the fork keyed up close, test the alinement of the naked end of the connecting-rod with the crank-pin at the inner and outer center. If it does not fall in line with the pin, liners must be introduced behind the brass on the short side, if it is an open stub, until the alinement is perfect. Forked-end connecting-rods are usual with the main connecting-rods of beam-engines, where they are made necessary by the support of the pin by a single beam. The use of a double beam with the pin between them makes this construction unnecessary.

The symmetrical connection of the cross-head of a beam-engine to the central beam of such engines would compel a connecting-rod forked at both ends. This would be troublesome and difficult, and for this reason it will be found that it is usual to use two short connecting-rods for this purpose. Each has two stub ends, and the same difficulty attaching to unequal wear requires to be guarded against here. Unequal length of these rods springs the cross-head, twists the beam, and gives general trouble. The same difficulty is to be guarded against in engines of the back-acting type for blowing or pumping, such as shown in Figs. 378 and 379, where the connecting-rods are attached to crank-pins outside of the fly-wheel from the wide cross-head. So critical is this difficulty from unequal length of two similar connecting-rods that the proper construction for such a case is to have the cross-head merely pinned to a boss on the rod so that it may yield and adjust itself to such slight inequalities of length. Many massive cross-heads have been cracked from inattention to this detail.

Small connecting-rods which can be hollow are frequently arranged to have a key at one end (or even in the middle) set up the brasses at both ends. A rod which passes through the hole in the bore bears against the brass at one end and against the key at the other. The term pitman sometimes applied to a connecting-rod should be limited to either a massive or long connecting-rod or to the connecting-rod which couples a vibrating beam or treadle to a revolving crank. The latter is its proper use, but it has been sanctioned by usage as a name for those wooden rods, stiffened by iron forgings on top and bottom, which are used as connecting-rods for the marine engines of the Western rivers. The mining origin of the term has been already referred to (paragraf 277).

CHAPTER XXIV.

CRANK-SHAFT. ECCENTRIC. FLY-WHEEL.

395. The Crank-Shaft. The effort of the steam-pressure PA upon the piston is transmitted to the crank-shaft through the crank-pin. The arrangement of the shaft as a whole must therefore be conditioned by the number of cylinders, and their arrangement respecting each other. The construction of the side-crank and center-crank designs also determines the arrangement of the cranks and pins upon the shaft. The illustrations of types chosen hitherto for illustration present these differ-

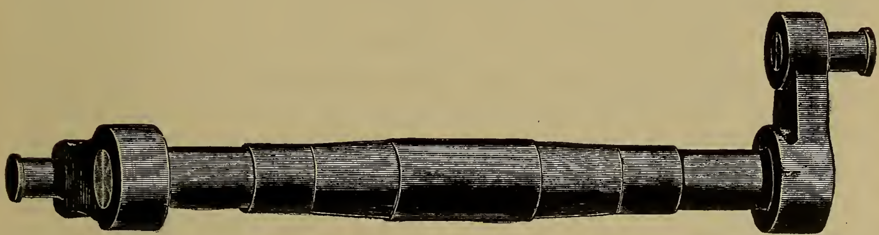


FIG. 540.

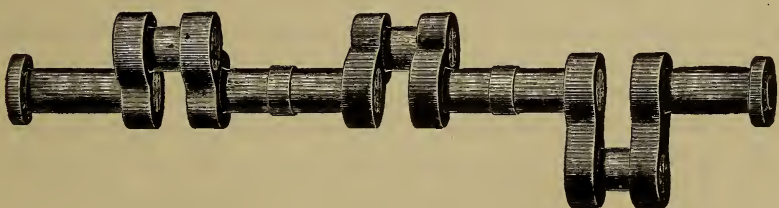


FIG. 541.

ences. The principal point to signalize is the distinction between single and multiple cranks; and the necessity for bearings close to the effort of the connecting-rod. The double crank produces less lack of balance and makes a stronger crank-pin. Fig. 540 shows the overhanging crank-pin construction of a cross-compound engine either horizontal or vertical; Fig. 541 gives the triple or three-throw crank-shaft intended to be supported on four bearings with each crank double.

The British name for the crank is the throw. Constructively it is a beam loaded at one end and fixed by the resistance to be overcome at the other. The pin is the point of application of the load.

396. The Crank-Pin. The crank-pin is usually of high-carbon steel. It requires to be very solidly inserted into the eye of the crank, and to this end three methods are usual. First, it may be forced in by a press. The hole in the eye is made cylindrical, and the end of the pin which enters the eye receives a very slight taper at the very end, but the cylindrical contact is practically of the same size in the eye and the pin. The pin is then coated with white lead and forced into the hole, which is a little too small for it. This is done either by hydraulic or screw presses exerting a force of 20 or 30 tons. The second method is to shrink the crank upon the pin by the method described in paragraf 382. The third plan is to have the pin and the hole taper, while the inner end of the pin is finished into a screw-thread on which fits a nut by which the tapers are drawn together. In some forms of disk-crank the pin may be held with a key. It is very usual to model the crank-pin with collars to prevent sidewise displacement of the brass of the connecting-rod upon it. On the other hand, many crank-pins are made without collars, and the shape of the brass keeps the connecting-rod from the plane of the crank, and a plate which bolts to the end of the pin forms a finish which the eye seems to demand, and keeps the connecting-rod from appearing to slip off. When collars are used it is common to fillet the corners rather than to give them a sharp angle where they join the bearing-surface. The pin is not only stronger, but there is less danger of a binding of the brasses.

397. The Crank. Whether single or double, cranks may be of cast iron or wrought iron or of steel. Since the shaft must be of wrought iron or steel, the continuous crank which is made in one piece with it, must be of the same metal. The cast-iron crank is therefore limited to cases where the shaft is built up. The ordinary form of cast-iron crank is shown in Fig. 542. Such crank requires to be secured to its shaft by means of steel keys inserted partly in the shaft and partly in the hub of the crank. It is usually more convenient to make use of two keys than to try to get sufficient shearing area in one. A much more usual form of the cast-iron crank is the disk-crank, such as shown in Fig. 526. This arrangement permits of balancing the weight of the crank itself on the other side of the center of motion, and furthermore gives a convenient space for additional metal which may serve to counterbalance the living force of the reciprocating parts. At high speeds, moreover, the disk-crank meets less resistance from the air. Where the disk-crank is not used in vertical engines a form of balanced crank, such as shown in Fig. 543, will be required to offset the weight or unbalanced effect of the mechanism. Some designers have built up cast-iron counter-weights upon a steel crank. Fig. 544A in the Appendix shows an arrangement of this sort.

When there are two cylinders to work upon one crank-shaft and it is to be a continuous or double crank, the double crank will be forged solid if the length of the stroke is not too great. The excess of metal will be cut out by slotting, and then the pin turned by mounting the shaft eccentrically in a lathe of sufficient size. Of course it is not easy to forge

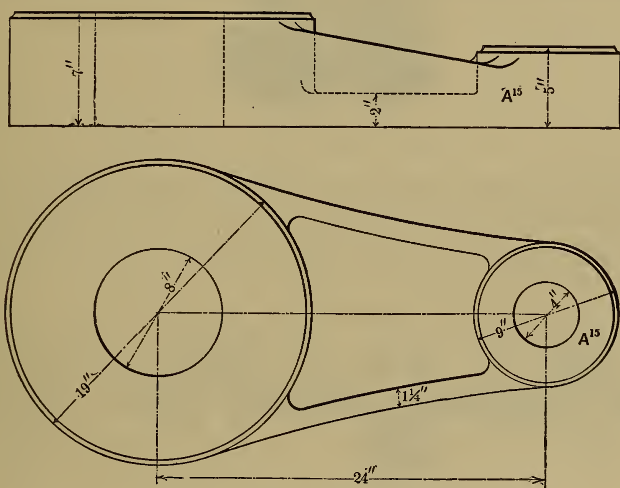


Fig. 542.

such cranks with short distance between them, nor is it easy to get them truly at right angles. Built-up crank-shafts, where the crank-pins and lengths of shafting are separate or fastened together by shrinking and keying, have been much used in marine practice (Fig. 545). For light service at high speeds and short stroke as in motor-vehicle practice, the drop-forged multiple crank shaft has had considerable acceptance.

398. The Locomotive Crank and Shaft. The ordinary American locomotive is constructed so as to be what is called outside-connected. The cylinders and driving mechanism are outside of the frames, and the crank-pins are inserted into proper bosses in the driving-wheels. Inside-connected engines, with the cylinders and mechanism between the frames, require cranked axles and have no pins on the drivers. The inside-connected designs have been most popular in Europe by reason of a supposed steadiness and because the effect of torsion between the two cylinders is exerted on a less length of axle. The cranked axle is, however, more difficult to forge. The inconvenience of having the principal parts of the mechanism clustered together under the hot boiler and between the frames, and the possibility of equal steadiness for the outside-connected design, have given the latter the preference in

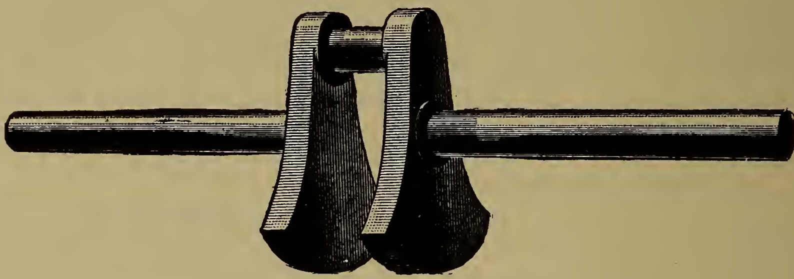
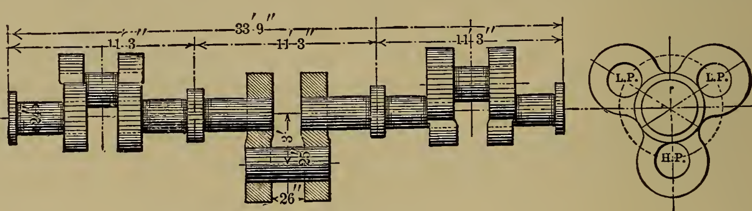
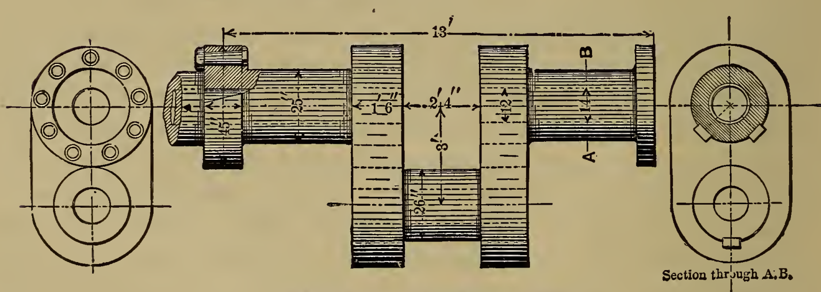


FIG. 543.

America, but for some compound designs of more than two cylinders it is unavoidable to use one cranked axle. The driving-wheels are pressed on the axle by heavy hydraulic pressure, and twisting is prevented by



Constructed by Messrs. John Elder & Co.



Constructed of Whitworths Fluid Pressed Steel.

FIG. 545.

keys. The crank-pin is also pressed into its boss. Where the length of the engine permits, the connecting-rod or main driving-rod acts upon the crank-pin of the main or forward driver. In short engines the connecting-rod will go to the rear driver. In the first case the main pin will have two bearing-surfaces on it, that for the main rod being nearer the face of the wheel. In the latter case the main-rod bearing will be outside of the bearing for the parallel or side rod. Where there are three drivers on a side the main bearing will be outside, and the side rod connecting the main to the front driver will require a pin-joint near the main stub so that no cross-strain shall be brought upon it from inequalities of level in the track. Locomotive crank-pins seem to undergo a structural change from the combined effect of vibration and shock to which they are subjected, so that it is a custom to force them out after a certain number of miles have been run and have them forged over and replaced, to prevent a sudden breakage on the road with its attendant disaster.

399. The Marine Crank-Shaft. For the ordinary paddle-wheel service in deep Eastern waters the crank is a double one, forged, and built up with inserted pin. The two halves are essentially alike, and with bearings close to the crank in the main frame and within and without the wheel. In a very few cases the two halves of the double crank are not in the same plane, but one is slightly behind the other in an offset eye. The object of this is to diminish the considerable danger in all long-crank engines lest the crank settle down upon the lower center by the action of the waves upon the propelling wheels when the engine is at rest, which makes it troublesome to start. In Western river-boat practice with side wheels the shaft is usually not continuous across the hull, but the two engines are separate and are separately handled. This gives greater maneuvering power in currents and for landing. Some special ferry-boats for railway service have also been constructed in this way. For marine engines which drive propellers at the stern it is apparent that the entire energy which propels the vessel must find an abutment against the lengthwise thrust of the screw in the construction of the shaft itself. This is done by means of what is called the thrust-bearing. This consists of a large bearing in which a sufficient number of grooves or rings is formed which fit corresponding collars upon the shaft. The area of these collars and their number are proportioned to the energy for which they must provide, and the contact-surfaces in the bearing are very carefully fitted with Babbitt or similar bearing metal. These bearings are also cored so that circulation of water can be provided to keep them cool. These thrust-bearings are usually placed close behind the engine, so as to be always under the careful scrutiny of those running the engine. As the engine will be located as a rule near the center of gravity of the hull, there will have to be a number of joints in the propelling shaft both for convenience of manufacture and for convenience of handling and repair. These sections will be joined by flanges carefully and strongly bolted together. Fig. 546 shows the construction of the thrust-shaft and propeller section of a marine engine, and Fig. 547 the provision made at the stern to permit the shaft to pass outwards through the hull. The joint is made water-tight by means of stuffing-boxes, and the actual bearing of the shaft is upon lignum vitæ or similar bearing material. Such shafts of large diameter are apt to be made hollow in best modern practice in order to secure strength with lightness and to eliminate the defects which in solid forging are apt to concentrate themselves at the center both of the ingot and of the forging which results from it.

400. The Main or Crank Bearing. The bearing of the crank-shaft close to the crank has to withstand all push and pull due to the steam

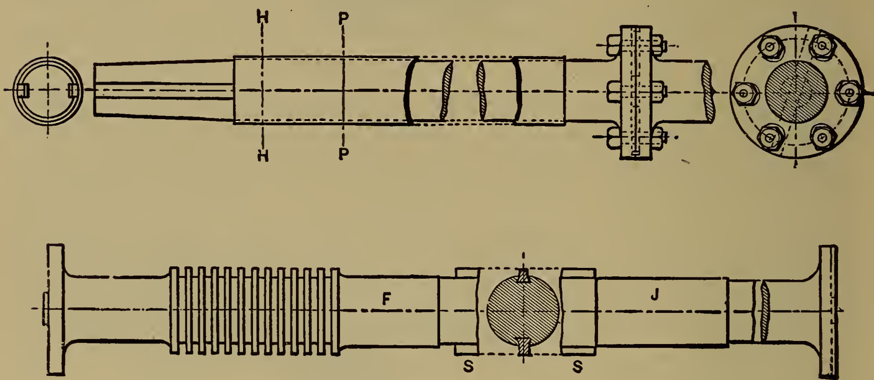


Fig. 546.

effort for which it is the fulcrum, and also the weight of the shaft, fly-wheel and attachments, and the pull of the belt, if one is used to take the power off from the shaft. Furthermore, the shaft turning in this bearing must remain very carefully in line both back and forth, and up and down.

To meet these requirements the main bearing must have a generous area of contact so that alternate pressure shall be unable to become so great as to squeeze out the lubricant from the contact-surfaces, and it must be capable of minute adjustment to compensate for wear. Fig. 548 shows a usual construction of such main bearing, in which, instead of two half-boxes as in the stub ends, the bearing-surface is made up of four segments. These segmental bearings are called quarter-boxes, and in the design shown are separately adjustable by means of wedges which come down through the massive cap of the bearing. The quarter-boxes are the ones which have to withstand the steam effort in a horizontal engine, while the lower one has to meet only weight. Fig. 549 also shows the combination of liners and wedges to keep the center

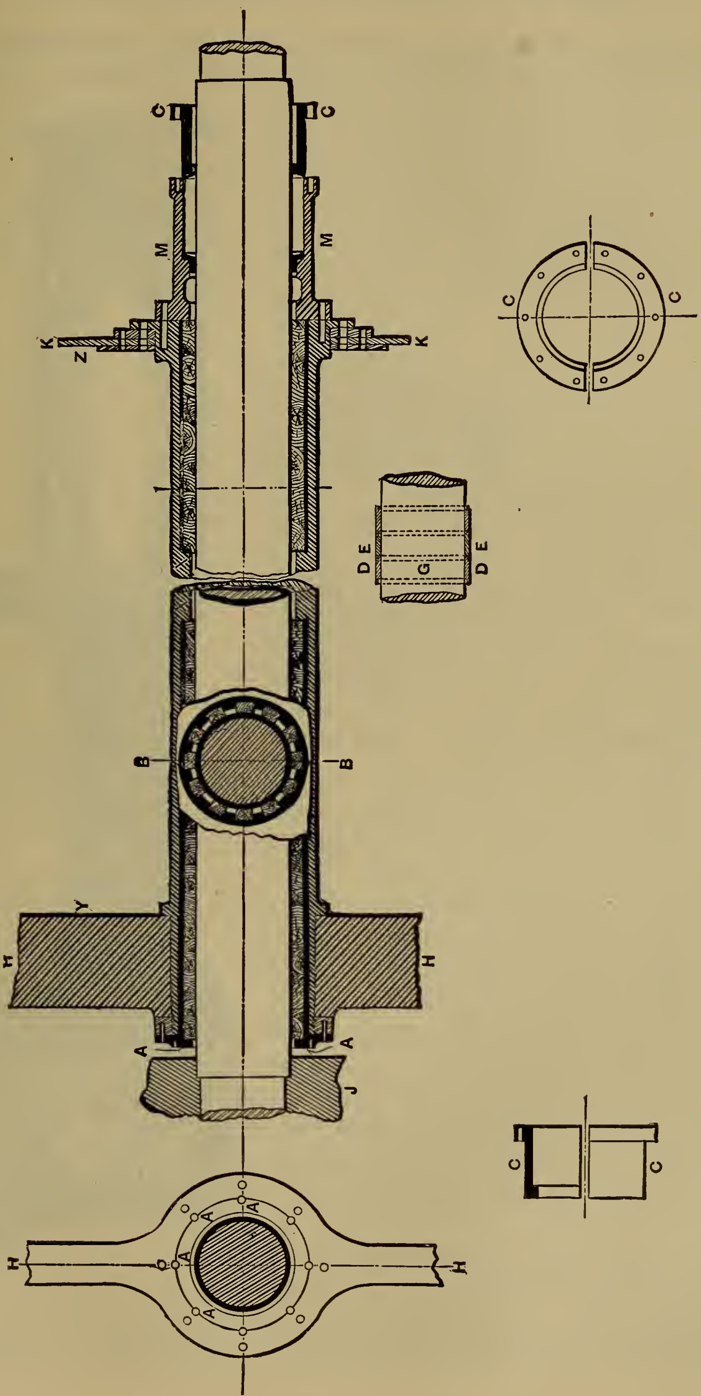


Fig. 547.

distance of the shaft from the cylinder a constant. It exemplifies a method of continuous oiling from a well below the bearing. Rings may be used instead of chains. Many different modifications of the wedge idea are to be met in the various bed-plate designs, such as set-screws through the face or side of the bearing and the like, but the same underlying principle is present in all. The main bearing requires special and abundant provision for oiling, to which reference will be made in proper course. The other type of construction is the removable shell which is replaced as it wears.

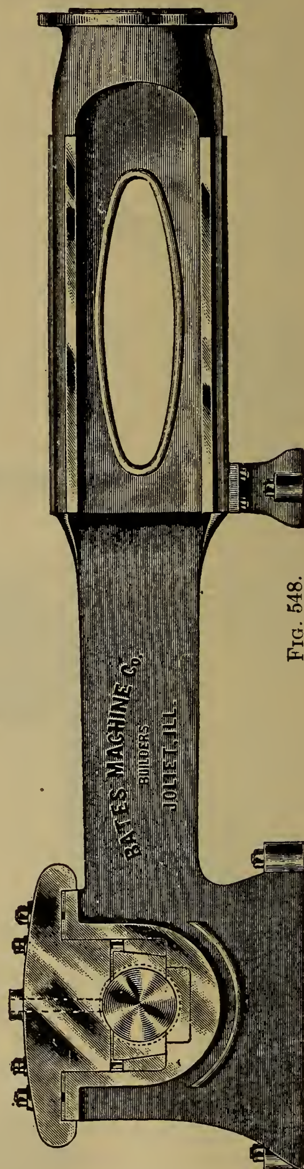


Fig. 548.

The outer bearing or outboard bearing of the engine-shaft has already been discussed with the necessity for its adjustment for proper alignment. It has to withstand only the weight of the shaft and the pull of the belt. Both bearings should have length enough to prevent the shaft from bending under the strains to which it is exposed when the diameter of the shaft has been intelligently calculated. One or the other bearing should have collars to prevent undesirable endwise motion. These collars, however, must not offer any danger from a seizing caused by expansion due to heat. Such a bearing is said to be "collar-bound," and excessive friction is the result.

401. The Eccentric. This eccentric, when not forming a part of the shaft-governor, will usually be placed just outside of the main bearing. It will be fastened to the shaft either by keying or by set-screws or by both. In many cases it is forged solid on the shaft. By reason of the diameter of the eccentric the stub construction is not usual or convenient, but the rod fits the disk by means of a bearing-surface which is called its strap. This strap is made in two halves which meet on a diameter at flanged surfaces by means of which the two halves are bolted together. The large area of contact due to the large diameter of the pin makes adjustment necessary only at long intervals as slow wear occurs, and this is done either by filing away the joint of the strap or by removing the liners of thin sheet metal, one by one, which were put in there when the joint was first fitted. To prevent sidewise motion of the eccentric strap, it is made either to fit in a groove in the face of the eccentric, or the eccentric fits in a groove in the strap. The latter plan has some advantages, since the strap thus forms a trough within which the oil

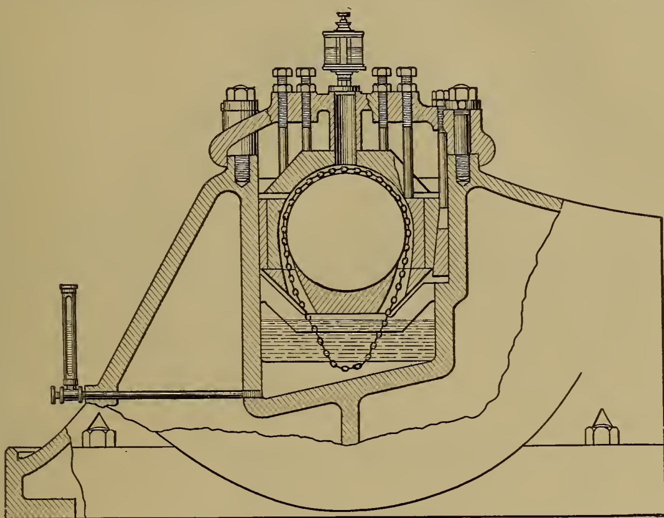


Fig. 549.

will gather and be retained at the bottom, whereas in the other arrangement the oil has a tendency to run off (Figs. 550 and 551).

It will be made apparent in the later treatment of valves that their operation is most convenient by the use of a crank. To get the desired short throw for a crank in a large diameter shaft by cutting its material away would be impossible as it would weaken it both against torsion and flexure. Hence the expedient of so enlarging the crank-pin of such crank to drive the valve that its diameter exceeds the diameter of the shaft enough to let such shaft pass full size through the pin which drives the valve. The eccentric disk is the result of this process. (See paragraf 420, Fig. 570.)

402. The Eccentric-Rod and Valve-Stem. The components of the crank-motion of the eccentric which are not needed to move the valve must be provided for as in the main connecting-rod. Hence there will be a joint of some sort between the eccentric-strap and the stuffing-box at which the valve-stem enters the valve-chest. In small and short engines the weight of the eccentric-rod will be small, so that it will be enough to provide a flexible or pin joint at the end of the valve-stem without providing a means to guide the latter except that provided by the stuffing-box. In heavier engines the end of the valve-stem may either be guided by a slide, or a rock-shaft must be interposed which will carry the valve-rod and from which the valve-stem will be driven. Where this rocking-shaft or vibrating lever is introduced it furnishes a very convenient means to modify the throw of eccentric and valve, and

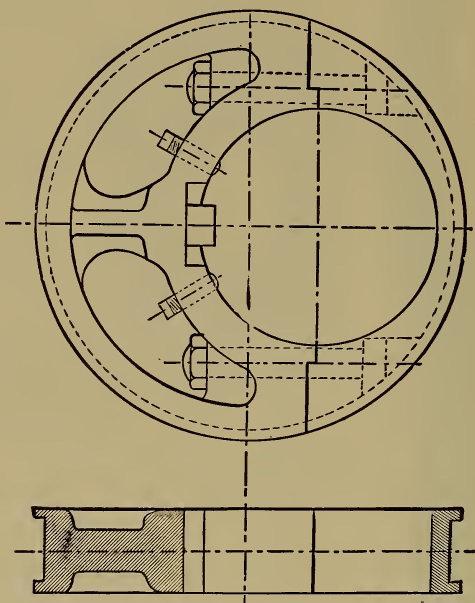


Fig. 550.

also gives opportunity for hooking and unhooking gear (see paragraf 461 and Figs. 618, 620, 622). The principal joints of such valve-rod and eccentric-rod may either be stub ends, or hardened steel pins may be used with hardened steel bushings which they fit accurately. The rubbing work is so small that such well-made work lasts indefinitely. The eccentric-rod is usually fastened to the eccentric-strap by screwing it into the latter with a jam-nut to prevent its working loose; in larger engines it will be a taper fit brought home with a key. In very long engines, such as are met with in river-boat practice, the eccentric-rod will be an open-work trussed structure of flat rods which ends in the single flat or square rod guided by the roller-frame by which it is unhooked. The locomotive-rods are usually flat, and are bolted sidewise into recesses made for them in a tail formed upon the inner eccentric-strap (Fig. 622).

The valve-stem is the name applied to the short rod which enters the steam-chest and actuates the valve. It will be either attached to the valve by means of a yoke which embraces the latter, or by a screw-joint with the necessary jam-nuts. The valve-rod is synonymous with the valve-stem except where the Stevens cut-off is used, where the valve-rod

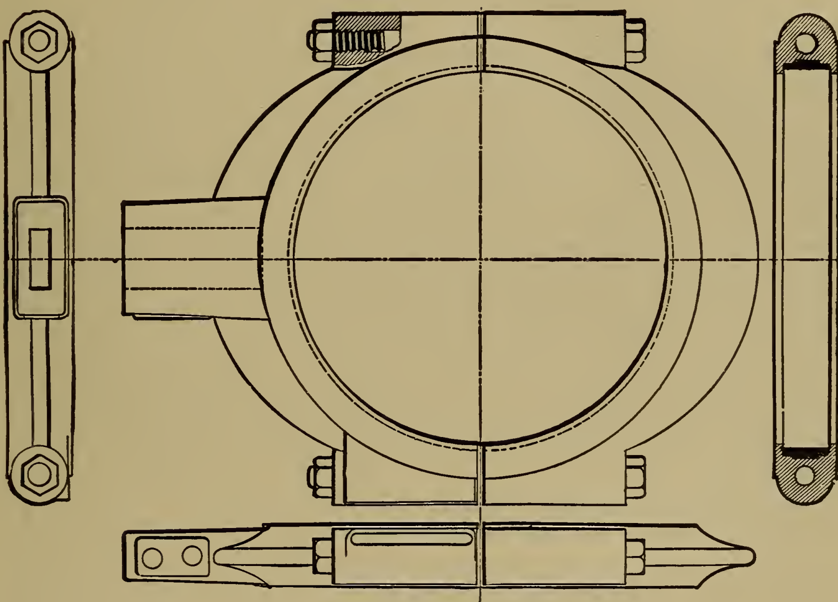


Fig. 551.

is the massive rod lifted by the toes, which carries a bracket or offset to which the valve-stem proper is attached by means of jam-nuts, whereby careful adjustment is made possible.

403. The Fly-Wheel. In early engines turning with a low number of revolutions, the fly-wheel required to be of large diameter, and was for this reason nearly always distinct from the wheel from which the power was taken off. In more modern engines the convenience of having the fly-wheel serve also as an element of the transmissive machinery has brought around the use of fly-band-wheels, where belts or ropes are used to take off the power from the engine-shaft. It is so much less the practice in recent years to use gearing in transmitting from the engine-shaft that the fly-wheel is very rarely a toothed wheel. In electric plants the generator mass always is made to discharge the fly-wheel function.

The function of the fly-wheel is threefold. First, to store up excess of energy received from the piston in one part of the stroke, and to give it out when the effort shall have grown less by expansion. Second, to equalize variation in the leverage with which the varying steam-effort acts upon the crank to revolve the shaft. Third, to give out or absorb energy when variation in the external load or resistance occurs suddenly. The fly-wheel is therefore an accumulator and an equalizer, and the reserve which it stores will be greater as its mass is greater and the leverage greater with which that mass acts. Since large mass means great weight, it is often convenient to increase the virtual radius of the wheel (mathematically its radius of gyration), and thus diminish weight which causes friction in the bearings. The objection to the large wheel is the space which it occupies vertically, and the complication in foundation which it causes. With large diameters centrifugal force in the rim becomes considerable, and may become equal to or surpass the tensile resistance of the material of the rim. For these reasons it will be found that smaller diameters prevail in modern engines, and that roughly the relation of four times the stroke of the engine is likely to approximate the diameter chosen. In early engines thirty-foot fly-wheels were often to be met, but now eighteen to twenty feet is a large diameter, and in center-crank high-speed engines six feet has become a large size.

The function of the fly-wheel as a regulator is quite distinct from that of the governor. The fly-wheel is to compensate for instantaneous variations, and give out or absorb energy, and maintain a constant speed under variations of the equality between effort and resistance which are too small to reach the governor and cause a variation of the cylinder-effort. For permanent variations, where the load is increased or diminished, the capacity of the fly-wheel is soon exhausted, and the engine will either increase or diminish its speed. The governor must then adjust its mechanism to bring the engine back to speed, and adjust the piston-effort to the new value of the resistance. It is often found in electric-railway power plants that wide variations occur in the current upon the line without the governor showing any appreciation of them. This is to be explained by the action of the fly-wheel and the absorption and giving out of energy under such instantaneous variations. The weight of the fly-wheel must be very largely determined by the character of the external resistance. A weight capable of equalizing and steadying the variations of the cylinder-pressure and of crank-leverage with a constant resistance would not be enough to serve as the necessary reservoir when heavy demands of power are made for short intervals. The best illustration of such wide variation of resistance is the rolling-mill engine, in which only the friction of the machinery is to be overcome when the train is empty, but in which the maximum power of the engine is taxed when the piece is between the passes and undergoing the action of the rolls. Rolling-mill-engine fly-wheels will have a weight of from thirty to fifty tons to meet this requirement, and in cable-railway and electric-railway practice, and also with the slow speeds of pumping-engines, very massive rotors are used.

404. The Mechanics of the Fly-Wheel. The discussions of Chapters XV and XVII are fundamental to a computation of the mass to be given to a fly-wheel. The effects of inertia brought out in paragraf 260, and the value of the net turning effort given in paragraf 263, and the variation of the impelling force due to working the steam expansively (paragraf 298) all enter as factors.

If from the diagrams of effort upon the crank-pin, in Figs. 365, 373, 375, 377, and from Fig. 374 in particular, a curve of efforts to turn the crank-pin be constructed, it will resemble Fig. 552. The length of the base-line AE being the semi-circumference of the crank-pin-travel in one revolution, the height of the ordinates should be reduced from those resulting from the indicator diagram on the piston area by the ratio of 2 to π to make the areas of effort in the two diagrams check with each other. From the beginning of the stroke at A to the point B the inertia of the piston and attached parts is being overcome and the uniform resistance RR' is greater than the effort. From the intersection of the effort curve with the resistance curve at B to the point F the effort is in excess, the cut-off having taken place at C. From F to the end the drop of steam-effort and the compression have neutralized the forward effect of the inertia, and the effort is again less than the resistance. F will be the point of maximum velocity during the stroke, since the effort has been in excess from B until the resistance becomes in excess, and B will be the point of minimum velocity with uniform resistance. Hence: let

R = the radius of gyration of the mass forming the fly-wheel, of arms and rim. This may be called the outside diameter of the wheel.

V = the mean velocity of a point at that distance from the axis; practically that of the outer periphery.

W = the weight of the rim, neglecting the arms: $= Mg$.

V_B = minimum velocity of rim; that at B in Fig. 552.

V_F = maximum velocity of rim; that at F in Fig. 552.

Then the energy resident in the revolving mass at B will be

$$E_B = \frac{WV^2_B}{2g};$$

and at the point F,

$$E_F = \frac{WV^2_F}{2g},$$

so that the increase from the one state to the other or during the angle or time represented by BE,

$$E_F - E_B = \frac{W}{2g}(V^2_F - V^2_B).$$

But the excess of steam-effort over the mean constant resistance must be source of such increase in energy; this is represented in the diagram by the area above the line RR', shown in double-section lines. Rankine called this ΔE, and it is quantitatively a work in foot-pounds which can be related to the average work of the stroke which

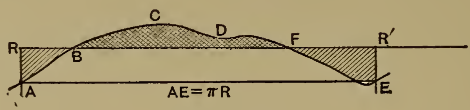


FIG. 552.

is supposed equal to the average resistance. Call $\Delta E = m \times$ work per stroke, or m times the area $ABCDFE$ for one stroke, which if multiplied by $2N$ for the strokes per minute, becomes

$$\Delta E \times 2N = \text{I.H.P.} \times 33{,}000 \times m,$$

whence

$$\Delta E = \frac{W}{2g}(V^2_F - V^2_B) = \frac{\text{I.H.P} \times 33{,}000 \times m}{2N}.$$

If now the variation in percentage of linear velocity on either side of the mean velocity V be denoted by K so that a variation of one per cent on either side makes $K = 0.02$, the mathematical expression for K becomes

$$K = \frac{V_F - V_B}{V}.$$

It may be defined as the coefficient of fluctuation of speed, so that

$$V_F - V_B = KV.$$

If the fluctuation be assumed to be cyclic in character, no error will be introduced by calling

$$V_F + V_B = 2V,$$

whence by multiplication

$$V^2_F - V^2_B = 2KV^2,$$

making the equation for ΔE above appear

$$\Delta E = \frac{W}{2g}(2KV^2) = \frac{\text{I.H.P.} \times 33{,}000 \times m}{2N},$$

from which

$$W = \frac{48{,}500{,}000 \times \text{I.H.P.} \times m}{KN^3R^2}$$

by substituting for V^2 its equivalent $(2\pi RN)^2$. The value of K will vary with the variation permissible in the service performed. In rough service it may be from 0.06 to 0.10. In factory practice with belt drives and no lighting service, from 0.02 to 0.04. In electric service involving lights, from 0.01 to 0.02, and ordinary variations: where variations are excessive and over wide limits it will be lowered to 0.0016 to 0.0025. By assuming sets of values for the ratio $\frac{m}{K}$ the weights per H.P. can be worked out for values of N and R. The factor m will vary with a wide range of condition respecting point of cut-off, speed, accelerating and retarding effort, number of cylinders, and relation of connecting-rod to crank (Chapter XV). The following tables* would appear to give fair average values from a wide generalization. The values for m in multiple cylinder engines cover a wider range of limit values — perhaps twice the average — than in single cylinder practice.

TABLE XIX.

VALUE OF $\triangle E$ OR m AS A FRACTION OF THE WORK PER STROKE.

Cut-off.	Single Cylinder.	Two Cylinders, Cranks Quartering.	Three Cylinders, Cranks at 120°.
0.1	0.35	0.088	0.040
0.2	0.33	0.082	0.037
0.4	0.31	0.078	0.034
0.6	0.29	0.072	0.032
0.8	0.28	0.070	0.031
Full stroke	0.27	0.068	0.030

FOR GAS OR INTERNAL COMBUSTION ENGINES.

Four-cycle, 1 stroke in 4	3.7 to 4.5	1.0 to 1.1	....
Four-cycle, missing alternate charge	8.5 to 9.8	2.1 to 2.5	....

The work stored by the revolving fly-wheel can be expressed in terms of the work done per stroke by the following analysis: since

$$\triangle E = \frac{W}{2g}(2KV^2),$$

and the work stored in the wheel is $\frac{WV^2}{2g}$, this latter can be written $= \frac{\triangle E}{2K}$.

The average work done per stroke is $W_a = \frac{\triangle E}{m}$. Hence the ratio of the stored work per stroke to the average work per stroke is

$$\frac{\frac{\triangle E}{2K}}{\frac{\triangle E}{m}} = \frac{m}{2K}.$$

If a value for K be assumed, then the quotient will be a factor giving the number of strokes whose work must be stored in the fly-wheel to give a steadiness corresponding to that assumption. If K be made one per cent, or the fluctuation be one-half revolution on either side of one hundred, the following table results. If K be n times one per cent, the numbers must be divided by n.

* From "Mechanics Applied to Engineering," John Goodman, London, 1904.

TABLE XX.

ENERGY STORED IN A FLY-WHEEL IN TERMS OF WORK PER SINGLE STROKE OF A DOUBLE-ACTING STEAM ENGINE WHEN K IS ONE PER CENT.

Cut-off.	Single Cylinder.	Two Cylinders, Cranks Quartering.	Three Cylinders, Cranks at 120°.
0.1	18	4.4	2.0
0.2	17	4.1	1.9
0.4	16	3.9	1.8
0.6	15	3.6	1.7
0.8	14	3.5	1.6
Full stroke	13	3.4	1.5

GAS OR INTERNAL COMBUSTION ENGINES.

Four-cycle, 1 stroke in 4	185 to 225	46. to 56	....
Four-cycle, missing alternate charge.	425 to 490	112. to 122	....

When the resistance external to the engine is variable and not a constant, the line RR' in Fig. 552 is not a straight line parallel to the base, but a curve with a variable height or ordinate. The resistance may also go through cycles which are not concurrent with the engine cycle. The method, however, is identical with respect to finding the excess $\triangle E$ and its ratio m to the average work as in the preceding case of uniform resistance.

405. The Stresses in Fly-Wheels. The fly-wheel in rapid revolution has its rim in tension by reason of centrifugal force. If the ring had no arms, it would be all equally in tension; but by reason of the arms resisting extension as the ring expands under strain, a cross-bending occurs between the arms if the wheel is solid, and if made up in segments this bending is concentrated close by the joints. In the second place, as the rim is the most massive part and tends to revolve uniformly, it will happen that when the resistance slows down the engine, the arms will be flexed by the effort of the rim to maintain uniform speed, and, on the other hand, the effort of the piston when the shaft has lagged behind will tend to bend the arms in the opposite direction. Both of these strains bring a very serious twisting effort upon the keys by which the wheel is secured to the shaft. If the wheel is a fly-band-wheel, the effort of the resistance comes directly to bend the arms. In the third place, initial strains of construction may be present in the wheel, which may superpose their effect upon the action of the other two strains. These can be greatly increased if the plane of the rim by bad machine-work should be out of the plane perpendicular to the shaft. A sort of gyroscopic action must occur from the tendency of the mass to revolve in the perpendicular plane. The strains from shrinkage in cast-iron wheels form a great objection to the use of solid wheels of large diameter of this material.

406. Solid and Segmental Fly-Wheels. Small cast-iron fly-wheels can be made all in one piece; the hub (by means of which the wheel is fastened to the shaft), the arms, and the rim being all cast at the same time. The arms may be straight or curved. When straight or curved they are of elliptical, oval, or fusiform section, with the long axis in the direction in which the wheel turns. The elongated section gives strength against the distortion of the rim as speed varies, and moreover opposes the least resistance to rapid motion through the air. The straight arm is carefully tapered from the hub to the rim, and is jointed to both surfaces by wide and generous fillets. The objection to the straight arm, which the curved arm is designed to avoid, is the strain of compression in the arms and of tension in the rim, which results when the larger mass in the rim cooling after the other parts have become solid contracts in such cooling. The straight arm cannot yield, but the curved arm allows a slight bending and relieves the rim from strain (Fig. 233). Skilled designers and careful handling in the foundry will diminish and almost eliminate these difficulties, so that the straight arm with carefully proportioned masses will be found characteristic of nearly all modern work in small sizes and is more workmanlike and pleasing to the eye.

The difficulty connected with the shipment of heavy cast-iron wheels in one piece, and the considerable extent of the contraction in cooling in large diameters, results in the practice of making the wheels in two halves. This is further a convenience in erecting the engine. The hub is divided, and each half receives an external flange construction so that the hub bolts together over the shaft, and the rim is similarly cut and flanged on its inside so that each half can be strongly bolted to the other (Figs. 553 and 555). The plane of these joints at hub and rim is usually different, so that the bolt-strains may not be entirely axial in both sets of bolts at once, and to diminish the difficulty from the tendency to fly into two halves by centrifugal force.

For cast-iron wheels of still larger diameter and heavier weight a segmental construction is usual both for convenience of shipment, handling, erection, and avoidance of shrinkage-strain (Fig. 554). The simple fly-wheel which does not have to be used as a band-wheel, and has a rim somewhat rectangular in section, will have an arm and a segment of the rim cast in one piece. The rim-segment will have a length of one-

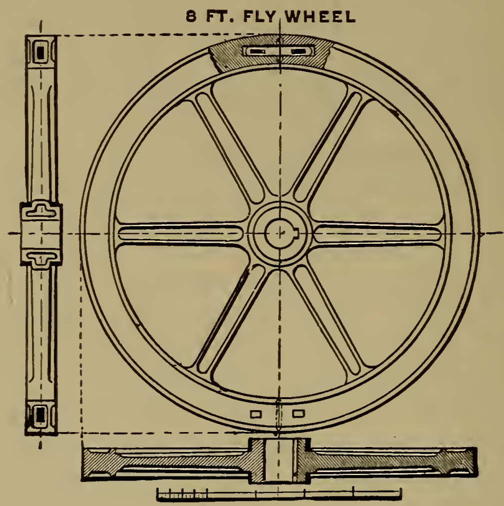


FIG. 553.

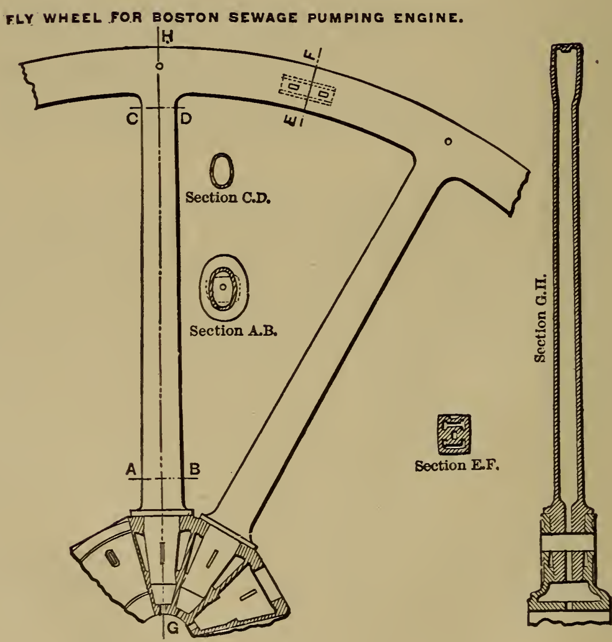


FIG. 554.

half the distance on each side of the arm necessary to reach the adjoining arm on each side, so as to have an appearance somewhat like a T with a circular cross-piece. The inner end of these arms is inserted in the proper sockets in a massive hub, to which they are secured by keys (Fig. 554). The rim-segments are joined together by careful fitting upon radial planes, and the rim made continuous by a joint which appears in several forms. A piece of wrought iron may be inserted into a recess in the interior of the rim, and taper keys or carefully fitted bolts driven through the rim keep this wrought iron a prisoner. A modification of this is to have two or four such prisoners let into recesses

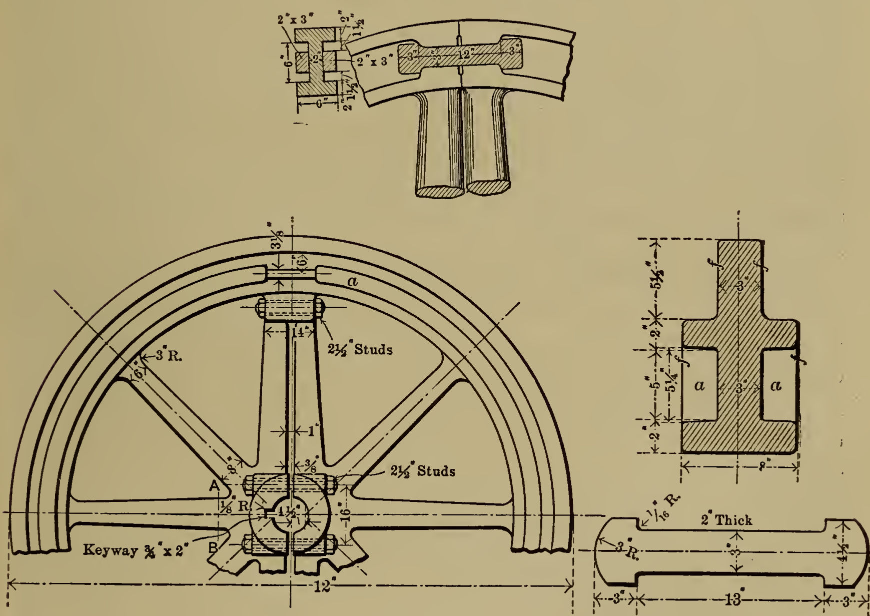


Fig. 555.

on the sides of the rim if there be but two, and into the inner and outer faces also if there be four. Even better than this is the use of wrought-iron prisoners which are inserted when red-hot into recesses in the rim, so that their contraction on cooling shall draw the joint together with a force which is measured by their cross-section. These prisoners may be of sections of an I, or they may be of the shape of an oval link. The recess which they fit enables them to be hammered solid while hot, and their projection or hold upon bosses in the recess forms the joint. Fig. 553 shows a two-part fly-wheel with interior prisoner, and Fig. 554 a segmental wheel. Fig. 555 shows an interesting form of rim-joint and its details, in which prisoners are used of I-section, carefully located in the axis of the stress, and drawing the two halves together at a point where the section is no less than at all the other points of the circumference. Stud-bolts through the arms act also to reinforce and stiffen the junction. The ribbed form of rim avoids some process weaknesses, but is of course unavailable for belt use. Any cantilever or fulcrum action at the joint is impossible when the two parts come together over the abutment at the end of the arms.

407. Fly-Band-Wheels. When the fly-wheel is to serve as a belt-wheel or in rope-driving, the rim requires it to be wide rather than deep radially. Such wheels moreover will usually be of large diameter, since the linear speed of the flexible material used in driving should be high. Such band-wheels can be made either by the segmental method shown in Fig. 554, or (which is perhaps more usual) the joint between the segments will be made at the ends of the arms. The arms will be cast solid with the hub and form a spider, and each arm will end in a sort of pad, which will form the bearing-surface for the bolts which unite the ring-segments to the arms and to each other. Such band-wheels have no initial strains from cooling.

408. Steel Fly-Wheels. The increased realization of the advantages of high speed (paragraf 280), and the desire to get increased power from an old engine or from a given volume occupied, have resulted in the design of wheels of steel to get the increased strength. These are either made all of steel plate, or of steel plate in the rims with a spider of cast-iron arms. The steel-plate wheel has a web or disk of plate, either flat if single, or dished if two plates are used. Where the rim is to be built up, steel-plate segments are laid on and riveted up to form a ring symmetrical with the web, the successive segments breaking joints around the circumference. The rivets are countersunk, and when the rim is thus built up the whole is turned up true. With the cast-iron spider construction, the rim is similarly built up, the arms having pads at their ends which become built in as the rim is formed. Steel is also used in composite wheels.

409. Composite Band-Wheels. Where great width of face is required, more than one set of arms becomes necessary to prevent a side flexure from unequal tension at different parts of the drum. The unnecessary weight of rim caused by the necessity for width if cast iron is used as material for the wheel has resulted in the construction of many wheels recently in which cast iron is either abandoned or used only incidentally. One design makes use of additional wrought iron or steel spokes to withstand tension; next, the use of wood built up in segments for the rim with a cast-iron hub and arms; and last of all, the use of steel plate.

This latter is either used flat or dished as a central web, and the rim is built up of the necessary number of plates, laid edgewise if the wheel requires no face, and laid tangentially if a wide face is required. Mass and strength have been gotten for the rim by the use of iron or steel wire wound around a cast-iron or other rim with sufficient tension to withstand centrifugal force and supply the mass desired.

410. Conclusion and General. — Marine engines require no fly-wheels, or rather the water-wheel and propeller serve this purpose. The locomotive requires no fly-wheel, since the driving-wheels and the living force of the engine and train serve this purpose. For rough work in furnaces, rolling-mills, and elsewhere, with quartering cranks a fly-wheel is often dispensed with.

Most fly-wheels of large engines have notches formed in their face to make convenient places in which a bar can be inserted in order to pry them over the centers if they should be caught there. Marine engines have usually special attachments of screw and worm-wheel driven by a small donkey-engine for turning them over in port for purposes of inspection and repairs. Large engines have sometimes a special engine to start them turning slowly from rest. Such engines have a small pinion meshing into teeth upon the fly-wheel, often internal to its rim. As these engines perform the same function as the bar in the notches above, they have been called barring-engines. As soon as the main engine receives its power and would turn the barring-engine at high speed, the governor on the latter throws the gears out of mesh, and stops the small engine.

Geared fly-wheels revolving faster than the engine-shaft have been proposed and used. When driven by belts they offer the advantage of compactness, and where the driven machinery turns faster than the engine they can apply their regulating effect more directly. They have been proposed as means of storing energy in central stations and upon railways with very steep gradients. The difficulties are those due to their friction even with roller-bearings, and the relatively small amount of energy which they will store. With steam-turbines the rotor and the generator supply all fly-wheel energy required.

CHAPTER XXV.

VALVES AND VALVE-GEARING.

415. The Valve-Gear and the Governor. The reciprocating engine has to have a device to admit the pressure of steam from the boiler upon the proper face of the piston and to release this pressure from the cylinder when its work has been done. This device will be some form of valve or valves, moving within the steam-tight steam-chest which will form part of the cylinder. To drive this valve or these valves, and to time their action relatively to the piston traverse, will require a mechanism, to which the name "valve-gear" is given. The valve-gear has also a most important function in regulating the weight or volume of steam which shall enter the cylinder from the boiler at each stroke, so as to admit only so much as is required to do the work of that stroke, without permitting the speed to fall off by admitting too little, or the engine to speed up by admitting too much. The automatic control of this latter function will rest in the governor, which therefore becomes closely related to the valve-gear. The difference of the function of the governor from that of the fly-wheel has already been discussed (paragraf 403); the governor will not drive the valve-gear as a rule, but will control the motion imparted to it from the engine-shaft.

The subject naturally divides itself into three parts:

The valve.

The gear for driving and timing the motion of the valve.

The gear for regulating the quantity of steam admitted.

The governor for controlling the valve-gear.

416. Engine-Valves. General. Lifting, or Poppet Valves. The opening which is alternately opened and closed by a valve is called its "port." The contact-surface which is so machined as to be tight is called the "face" on the valve and the "seat" upon the stationary part of the casting.

A valve may open and close its port by lifting from off its seat, or by sliding to one side so as to leave the passage free. The sliding-valve requires to be lubricated; the lifting-valve does not. Hence at high temperatures, as in gas-engines and with highly superheated steam, the lifting-valve offers many advantages.

The lifting-valve is often called the "poppet"-valve, — either because it appears to "pop up" as it opens, or from a fancied analogy to the motion of the figures in a puppet-show. For small areas, it will be a simple round disk upon the end of its rod or stem, the face being either flat or conical (Fig. 560). In these forms they are often called mushroom-valves. A simple calculation for lift in terms of port diameter shows the valve should lift one-quarter the diameter of its port in the flat-face type to open full area of the port.* Such valves rarely lift as high as this on account of noise in closing and of leakage during the time of closure.

FIG. 560.

The effective area of the port should be so chosen that the velocity through the opening should not reduce the pressure on the lower pressure side of it by the process called "wire-drawing" with attendant loss. This will be secured if the velocity is less than 100 linear feet per second † (paragraf 236) or 6000 feet per minute.

For large port areas, the balanced or double-seated poppet-valve will be used (Fig. 561) by reason of the lessened power required to lift it from its seat and its quieter closure. The cut shows the steam inlet and exhaust outlet valves for a vertical river-boat beam-engine. The cylinder connects to the space between the two disks as shown at the left hand, and the partition at the center of the right-hand cut separates the pressure and the vacuum spaces from each other. The steam-pressure on the exhaust or vacuum side is therefore from within upwards on the upper disk and downwards on the lower; while on the boiler-pressure side it is upwards upon the lower and downwards upon the upper. It will be apparent, therefore, that valves so constructed can be made either perfectly balanced, underbalanced, or overbalanced, according to the area and the direction of the pressure. They are most frequently slightly underbalanced in river-boat engines, where they are much used, because it is convenient to construct the valve-seat of the upper valve large enough so that the lower valve will pass through it. This means that the small base of the cone in the upper seat shall be just larger than the large base of the disk which

* Let d = diameter of the port, and l the lift. Then to make the surface of the cylinder of height l and diameter d equal the area of diameter d the equality must exist:

$$\pi d \times l = \frac{\pi d^2}{4} \therefore l = \frac{d}{4}.$$

† The volume $V = 2nv$ filled per minute when the engine turns n times per minute, filling the volume v per stroke, when divided by the port area, gives the linear velocity; since volume = area × length.

closes the opening in the lower seat. Balanced lifting-valves of this class open a comparatively large area for the passage of steam, and have no friction except that of the stems which pass out of the steam-chest through the stuffing-box.

The objections to them, as applied to engines in which the valve-stems are parallel to the piston-rod as in vertical engines, are the excessive clearance volume which is entailed, and the difficulty of making them

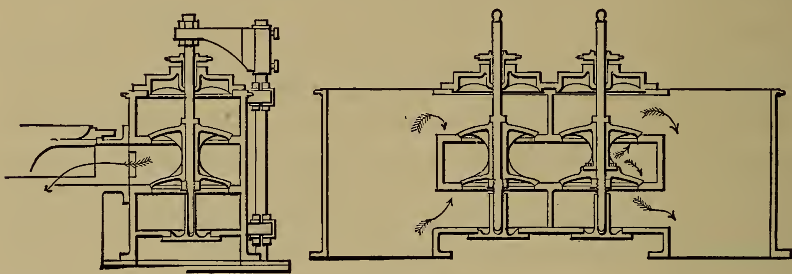


FIG. 561.

so that they will not leak. In horizontal engines, where the poppet-valves can lift at right angles to the diameter of the cylinder, the loss from clearance need not be so great (Fig. 610).

Poppet-valves are most usually operated by cams either of the revolving type (Fig. 608) as in the gas-engine, or of the rocking or oscillating type in river-boat practice (Fig. 389) or in blowing-engines (Fig. 361). The slow rotative speed in the latter classes favors the cam-gear (paragraf 451).

Poppet-valves are subject to erosion on their contact-faces from the passage of steam or gas over these areas; they suffer from deformation from heat and pressure, and from the impact effect of rapid and hard closure.

Furthermore, in the simple or mushroom type they control only one port area; or if two disks are fitted for balance, they must both operate in one chest or perform one function. If one valve-stem is to carry two disks to perform differing functions they will be in separate valve-chests, and expansion or contraction of the stem will affect the tightness of the fit upon the seats. Hence poppet or lifting valves are usual only in valve-gear of multiple valves (paragraf 418), with attendant expense for the gear. When lubrication can be effective, the sliding-valve is the cheaper to make and maintain.

417. Engine-Valves. Sliding, Rocking, or Revolving Valves. The other type of valve opens its port without lifting off the seat, but by sliding upon it across the opening and exposing it. The pressure is usually downward upon such sliding-valve, pressing it to its seat, so that it has to be moved under such pressure. The contact-faces, however, are not exposed to abrasive wear from the steam, but the faces of valve and seat wear each other mechanically if not kept apart by a film of lubricant.

Sliding-valves may be plane blocks sliding upon a plane seat; or the seat may be a segment of a cylinder, or a cone surface, upon which slides a cylinder or a cone which fits the seat steam-tight. Such cylinders or cones may be complete or partial circumferentially; they may expose the port by a rocking or reciprocating motion, or the valve may revolve continuously in one direction. The rocking or alternating motion is the usual one. The plane surface with a plane block reciprocating on it may be called a surface of a hollow cylinder of infinite radius, on which rocks a cylinder also of infinite radius, thus making the rocking motion the general case. This is the common plug-cock when used in taper or conical-frustum form, rocked back and forth by a lever which revolves it part way around its axis. It is a very early form, and is still used in its old form in some small engines for the sake of cheapness, and in a modified form in some large and elaborate designs. (Fig. 499.) The objections to the old plug-cock valve are:

1. The valve and its casing cannot usually be cylindrical, because unequal expansion by heat is apt to cause the casing to seize the plug with a firmness which will cause valve-rods and pins to buckle and shear before the valve will turn. This occurs because the casing is exposed to radiation to the outer air and protects the plug from its action. The plug will therefore be hotter than the casing, and if fitted snugly when cold they will seize together when hot.

2. If fitted loosely enough not to seize, cylindrical plug-valves will leak. To prevent this they are usually made slightly taper, so that just the necessary friction and tightness may be secured by adjusting the plug lengthwise in its conical seat or casing. These taper fits are not so easy to make perfect except with special machinery. Even then in large sizes the large end expands more than the small, and for a given angular motion the large end slides over a greater space than the small. This tends to produce unequal wear and leakage. The taper plug can be refitted to its seat when worn, which cannot be done with a cylindrical fit. The taper is usually one in sixty-four or thirty-two.

The vibrating-valve can be made, however, into a very satisfactory arrangement by either making the plug proper a shell which is independent of the axis of the cylinder in which it turns — this is the Corliss valve — or else by cutting away all of the plug except just the surface required to close the opening through the casing which the valve must control. This is a feature of the Wheelock valve and of many derivatives and modifications of the Corliss (see Figs. 562 and 563). They present the advantages which belong to an arrangement which opens a wide area by a comparatively small motion of the valve.

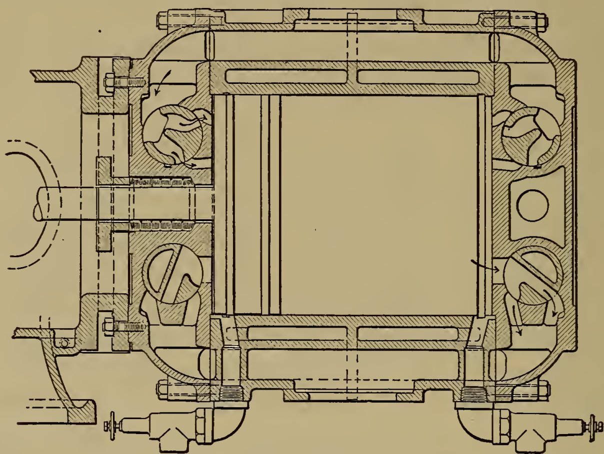


Fig. 562.

It will be obvious that these plug or rocking valves can be used either for simple or single functions; or they may be made to discharge several,

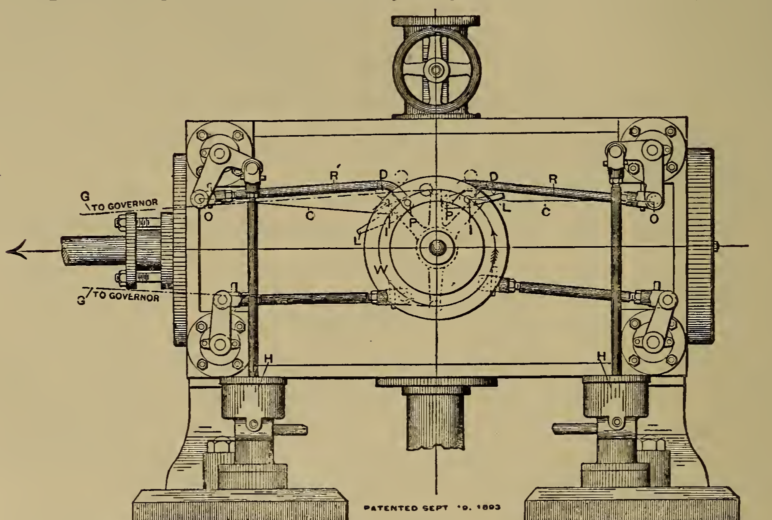


Fig. 563.

since separate ports may be made in the casing coming from different directions, and leading in others.

The great advantage of the flat slide-valve and seat is the ease and certainty with which plane surfaces can be made in practice, and that when the two surfaces are true and of homogeneous material the sliding of the valve upon its seat tends to wear the contact always closer and diminish leakage. It is this practical consideration which has had much to do with the abundant distribution of the flat sliding-valve. Figs. 568 and 616 show cross-sections of such valves.

418. The Timing of the Valve. One, Two, Three, and Four-Valve Systems. The elementary function of the valve and its operating gear is to time the opening and closure of the ports as respects the piston motion. The pressure must be admitted to the working face of the piston from the boiler the moment it is ready to move from its either dead-center; the exhaust must be opened at the other end of the cylinder at the same moment to let the used steam escape. The exhaust must keep open throughout the stroke to let the steam escape freely, but must close at such point in the piston traverse as shall entrap and compress the steam desired to produce a cushioning effect (paragraf 262) and bring the reciprocating parts to rest and help absorb their inertia without inconvenient stress upon the crank-pin. These three functions will be comparatively constant, whatever the load. The fourth function will be to time the cut-off or arrest of admission so as to vary the degree of expansion (paragraf 298) in accordance with best economy and the load to be borne for that stroke. This will be essentially a function timed variably in the piston traverse. It will be noted therefore in detail from the succeeding paragrafs that the timing function of the valve-gear relatively to the piston travel will be to secure at both ends:

1. Constant steam-lead; that is, constant admission at the dead-center or before the piston is at its dead-center.
2. Constant release of exhaust.
3. Constant compression of the exhaust.
4. Variable cut-off of admission (or constant in the simplest case).

In vertical engines, it may be desirable not to have these four functions the same at both ends of the cylinder, but usually they will be so.

The simplest case will therefore be that in which the admission is also constant. This could be realized by four plug-cocks, one at each corner of the cylinder as it were, or two for each end, as in the Corliss or four-valve design of Fig. 562. The upper pair at the two ends lead boiler-steam in alternately; the lower pair lead exhaust-steam out. They act in diagonal pairs; that is, the upper at one end is open when the lower is open at the other side of the piston. In Fig. 562 these four valves are made double-ported so as to open a large area for a small rocking motion, and are in the heads of the cylinder. In Fig. 563 the admission-valves are double-ported and the exhaust are single; the valves are in the cylinder-casting itself, and not in the heads. This construction, whatever the driving-gear, lends itself easily to all four functions, and particularly to constancy of the first three and variability of the fourth if desired. The gear must be different in the last case.

The constancy of the first three, and particularly of Nos. 2 and 3, can be secured by keeping the single-function cock for the steam-inlet, or by using lifting-poppets for admission, while using a three-way valve for the exhaust. Such valve is illustrated in Figs. 564, 565, only for this use the outlet marked to boiler should be marked to Head End of Cylinder, and that to exhaust marked to Crank End of Cylinder, while the outlet to cylinder should be changed to Exhaust Pipe. Such

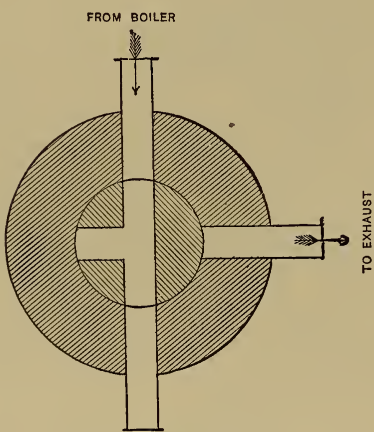


Fig. 564.

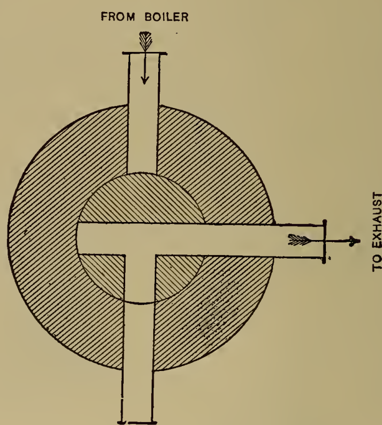


Fig. 565.

engines are called Three-valve Engines. The steam-valve closure can be varied, while the exhaust functions are untouched. The third class of two-valve engines would be those in which there was a valve for each end of a double-acting engine, having the construction and operation of the three-way cock as it is presented in Fig. 564 for the inlet or admission functions Nos. 1 and 4, and for the exhaust functions Nos. 2 and 3 in Fig. 565. One of these only is required for a single-acting engine. But the important consequence must be noted, that the function of Nos. 1 and 4 has been so tied to Nos. 2 and 3 that to vary No. 1 or No. 4 is also to vary Nos. 3 and 4.

If now instead of the three-way cock, the four-way design is used as in Figs. 566 and 567, it will appear that one valve only will do all that is required. Fig. 566 shows steam entering to the head end while exhaust is leaving from the crank end. But the same inflexibility attaches as to the previous case, since the same valve does both the admission and exhaust opening and closure. The simplicity and cheapness of the single-valve construction have given it its wide adoption as a system, in spite of its theoretical drawbacks. Two-valve types will appear in later paragraphs as solutions to secure the greater economy of variable cut-off with varying loads. The wide distribution of the single-valve engine has made it the simplest and typical form to begin with. In the form of the development of the cylinder or cone into a surface or valve-seat which is plane, with a flat block sliding on it, the boiler inlet is

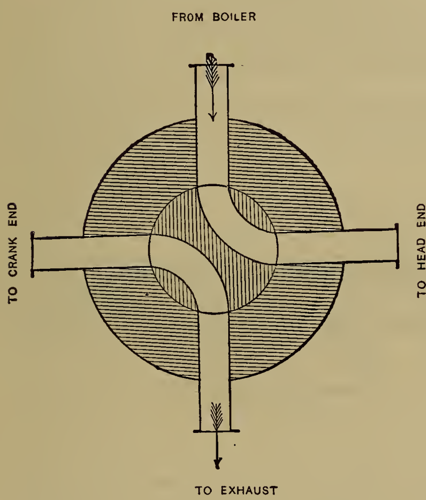


Fig. 566.

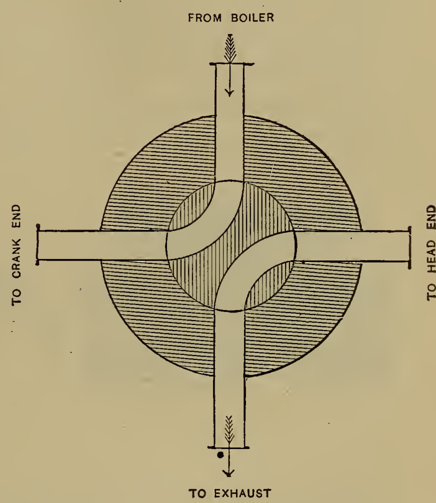


Fig. 567.

the steam-pipe into the valve-chest. Opposite to it and below the valve is the outlet ending in the exhaust-pipe. The two ports leading to the two ends of the cylinder are symmetrically disposed on each side of the central exhaust-port, as in Figs. 568, 569. What dimensions shall be given to these parts?

419. Plain Slide-Valve, Taking Steam Full Stroke. The simple case forms the starting-point, where all functions are constant, and are made to last throughout the full stroke. That is, the engine is to use its steam non-expansively (paragraf 296). There will be no interval when the cylinder is not receiving steam except at the dead-points.

As the valve stands in Fig. 569, its length from out to out horizontally is just the length from out to out of the steam-ports. Both ports are closed, but the motion of the valve in either direction will cause steam from the boiler to pass into one or the other of the ports to reach the end of the cylinder and drive the piston. It would appear then that the position of the valve shown in Fig. 569 is that which belongs to the two dead-centers of the piston.

It will be further seen that the hollow in the under side of the valve has a net length the same as the length between the inner edges of the steam-ports. Hence when the valve moves in either direction so as to admit steam by its outer edge to either steam-port, by that same motion the port at the other end is opened by the inner edge to allow steam to escape from the other end of the cylinder into the hollow of the valve which is always in connection with the exhaust-port and pipe. The distance between the edges of the steam-ports out to out is immaterial within limits, since the only effect of separating these outer edges is to lengthen the valve. To do so, however, is to increase the area upon which pressure of steam acts to press the valve to its seat, and hence to increase the force necessary to slide the valve. The width of the ports

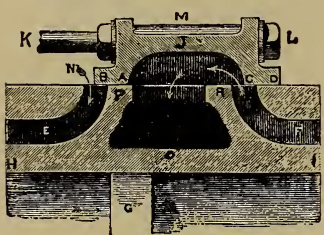


Fig. 568.

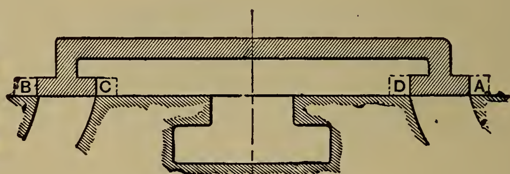


Fig. 569.

is fixed by the area which they must have in order to pass the steam which the engine requires per stroke without imposing an excessive linear velocity for that steam. The length of the port in the direction perpendicular to the plane of the paper is conditioned by the diameter of the cylinder, which of course it cannot exceed. It can at best be equal to that diameter, but it is more usual to make it somewhat less. With the length thus fixed the area of the port should be such that the linear velocity of the steam through the port should not exceed 100–150 feet per second, or 6000–8000 feet per minute.

It is of advantage not to make the port too wide in the direction of the motion of the valve to the right or left, since it will be apparent that the motion of the valve from this central position to the right or left should be equal to the width of the port in order to open it wide. In other words, the throw of the valve and the port width should be the same under the conditions now being considered. If the valve-throw from its central position is greater than the port width, an unnecessary force is expended to slide the valve. If the port width is not uncovered by the throw of the valve, an unnecessary surface is exposed below the valve to the steam on its way from the valve to the piston, causing losses by radiation, by contact, by condensation, and by unnecessary clearance-volume, which the steam fills to no purpose. The throw of the valve is the distance which it moves from its central position in each direction. The travel is the distance which it moves from its extreme position at either side to the other extreme position, and is therefore twice the throw. If the valve is operated by a crank or a modification of it, the radius of the crank will be equal to the throw, and also equal to the port-opening. It is susceptible to demonstration that the volume of the cylinder and the area of the port-opening increase according to the same law when the motion of each is controlled by a crank;* but the exceeding convenience of the crank induced its adoption before this theoretical peculiarity had been elaborated, and the discussions of paragraf 258 have shown that it is a very effective device for transforming rotary and reciprocating motion.

420. The Eccentric is a Crank. The throw of most engine-valves will be comparatively small as compared with the stroke of the piston; and in engines of considerable size, when the shaft is of a large diameter, it becomes inconvenient and impossible to cut away the large shaft so as to get the small crank in the middle of it. It is furthermore inconvenient to drive the valve in most engines from a crank at the end of the shaft. It does not affect the peculiarity of a crank to enlarge its pin. So that if the typical crank AB shown in heavy lines in Fig. 570 have its crank-

* By reference to Fig. 580 it will be apparent that if the angular velocity of the crank-pin be ω, the space described by it will be $R\omega$, and the velocity of a horizontal piston will be $R\omega \cos \omega$. If the area of the piston be A, the volume to be filled in a time dt will be

$$AR\omega \cos \omega \, dt. \quad \ldots \ldots \ldots \ldots \quad (1)$$

If x denote the linear velocity of flow of steam through the port in the valve-seat whose length is l, and r be the radius of the valve-crank, then during the same time dt the motion from its central position for the same angular velocity ω will be $r \cos \omega \, dt$, opening an area $lr \cos \omega \, dt$ through which a volume of steam passes equal to

$$xlr \cos \omega \, dt \quad \ldots \ldots \ldots \ldots \quad (2)$$

Since these should be equal to each other, by equating (1) and (2) the value of x becomes

$$x = \frac{AR}{lr}(\omega), \quad \ldots \ldots \ldots \ldots \quad (3)$$

which is a quantity involving the crank-pin or rotative speed only, which is common to both cranks, and the arbitrary constants which fix the linear velocity of the steam. In other words, the steam enters at all angles with the velocity determined by cylinder-volume and port-area, and not by variation in relation of the crank-angles.

pin successively enlarged until the diameter of the latter becomes so great that the circle representing it surrounds the shaft which is the center of motion of the crank, there will be no difference produced in the motion of such a pin as it revolves around the original center of motion. The enlargement of the pin has produced an *eccentric,* in which the distance between the center of motion and the center of figure is the radius of the original crank. The valve driven by an eccentric is therefore driven by a crank (see paragraf 401), and the mechanism has the further advantage that the direction of the center-line of the crank can very easily be changed with respect to the engine-crank, should it be desirable to alter and adjust the angular relation between these two. When valves are not driven by a crank or eccentric, it will be found that the motion will be given either by cams or by such a combination of rods or links as to constitute a link-motion. These methods of driving valves will be discussed in proper course. Fig. 233 shows the valve driven by a crank from the end of the shaft, but in the majority of engines where the valve is driven directly it will be found that the eccentric is used.

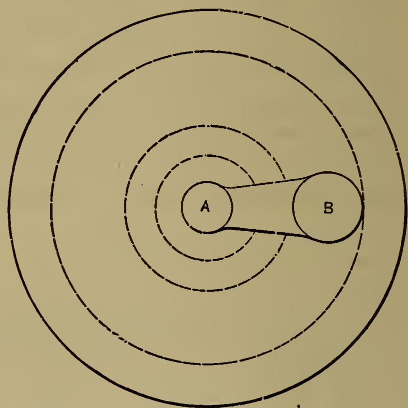


FIG. 570.

421. Setting of a Plane Slide-Valve working Non-expansively. From an inspection of Fig. 568 it will be observed that the valve has the shape somewhat like a letter D, resting with its straight side upon the seat. For this reason this valve has been called the D slide-valve (the German name is Muschelschieber, or shell-slider). It will be observed that if the piston is at its dead-center at the right of the page in Fig. 569, the valve should move towards the left to admit steam to drive it. If the piston is at the left of the page, the valve should move towards the right. It has been further shown that when the engine-crank is at one of its dead-centers, and in a horizontal line in a horizontal engine, the valve is in its central position with its crank therefore standing vertically up and down. The fair conclusion then is that in an engine working non-expansively the valve-crank is 90 degrees distant from the engine-crank. Is it to be 90° ahead or behind?

When the engine throws over (paragraf 289) and the piston is on its dead center at the right of the page, it is obvious that the valve has been at its right-hand end and has returned to its central position from the right to reach the position shown in Fig. 569. This follows because it has been admitting steam for the stroke of the piston from left to right, and has closed the port at the left at the end of the stroke by coming from the right. It is therefore to admit steam to the right-hand port by moving towards the left, which it can only do when the crank driving the valve is standing vertically upwards. It is assumed that the engine-shaft is at the left as the observer faces Fig. 569, and that the rotation is contrary to the hands of a clock. It is further assumed that the length of the rod connecting the valve-crank or eccentric to the valve is of exactly the right length, and that the valve is connected to the eccentric without the interposition of a vibrating arm or rock-shaft, which would reverse the motion imparted by the valve-crank. Under these assumptions the valve or crank is to stand with its center-line making an angle of 90 degrees with the engine-crank ahead of the engine-crank in the direction in which the engine is to turn. Hence the directions for setting the valve for a non-expansive engine of this sort are:

1. See that the valve-rods are of the right length so that the valve opens the ports equally at both ends of its throw. This is called making the valve run "square."

2. Set the main crank of the engine on either dead-center. This can be done either by eye, or more exactly by the following process. Turn the engine over until the crank is nearly at its dead-center and scratch a mark on cross-head and guide which shall indicate such position. Take a beam-compass or trammel, such as shown in Fig. 571, and put one point in a center-punch mark made on the rim of the fly-wheel while the other end rests on a similar prick-punch mark on the frame or bed-plate of the engine. Then turn the engine-crank past its center until the mark scratched on cross-head and guides comes to coincidence again, and the trammel in the fixed point on the bed-plate locates by its other end a second point in the fly-wheel rim. It is apparent that the first and second of these points in the rim indicate two angular positions equally distant from the dead-center on opposite sides of it. The point half-way between them on the rim should be the point in which the same trammel standing with one end in the fixed point on the bed-plate should reach when the engine is on its true dead-center.

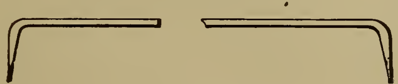


Fig. 571.

3. Slip the eccentric around the shaft in the direction in which the engine is to turn until it is 90 degrees ahead of the engine-crank, if this can be observed. If not, it is reached when the valve is in its central

position, line and line with the edges of the steam-port. Then the eccentric is made fast.

4. Turn the engine through 180 degrees to bring the main crank at its other dead-center to test the accuracy of the adjustment.

If the engine has a rock-shaft which reverses the motion from the eccentric, the eccentric should be set 90 degrees behind the engine, or at a position 180 degrees distant from that which it occupies when the motion is direct.

If the engine throws under instead of over, the eccentric is still 90 degrees ahead of the crank, in the direction in which the engine is to turn, but is 180 degrees distant from the position which it occupies when the engine throws over.

The expansion of the valve-rod by heat must not be overlooked in its effect upon the length of such rods. It will lengthen the rods of a valve directly connected, and either lengthen, shorten, or be without effect upon the rods which are connected to a rock-shaft. If the rod from the eccentric to the rock-shaft and from the rock-shaft to the valve were of the same length and of the same temperature, the effect of expansion would be compensated.

422. The B Valve. In some forms of engines, particularly pumps, it is desirable that the motion of the valve to admit steam should be in the same direction as that which the piston had before it completed its stroke. In the D valve these motions are opposite. The valve to meet this condition must differ slightly from the D valve in its shape, and from the form which it takes it is called the B valve (Fig. 572). It admits steam into the one hollow of the B by sliding past the end of the seat or over the edge of an outer port. The other hollow is by this motion put into communication with the other steam-port and the exhaust, so that its functions are the same as those of two D valves, with the exception that the outer edge of the valve does not act. The details for setting the directly connected B valve are the same as for setting the D valve with rock-shaft.

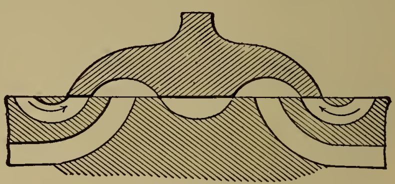


Fig. 572

423. Engine to Use its Steam Expansively at Constant Cut-off. Lap in the Slide-Valve. The first modification of the previous simple case is where the admission of steam is not to be during full stroke, but the steam is to expand at the end of each piston traverse. This must be secured by making therefore a period of piston motion during which no steam enters.

The valve of Fig. 570 cannot be made to do this by reason of its length being only that from out to out of the ports. The valve must be lengthened in order that it may close one port and be still sliding upon its seat while the piston is moving towards the end of its stroke, and so that it shall just reach the position of opening the new port at the other end as the piston comes to rest at the dead-center. This addition of length must be symmetrical on both sides of the center-line, and the increase at each end over the fundamental length of the non-expansive valve is called its lap, *ab* in Fig. 573.

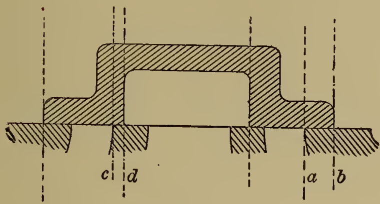


Fig. 573.

Outside or steam lap may be defined as the amount which the valve standing in its central position projects over or laps beyond the outer or steam edge of the port at each end.

424. Effects of the Lap. The effects of the lap are:

1. To accomplish its purpose and compel the steam in the cylinder to work expansively, or to produce the cut-off of admission before the end of the stroke. It will be apparent from Fig. 574, which shows a valve on its seat moving from right to left and distant from its central position a distance equal to the lap on the left-hand end, that it has just cut off admission from the left-hand end of the cylinder, and must move to the left through a distance equal to twice the lap before it can open the right-hand port. It must open this latter port at the instant that the piston is ready to begin its stroke from right to left. Consequently the piston will have moved without admission during a period of angular motion at the end of its stroke corresponding to that required to move the valve through twice the lap. This is an angular motion corresponding to twice the angle whose sine is the lap. This being so, the instructions for setting the slide-valve without lap or without expansive working (paragraf 421) require to be modified. The valve-crank or eccentric is ahead of the main engine-crank not only the 90 degrees there deduced, but is to be ahead 90 degrees plus an angle whose sine is the lap.

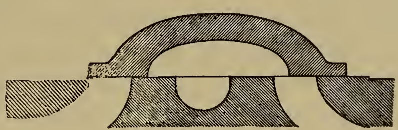


Fig. 574.

2. Hence the second effect of the lap is to set the valve-crank forward and prevent the valve being in its central position when the engine is at its dead-center.

3. The consequence of this second peculiarity is that the opening of the exhaust-hollow in the valve to the two ends of the cylinder becomes displaced, and does not take place as heretofore when the engine passes its centers. The exhaust on the expanding or completed stroke is preopened, because the valve passes its central position before the piston reaches the end of its stroke. It must do this because it has to slide through a space equal to the lap in order to open the valve at dead-center for the ensuing stroke. This has also preclosed the exhaust-port at the right-hand end by the same action and for the same reason. The preclosure of the exhaust is of no great disadvantage within limits, inasmuch as by this action the entrapped exhaust undergoes compression after its outlet is closed and there is produced the cushion which was referred to as desirable in paragraf 262. The preopening on the expanding side, however, is absolute loss, since tension of the driving steam which should have followed the piston clear to the end is released into the exhaust, and is wasted.

425. Inside Lap. When the term lap is used without qualification it means lap added to the outside or steam edges of the valve. In order to prevent pre-release of the expanding steam, from too early opening of the hollow of the valve to the steam-port, the valve-face can be widened towards the inside by adding metal which shall narrow the opening into the hollow. The normal valve has its hollow of the same length as the distance between the inner edges of the steam-ports. When the valve stands in its central position, the distance by which the length of the hollow is less than the distance between the inner edges of the ports amounts to twice the inside lap. Or, in other words, the inside lap is the distance which the valve must move from its central position in either direction in order to open the corresponding end of the cylinder to the exhaust-port (see *cd*, Fig. 573).

426. Effect of Inside Lap. The effects of inside lap are:

1. To prevent pre-release of expanding steam before the stroke is completed.

2. To close the exhaust-outlet from the cylinder before the exhaust-stroke is completed. This produces a compression. The effect of this compression on the practical working of the engine is fourfold.

(1) It serves to produce a spring or cushion of elastic steam which serves to absorb living force in the reciprocating parts and bring the latter to rest by a gradual force exerted to take up lost motion in the joints in the direction in which the next working stroke is to strain them. Without this cushion the living force of the reciprocating parts must be absorbed by the crank-pin, which will produce tension on the joints

just previous to the compression-stroke, and compression of the joints just previous to the tension-stroke. The steam-cushion makes the engine run more quietly (paragraf 262).

(2) This compressed steam after exhaust-closure fills the clearance and port passage with steam otherwise wasted, so that the entering steam when the valve opens does not have to fill such waste room. Generally the compression should be so calculated that the final pressure of the steam compressed into the clearance-volume nearly equals the pressure of the steam coming from the boiler.

(3) The compression caused by inside lap exerts an upward pressure upon the valve which tends to counteract the downward pressure from the boiler, and thus makes the valve move more easily upon its seat for that part of the stroke during which the compression occurs.

(4) The effect of the compression of the exhaust-steam is to raise its temperature and with it the temperature of the cylinder-walls. This heat is due to the absorption by the steam of the work done in compressing it, and consequently the entering steam on the new stroke undergoes less condensation in heating the metal.

Excessive compression due to excessive inside lap or too early closure of the exhaust-port diminishes the power of the engine to the extent represented by the unnecessary work done in compression. This may be enough to lift the valve off its seat, which will be shown by a knock or slam when the valve opens and it comes down into contact with its seat.

427. Exhaust-Clearance, or Negative Exhaust-Lap. It will be apparent that the inside or exhaust lap will make the exhaust sluggish by reason of its tendency to contract the exhaust-passage and produce the effect called wire-drawing. This danger is most to be dreaded in engines of high rotative speed (paragraf 280); and to avoid this difficulty in engines of this class, designers have sometimes lengthened the hollow in the valve, so that when it stands in its central position the distance between the edges of the hollow is *greater* than the distance between the inner edges of the steam-ports. The distance which the hollow lacks to enable it to meet the inner edge of the port is called exhaust clearance or inside clearance or negative exhaust-lap. Its use is restricted to high-speed engines and to those in which the expansion by means of lap is not carried very far. The difficulty would be from the too early release of the expanding steam. Negative exhaust-lap is sometimes called also exhaust-lead. Its effects are to free the exhaust and to diminish back-pressure at the beginning of the return-stroke.

428. Lead in the Slide-Valve. In slow-moving engines the port may open to admit fresh steam just as the piston begins to move. In faster-moving engines, on account of the time taken by the steam to fill the passages and clearance, it is usually better to open the valve a small angle of the crank in advance of the dead-center. This in effect makes the valve lead the piston, so as to bring full boiler-pressure on the piston at the very beginning. The definition of steam-lead, or simply lead, is the amount which the steam-port is open when the piston is at its dead-center ready to begin its stroke. Referring to Figs. 568 and 575, the port-opening between the edge of the valve and the edge of the port is

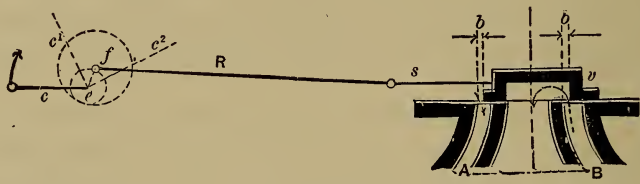


FIG. 575.

the lead. It will be seen at once that the lead is a matter of adjustment merely, while the lap is a matter of construction of the valve. Lead may be varied, but the slide-valve lap cannot.

It will be apparent that the effect of lead on the setting of the valve-crank or eccentric will be to increase still further the angular advance of the valve-crank beyond the 90-degree advance discussed in paragrafs 421, 424. The setting of a valve having both lap and lead requires that, after the valve has been set in its central position with the valve-crank 90 degrees ahead of the main crank, the eccentric is to be set forward in the direction in which the engine is to move through an angle whose sine is the lap plus the desired lead.

429. Effects of Lead. The effects of sliding the eccentric forward in order to give lead at the steam-edges are five:

1. To increase the angular advance and modify the setting adjustment.

2. To increase the expansive working by causing the steam-edge of the valve to close the admission-port by so much earlier as the valve has to move before the piston reaches its dead-center in order to give the determined lead.

3. The effect which these two phenomena have is to increase the distortion of the exhaust period. The pre-release and compression are increased, or the effect of the inside lap is neutralized at one end and increased at the other.

4. The clearance-volume and port-passages are filled with steam

entering the cylinder before the piston reaches its dead-center, so that full boiler-pressure comes on the piston at the very beginning.

5. The living force of the reciprocating parts is arrested by this cushion of live steam from the boiler. The effect is the same as if it were done by the exhaust-cushion, but it is produced by steam which must be paid for instead of by steam which would otherwise be wasted.

430. Setting of Slide-Valve without Access to Valve-Chest. Setting by Sound. The slide-valve of an engine works in a valve-chest. This is a box either cast in one piece with the cylinder and arranged with lids or bonnets by which access can be had to the valve and seat, or else the valve-chest is cast separately and secured to the cylinder-casting by carefully made steam-tight joints, which are kept tight by bolts. The opening of the valve or steam-chest for the purpose of setting the valve is to be prevented when possible, inasmuch as to break a satisfactory steam-joint is a thing which is to be avoided. It is by no means difficult to transfer the motion of the valve to reference-points outside of the valve-chest so as to avoid the necessity for getting into the chest. This is easily done by means of a trammel as presented in Figs. 571 and 576.

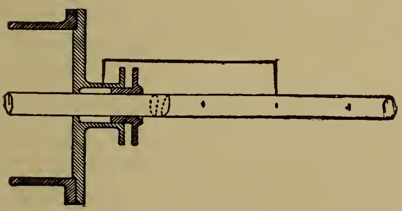


Fig. 576

It will be apparent that if one end of the trammel is placed in a center-punch mark on the valve-chest at such a point as, for example, on the stuffing-box, the other end can be used to fix upon the valve-stem itself a point which shall indicate the position of the valve within. If the engine be turned over by hand so that the valve-stem is made to travel to the right, the trammel can be made to locate a point on the valve-stem in which the outer end of it fits when the valve is all the way over to the right (Fig. 576). If the engine be turned over further, so that the valve-stem slides to the left, the trammel locates a point on the stem which belongs to the extreme of the travel to the left. Half-way between these a point can be marked by a center-punch, and when the point of the trammel lies in that punch-mark, the valve is in its central position. The engine being located with its piston upon its dead-center, the eccentric can be slid around until the valve is in its central position, and with the trammel in place the valve-stem can be moved through a distance, first, to bring the lap line and line with the port edge, and, secondly, the further distance proper for the desired lead. As in paragraf 421, due regard must be had to direction of motion and to the possible reversal of connection by rock-shaft or otherwise.

Another method of setting the valve without taking off the valve-chest lid or bonnet is to depend upon the regular pulsations of the exhaust, as they furnish an indication to the ear. Their regularity in time indicates a symmetrical motion of the valve, and their regularity in intensity or volume indicates admission of steam and expansion symmetrically at the two ends. This is much the most sensitive method in two-cylinder engines with quartering cranks, as in the locomotive.

CHAPTER XXVI.

VALVE-GEARING. DESIGN. SPECIAL FORMS.

435. The Zeuner Polar Diagram for Slide-Valve Design. The amount of port-opening, and the timing of the events in a valve motion which has been designed, can be laid down graphically from the motion of the driving-crank. The distance the valve has moved from its central position is the sine of the angle through which it has swept counted from the 90-degree point in a horizontal engine. (Figs. 577, 578, 579, Appendix).

A more convenient plan has been worked out by the late Professor Zeuner, which not only gives the facts for an existent gear, but can also be used in design to make the valve meet assigned conditions in advance. Modifications have also been made by other investigators.

The basis of the Zeuner analysis is to compute the value for the distance which the valve has moved from its central or symmetrical position corresponding to any crank-angle, or when the main crank and valve-crank have moved through an angle ω. If r is the radius of the valve-crank, l and l_1 the lengths of the valve-rod to the knuckle-joint and the valve-stem proper respectively, then from the diagram (Fig. 580)

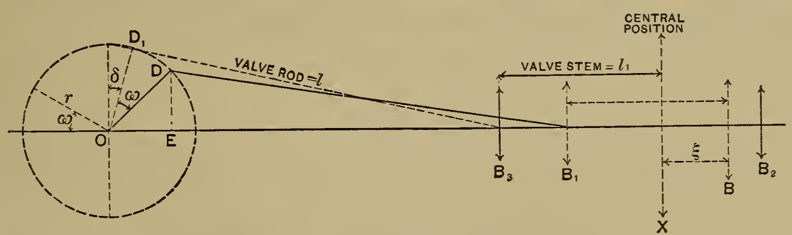


FIG. 580.

in which the line through X fixes the central position of the valve, the length BX may represent the space the valve has gone to the right for the angles δ and ω. If this distance be called ξ, then

$$\xi = BX = OB - OX.$$

To find a value for OB:

$$\begin{aligned} OB &= OE + EB_1 + BB_1 \\ &= OE + BB_1 + \sqrt{DB_1^2 - DE^2} \\ &= r \sin(\omega + \delta) + l_1 + \sqrt{l^2 - r^2 \cos^2(\omega + \delta)}. \end{aligned}$$

But $\left(l-\frac{r^2\cos^2(\omega+\delta)}{2l}\right)^2 = l^2 - r^2\cos^2(\omega+\delta) + \frac{r^4\cos^4(\omega+\delta)}{4l^2}.$

$$\therefore OB = r\sin(\omega+\delta) + l_1 + l - \frac{r^2\cos^2(\omega+\delta)}{2l},$$

when the last term is neglected as being so small a quantity as to be negligible.

Similarly, a value for OX is

$$OX = \frac{OB_3 + OB_2}{2} = l_1 + l - \frac{r^2\cos^2\delta}{2l}.$$

Combining these values and substituting,

$$BX = \xi = r\sin(\omega+\delta) + l_1 + l - \frac{r^2\cos^2(\omega+\delta)}{2l} - \left[l_1 + l - \frac{r^2\cos^2\delta}{2l}\right]$$

$$= r\sin(\omega+\delta) + \frac{r^2}{2l}\left[\cos^2\delta - \cos^2(\omega+\delta)\right]$$

$$= r\sin\delta\cos\omega + r\cos\delta\sin\omega + F.$$

If $$r\sin\delta = A$$

and $$r\cos\delta = B,$$

$$\xi = A\cos\omega + B\sin\omega + F.$$

F is a term to include the motion due to the angularity of the valve-rod. It is a small quantity, because the length l is always great compared to r, and the cosines of the angles are small, and their squares smaller. The quantity F may therefore be dropped for convenience, or treated as a "missing quantity."

The equation for ξ will give also the value of the radius vector if a pole be taken in the circumference of a circle the co-ordinates of whose center (Fig. 581) are $OB = a = \frac{A}{2}$ and $BC = b = \frac{B}{2}$, and whose diameter is r. For if P_2 be any point whose radius vector is OP_2 and the angle $MOP = \omega$, then if

$$OM = \xi\cos\omega,$$

and $$MP = \xi\sin\omega,$$

it can be proved that

$$CN^2 + NP^2 = CP^2 = \left(\frac{r}{2}\right)^2,$$

$$(OM - OB)^2 + (MP - MN)^2 = \left(\frac{r}{2}\right)^2,$$

$$(\xi\cos\omega - a)^2 + (\xi\sin\omega - b)^2 = \left(\frac{r}{2}\right)^2,$$

$$\xi^2\cos^2\omega - 2a\xi\cos\omega + a^2 + \xi^2\sin^2\omega - 2b\xi\sin\omega + b^2 = \left(\frac{r}{2}\right)^2,$$

$$\xi^2(\cos^2\omega + \sin^2\omega) = 2(a\cos\omega + b\sin\omega),$$

whence

$$\xi = 2a\cos\omega + 2b\sin\omega;$$

and if $$2a = A \text{ and } 2b = B,$$

$$\xi = A\cos\omega + B\sin\omega.$$

Or in other words, the motion of the valve from its central position may be represented by, or replaced by, the length of the radius vector of a polar circle whose diameter is the throw of the valve. The radius vector when ω is zero determines the angle P_3OR, because the main crank being at its dead-center the valve should have a radius vector equal to the sum of the lap and the lead.

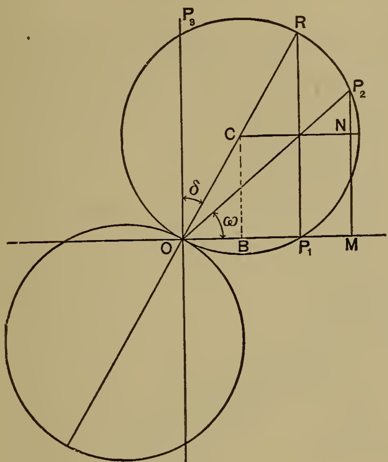


FIG. 581.

436. Graphical Solution by the Zeuner Polar Diagram. The equations of the Zeuner analysis are practically never used, but the solution is always found graphically by drawing the necessary circles full size. In Figs. 582 and 583 the radius ON is the throw of the valve, equal to the eccentricity of the eccentric. If there is no lap, the value of the radius vector is zero at the dead-center. Hence for a horizontal engine, the position of the valve-circle with a diameter equal to the throw will be that of Fig. 582. When there is a lap, then the valve will be a distance from its central position equal to that lap when it is just ready to open

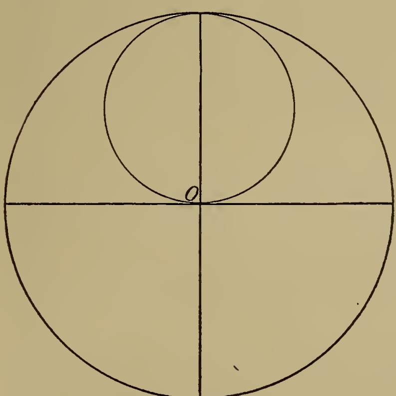


FIG. 582.

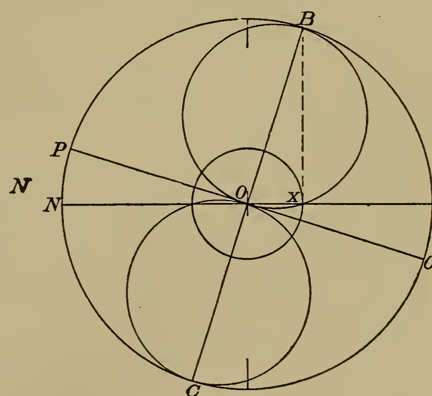


FIG. 583.

the port. The distance OX in Fig. 583 will be the polar radius vector in this case, and it is apparent that the valve-crank should be 90 degrees plus an angle whose sine is the lap ahead of the main engine-crank; hence OX will be the lap and OB the required angular advance. If the engine has lead as well as lap, the valve must be distant from its central position a distance equal to the sum of the lap and the lead. The radius vector at the dead-center must then have a value represented by Oy when xy is the lead (Fig. 584).

In these illustrations the engine-crank is to be assumed as belonging to a horizontal engine on its inner dead-center, and the rotation to be opposite in direction to that of the hands of a clock. The maximum throw is reached when the crank of the engine is in the position OB, and beyond this angle the valve starts to come back and close the admission. The closure of admission will occur when the valve on its return towards its central position is distant from it a length equal to the lap (paragrafs 85 and 86); hence if with O as a center and with Ox as a radius a circle be drawn, it will intersect the circle whose diameter OB equals the throw, and which is called the valve-circle, at points which will indicate the angles at which the valve begins to open and at which it closes. The radius drawn through Z (Fig. 584) gives the crank-angle at which the inlet-valve opens, and a radius drawn through W gives the crank-angle at which admission ceases or the cut-off takes place. It is obvious that in a valve with lead the valve would open before the piston reaches its dead-center.

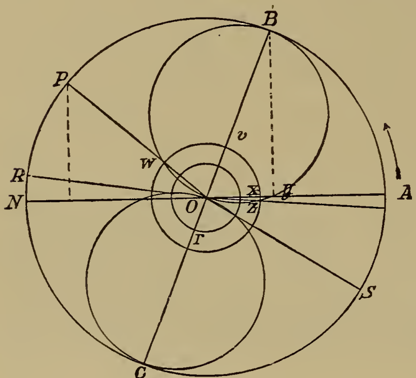


Fig. 584.

A strict adherence to the Zeuner method would have the circle described on OB conceived as attached to the engine-shaft, and the crank when at its dead-center to lie in the position ON. It is so much easier to cause the radius to swing through equal angles in the contrary direction, while the valve-circle remains fixed, that this method is preferred for practical use.

If the diameter BO be produced beyond O to C, and a second circle of equal diameter be drawn upon OC as such diameter, a circle is given whose radius vectors give the exhaust events. If there is no inside lap, the exhaust opens and shuts when the radius vector of this second circle is zero, which is the position when the crank is at OP and OQ (Fig. 587). If there is an inside lap, the port will not open until the valve has moved through that lap. Hence the effective opening will begin only when the radius vector for the secondary circle exceeds the lap. Therefore if with O as a center and inside lap Or as a radius a circle be drawn, its intersections with the secondary circle will give the crank-angle at which the exhaust on the working-stroke and compression on the exhaust-stroke begin.

437. **Use of the Zeuner Polar Diagram.** The Zeuner polar diagram not only gives all information which is given by a motion-curve, but it furthermore enables the user to design the valve and seat to fulfill specified conditions. In Fig. 584 the throw lap and lead are the given data. The angle AOB shows the advance of the eccentric disk beyond 90 degrees proper for such lap and lead, and the cut-off takes place at

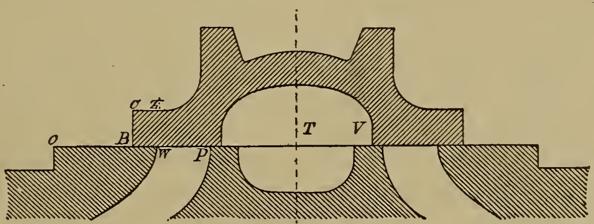


FIG. 585.

an angle AOP from the beginning of the stroke at dead-center. The design of a valve and its seat proper for the conditions assumed will give a drawing such as Fig. 585.

It is desirable that the valve in sliding upon its seat should come to the edge of it in its extreme throw, in order that the wear of the valve and seat may be uniform all over their surfaces of contact. Starting, therefore, at the point O, which marks the extreme edge of the seat, a distance OB is laid off equal to the throw OB (Fig. 584), which is the radius of the valve-crank, and the maximum distance it can throw from its central position. The point B will be the beginning of the valve, since it projects over the port a distance Ox equal to BW and equal to the lap, and consequently at the point Z in Fig. 585 the outer edge of the port should begin. With the assumed throw of the valve and the assumed lap the port can never be opened wider than the distance vB in Fig. 584. Hence the indicated size for the port WP, Fig. 585, is the length vB in Fig. 584. If the port were made larger, the throw chosen would not open it wide, since BP equals OB; and if WP were less than vB, the valve in sliding would go beyond the point P, which is unnecessary. The calculation or design must be checked at this point to ascertain whether the area of the port-opening given by the product of WP multiplied by the permissible length in the direction perpendicular to WP gives an area sufficient or unnecessarily large to admit the quantity of steam required in the cylinder per stroke according to the calculation made in paragraf 236.

A valve having no inside lap will have the inner edge of its working face which is the beginning of the exhaust-hollow line and line with the edge W of the port. If there is an inside lap, represented by Ov in Fig. 584, it will give a projection of the valve-face beyond the port-edge P.

The steam-port must be separated from the exhaust-port by a partition. The amount of metal in this partition is immaterial provided only it is enough to secure stiffness. It is often one-half the port width PW or vB, when there is no reason for making it anything else. The only effect of metal in this bridge or partition is to lengthen the valve and increase the power required to slide it on its seat (paragraf 445).

The inner edge of the exhaust-hollow travels to the right or left a distance equal to OB, or the throw of the valve. It is desirable that when it is moved all the way to the right the hollow face shall still leave between its edge and the point or line V a space TV, equal to or larger than the port PW, discharging into that hollow from the right-hand port. That is, the motion of the valve to T should not constrict or reduce the passage through which the exhaust is escaping any further than it is necessarily reduced by the fixed opening corresponding to PW. This fixes the right-hand edge of the exhaust port in the seat, and the rest of the valve and seat is made symmetrical with the left-hand half already constructed.

438. Valve-gear Problems and Design. It is beyond the present purpose to follow the use of the Zeuner valve-diagram further. It can be made to solve problems covering all quantities when a few assumptions or data are made or given. The point of cut-off is one of the most usual data, as the engineer in most cases desires to work his steam with a certain expansion. The lead is another fundamental assumption. But a variety of combinations is possible involving the throw of the valve, the port-opening for a certain position of the crank, the release and the compression; and for these the student is referred to special treatises.

For the immediate purpose in hand, however, the special problem will be considered in which the throw, point of cut-off, and lead are given and it is required to find the angular advance and lap to fill the condition assumed.

In Fig. 586 let the horizontal line AB represent the whole travel of the valve in a horizontal engine. Bisect the line at O and with OA as radius describe a circle on AB as diameter. Draw the radius OC, representing the angle from B as dead-center at which it is proposed to have the cut-off take place. This is the point of closure of the valve; and since there is to be a lead, the valve should open before the crank reaches the position OB. The amount of angular motion before the crank reaches OB should be that through which the center of the crank should move in subtending an angle measured by that lead. If, therefore, with B as a center and the radius Bl equal to the assumed lead, a circle be drawn, the radius OD will give the position at which the valve crank stands when the port just begins to open. The maximum radius vector will be at a point half-way between these two crank-positions OD and OC, so that if the angle DOC is bisected by any of the usual geometric methods and the position OE thus determined, the

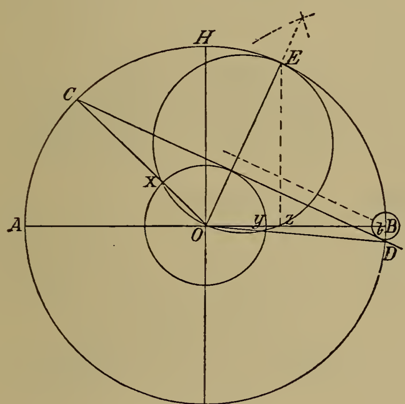


FIG. 586

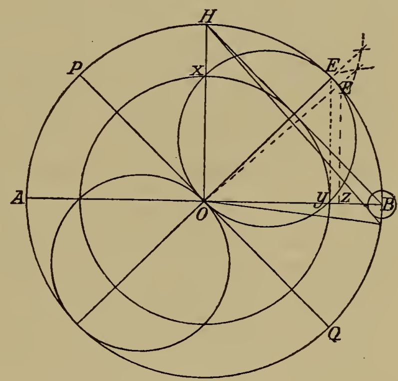


FIG. 587.

angle HOE will be the angular advance ahead of 90 degrees at which the valve-crank should stand. The length OX cut off from the line OC by the valve-circle just drawn is the value of the radius vector, or distance of the valve from its central position, when the steam-edge coming back over the port cuts off admission. OX is therefore the value of the lap, and a circle drawn with O as centre and OX as radius will be the lap-circle for that diagram. It is a matter of simple geometric proof to show by similar triangles that the length yz, which represents the lead in its customary position, is the same as the length Bl used as an expedient in construction.

439. Limitations of the Single Slide-valve. The cheapness, convenience, simplicity, and permanency of the single slide-valve are the great inducements to its use. It can be shown, however, by the method pursued in paragraf 438 that the limit of expansive working with a single valve performing all functions is reached before it is demanded that these be cut off at half-stroke. If the conditions of cut-off at half-stroke be imposed and the method of paragraph 438 be followed, it must be apparent from Fig. 587 that the angle HOE will be a little greater than 45 degrees if there is a lead, and will be 45 degrees if there is none. The drawing of the circle of the lap with a radius OX determined by that valve-circle will give a lap so large in relation to the throw OH that the port-opening becomes absurdly small.

The matter is not materially helped by increasing the throw, because the lap increases with the throw.

Furthermore, if there is no inside lap, the exhaust-opening and closure will take place at angles represented by the lines OP and OQ, which, it will be seen, are 45 degrees from the crank-position which belongs to the ends of the stroke. The exhaust events have thus become distorted so that successful working becomes impossible. Hence with high expansion, secured by the expedient of increasing the angular advance of the valve-crank, the limit is certainly reached before two expansions are secured. A much higher degree of expansion is desired in all fly-wheel engines. How shall it be secured?

440. Valve-gear for High Degrees of Expansion. Two-valve Systems. The advantages of the single valve induce an effort to make use of it if possible. Particularly in such mechanisms as that of the locomotive and the marine engine, where complication is to be avoided, it is desirable to retain the single slide.

(1) The first method is to design the engine or its gear so that, as it is desired to cut off earlier, the throw of the valve should be lessened. It will be apparent that if the throw were equal to the lap or less than it, the valve would not move from its central position far enough to uncover either port. This makes the cut-off before the stroke begins, and is the limit. If a port of extra width or length will give area sufficient to let steam through in sufficient quantities to give the engine the necessary power, the admission will stop earlier and earlier as the throw diminishes; but, the angular advance remaining constant, the exhaust-ports open and close at the required angle for which they were designed. This principle underlies many designs of automatic cut-off engines, in which the governor varies the throw of the valve as the speed varies. It is also one of the underlying features of the link-motion used as a cut-off gear on locomotives.

(2) The second method to secure high expansion is to use two slide-valves. These may work in the same valve-chest or in different valve-chests. When the two valves work in the same valve-chest it is usual to have them operated by separate eccentrics, and to divide the functions of distribution between them. The valve nearest the seat will control the exhaust-port openings entirely, and will be driven by an eccentric

having a comparatively small angular advance. This principal valve will be called the main, or distribution, valve. The second valve will be driven by its own eccentric, set at a considerable angular advance, and will have no exhaust functions. Its business will be solely to cut off admission of steam into the ports or passages through the main valve whereby steam is admitted to the ports in the seat (Fig. 588). This valve will be called the cut-off valve, and from the fact that it slides or rides in the back of the main valve this arrangement is often called the riding cut-off. The main valve requires to be prolonged so as to form the seat and port for the cut-off valve. The inner edge of this port performs cut-off functions late in the stroke, and prevents the cut-off valve from opening by its return motion the port which its greater angular advance has caused it to close. This riding cut-off arrangement lends itself easily to automatic adjustment of cut-off, since the release and compression are provided for at fixed points by the lower or main valve, and admission can be varied within very wide limits without affecting these exhaust functions. It may be a solid block, or in two separate blocks adjustable on their rod, as in the type of Meyer valve selected in Fig. 588.

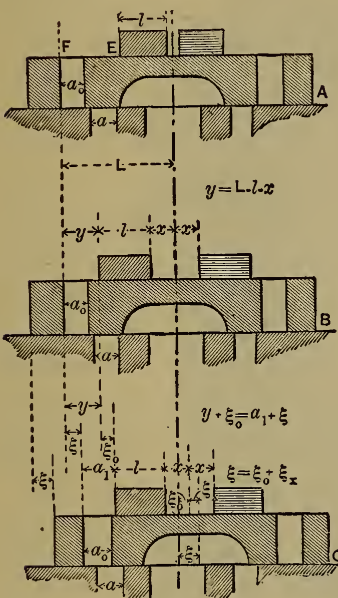


Fig. 588.

A modification of this system has a partition between the two valves, with one or two rectangular openings in it (Fig. 589). Steam from the boiler passes to the main valve below the partition when the cut-off valve uncovers the opening. The objection to this arrangement is that the steam which surrounds the main valve partakes of the expansion after the admission is cut off by the upper valve, and the work of such expanding steam is lost. In the riding cut-off, to diminish this loss as

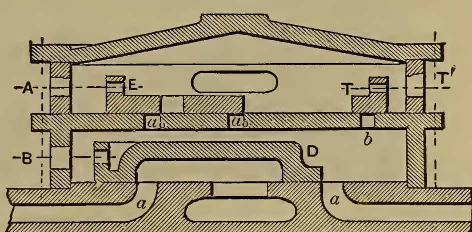


Fig. 589.

far as possible, the thickness or height of the main valve is kept as small as consistent with giving the exhaust-hollow the depth which it requires. The small port *b* is a hand opening, or by-pass, to let steam enter the

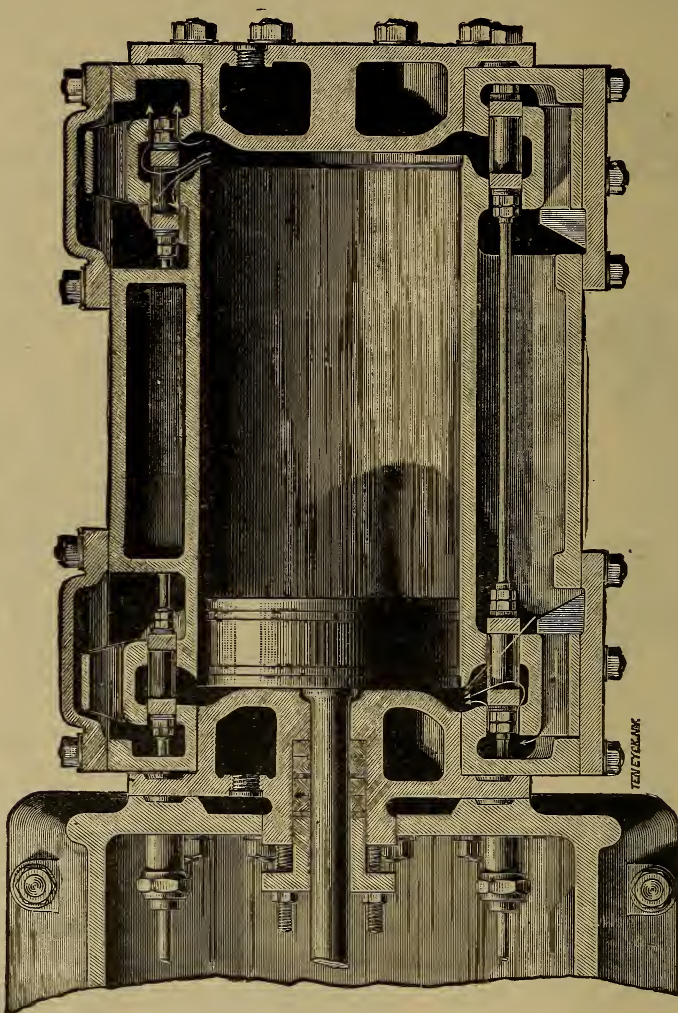


Fig. 590.

main valve-chest if the engine shall have stopped with the upper port closed.

When two valves are used in separate chests, the exhaust functions belong to one and the steam functions to the other. This makes both design and variation of cut-off exceedingly simple. Fig. 590, showing a section of the Porter-Allen engine, presents the features of a valve-gear of this sort in which the two valve-chests are on opposite sides of the cylinder and both are slide-valves. It will be easily seen from paragrafs 436 to 438 that the steam-valve need be planned only for lead, lap, and throw, and the exhaust-valve for throw and lap, when compression and release are fixed.

441. Three and Four-valve Gears. When the principle of one valve has been abandoned it becomes very simple to design valve-gearing with a steam-valve for each end and an exhaust-valve for both ends, making a three-valve engine, or a separate valve for steam and exhaust at each end, making the four-valve engine (paragraf 418). In both the three and the four-valve engine the same advantages are derived, having the compression and release occur at fixed points in the stroke, while the cut-off and expansion can be varied automatically or by hand without interfering with the exhaust functions. The forms taken by the three and four-valve systems present almost every combination of lifting and sliding-valves which can be made. The typical river-boat engine, the older blowing-engines, and the older beam-pumping engines illustrate types of lifting-valves, and the Corliss engine and its imitators illustrate the cylindrical sliding-valve to accomplish these same results.

442. Shortening Steam-passages. In the typical slide-valve which has been discussed hitherto the valve has been considered as short as consistent with adequate area for ports and for the exhaust-hollow. This results in engines of long stroke in a considerable length from the port at the valve-seat to the end of the cylinder. As a rule in this design the valve is in the middle of the length of the cylinder, although this is not necessary if it be more convenient to have one passage longer than the other. The objections to the long passages from the valve-seat are:

1. The friction which they oppose to the passage of the steam. These passages are usually moulded in the cylinder-casting by means of cores of proper shape, and their surfaces will be rough. The effect of this friction is to increase the difference of pressure which exists in the boiler or valve-chest and in the cylinder.

2. The long passage cools the steam by contact with its sides. This cooling is first by ordinary radiation, but more important than this is the cooling which follows when the passage is in communication with the exhaust-port. The lower-pressure steam, carrying perhaps a mist of watery particles, will absorb heat very rapidly during that part of the stroke in which it is serving as an exhaust-passage. The longer the surface the more heat will be required from the entering steam on the next stroke to heat the metal up to the temperature of the entering steam.

As has been heretofore observed, there is no difficulty in lengthening the valve and thereby shortening the passage. The size of the exhaust-hollow and the bridges which separate the ports produce no effect upon the distribution. The only objection is that increased length of valve gives an increased area for pressure, and consequently makes the valve demand more power to move it. In early designs for low-pressure steam, where this matter was of little moment, engines with very long slide-valves will be found. They have been sometimes called Murdoch's valves. With high-pressure steam the difficulty has been met in another way. When the single eccentric is to drive a valve

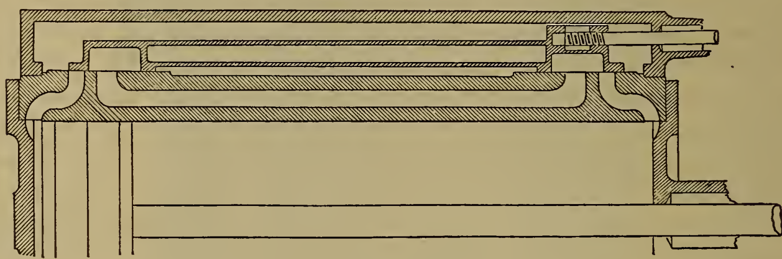


Fig. 591.

performing all valve-functions the usual plan is presented in Fig. 591. It will be observed that while the valve acts as one it is really made in two parts, or like a B valve. The exhaust-port is divided by its special bridge, and the admission of steam is controlled by the left-hand edge of the left half of the valve and by the right edge of the right-hand half. The length of passage is thrown into the exhaust, where it makes no difference, and the length of the steam-port at each end is reduced to the shortest possible line. By joining the two halves by a rod surrounded by the steam in the chest there is no pressure on the valve between the active parts at each end. The system of Fig. 590 illustrates the same short-port idea, and in the Corliss valves of Fig. 562 the minimum port surface is secured. This same result is sometimes attained by making the valve resemble a low and broad letter H in plan. The uprights of the H are the working-parts of the valve, and the cross-bar between is made hollow and fits the exhaust-port, whose length is at right angles to the length of the steam-ports (Fig. 592). Cut-off valves which ride on the back of the main valve can be designed this way.

443. Shortening the Throw of the Valve. Allen Valve. Another expedient for diminishing the power absorbed by an engine in working its own valves has been to shorten the throw or travel of the valve upon its seat. This must be done without constricting the port-area, which is an essential condition. If by keeping the throw of the eccentric larger than the throw of the valve, there is yet opened an equal port-area with such reduced throw, there has been given to the eccentric a mechanical advantage to overcome the pressure which holds the valve at its seat. Fig. 593 shows the Allen slide-valve whose characteristic is the passage or hollow through the shell over the exhaust-port, and the use of a comparatively short seat or a seat with more than three ports in it. From the construction and the proportions it will be seen that when the steam-edge of the valve uncovers the left-hand port by a motion of the valve from its central position, the hollow in the shell is by that same motion brought into communication with the steam-pressure at the other end of the valve. In consequence of this a given motion opens twice as much port for the passage of steam into the cylinder as would be the case in the ordinary valve. Steam-pressure is thus very rapidly established as the valve moves a less distance than the port width which it is made to serve. This double ported principle will be seen also in Figs. 562, 602, 607.

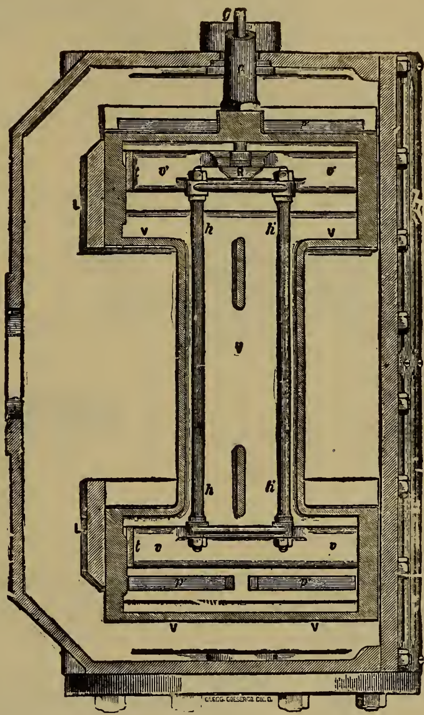


Fig. 592.

444. Gridiron Slide-valve. It will be immediately apparent that if the slide-valve be constructed with alternate holes and solid bars each of one inch in width which match similar holes and bars in the seat, that the motion of one inch which brings the holes in the valve to match the holes in the seat will open an area of port as many times one inch in the direction of motion as there are holes in either valve or seat. Fig. 595 shows a slide-valve of this construction, intended for steam only. It is called from its resemblance a gridiron slide-valve, and the principle of having many openings into the steam-passage gives it also the name of multiported valve. The adoption of this principle will be

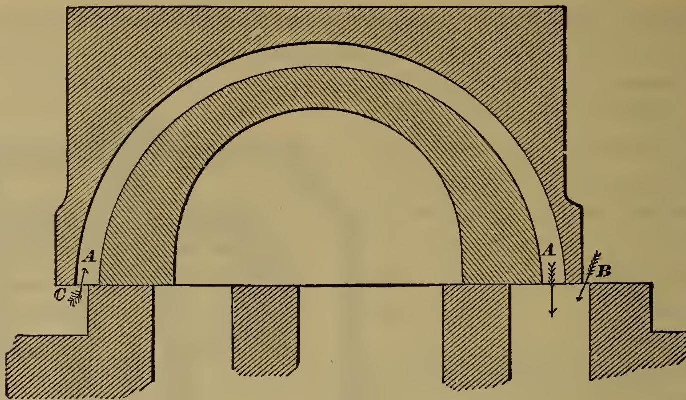


Fig. 593.

Fig. 595.

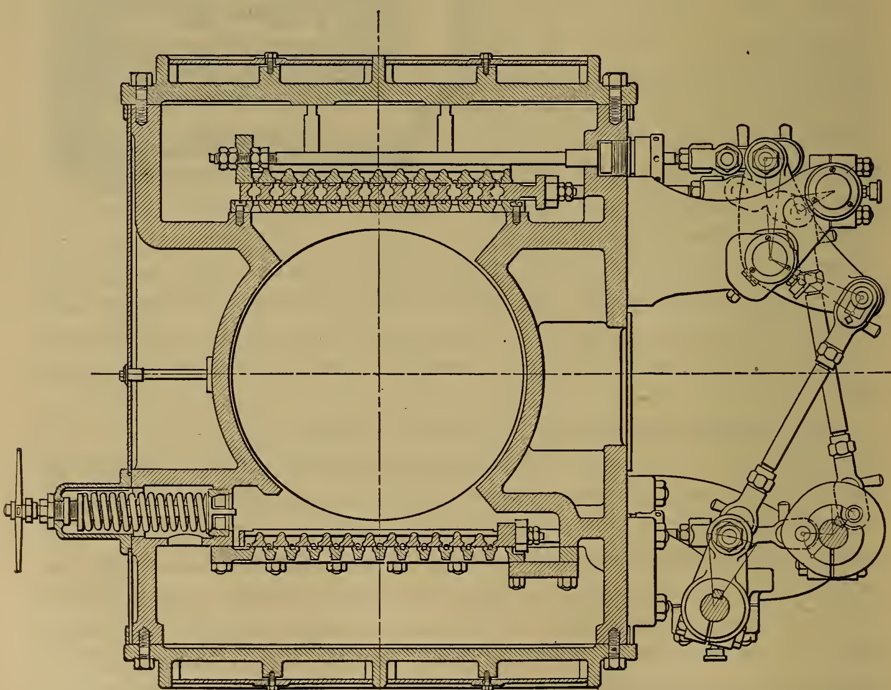


Fig. 596.

observed in many large engines, and is particularly useful for high pressures.

Fig. 596 for example shows the very short-port passage and yet the large area for both steam and exhaust in a four-valve engine in which the gridiron principle is applied. The valves can slide crosswise, due to their short travel, which results in a compact driving mechanism.

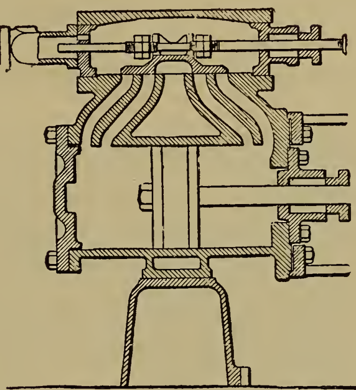


Fig. 597.

The multiported valve-seat can also be made to serve another purpose, which will be understood from Fig. 597. The division in the port is carried all the way down to the bore, so that the piston in its movement towards the head of the cylinder closes the inner port at either side before it reaches the end of the stroke. This inner port has been the principal dependence as an exhaust-port, so that when it is closed by the piston a very energetic compression is the result, and the piston is arrested by a cushion of the exhaust steam. On the steam end the admission is gradually cut off by the closure first of one port and then of the other. This principle of using the piston as a valve to close ports in the bore of the cylinder is the principle underlying several designs of so-called valveless engines. A modification of it is to be noted in Fig. 405.

CHAPTER XXVII.

VALVE GEARING, BALANCED VALVES, CAM AND TRIP VALVE GEAR.

445. Balancing Slide-valves. The power necessary to slide a valve upon its seat is measured by the area of the valve multiplied by the pressure upon that area, and by a factor expressing the coefficient of friction between the valve and its seat. The pressure is the net pressure or the algebraic sum of the downward pressure on the valve and the upward pressure exerted from the cylinder against the under side of the valve. It is not a difficult matter to make this calculation for assumed conditions.

To evaluate the pressures to be balanced on a slide-valve, suppose a locomotive-valve 17 inches wide and $10\frac{1}{2}$ inches long, having a lap of 1 inch, a travel of 4 inches, a port of $14\frac{1}{2}$ by $1\frac{1}{4}$ inches, and an exhaust hollow of $6 \times 14\frac{1}{2}$ inches. Let the valve be working with the lever in the eight-inch notch, or cutting off at one-third stroke. The following areas and pressures will prevail:

Port-area	= 18 square inches	(1)
Exhaust-hollow	= 87 square inches	(2)
Gauge-pressure	= 160 lbs. per sq. inch	(3)
Back-pressure	= 5 lbs, per sq. inch	(4)
Mean pressure	= 110 lbs. per sq. inch	(5)
Cushion-pressure	= 40 lbs. per sq. inch	(6)

The downward pressure during admission, or one-third stroke, is exerted on $10 \times 17 = 170$ square inches.

After cut-off, during expansion, or two-third stroke, the whole area receives pressure $= 10\frac{1}{2} \times 17 = 178$ square inches. The average area or 174 square inches receives a downward pressure of 160 lbs. per square inch = 27,840 lbs., or nearly 14 tons.

Upward pressure from the cylinder relieves this somewhat.

In the one third during admission	(2)	has	(4)	upon it	=	435 lbs.
" " second third,	(1)	has	(5)	upon it	=	1980 "
" " " "	(2)	"	(4)	" "	=	435 "
" " third third	(1)	"	(6)	" "	=	720 "
" " " "	(1)	"	(5)	" "	=	1980 "
				Summation	=	5550 lbs.

or an average through the stroke of = 1850 "

Subtracting this from the average downward pressure 27,840 − 1850 = 25,990 lbs. net average downward pressure.

The work in foot-pounds per minute to slide the valve upon its seat will be the product of the pressure factor multiplied by the feet per minute through which this resistance to motion is overcome. It will be seen that the previous discussions have shown how these pressures may be kept as small as possible and how the motion or travel can be diminished. With high pressures and large volumes of cylinders to be filled steps must be taken to diminish pressure on the valve, since the

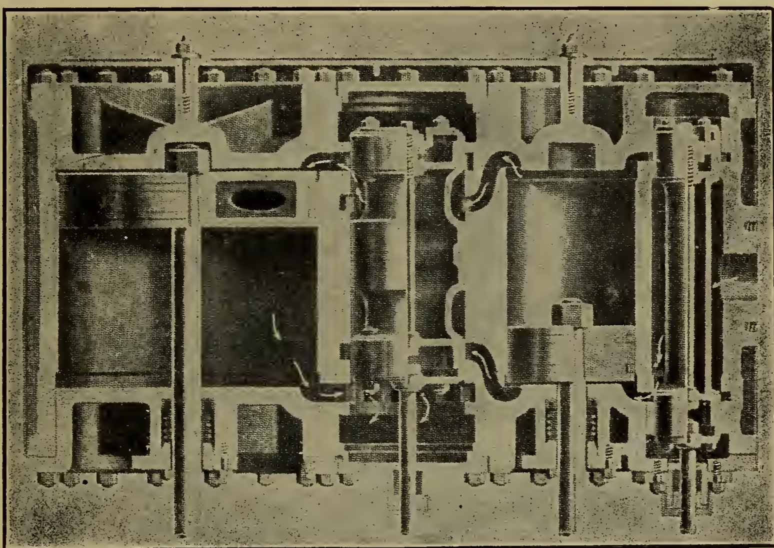


Fig. 599.

feet of travel must remain always a considerable quantity. The most satisfactory method for accomplishing this result gives rise to what are called balanced valves. The most effective methods to secure this balancing may be grouped into:

1. The Piston Valve System.
2. The Pressure Plate System.
3. Counter Pressure Systems.

In both the latter are several sub-divisions. In the last are the balanced poppet systems.

446. Piston-Valves. The simplest form of balanced-valve is what is called the piston-valve, which is shown in the section of the valve and chest in Figs. 599 and 510. The piston-valve consists of an ordinary shell or D valve, which has been made to revolve around its valve-stem as a center so as to generate a volume of revolution. The plane faces of the typical slide-valve become the surfaces of a cylinder, and the plane valve-seat must become a hollow cylinder which the valve-faces fit like a piston. By referring to Fig. 600 it will be apparent that steam-pressure is equalized upon the two end-faces of the cylindrical valve, and the contact of the valve with its bore prevents any pressure other than friction from getting at the valve sidewise. The only resistance to the motion of such a valve is its own friction, no matter how great the steam-pressure may be. The valve may be arranged with double pistons as in Fig. 510, or with single pistons as in Fig. 600.

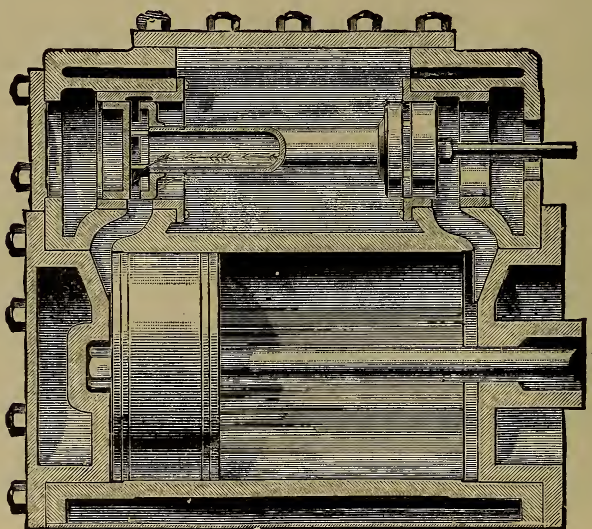


Fig. 600.

The objections to the piston-valve are the difficulties from leakage and wear. The pistons cannot fit tight in their bore, because unequal expansion would cause them to be seized by the bore when the latter was cold and the pistons were hot. To prevent excessive leakage they must therefore be made steam-tight by means of spring-rings whose elasticity causes them to spring out against the bore while they fit grooves in the piston tightly enough to prevent leakage around them (paragraf 381). These spring-rings must be prevented from catching in the ports over which they slide by bridges of metal which prevent their enlargement when the rings are opposite the ports. These rings, however, cause friction and wear. When the piston-valve is used without rings great care must be taken to prevent the possibility of difference of temperature between the piston and its bore. Unequal expansion would cause the piston to be cramped by the bore, and this must be prevented by jacketing the latter with extreme care. Fig. 600 shows this precaution taken. The piston-valve precludes great reduction of the clearances and the shortening of the passages. For very large marine engines, in order to diminish the diameter of the valves and at the same time to shorten certain connections, two sets of piston-valves are used, working together from a common valve-rod. The piston-valve is extensively used on locomotives, steam-hammers, rock-drills, and the like. Its great advantage for steam-hammer practice in large sizes comes from the ease with which the valve can be worked by hand. A favorite form of such piston-valves is rectangular or square instead of round. Round pistons are easier to make and to pack.

447. Pressure-plate Systems. The pressure-plate system aims to secure the release of the valve from unbalanced steam-pressure by

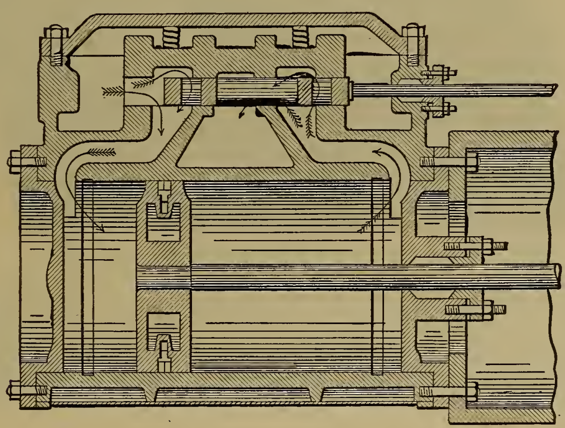


FIG. 601.

receiving that pressure on a plate which is supported positively in the steam-chest and underneath which the valve shall slide. The principle is fundamentally the same as the piston-valve; but the valve can be flat, and need only be steam-tight on its top and bottom, where it touches the seat and pressure-plates, respectively.

The pressure will be on the ends equally; and the sides of the valve to be finished need only be the top and bottom, which slide on the seat and the lower face of the pressure-plate. The valve is a flat block, and its faces can be scraped true. Clearances are diminished and the lengths of the passages. The system will be found in very extensive use (Fig. 601).

There are three great systems of arranging the pressure-plate principle. First, the fixed plate, non-adjustable. In this design the fixed plate rests upon lugs or ledges in the sides of the steam-chest or on its bottom,

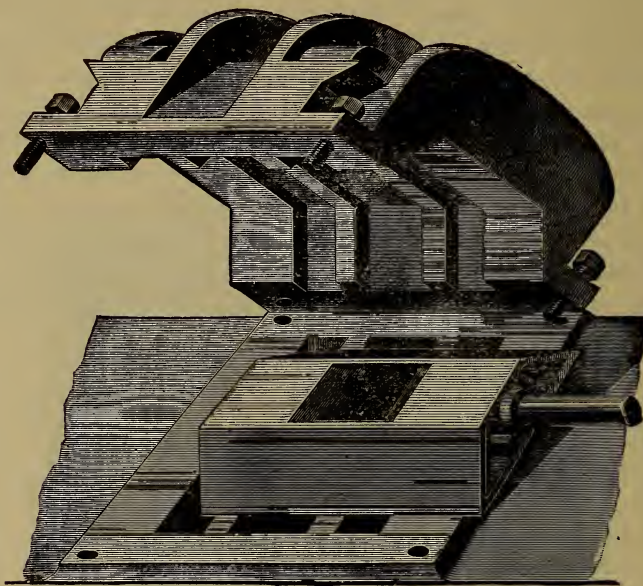


FIG. 602.

the length of the plate being such that the valve will always remain under it as it travels (Fig. 602 and right hand of Fig. 604); or the plate may be the top of the steam-chest. The valve may have no

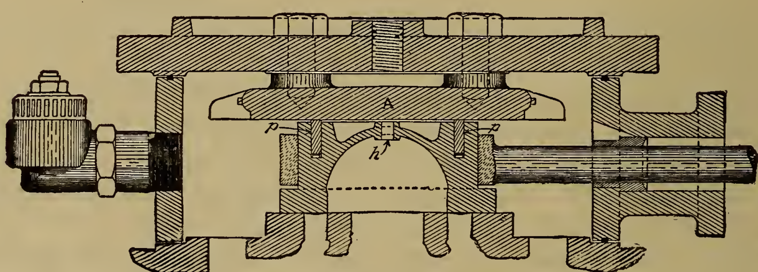


FIG. 603.

packing provision as in Fig. 602, or an adjustment may be provided as follows: To the back or top of the valve is fitted a spring-ring or an equivalent device, which fits the valve and the plate in such a way that no steam can reach the top of the valve by reason of the contact continuously made between the valve and the plate through the spring-ring. It will thus be seen that the valve is in equilibrium of steam-pressures all around it except on its back, and the only resistance to its motion is the friction caused by the elasticity of the packing device. It is frequently arranged to have the space inclosed by the packing-ring communicate with the exhaust through a hole, so that if the packing-ring should leak, the leakage would be into the exhaust-pipe, and it would be impossible for pressure to get upon the valve. Fig. 603 shows the balanced valve of this fixed pressure-plate and adjustable-ring type. The design chosen for exhibition is one which has been

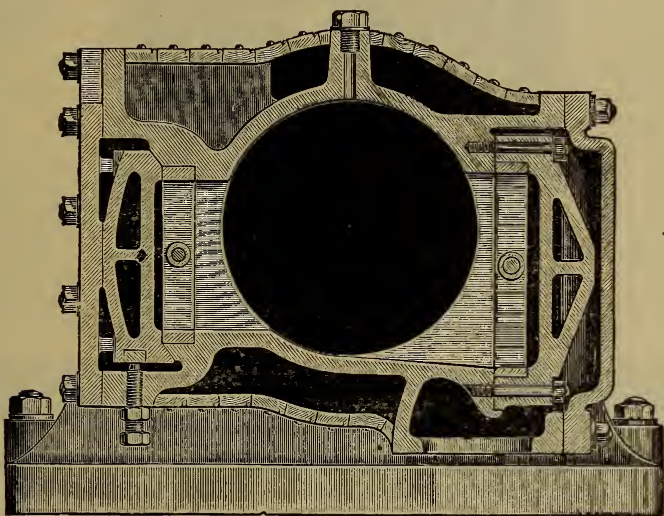


Fig. 604.

much used in locomotive service. In Fig. 602 the pressure-plate has relief-valves on its back which can open by excess of pressure from below, due to water forced back from the cylinder through the extra length of the ports. The piston-valve and the inelastic pressure-plate do not allow this displacement of the valve by excess of upward pressure. In Fig. 601 the pressure plate is held down upon its fixed supports by springs in addition to the steam-pressure, so as not to lift and clatter when steam is shut off.

The second type of pressure-plate systems is shown at the left hand of Fig. 604. In this design the valve is a solid block, but the pressure-plate which fits upon its back is arranged to be supported upon two inclined planes. The pressure-plate is also made to slope at the same angle so that by means of the adjusting-screw the inclined planes are made to slide over each other; the surface which bears on the valve remains always parallel to itself. It will be apparent that any desired

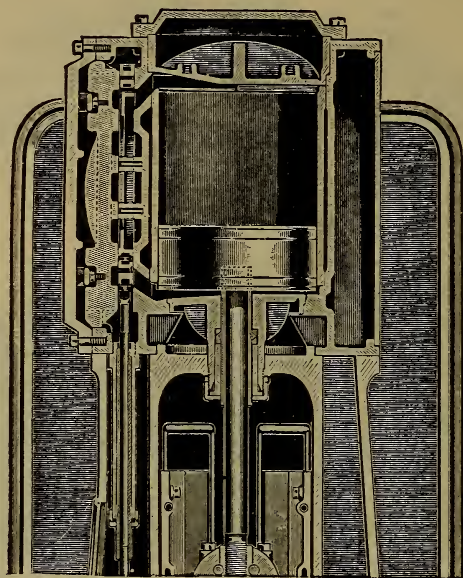


Fig. 605.

pressure of contact can be made between the sliding-valve and the pressure-plate while the pressure of steam is kept from the back of the valve.

In Fig. 598A in the appendix is another form of adjustable pressure-plate, adjusting lengthwise. Or, again, the pressure-plate may bear upon flat surfaces, from which it is held away by packing-strips, which are taken away as wear may render this course necessary (Fig. 605), or for exhaust-valves the construction of the right hand of Fig. 604 may be used, where the pressure-plate is bolted in place, and is relieved by surfacing anew as it wears.

The third pressure-plate system has been adopted for rougher grades of work than the two preceding. The pressure-plate is a disk or plate of steel or similar flexible and resilient metal. It is so calculated that the pressure upon it shall force it down upon the valve which slides under it, but its resistance to such flexure shall be sufficient to permit only the desired nip or squeeze to reach the valve and prevent leakage of pressure between the valve and the plate (Fig. 606). In some old designs this flexible pressure-plate was supported at its central point by a stud which came up through the top of the steam-chest, and could be adjusted from outside with steam upon the valve within.

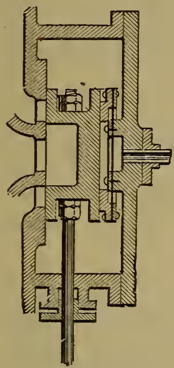


Fig. 606.

The first two systems are the most usual.

448. Valves taking Steam Internally. Closely resembling the piston-valve and pressure-plate modifications of it is the third type of balanced valves. In these, provision is made to have the upward pressure exerted by the steam upon the area made just enough less than the area exposed to downward pressure so as not to lift the valve. The unbalanced force tends to keep the valve upon its seat. This compels the use of a hollow valve whose upper side shall either fit against a surface, adjustable or fixed, which shall serve to keep pressure away from the outside of the hollow valve on the side corresponding and opposite to its seat, or a scheme is used in which the pressure is balanced around it (Fig. 607). Here it will be observed that the steam enters within the hollow valve from below and presses upward over the under side of the shell. This would lift the valve away from contact with the seat were it not for the

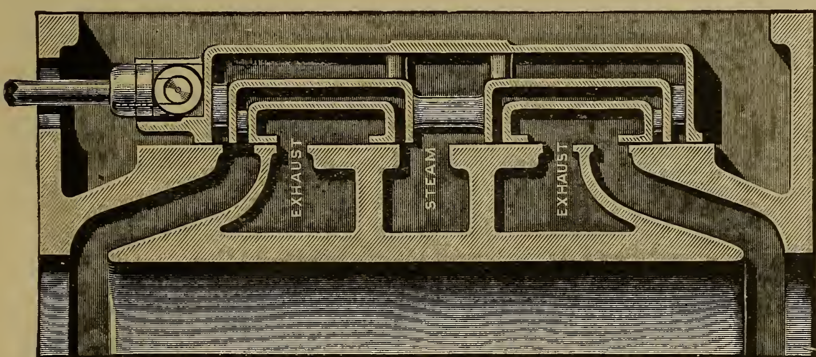


Fig. 607.

unbalanced pressure in the Allen thoroughfares over the two exhaust ports. This is practically a hollow piston-valve, with pressure inverted. These types run naturally into the pressure-plate system with steam on the ends only.

449. Valves with Counter-pressure. A very simple arrangement of counterbalance was applied to many Worthington pumping-engines of large size. The valve was attached by a link and pin-joints to a piston. This piston fitted a vertical cylinder directly over the valve, the area of the piston being so calculated that it was not quite enough to lift the valve even when the latter had its maximum pressure underneath it. The cylinder needs only to be long enough to permit the swing-link to follow the valve as it moves back and forth. This principle of counterpoising pressure by means of a piston has also been used in massive vertical engines to provide for the weight of the valves and their rods and to keep the strain always in one direction. It can also be similarly used to counterbalance the weight of the mechanism of the engine itself.

Locomotives have been fitted with a nest of rollers which lie in a groove below the surface on each side of the valve. They come just to the level of the seat, and carry the downward pressure to a degree without allowing leakage below the valve and between it and its seat. They form a roller-bearing.

450. Poppet-valves. It is one of the great advantages of the double-seated lifting-valves that they are nearly, if not entirely, balanced, as discussed under paragraf 416. Their limitation when massive to low rotative speeds has restricted their use as a means of relieving the pressures to be overcome at high speeds of rotation.

451. Cam Valve-gears. The convenience of a cam as a means to operate the valve distributing steam to the engine-cylinder was early appreciated. The profile of the cam, whether revolving continuously or vibrating through an arc or a part of a circle, can be designed to give to the valve exactly the desired motions and at the desired times. It can moreover hold the valve open or shut while continuous motion of other elements or organs is in progress, which the crank motion does not permit; and furthermore it permits the sudden or rapid closure of the valve by gravity or by a spring when the profile of the cam lets go of the stem.

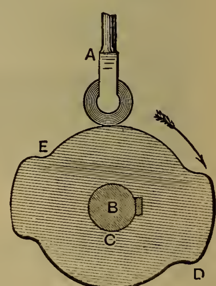


Fig. 608.

Cam-motions are of two great classes. In the first the cam-shaft revolves continuously in one direction. In the second the cam-shaft is a rock-shaft vibrating through an angle first in one direction and then in the other. In cam valve-gears of the first class there are two arrangements usual. In the first arrangement the cam bears against a roller in contact with its exterior surface; such are called outside cams (Fig. 608). The roller is conveniently mounted in the end of the valve-stem, and can be in the plane of such valve-stem, or the cam may bear against the end of a pivoted lever which actuates the stem (Fig. 610). In the other arrangement the roller fits in a groove in the side of a cam-plate.

The outside cam-motion works the valve in one direction only; for the return motion either gravity or a spring must be depended on, or else there must be a roller or yoke opposite to the first one to bring the rod back with a motion similar to that caused by the cam against the first roller. This outside cam arrangement has the advantage that the roller always turns in the same direction. The two-roller or yoke-plan has a grave objection from the difficulty that wear prevents the distance between the roller or yoke surfaces remaining always the same as the net diameter of the cam at every point. If the roller or yoke does not touch the cam continuously, there is a jar and shock followed by wear at the points where such contact begins.

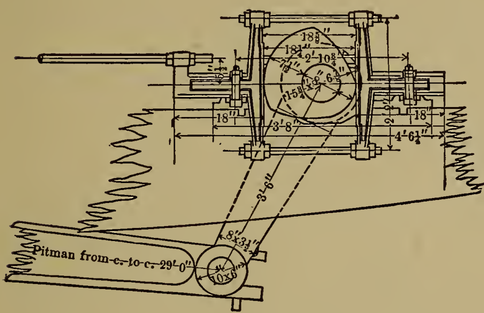


Fig. 609.

The side-cam arrangement where the roller fits in a groove has the

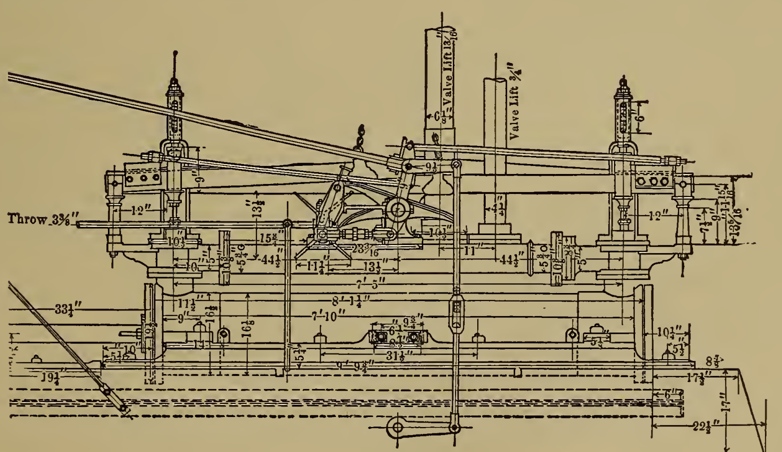


Fig. 610.

entire motion of the valve effected by that one roller and groove. This is called the box-cam system. The difficulty is that the inside surface of the groove drives the roller on the lifting stroke, and the outside surface of the groove drives the roller on the reverse stroke. Hence at each reversal of motion the rotation of the roller must be instantly reversed, and the inertia of the roller resisting this reversal prevents perfect rolling contact at those points of reversal, and wear and rattle ensue. The difficulties from the inertia of the rollers and the mass of the valve-rods have limited in the past the use of cams to relatively low rotative speeds. Recently a design has been brought forward in which the reverse motion of the valve-stem is caused by a pressure of air or steam upon a piston on the rod, so that the mass to be moved by the cam is reduced to its lowest terms, and the pressure on all pin-joints is kept constantly in one direction (Fig. 613).

When the cam rocks or vibrates upon an oscillating shaft instead of revolving continuously (Fig. 610) it drives in one direction only, and the weight of the rod or a spring, or both, must be used to return the valve. This rocking or oscillating cam is almost a distinctive peculiarity of the beam-engines used in deep-river-boat practice of Eastern America (Fig. 389). These are four-valve engines, the two exhaust-valves being on the right-hand side and the two steam-valves on the left, as the observer faces the engine (Fig. 561). Eccentrics on the water-wheel shaft transmit a reciprocating motion to cranks upon a rock-shaft which crosses the front of the engine and is divided into two sections at the middle bearing. The valve-rods are lifted by the profiles of curved cams or wipers which bear against the horizontal surfaces of toes which are lifted by the rocking of the rock-shaft. The exhaust-valves must be open full stroke, and consequently the plane of the two wipers has almost a common tangent at the dead-centers, so that the cam on one side will have just closed the valve at the upper end of the cylinder when the cam on the other is to open the valve at the lower end. The steam cams make an angle with each other, so that there will be an interval between the closing of one and the opening of the valve at the other end. This gives the interval for expansion at the end of each stroke, while securing admission at the proper point. The eccentric of the steam end of the rock-shaft is usually of greater throw than the exhaust side, so as to give greater amplitude to the motion of the cams, and the steam-cams are made longer than the exhaust-cams, in order to give gentle curves for their action, and yet open the valves quickly and close them promptly. This valve-motion of wipers and toes was first proposed by the Messrs. Stevens in 1848, and is usually known as the Stevens cut-off.

The rocking-cam also appears in Figs. 610 and 614 for actuating the valves of inclined cylinders by bearing against the under side of a pivoted lever to which the poppet-valve is attached.

The Western river-steamboat with horizontal engines for its water-wheel usually operates its valves by continuously moving cams on the water-wheel shaft, which bear against the surfaces in a frame to which the revolving rod is attached. The relatively slow motion gives rise to no difficulty with this type of gear (Figs. 609, 614).

Cam motions can be made adjustable or variable by arranging to have

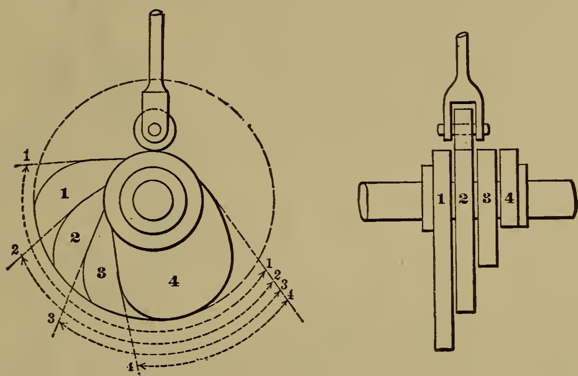


FIG. 611.

the profile of the cam variable. This is usually done by having the cam made up of several layers which are movable under the roller in such a way that the acting face of the cam can be made shorter or longer

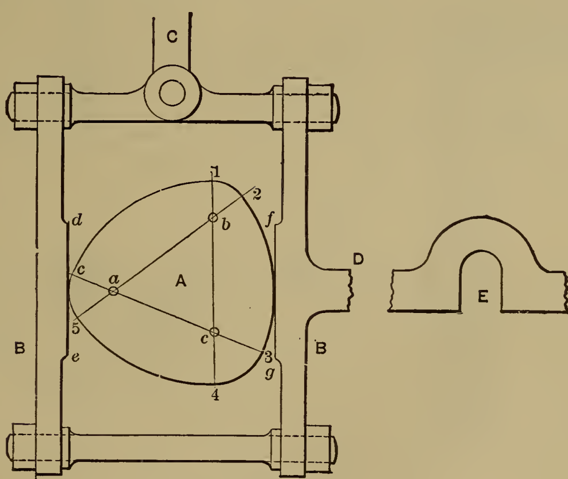


FIG. 612.

at the pleasure of the operator. Or the cam is made of a varying profile at different sections of its considerable face, and different parts are brought under the roller or valve-lever (Fig. 611).

It is not often attempted in cam valve-gears to make one cam and one valve perform all the valve-functions. Cam valve-gears are usually three or four-valve designs. The cam gear is usually worked with

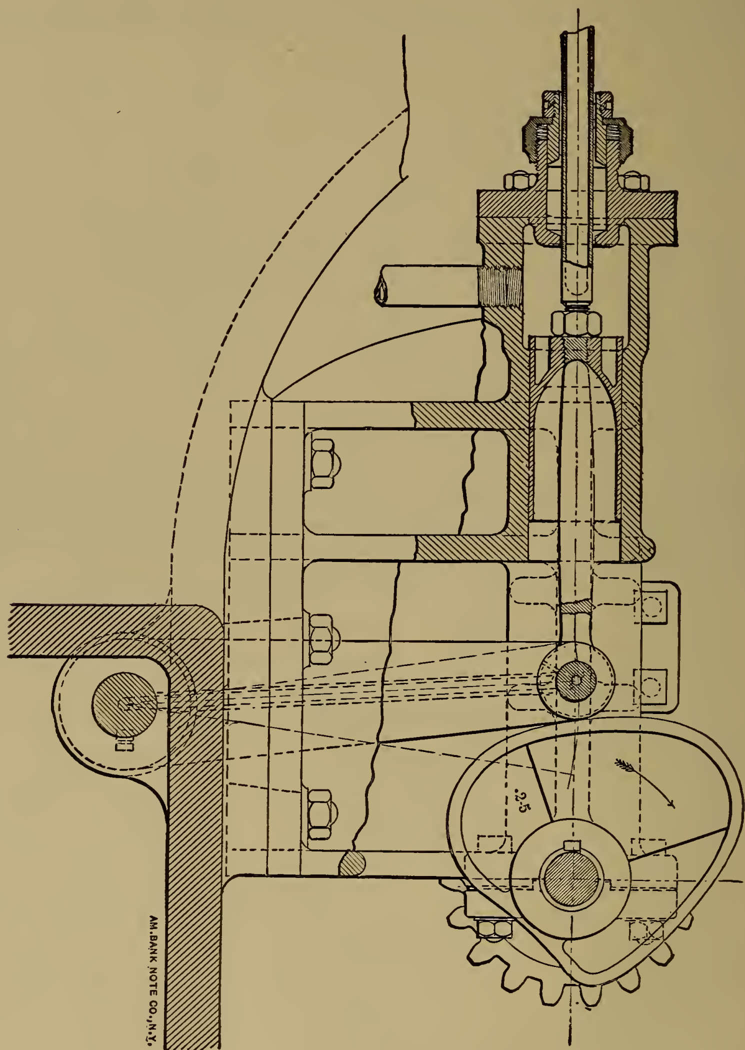


Fig. 613.

poppet-valves, because the valve must oppose the least resistance to motion, and must be balanced so as to be self-closing. Examples of cam valve-gear will be found in Figs. 609, 610, 613 and 614. Fig. 612

shows an arrangement which holds the valve open or shut through a considerable angle of the rotation of the shaft.

452. Trip or Releasing Valve-gears. Belonging to the same general class as the cam gears are those in which a detent or catch which is pushed or pulled to open the valve is released when the valve is to be closed, so that it returns to its closed position independent of the operating mechanism. This return is effected usually by a weight or a spring or both, so that the valve is closed more quickly than it could be if the connection of the valve to the operating mechanism were positive. This principle of trip or release gears is identified in America with the

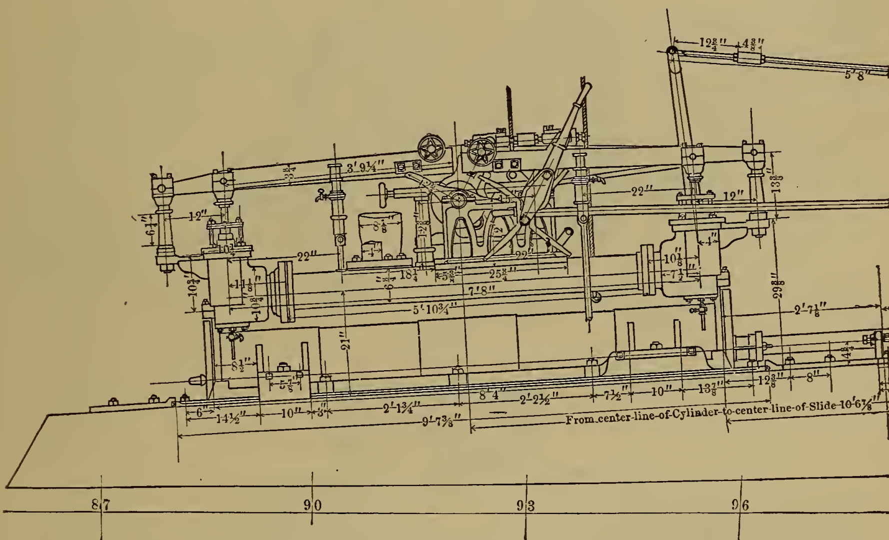


FIG. 614.

name of Frederick E. Sickles (1841), but has received its greatest development under the name of Corliss (1849), with whose name it is best identified in Europe. The original Sickles cut-off was applied to poppet-valves lifted by cams. When the cam had lifted the valve-stem to the desired point, a latch connection between the stem and the lifting mechanism was released and the valve closed independently by dropping. As the lifting mechanism descended it displaced the latch or detent until it had passed the latter, when, by a spring, the latch came forward into position to be caught by the lifting mechanism when it was to make its next stroke. As applied to early engines the Sickles principle was arranged to have the latch release adjustable by hand. It becomes easy to have this adjustment made automatically by the governor, and this easy adaptation has been a great stimulus for the development of this class of valve-gear. A type of gear which presents the trip-and-release mechanism in its simplest form is that which is identified with the Greene engine, Fig. 615. It will be seen that the slide *J* traverses back and forth driven by an eccentric. This slide carries the latches *GG′*, which are held upwards by means of springs and are inclined upon their upper faces. It will be apparent that as the sharp corner of the latch catches the end of the arm *B* it will swing it upon one stroke, but will be depressed by it on the other. As soon as it has passed by, however, the spring under the latch will force it up

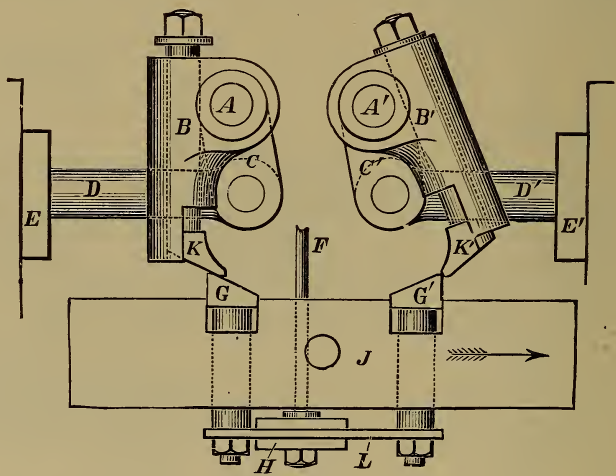


Fig. 615.

so that it will be ready to catch and swing the arm *B* on the next stroke. If the governor mechanism be attached to the rod *F*, it is obvious that, if it acts to depress the latches *G*, the arm *B* moves the valve-stem *D* through a less angle, and lets it go so much the sooner. The exhaust-valves of the Greene engine are operated independently underneath the cylinder, one at each end.

453. Corliss Valve-gears. The Corliss valve-gear involves four separate features. The first is the rocking cylindrical valve, of special construction to eliminate sticking fast by expansion. The second is the operation of both steam-valves and exhaust-valves from a rocking wrist-plate; the third is the trip or release of the steam-valves by the governor, and the fourth is the closure of the steam-valves by gravity or a spring.

The Corliss system uses four valves, two of them for steam above the axis of the cylinder, and two for exhaust below. In the form shown in

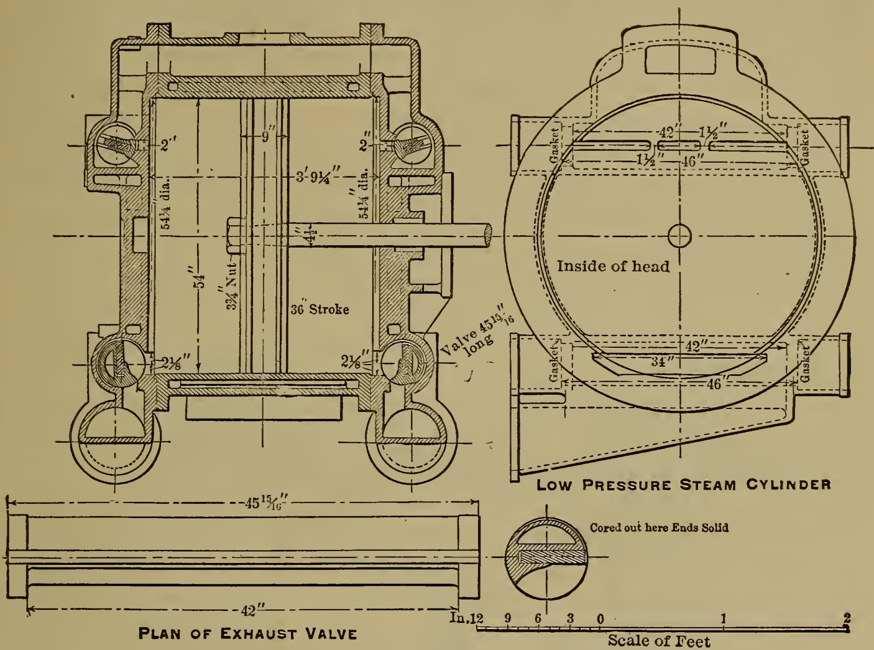


Fig. 616.

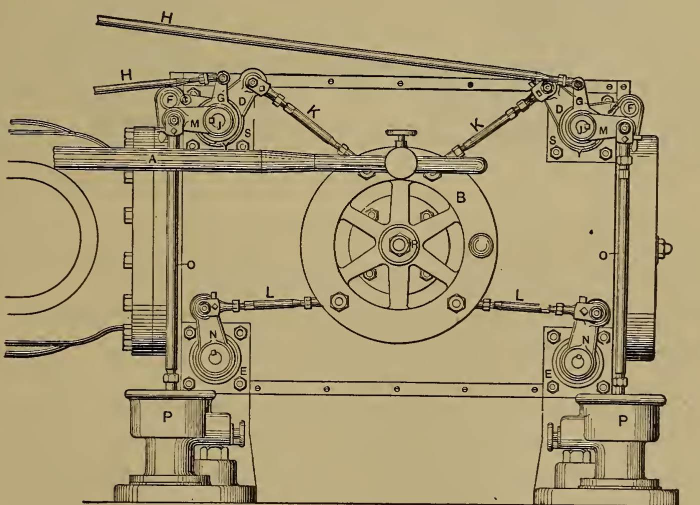


Fig. 617.

Figs. 562 and 616 they are made directly in the cylinder-heads. It is also quite usual to put them upon the sides, as shown in Fig. 563. The characteristic of the valve is that the spindle by which the valve is caused to rotate is not fastened to the valve, but is independent of it. This takes away the objection to the cylindrical cock-valve. The valve need not fit tight all about its casing, but must turn when its spindle is turned from without. On the type shown it will be apparent that ports and passages are of the smallest possible length and surface, thus taking advantage of the features discussed in paragraf 417.

The second feature of the Corliss valve-gear is the operation of the

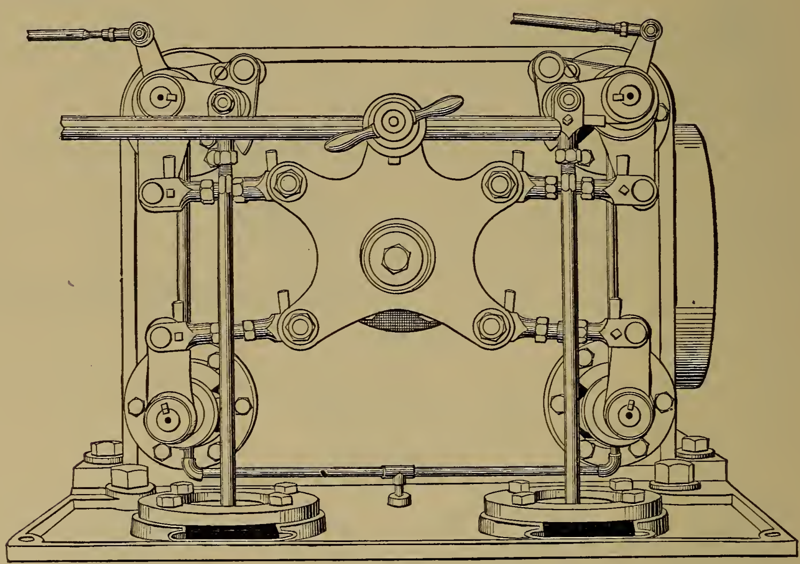


Fig. 618.

four valves from points on a wrist-plate which is made to vibrate back and forth through a considerable angle by its connection with the eccentric. The two upper valves, Fig. 617, are connected near the vertical diameter of the wrist-plate, and the two lower or exhaust-valves connected nearer to the horizontal diameter. It will be apparent from this peculiarity that the steam-valves will be opened rapidly as the engine passes its dead-points, while the exhaust-valve will be held wide open during that fraction of the angular motion of the wrist-plate during which the link of the exhaust-valve is coming up into line with the center of the wrist-plate and is passing beyond and above that line and is returning to its original position. The other steam and the other exhaust-valve are without effect on their openings during the stroke in which their mates are in action. The valves usually work in diagonal pairs. The wrist-plate appears in many forms in the various makes of engine. Fig. 618 shows a type where a larger throw is given to the steam-valves than to the exhaust-valves to extend the period during which cut-off may take place. The eccentric driving the wrist-plate will be 90° ahead of the crank so as to operate the exhaust valves at the right time. Since the trip can only take place during the pushing stroke of the steam-rods, it cannot take place later than one-half or three-eighths stroke when the whole wrist-plate is driven by the one eccentric. This is the basis for two wrist-plates and two eccentrics, the steam-plate having much the larger throw and set at a different angular advance. By doing this the range of variability of the cut-off is much wider, without changing the form of the indicator card.

The third feature of the Corliss gear is the release or trip of the steam-valve rods, by which the hold of the wrist-plate upon the valve is dropped and the valve closed suddenly by weight or spring. The peculiarities of the method followed in the detail of this release-gear differentiates many of the various Corliss engines from each other. In the very earliest forms the detent was thrown off as the valve-rod moved up an inclined plane or wedge. Another form throws a cam or eccentric into engagement with a curved arm or toe, and the pressure of these upon each other forces the rod to let go of the catch on the arm of the valve. In all of these forms the adjustment of the gear is usually made by the governor, so that the speed of the engine varies the length of admission by causing the cut-off when the trip occurs. The exhaust-valves are positively connected to the wrist-plate so that release and compression are constant, while cut-off and expansion vary according to the work of the engine. Fig. 619 shows the trip mechanism and Fig. 607 shows the same parts. J is the valve-spindle, lifted by a hook or catch on F. G causes the release or trip, as controlled by the governor connected through the rods H.

The fourth feature of the Corliss valve-gear is the closure of the valve by a weight or a spring with dash-pots. The dash-pot is a cylinder in which fits a piston nearly or entirely air-tight. (Fig. 617.) As the valve is lifted it lifts the piston in the dash-pot. Air enters below in the weighted dash-pot and when the valve is released the weight of the piston, with or without the help of additional weights, closes the valve, when the retarded escape of air in the dash-pot arrests the motion without excessive shock. In the vacuum dash-pot the piston fits tightly, and the lift of the piston, creating a partial vacuum below itself in the dash-pot, causes atmospheric pressure to become the weight or spring which closes the valve. The compression of the air remaining below the piston performs the cushioning necessary to prevent shock.

454. Advantages of Trip Valve-gear. The advantages of the Corliss and other trip valve-gear are:

1. The degree of expansion or the point of cut-off can be varied through a wide range of load variation while the release and compression are kept constant.

2. Quick opening of inlet-valves establishes full boiler-pressure in the cylinder early in the stroke. It is a feature of most of these gears

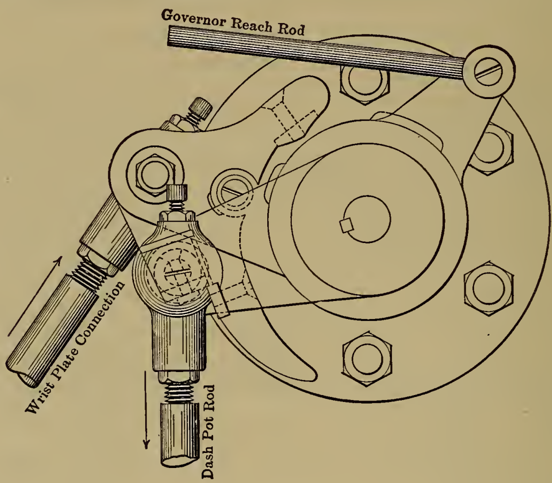


FIG. 619.

that their construction and operation give large port-areas and little friction through the valves.

3. The inlet-valve closes quickly. The effect of this is to increase the area of the work-diagram by giving a sharp corner at the point of cut-off instead of a rounded curve. A rounded corner at this point has the effect of gradually lowering the pressure from wire-drawing as a consequence of gradual closure.

4. This type of gear, being specially adapted for engines which are designed to be regulated by varying admission, gives the advantage of making the terminal pressure low.

5. The ease of adjustment of such independent valves if variation at the two ends of the cylinder should be desirable.

6. The small motion and the period of rest for the steam-valves after closure diminish the friction of the valve-gear and the attendant loss of power.

455. Disadvantages of Trip Valve-gear. The trip valve-gear, being almost always a multiple valve-gear, is open to the following objections:

1. The complication and number of parts in most of the gears.
2. The expense of most of the engines fitted with complicated gear.
3. The limitation in rotative speed or number of revolutions imposed by the necessity of engaging the catches and valves. The inertia of the masses precludes an instantaneous action in response to the spring or weight, and the snap-and-catch action becomes noisy when the springs or similar devices have stiffness sufficient to make them positive at speeds faster than 150 revolutions per minute.
4. The trip and the cam valve-gear have this objection in common, that the release of the catch and closure of the valve must be effected by the governor during that stroke of the valve-rod which nominally opens the valve. In other words, in a lifting-valve the release must take place before the valve reaches its point of greatest opening. This limits the range of the ordinary gears of this class with respect to their ability to adjust the point of cut-off, and is only remedied by the second wrist-plate and its attendant complication and cost.
5. The variable stroke or traverse of the valve. The seat wears through the angle of most frequent motion more than beyond this range, and this forms shoulders which are hit by the valve on its wider swing.

456. Sundry Valve-gears. The method of operating the valves of a non-expansive engine by a steam-cylinder, which is used in direct-acting pumps and has been discussed in paragraf 149, should be referred to again in this connection. See also Fig. 697A in the Appendix for a riding cut-off valve thrown by steam. The cataract of the Cornish pumping engine is another system of valve operation.

CHAPTER XXVIII.

REVERSING VALVE-GEARS. LINK-MOTIONS.

460. Reversing-gears with One Eccentric. It will be apparent from discussions in parts of Chapter XXV, which have treated of the setting of valves, that if the valve had neither lap nor lead, so that the angular advance of the valve-crank was 90 degrees ahead of the engine-crank in order to go forward, it could not be at the same time 90 degrees ahead of the same crank which was to turn backward. A very simple reversing-gear for a valve of this type can be made by having the valve-stem driven from a rocker-arm and so constructed that the rod of the eccentric can be geared to it either on the same side of its center of motion as the valve-stem or at will upon the opposite side. From the discussion in paragraf 421 it will be apparent that when the motion of the eccentric-rod is reversed by the rock-shaft the engine will turn in one direction, and when it is not so reversed it will turn in the other. When the valve has lap or lead or both, and is intended to work expansively, the valve-crank is $90° + $ an angle a, which represents the angle AOE in Fig. 584, ahead of the main crank. Hence the position of the center-lines of eccentrics for forward and backward motion will be distant from each other an angle represented by $180° - 2a$, and a reversing motion by the method just described is impossible.

There are two methods of reversing an engine using one eccentric. The first is to have the eccentric loose upon the shaft and free to move independently of that shaft between two stops which are bolted, keyed, or otherwise secured to the shaft. The loose eccentric has a corresponding lug or projection which engages with these stops. The angular distance between the stops upon the shaft is so adjusted that when the first one engages with the lug upon the eccentric-disk, the relation of the eccentric-crank to the main crank is that which adjusts the valve-gear to distribute steam for forward motion. The resistance of the valve as the engine turns in one direction keeps the lug and first stop continuously in contact. If the engine-shaft be turned in the opposite direction by operating its valves by hand, the first stop will leave contact with the lug, and the eccentric will stand still by reason of the friction of its attachments until the second stop on the shaft comes in contact with the lug on the eccentric. The adjustment of the second stop is such that when it touches the lug the relation between the eccentric and the main crank is that for distributing steam for backward motion. This arrangement of loose eccentric with lug and stops on the shaft has been a very favorite design for ferry-boat engines. The working of such boats in and out of slips is done by hand-working of the valves in any case, and their comparatively slow rotative speed and the large masses in the disks and rods lend themselves to this arrangement.

The second single reversing-gear adjusts the eccentric through the angle $180° - 2a$ by having the latter borne upon a sleeve to which it is feathered, so that it must rotate with it and the shaft, while the sleeve can be slid lengthwise on the shaft under the eccentric. This sliding

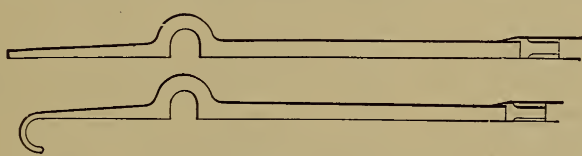


FIG. 620.

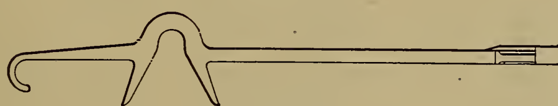


FIG. 621.

of the sleeve is done by a lever which has a latch attachment so that it can be locked in the desired position. The sleeve has a spiral slot cut in it, the slot subtending the angle of $180° - 2a$ and fitting a radial pin projecting from the shaft. It will be obvious that when the sleeve is slid lengthwise along the shaft the slot and pin will twist the sleeve through the angle $180° - 2a$, and carry the eccentric through that same angle. The latch prevents readjustment except at the will of the runner. This makes a very compact reversing-gear, but is limited to engines of small size (Fig. 409). It is much used in small launches and in geared road-rollers or traction-engines.

461. Reversing-gears with Two Eccentrics. Gab-hooks. It makes so much simpler a reversing-gear to use two eccentrics, one set $90° + a$ ahead of the crank for forward motion, and the other set $90° + a$ behind it, which becomes that same angle ahead of the crank for backward motion, that this type of reversing-gear is much the most usual. There is a rod from each eccentric which is to be hooked and geared to the valve-stem at will, and the method of bringing the forward and backward eccentric-rod into gear with the valve-stem constitutes the differentiating feature of all forms of motions.

The simplest and oldest device to attach the eccentric-rods to the valve-stem is a hook. This hook, called a gab or gab-hook, is simply a hole which fits a pin on the valve-stem or on a rocker-arm connecting to it, which hole has one side cut out and away so that it can be lowered down upon the pin or lifted off from it. Of course the two hooks must not be engaged with the same pin at once, and many different methods are used to take care of the hook and rod which for the time being are not to engage with the pin. The simplest is a lifting-roller so adjusted that when brought against the under side of the rod it lifts it above the plane in which the pin travels. This may be done also by lifting a suspending-link. Other devices involve the use of cams or bars which shut down over the sides of the hook and fill up the hole, and are held in place by a latch or snap. With these the eccentric-rod may slide upon the pin itself, but when these appliances are in place the hook is closed and takes no hold of the pin. Figs. 620 and 621 present certain forms of gab-hooks.

The objections to the gab-hook are three:

1. The engine reversed by this means must have a low speed of rotation.

2. The engine has to be of such a character that the reversal can be leisurely. It is not convenient to reverse at speed with a gab-hook, but the engine must be turning slowly when the hook is dropped upon the pin.

3. The engine must be of such a character that it can be started by hand-working of its valves. The reason for this is that there is but one position of the main crank in which the hook of the forward gear and that of the backward gear coincide, so that either can be dropped upon the pin and operate the valve properly to pass from forward to backward motion. This position is the dead-center, either outward or inward, on which the engine-runner would never stop his engine if it could be helped, so that hand-starting is compulsory.

To avoid the objections to the ordinary gab-hook so that the engine might be reversed at speed and started in the reverse direction without hand-working, the mouth of the hook was widened and the sides lengthened so that it took somewhat the shape of an inverted letter V. The distance apart of the horns of this V hook was made equal to the travel of the eccentric-rod plus the diameter of the pin, so that, no matter where in its course the pin might happen to be, the sides of the hook, pressed upon the pin, would slide it in the direction of motion until it caught into the hook proper at the foot of the V. These V hooks were early solutions of the problem of reversing the locomotive engine. Figs. 621 and 614 give the general appearance of such hooks.

462. Link-motion of Stephenson or Howe. The difficulties attendant upon large-size hook-gears for reversing when they came to be applied in high-speed practice brought about the development of what have been called the link-motions. If the forward and the backward hook be made to face each other so that one hooks upon the pin from the

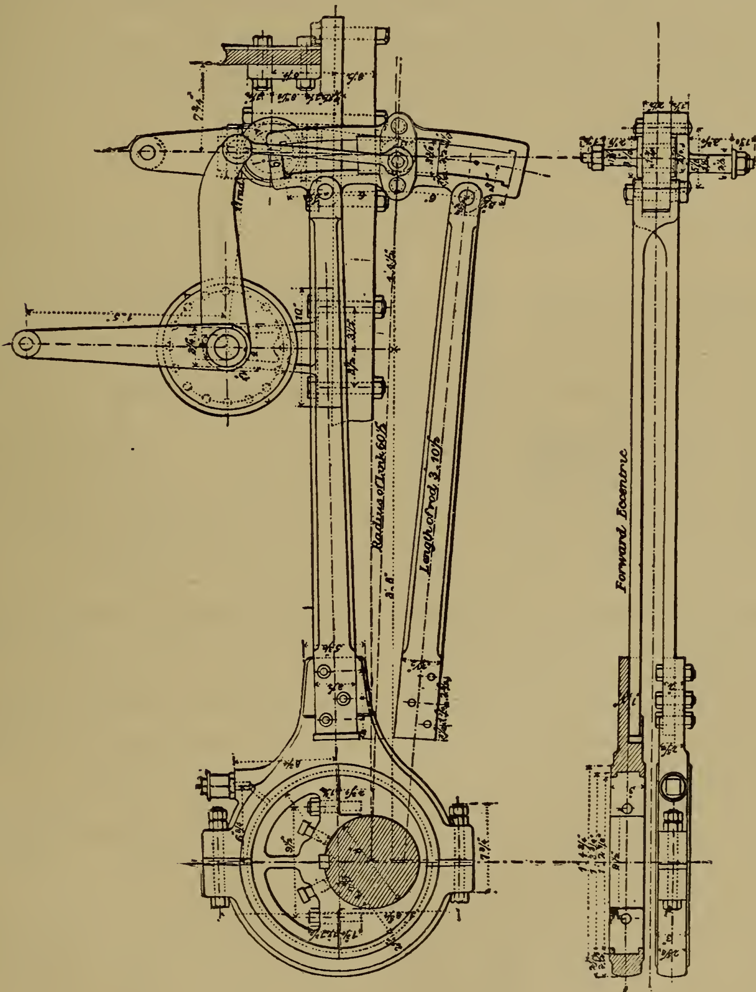


Fig. 622.

top and the other from the bottom, and if these two hooks be joined together on their outer and inner edges by two arcs of circles struck from the center of the shaft, there will be derived the Stephenson link. The upper hook of the old gear becomes that part of the slot in the link just behind the joint of the forward eccentric-rod to the link, and the lower hook the similar part of the link-slot behind the joint to the backward eccentric. The curved profile of the link keeps the two eccentrics from undesired motion, and the pin of the valve-stem fits in a suitable block in the slot of the link, so that the latter is always ready to be moved to bring either forward or backward eccentric to drive the valve, while the eccentric not required simply vibrates the link around its virtual center without affecting the valve. Fig. 622 shows the typical Stephenson link-motion as designed for locomotive practice and Fig. 623 its skeleton diagram. There require to be attached to the link con-

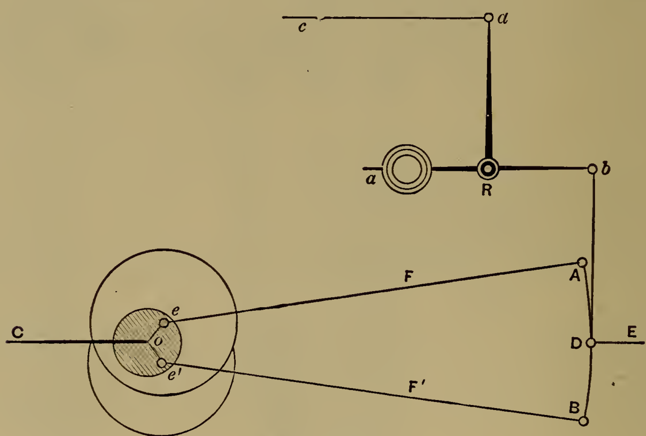


FIG. 623.

venient connections to bring the forward or backward eccentric into line with the pin and to hold it at the desired position, but their construction requires no explanation.

This simple form of reversing-gear was applied to early locomotives turned out in England by Stephenson, but strong claims have been advanced by a William Howe for its first suggestion in 1843.

463. Features of the Stephenson Link-motion. The Stephenson link-motion has certain peculiarities. If the valve had neither lap nor lead, so that the two eccentrics were 180° apart, it would be apparent that the link would vibrate around a virtual axis at its middle point, and that when the pin connected with the valve-stem coincided with that axis, the valve would have no motion. The angle between the eccentrics is not 180°, on account of lap and lead, and hence when both eccentrics are near the horizontal line in a horizontal engine they are each moving the link in the same direction at top and bottom. That motion, however, is usually so small that it does not uncover the laps over the port; or in other words, cut-off takes place before the stroke begins. At intermediate points above and below the center the travel of the valve is less than full throw of the eccentric, and by reference to Chapters XXVI and XXIX it will be apparent that earlier cut-off and greater expansion will be secured by this diminished throw, and yet without seriously distorting the exhaust events, since the angular advance is not disturbed. It is no disadvantage in locomotive practice to have compression increase with earlier cut-off. The heavy duty of the locomotive is in starting its load from rest, and at very high speeds on a level track the engine is doing much less work, so that it can be operated at earlier cut-off. The compression is a decided advantage at the high rotative speeds. It is the simplicity of combining the variable cut-off gear (which is desirable) with the reversing-gear (which is necessary) and in one mechanism so simple as to be operated with one lever, which has given the Stephenson link-motion its popularity for the locomotive.

The only theoretical objection to be urged against the Stephenson link is its slight inaccuracy, which produces a variation of the lead at different points of cut-off. By reason of the fact that the link is raised and lowered, carrying with it the rods, the latter are shifted around their eccentric disks. It will be seen that when the angle is varied which the eccentric rod makes with the line through the dead-center of the engine crank, from which angles are counted, there will be of necessity a motion of the valve at dead-centers of the engine-crank, since the effect produced is to diminish the angle $90° - a$, which is therefore the same as increasing the angular $90° + a$, which measures the angular advance ahead of the crank. To increase this angular advance ahead of the crank is a thing which is done where the lead is to be increased (paragraf 429). The lead, therefore, increases as the cut-off is made earlier. The following table shows the extent of these variations in a standard locomotive gear, especially when a little worn:

Notch in Sector of Lever.	Travel of Valve.	Port-opening.		Point of Cut-off.		Point of Release.		Lead.
		Forward.	Backward.	Forward.	Backward.	Forward.	Backward.	
20	5	$1\frac{1}{4}$	$1\frac{1}{4}$	$20\frac{5}{8}$	$20\frac{1}{2}$	$23\frac{1}{8}$	$23\frac{1}{8}$	$\frac{1}{16}$
19	$4\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$19\frac{5}{8}$	$19\frac{1}{2}$	$22\frac{13}{16}$	$22\frac{3}{4}$	$\frac{3}{32}$
18	4	$1\frac{1}{8}$	$1\frac{3}{32}$	$18\frac{1}{4}$	$18\frac{1}{8}$	$22\frac{3}{8}$	$22\frac{1}{4}$	$\frac{1}{8}$
16	$3\frac{1}{2}$	$\frac{7}{8}$	$\frac{27}{32}$	$16\frac{1}{2}$	$16\frac{1}{2}$	$21\frac{11}{16}$	$21\frac{9}{16}$	$\frac{5}{32}$
14	$3\frac{1}{8}$	$\frac{11}{16}$	$\frac{41}{64}$	14	14	$20\frac{11}{16}$	$20\frac{9}{16}$	$\frac{3}{16}$
12	$2\frac{7}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	12	$11\frac{3}{4}$	$19\frac{3}{4}$	$19\frac{9}{16}$	$\frac{7}{32}$
9	$2\frac{5}{8}$	$\frac{7}{16}$	$\frac{13}{32}$	$9\frac{1}{2}$	$9\frac{1}{2}$	$18\frac{5}{8}$	$18\frac{5}{16}$	$\frac{1}{4}$
8	$2\frac{1}{2}$	$\frac{11}{32}$	$\frac{5}{16}$	8	$6\frac{3}{4}$	17	$16\frac{5}{8}$	$\frac{9}{32}$

The throw of the eccentrics was 5 inches, the steam-ports $1\frac{1}{4}$ inches and the exhaust-ports $2\frac{3}{4}$ inches wide. The lap was $\frac{7}{8}$ of an inch outside and $\frac{1}{16}$ inside.

464. Gooch's Link-motion. To counteract the difficulties of the Stephenson motion caused by shifting it around the eccentric and shaft, Sir Daniel Gooch, a railway motive-power engineer of England, reversed the Stephenson link and made the valve-stem pin slide up and down in the link which was suspended, so as to be unable to rise or fall (Fig. 190). It is obvious that the reversing and cut-off action are

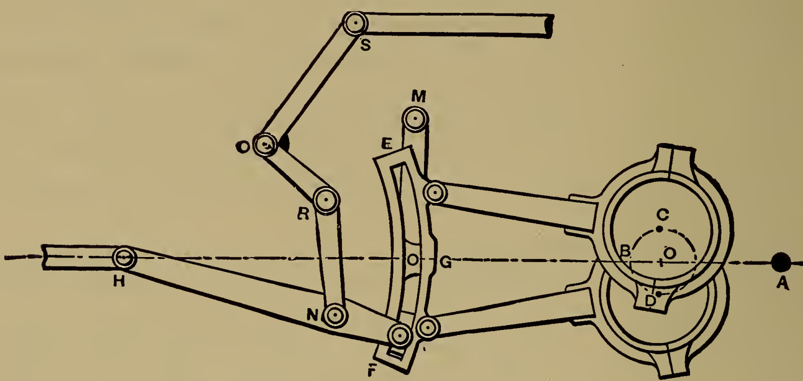
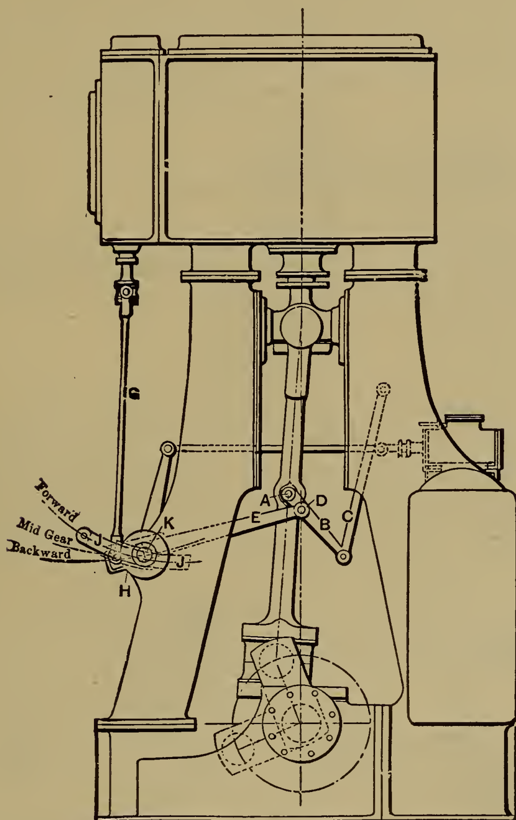


Fig. 624.

retained, but the variation in the lead is eliminated. This motion has never been popular in America, since for its satisfactory working the valve-stem element HNF must have considerable length, and the design of American locomotives makes it difficult to secure this. If the curvature of the link has a short radius, irregularity in the valve-motion is introduced.

465. Allan's Link-motion. The Allan link-motion combines the characteristic features of the Stephenson and Gooch. The link and valve-stem are swung from opposite ends of a lever pivoted at or near its center, and variation in position of the stem and link is produced by lifting one and lowering the other. The advantage of this form is the straight profile of the link, which makes it easy to machine in the shops, but, like the Gooch link, it has never met much acceptance among American locomotive-builders, where the type of outside-cylinder engine, which is preferred, makes it more usual to put the valve mechanism between the frames under the boiler. It has the advantage over the other two that the weight of link and of valve-stem partly balance each other so that counter-balancing weights or springs are not required as in the other forms.

466. Radial Valve-gear. Joy's Valve-gear. Variations have been made upon the link-motion hitherto discussed in the effort to do away with one of the eccentrics or both. The eccentrics in fast-running engines are sources of friction by reason of their large diameter, and they not infrequently give trouble from heating. The general name of radial valve-gear has been applied to such valve-motions as transmit a motion to the valve-stem from an arm, one end of which moves in a



JOY'S VALVE GEAR

FIG. 625.

closed curve and which has another point constrained to move in either an open or a closed curve by its connection with the frame through levers or slides. The closed curve described by the first point is usually a circle, an oval, or an ellipse, and motion is imparted from an eccentric or a crank or by the connecting-rod.

The best-known valve-motion of this type is Joy's valve-gear, shown in Fig. 625 as applied to a marine engine, and in Fig. 626 in outline. It has been used quite a little in both marine and locomotive service.

It will be seen that the motion originates from a point on the connecting-rod which gives, from its connection to a secondary link, a reduced motion to the lever which drives the valve-stem. The point D describes an oval. The other end of the link slides in a curved path to provide for the back-and-forth motion of its first end. The reversing effect is

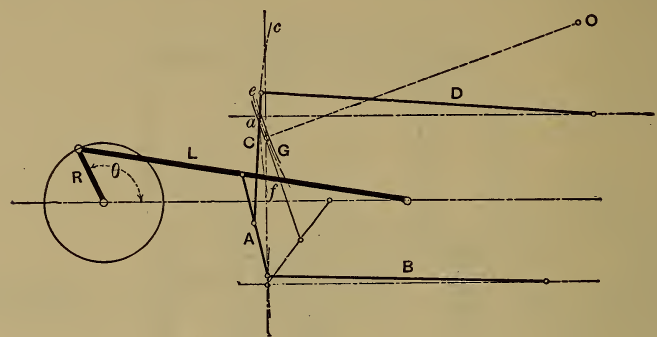


FIG. 626

caused by the angle at which the curved slot is inclined. A variation in the point of cut-off is produced by the variation in the throw of the valve-stem, which is least when the curved slide is midway between its forward and backward position. The slide may be replaced by causing the point K to vibrate from its connection to a link whose

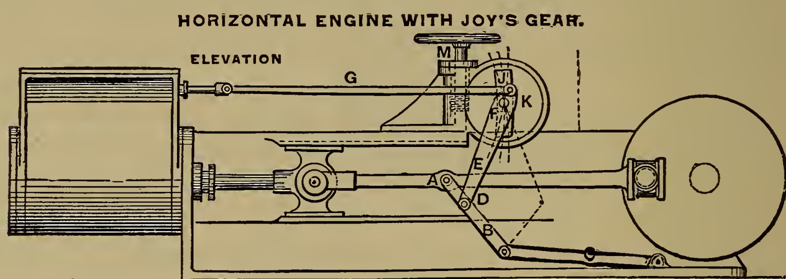


FIG. 627.

radius is the same as that used in describing the slide J in Fig. 625. The Joy valve-gear is made up entirely of pin-joints for the moving parts, and gives equal lead, cut-off, and port-opening in both gears. The objection to it in locomotive practice is the exposed position of the links outside of the frames, where accidental injuries are most likely to affect them. Fig. 627 shows this gear applied to a stationary engine.

Other gears in this class are those of Marshall, Brown, Hackworth, and Angstrom.

467. Walschaert Valve-gear. Almost the only other form of valve-gear which has contested with Joy's the sole acceptance with locomotive-builders is a Swiss motion which bears the above name of its inventor. Fig. 628 shows that the double motion is derived partly from the engine cross-head and partly from a crank or eccentric 90° from the main engine-crank. The valve gets an aggregate motion from the cross-head and from the curved link, and reversing is effected by reversing the motion derived from the eccentric-rod when the sliding-block is on one

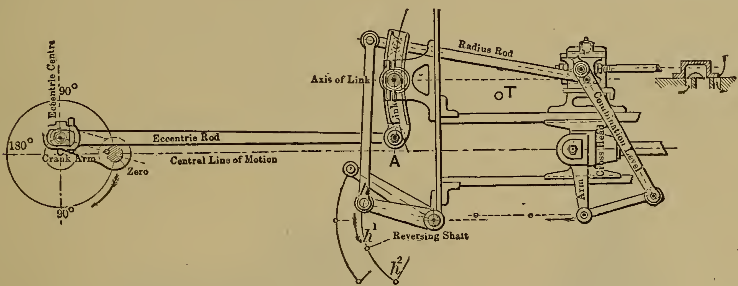


FIG. 628.

side or the other of the fixed center of motion of the curved link. It will be seen that such a gear produces no variation in the lead. Fig. 629 shows the American form of this gear as put out by the Baldwin Locomotive Works.

468. Allen Link-motion. The link-motion first proposed by Allen is the one which in a modified form is known as the Pius Fink gear. It has no eccentric-rod properly so called, but the link is an integral part of the back strap. The half of the strap which carries the link has a fulcrum-pin by which it is attached to the engine-frame above or below the shaft, so that the motion of the center of the link is an aggregation of the back-and-forth motion of the strap as a whole, and the up-and-down motions caused by the constraint of the fulcrum-pin which prevents undesired motion of the point where it is attached. Fig. 630 shows the Allen link. If the engine is not intended to reverse, but variation in point of cut-off only is desired, the slot in the upper half above the fulcrum-pin only is needed. As the valve-stem approaches the center-line of the shaft, its motion diminishes. In the Porter-Allen engine the separate exhaust-valve is driven from a fixed point near the end of the slot, giving constant travel, release, and compression. The eccentric of the Allen link is set opposite or at 180° with respect to the crank.

469. Link-motion for Riding Cut-off Valves. It adds considerably to the complication of a valve-gearing which uses an independent cut-

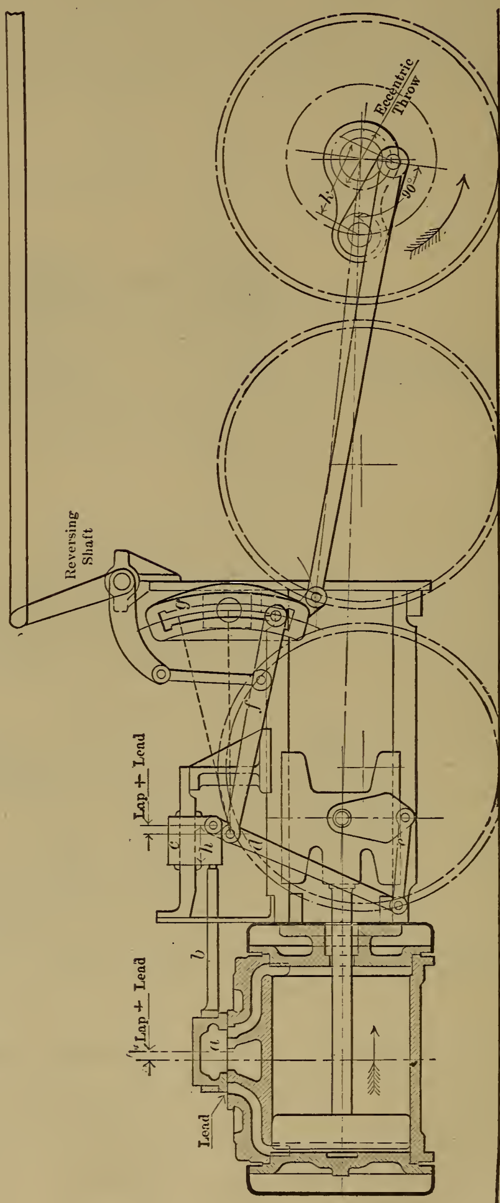


FIG. 629.

off valve when it is required to reverse the motion of both valves. The cut-off valve may have its independent link-motion coupled to the reversing-levers so that one motion reverses both the main and the cut-off valve. To avoid this complication many designers have arranged the cut-off valve to work with an eccentric 180° distant from the main crank, so that the cut-off valve works equally well with forward and with backward motion of the main valve.

Link motions for locomotives operated with cut-off valves are identified with the names of Polonceau, Gonzenbach, and Meyer. The student is referred to special treatises for study of their peculiarities.

470. Power Reversing-gears. The Stephenson link-motion has been a favorite valve-gear for marine engines, for reversing rolling-mill engines and similar massive designs. The weights and masses to be

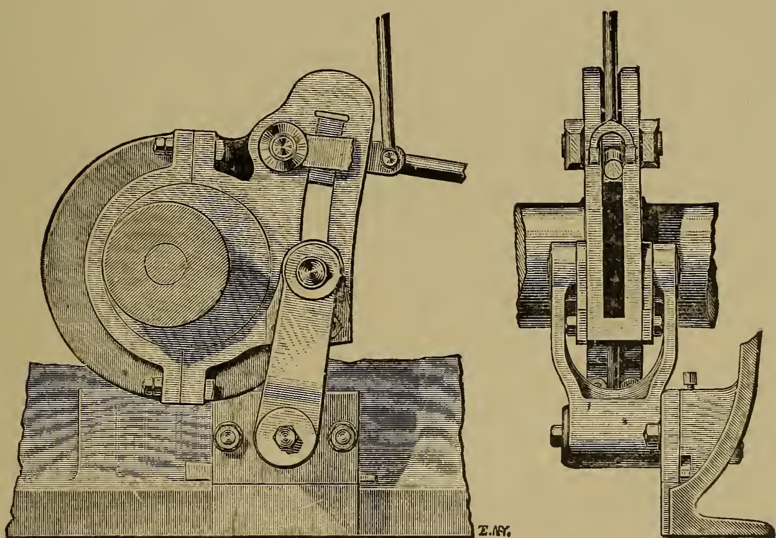


FIG. 630.

moved and the necessity for quick action have compelled designers to apply mechanical power to reverse the link-motion. Steam power or hydraulic pressure have been the usual methods. Steam power has been applied first by means of a reversing-engine on whose shaft was a screw. The nut of this screw travelling in one direction or the other moved the link into forward or backward gear. The second plan is to attach the rod of the tumbling or rock-shaft to a steam-piston in a cylinder. This would be a too rapid reversing motion, so that it must be controlled for speed and the piston must be held still or latched at the desired point of the motion of the link. This is attained by attaching to a prolongation of the piston-rod a second piston which moves in a cylinder filled with water or oil at both ends. The motion of this oil-piston from one end of the cylinder to the other will be controlled by the passage of the oil through a pipe connecting the two ends of the cylinder through a valve. The velocity of motion is controlled by the greater or less opening of this valve, and when the valve is shut the piston is locked in place and the link is held. The third form applies the principle of steam steering-engines to the link-motion. The motion of the engine to throw the link is continually closing the admission-valve of this auxiliary engine, so that continuous motion of the hand is necessary to keep the link moving. When the hand stops the engine stops. This prevents the attendant from jumping the valve-gear.

Hydraulic-pressure reverse-gear is available where water under sufficient pressure can be had from pumps or accumulator. The power cylinder is sufficient with hydraulic pressure, since a closure of both inlet and outlet valves to the piston locks it rigidly in place and holds the link at the desired position. The piston-rod of the hydraulic cylinder either throws the link directly, or operates a tumbling or rock-shaft to which the link is connected by rods.

CHAPTER XXIX.

VALVE-GEARS FOR VARIABLE CUT-OFF.

471. Variable Cut-off and Throttling Control. The treatment in Chapter XVII concerning expansive working of the steam (paragrafs 297–299, and 304, 305) should have made it clear that as the work varies the area of the indicator card should be made to vary with it, so that the speed of the engine shall be kept constant. This might be done either by throttling the steam-pipe area by a valve without modifying the timing of the point of cut-off; or the point of cut-off might be varied without affecting the initial pressure of the entering steam. The first or throttling system is independent of the design or functioning of the valve-gear: the second system works through the valve-gear and the design of the latter is conditioned by this fact. The argument of paragrafs 299 and 305 showed this automatic control by the governor of the degree of expansion to be of advantage when the expense of first cost was justifiable. Hence a study of the methods by which the governor of the engine can be made to vary the expansion may precede a study of the governor mechanism itself. The same methods apply also to varying the point of cut-off or degree of expansion by hand or through human intervention. It will not be worth while to use an automatic governor control of cut-off in engines of propulsion, such as the locomotive, the motor-vehicle, or the marine engine; nor in engines for pumping or for hoisting where the work does not vary irregularly; nor for many uses where the variation is in starting only, and while the engine man will be in attendance in any case. The same is true where close regulation to speed is not necessary or where fuel economy is not significant.

In general there will be four methods used to attain variable cut-off whether by hand or by governor:

1. To vary the throw of the valve, in the sliding type.
2. To vary the lap of the valve, in the sliding type.
3. To vary the angular advance of the valve.
4. To vary the point of trip or release, in both sliding and lifting type.

472. Cut-off Varies by Varying Throw of the Valve. It will have been apparent from the solution of the problem in paragrafs 438, 439, that with the lap fixed by the invariable length of the valve, the shorter the throw relatively to this lap, the earlier the cut-off. Hence with all the link motions which have a reversing function the cut-off becomes earlier as the gear is hooked up, making the throw less. This is done by hand, or in such designs as the Allen or Fink link of paragraf 468 it may become automatic by causing the governor, as the engine speeds up, to lower the valve-stem operated by the slot in the link, so as to diminish the throw. The slider in that slot can be raised and lowered by hand, if desired, by having it mounted upon a screw.

The second great method of securing cut-off by varying throw is to arrange the eccentric so that the effective valve-crank can be made less or more. In the discussion of shaft-governors hereafter it will be seen that it is quite easy to vary the eccentricity of the eccentric without changing its angular advance by means of an equilibrium between revolving weights and springs (Fig. 651). The eccentric can be made to have a variable eccentricity by mounting it upon the outside of another eccentric which shall be adjustable under the outer one. The effective eccentricity of the outer one can thus be varied between the sum and the differences of the eccentricities of the two eccentrics. These two systems fall short of the ideal in that the exhaust functions are modified also as the cut-off varies, and release and compression are not constant with variable cut-off.

The third method of varying the throw is to be met in cam valve-gears. The profile of the cam can be made to be different at different transverse sections. A mechanism which slides the cam underneath the roller which it drives will cause the valve to open farther and remain open longer when the valve-stem is driven by the wider and more prominent profile of the cam (Fig. 611). This lengthwise sliding of the cam on the cam-shaft can be done by hand or by a governor.

473. Cut-off Varies by Varying Lap of Valve. The discussion of lap in Chapter XXV, which showed it to be a matter of construction of the valve itself, might appear to indicate that the lap of a given valve is not a variable. This is true for a valve-gear dependent upon one valve only, or in which the steam-ports and cut-off edges of the valves are parallel. The discussion in paragraf 438 will have shown that as the lap is increased the cut-off becomes earlier.

Fig. 631 shows the form of riding cut-off valve discussed in paragraf 440, having the simple expedient of making the valve in two parts, which are attached to the valve-rod by being fitted to screws on that rod. It will be noticed that one screw is right-handed and the other is left-handed. When the rod is turned around its axis by a hand-wheel or through similar means outside of the valve-chest, the two blocks are drawn together or separated according as that motion is right-handed or left-handed. A swivel-joint in the valve-rod permits this motion of adjustment, and an indicator bearing a graduated scale can easily be attached to the valve-stem connection, so as to indicate the effective length of the valve from out to out, and the point of cut-off which belongs to each particular length of the valve. This is called the Meyer cut-off.

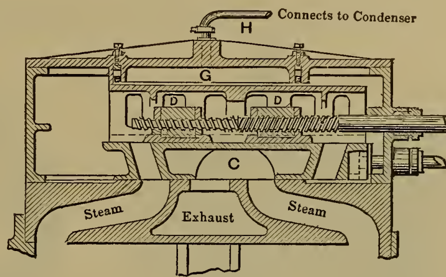


Fig. 631.

For adjustment of cut-off over a wide range, this compels a number of turns of the screw and stem, and hence is an inconvenient thing to effect by the governor. To get around this, the ports in the main valve have been inclined to each other on the upper face, and the riding-valve made trapezoidal in plan. It will be apparent when the spindle AB in Fig. 632 is revolved through a small angle, the effective length of the valve between cut-off edges is varied, and the cut-off made early or late. The riding-valve can be plane (Fig. 632) or cylindrical (Fig. 633).

A scheme for securing an equivalent for the variation of the lap is represented in Fig. 634, in which it will be observed that the steam-edge of the port is made with a false seat to which motion can be imparted through the rod C. As cut-off takes place with the outer edge of the valve as it approaches its central position from its extreme throw, it will be apparent that to have the valve-seat moved to meet the valve is to produce the same effect as lengthening the lap of the valve over a stationary port. It is only necessary that provision should be made to vary the angular advance of the eccentric which drives the rod C. This shows a balanced valve also (paragraf 447), as well as Fig. 631.

474. Cut-off Varies by Varying Angular Advance of Eccentric. To avail of this method to vary the cut-off, the eccentric cannot be positively fastened to the shaft. There must be some provision similar to the methods described in paragraf 460 to adjust the relation of the eccentric to the crank, or the mechanism of the shaft-governor (see Chapter XXX) must be so connected to the eccentric as to produce this effect. The objection will be that, while cut-off will become earlier with increasing angular advance, the exhaust events are distorted. An exception of note is to be met in the valve-gearing of the Buckeye engine, in which the ingenious expedient has been adopted of mounting the cut-off valve mechanism upon a rocking-arm which is a part of the main-valve gear. Increasing degree of expansion without interference with other functions follows from simple change of the angular advance.

475. Cut-off Varies by Varying Point of Release or Trip. This form of variable cut-off gear has been fully discussed in Chapter XXVII. The primary intent of most trip-gears is to have the period of the release of the admission-valve variable at will while keeping release and com-

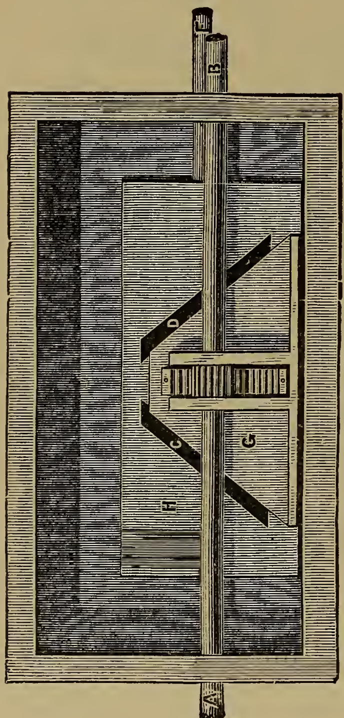


Fig. 632.

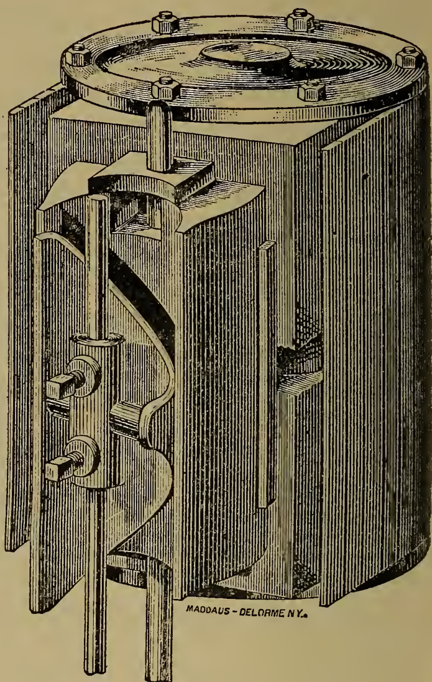


Fig. 633.

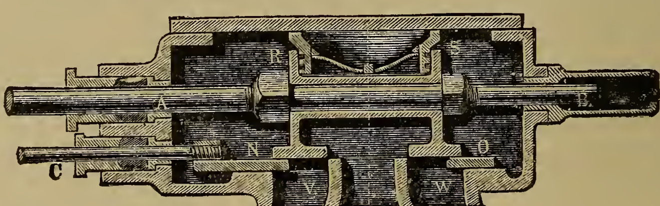


Fig. 634.

pression constant. The cam valve-gears can be similarly made variable by so arranging the cam itself or the lever which it operates that an adjusting mechanism shall cause it to come out of contact at the desired point of the stroke. The methods for accomplishing this result are very numerous and can be quite simple.

CHAPTER XXX.

GOVERNING AND GOVERNORS FOR STEAM ENGINES.

480. The Problem of Governing. In paragraf 403 the distinction was made between the function of the fly-wheel and that of the governor in the steam engine. The governor is to keep the engine to its mean speed under all variations of load; the fly-wheel is to prevent variations from the mean speed during a stroke or when the mean effort is adjusted to the mean resistance for a given stroke or a small number of them.

The governor problem appears in two categories: The first is to prevent the engine from racing or running away when the load is suddenly taken off; the second is to keep the engine to an assigned speed under all variations of load with a definite limit of excess or deficiency.

A governor which keeps the engine making its revolution in the same or equal time under all variations of the load is called an *isochronous* or equal-time governor. The governor which only precludes racing need not be isochronous. Perfect isochronism is the ideal; practical isochronism is realized when the variation from speed is only two per cent.

While governors are intended to control the energy delivered to the cylinder, it is not convenient to make them do this directly. It is easier to have the variation of speed made the element or factor which puts the governor into action. The less the variation of speed required to affect the governor, the more sensitive it is.

481. Classifications of Governors. Steam-engine governors may be variously classed. They may act to throttle the steam in the pipe (paragraf 304), or they may act to vary the duration of the admission but not its pressure (paragraf 305). The first class will be called throttling-governors, the second class cut-off governors.

Governors are nearly always founded upon an equilibrium or balance of forces at the desired or normal speed, so that the disturbance of the equilibrium due to a change of speed calls for an adjustment of their mechanism, and the motion of the adjustment alters the distribution. A direct relation between speed and centrifugal force has long induced designers to plan their governors in dependence upon the energy generated in revolving weights by centrifugal force. A second classification, therefore, would be to divide governors into centrifugal, inertia, and resistance governors according as variation in speed is desired to produce a variation in equilibrium between these forces and some other in opposition to them. The class of centrifugal governors may be divided into two according as the acceleration due to centrifugal force is balanced by the force of gravity or the tension of springs. The spring-governors are sometimes called balanced governors, because most of them will work in any position. The resistance-governor is operated by a variation in the resistance offered to motion by some part or organ of its construction. This is most usually done by the use of a fluid, when the governor becomes a fluid governor; or by a braking action which is stronger or weaker than the normal according as the speed increases or diminishes. The inertia-governors depend upon the principle that the variation in inertia of a revolving mass follows instantly upon a tendency to vary the speed; and change in position following the change of inertia adjusts the mechanism.

Governors may be classified again according to the method adopted to effect change in the valve-gear or the distribution. Under this grouping they would appear in three classes: The first and most generally used might be called position-governors, in which the weights or masses produce their effect to diminish or increase the energy admitted to the cylinder as the position of these weights is varied by the preponderance of weight or some other of the forces which are in equilibrium at normal speed.

The second group would be called disengagement-governors. These are of several types, but their underlying principle is that at normal speeds the governor is without effect upon the regulating-train or is disengaged from it. As the speed varies above and below that of normal rate the governor engages or puts in motion a train of mechanism whereby the supply of energy is diminished or increased. This is a specially useful type of governor for water-wheel motors, but can easily be applied to engines. It will be seen, however, that it is likely to be a better type for safety against racing than to secure continuous isochronism.

The third group in this class will be called differential governors. In this class a certain normal speed is fixed by braking, or a uniform resistance or a separate mechanism, and when the governor revolves at this speed it is without effect upon the regulating-train. Above that speed or below it the difference causes a motion of readjustment to take place, and this difference according as it is positive or negative closes or opens the supply of energy.

Governors may be divided again according to the arrangement or disposition of their mechanism. This gives rise to the division into spindle-governors, which revolve around a vertical axis or spindle, and shaft-governors, which revolve in a vertical plane with the main shaft of the engine as their axis, or around an independent horizontal axis. The class of shaft-governors requires no connecting mechanism between the engine-shaft and the governor. The spindlė-governors are connectėd to the engine-shaft by belting or gearing or both.

482. The Mechanics of the Governor. The conical pendulum was applied as a governor by James Watt, and is often called the fly-ball or Watt governor in the form in which he used it as a means to prevent racing. Fig. 640 shows three types

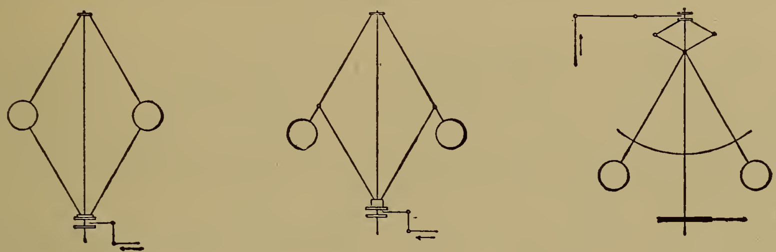


Fig. 640.

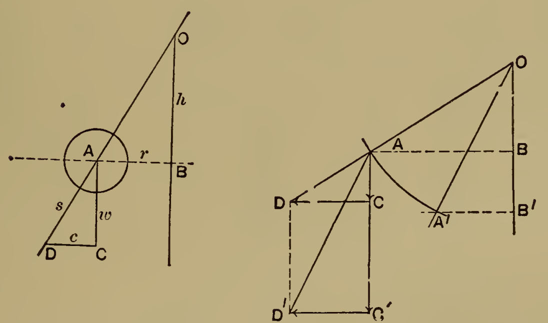


Fig. 641.

of arrangement, the rise of the plane of the balls as they revolve, lifting the sliding collar and closing the valves to which it is connected. In Fig. 641 let:

w = the weight of the ball.
h = the height of O above AB in inches.
H = the height of O above AB in feet.
r = radius of ball path in feet.
c = centrifugal force acting horizontally.
n = number of revolutions of per second.
N = 60 n = number of revolution of ball per minute.
v = linear velocity of balls in feet per second.

Then, from similar triangles

$$\frac{H}{r} = \frac{w}{c} = \frac{w}{\dfrac{wv^2}{gr}} = \frac{gr}{v^2} \text{ and } H = \frac{gr^2}{v^2}$$

But since $v = 2\,\pi rn$

$$H = \frac{gr^2}{4\pi^2 r^2 n^2} = \frac{0.816}{n^2}$$

whence

$$h = \frac{35230}{N^2}.$$

That is, the height h depends only on N, and is independent of the weight of the balls, length of arms, or other structural details. If these values for h be computed for successive changes in N of 10 revolutions per minute, the following table results, from which it appears that the change of height which is depended on to close the valve falls off rapidly as the height decreases; or that regulation is much more prompt as the height h is kept large. That is the same as to say that the governor is more sensitive as the height h is kept large, or the angle AOB at the vertex is kept small. This might have been foreseen, since the centrifugal force varies with the speed, and the centripetal force does not.

R.p.m. — N	Height h in Inches.	Change $h' - h'$.
60	9.79	....
70	7.19	2.60
80	5.51	1.68
90	4.35	1.16
100	3.52	0.83
110	2.91	0.61
120	2.45	0.46

To keep the apex angle AOB small, and the height h large, the expedient of an extraneous force additional to the action of gravity on the balls has been introduced to *load* the governor. That is by a weight on the spindle or by a spring the former force w becomes $w' = AC'$ in the right hand of Fig. 641. Then OB becomes $OB' = h'$ and if the extra load on each ball be called $\frac{W}{2}$, so that $w' = w + \frac{W}{2} = mw$

$$p = \frac{35230}{N^2}\left(\frac{w + \frac{W}{2}}{w}\right) = \frac{35230}{N^2}\left(1 + \frac{m}{2}\right).$$

The governor can now be speeded up with advantage and yet keep the same power, as the weight of the extra load is not exposed to centrifugal action. If the loading weight be applied by a leverage of any sort to give it a mechanical advantage, and it moves y times as fast as the balls do, then $h = \frac{35230}{N^2}\left(1 + \frac{ym}{2}\right)$

In the usual Porter governor (Figs. 500, 654A), the links are made of equal length, making y therefore equal to 2; so that

$$p' = \frac{35230}{N^2}(1 + m).$$

If the extra loading is done by a spring instead of a weight the simplest case is where the spring has no initial tension when the balls are down, and it is compressed upon their rise. Then if the link is l feet long, and the height when the spring is compressed to balance the centrifugal excess over the weight is H feet, then $l - H = \frac{y}{2}$ and the compression in the spring compressed y feet may be called Py. Then as before

$$\frac{H}{r} = \frac{W + Py}{c} = \frac{W + 2\,Pl - 2\,PH}{0.00034\,Wrn^2};$$

hence

$$H(0.00034\,Wn^2 + 2\,P) = W + 2\,Pl$$

In a horizontal governor W disappears from the numerator so that

$$H = \frac{2\,Pl}{0.00034\,Wn^2 + 2\,P}$$

for such horizontal spring governor.

No gravity governor of the simple conical pendulum type can be truly isochronous. It can be made sensitive, and so approach isochronism.

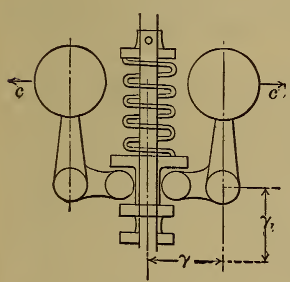


FIG. 642.

The crossed arm governor, approximating to the parabolic arc for the path of the balls is approximately isochronous (Fig. 643). The nearer the points of pivot of the arms are to the spindle the more stable or sluggish the governor. The vertical center of the arms is their intersection.

By eliminating gravity and substituting spring action a true isochronism can be secured. If the type be chosen which is used in the Parsons-Westinghouse turbine (Fig. 473), and shown in diagram Fig. 642, let c be the force on each ball due to centrifugal effect and p the pressure from the spring upon that ball to draw it back. Within a restricted range of flexure of the spring p will increase directly as r; so that p can be called Kr, where K is a constant dependent on the stiffness of the spring. The link radius r_l is often made equal to r, but may be greater or less. Calling $r_l = ar$, then:

$$cr_l = pr \text{ and } 0.00034\,Wr^2an^2 = Kr^2 \text{ giving } n^2 = \frac{K}{0.00034\,Wa}$$

Now the weight W of the balls is constant for any given governor, and a is a constant for that design; or for that governor the quantity n^2 or n is a constant. From which the deduction is that there is only one speed at which there is equilibrium of spring pressure and centrifugal force. Above that the balls tend to fly out; within it, they tend to come in by the spring. Any speed change from the normal sends the valve open wide or tight shut, and tends to make the engine keep to the desired speed. In other words the governor is always at work except at the normal speed. This is the reason for the vogue of the shaft governor.

483. Defects of Governors. The mass or weight required to overcome the resistances of the valve or gear which the governor adjusts, or which give power to the governor, is the occasion for an inertia effect. This slows down the response of the governor to change of speed, and makes it sluggish. The same mass or weight causes friction at the joints or rubbing surfaces; lack of effective lubrication also tends to sluggishness. Lack of power to operate its valves and gear is also a defect. Hypersensitiveness in governors intended to be astatic, or give isochronism in the engine, keeps the governor chasing over its range of position, so that it is never at the point of control at mean speed desired, but is "hunting" the engine, and always out of its place. This is corrected by dash-pots, which will be seen on many of the illustrated engines. To reduce the defects of sluggishness and inertia, the following designs are in use.

484. The Loaded Governor. Porter's Governor. The discussion of paragraf 482 showed that an extra weight on the axis of the governor spindle, which was not exposed to the centrifugal effect, enabled the governor to have a higher rotative speed and sensitiveness, but without the injury of an inertia effect, or adding to friction. Figs. 644 and 654A show loaded governors. The pear-form of the weight is characteristic of the Porter design.

485. The Parabolic Governor. The parabola has the property that its subnormal is constant and equal to the parameter. If the balls be made to travel outward upon an arc of a true parabola, the height h in the treatment of paragraf 482 is this constant subnormal: hence at any speed, the balls will be indifferently at the extreme outward position, tending by their position to shut steam off completely; or they will be all the way in, opening the steam admission wide, and this without any change in the value of n or N. Such a governor will readjust the steam distribution with startling rapidity, but it is hypersensitive, or too sensitive to be of practical use. The effect of sudden changes of load with such a governor would be to introduce momentary departures from the normal or mean speed. This difficulty of the exact parabolic governor is corrected in two ways. First, by attaching a dash-pot to the governor-spindle, and secondly, by the use of approximate parabolas for the path of the balls. The dash-pot method attaches to the adjusting spindle a piston which fits in a small cylinder filled with oil. The resistance offered by the oil to displacement from one end of the little cylinder to the other through and around the piston serves as a brake, to prevent jumping or racing or hunting, while no real resistance is offered to changes of position. The approximate parabolic governor is sometimes called the cross-armed governor. The suspending links are hung from points which are the centers of a circle whose radius is the radius of curvature of the parabola for that part of its arc over which the ball is to travel (*BC* in Fig. 643). This type, first introduced by

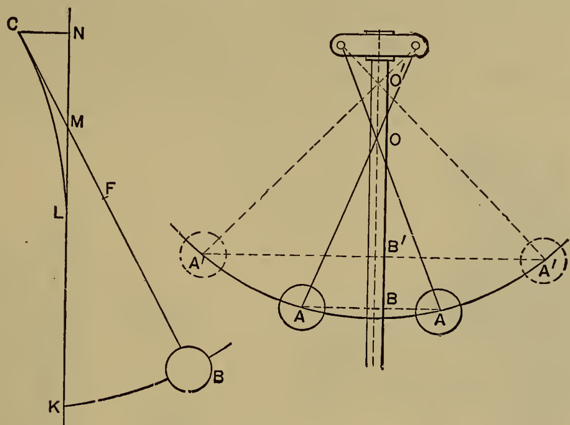


Fig. 643.

Farcot in France, has been widely used. Greater power can be given to such a governor by loading it. Fig. 644 shows the Steinlen loaded approximate parabolic governor.

486. Balanced Governor without Spring. Many forms of governor have been devised to secure an approach to isochronism by aiming at balancing the effect of gravity in part and thus make the governor more acutely sensitive to changes of speed. The direction in which this has been sought most frequently is to connect a second smaller weight to the suspending link on the opposite side of the vertical spindle. This arrangement has taken many forms, but perhaps that shown in Fig. 645, which shows the Buss governor, presents a European type as well known to Americans as any other of its class. The Babcock & Wilcox governor, shown in Fig. 646, will stand as representative of another solution, in which the weight of the balls is eliminated from the forces in action by the connection through the radius-rods *P* to the revolving spindle.

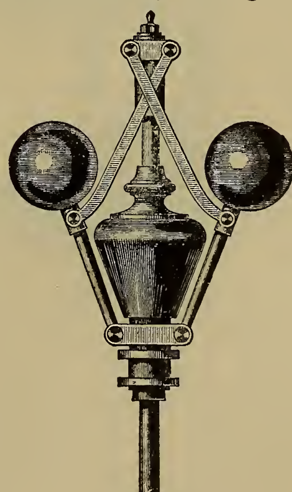


Fig. 644.

Since the lengths of the rods *n* and *P* can be so related to each other that *P* shall be one-half the length of *n*, a parallel motion will be formed so that the balls fly in and out, not in arcs of circles as in previous spindle designs, but in a horizontal plane. They do not have to be lifted, therefore, in order to travel in a larger circle, and an increased speed is not needed to maintain them in their advanced position. That there may be a force to bring them in, the spindle is lifted by the weight W operating through a bent lever. The proportions of this lever and

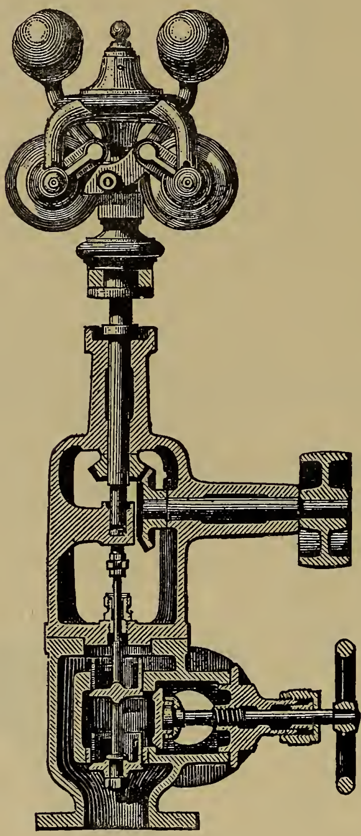


FIG. 645.

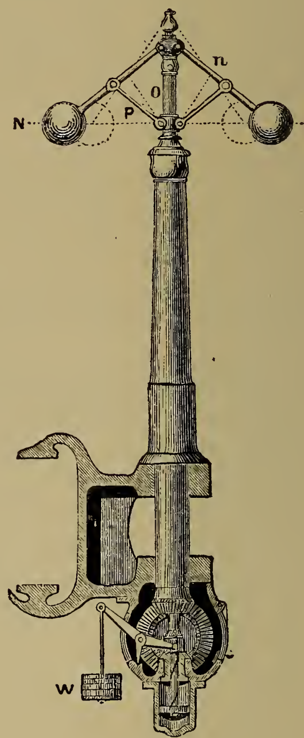


FIG. 646.

the variation of its arms are so adjusted that the centrifugal force at any given speed will just balance the weight in all its positions. Any increase in speed will cause the balls to preponderate, and a diminution of speed will cause the weight to preponderate. By connecting the spindle to the cut-off mechanism, the cut-off will be changed until the speed comes again to the standard, where the force resident in the weight balances the downward pressure on the spindle due to the centrifugal force of the balls. By increasing the weight W or diminishing it the desired speed can be varied. The dash-pot serves to prevent instability or jumping.

487. Balanced or Spring Governors. A much nearer approach to isochronism is made by those forms of governor which substitute a spring for the force of gravity to draw in the balls and open the valve when the speed falls. This has been a very fruitful field for governor designs,

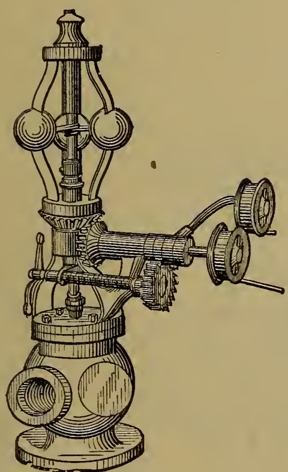


FIG. 647.

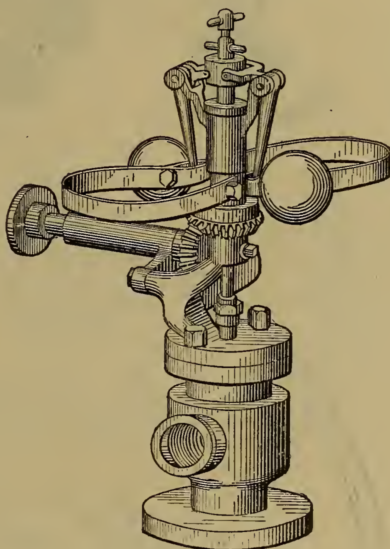


FIG. 648.

and successful spindle-governors and all shaft-governors depend on this principle. They approach isochronism more closely because the tension of the spring can be made to increase as the centrifugal acceleration increases, so that the revolving weight and the spring are in equilibrium only at the normal speed.

Early forms of successful spring-governors of the spindle type are Pickering's, Fig. 647, and Waters's, Fig. 648, Gardner's and Wright's, Fig. 649. In Pickering's governors the jointed link of the typical fly-ball spindle-governor is replaced by a flat steel blade to which the balls are secured rigidly through their centre of gravity. There are usually three balls, and the curve of the springs is such that in action they take the curve known as the cyma-reversa. In the Waters governor the balls are similarly mounted on flat-blade springs which are bent before fixing to the spindle into the form shown. The object in both cases is to get a balance between centrifugal force and the resilience of

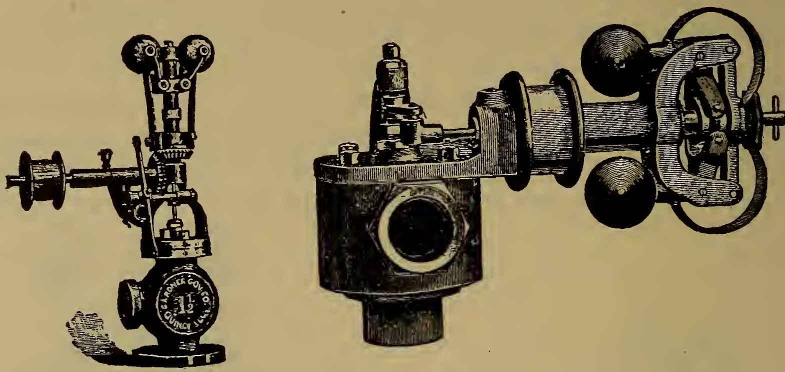


Fig. 649.

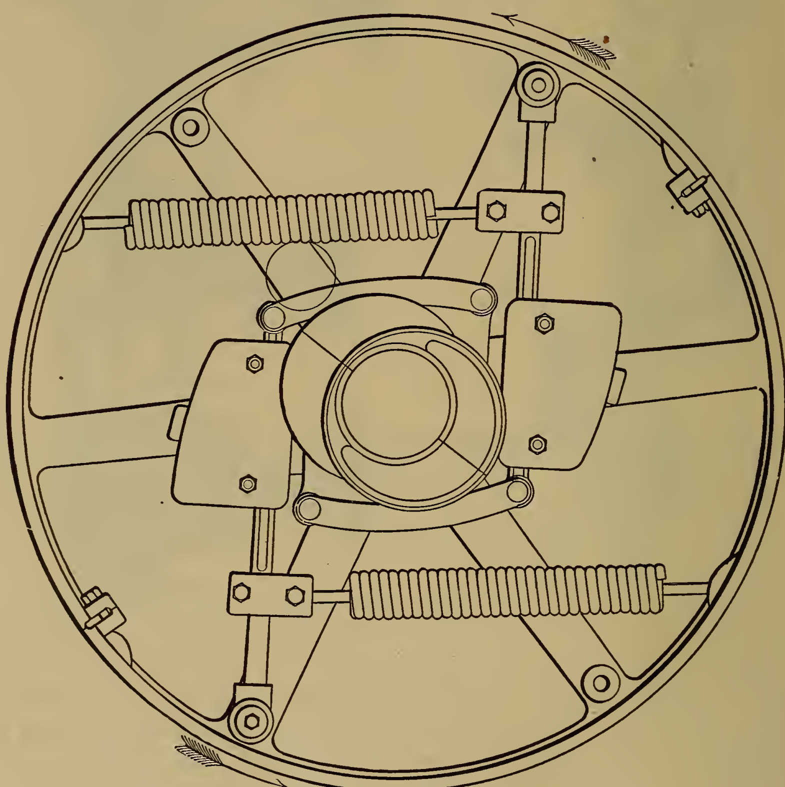


Fig. 650.

the spring at the normal speed only, and a preponderance of one effect over the other at all other speeds. In these designs the balls are small and light and revolve at high speed; and the carrying of the balls upon the springs eliminates friction at pin-joints.

The first spring governor using an initial tension of the springs was patented by Chas. T. Porter in 1861. His claim was for the idea of

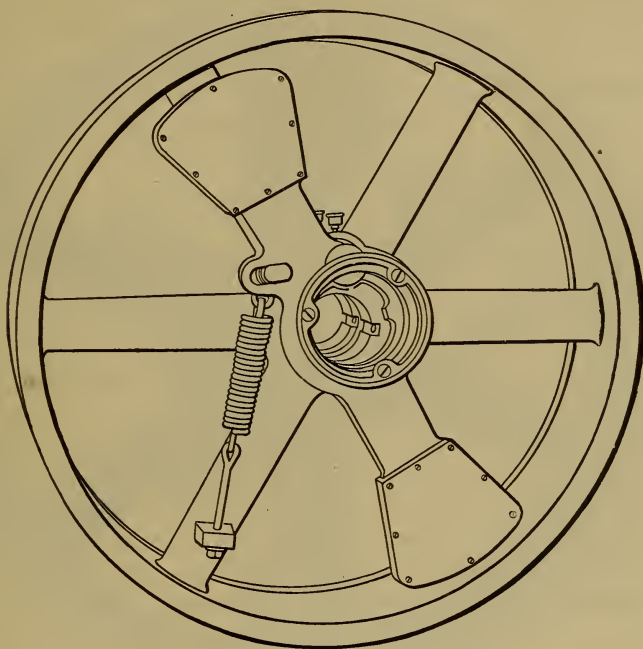


Fig. 651.

giving to the spring of a centrifugal governor an initial deflection of such amount that in every position of the balls the radius of the circle described by them and the distance through which the spring is deflected shall bear a nearly constant ratio to each other (see paragraf 482).

488. Shaft-governors. When the vertical-spindle idea is abandoned and the revolving mass is attached to the horizontal shaft of the engine so that it turns in a vertical plane, the balanced and spring principle is a necessity, and gravity must be eliminated. The methods pursued in the design of shaft-governors differ very widely, while yet possessing much in common. Two pivoted masses or weights are disposed symmetrically on the two sides of the shaft, and their tendency to fly outwards is resisted by springs either in simple spiral form or in flat-leaf form. The outward motion of the weights closes the admission-valve earlier, and the inward preponderance of the springs closes it later. Equilibrium exists only at a certain fixed speed, and that speed can be varied by varying the spring tension. In Fig. 650 the radial travel of the weights rotates the loose eccentric and alters the angular advance. It is perhaps more usual to swing the eccentric across the shaft so as to diminish its effective eccentricity and the throw and travel of the valve as the weights go out. This is the method of the governor in Fig. 651. Fig. 653A in the appendix shows a shaft-governor of this type in its position of early cut-off on the right and latest cut-off on the left.

489. Inertia-governors. Fig. 651 will serve as a type of the governors which are planned to produce their controlling effect by the change of position which will occur when a weighted lever *B*, pivoted at *P*, finds that the fly-wheel which carries it is lagging behind or overrunning the normal speed. At the normal speed a weighted lever occupies a certain position between the stops shown in the cut in equilibrium with the spring tension, which at rest would hold it against one of them. When the load varies the speed of the fly-wheel, the revolving weights keep on at their previous speed, thus changing the relation between the lever and the fly-wheel, and adjusting the admission mechanism until the normal speed is regained. This can also be done by mounting the weighted arm nearer the circumference of the fly-wheel, or balancing the drag or lag of the weight due to inertia by a proper spring.

The instability of inertia-governors, which is the consequence of their sensitiveness, makes it necessary that many of the forms should be steadied from too rapid fluctuation by dash-pots (E in Fig. 652).

490. Spindle and Shaft-governors Compared. The shaft-governor must be a cut-off governor. The spindle-governor may be either a throttling or a cut-off governor. The shaft-governor turns at the speed of the engine, and is valuable only at high rotative speeds. The spindle-governor can turn faster than the engine if desired, and can work at low rotative speeds. In some recent designs the shaft-governor has been geared from the main shaft so as to be run at a different speed. The shaft-governor is compact, and is directly connected to the engine-shaft, and therefore prompt in action. The spindle-governor is connected either by belt or shaft to the main shaft, and a breakage of such belt or the accident of its slipping or running off its pulley permits the engine uncontrolled to run away. The balls drop as the governor ceases to turn, and the valves open wide, letting full power on the engine.

491. Resistance-governors. The class of resistance-governors is less in use under high-speed conditions than it was when rotative speeds were low. A very successful form of such governor was one in which the opening of the throttle-valve was controlled by a rod attached to a weighted piston in a little cylinder. A small pump operated by the engine-shaft forced oil or water under this piston, while a graduated orifice permitted it to flow back into the suction of the pump. When the engine speeded up, the oil or the water was pumped into the cylinder faster than it could flow out, so that the piston was lifted and the energy

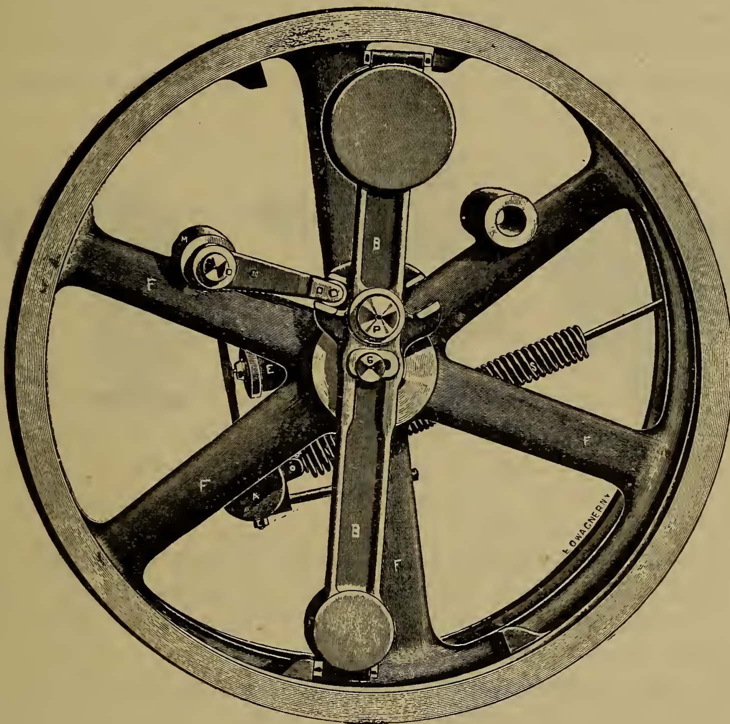


Fig. 652.

reduced. When the pump and engine workea too slowly the weighted piston descended and more energy was admitted to the cylinder.

Another form of resistance-governors has a propeller-wheel revolving in oil within the cylindrical casing. The revolution of the inclined blades tends to force the propeller shaft lengthwise, and this tendency is resisted by weight or spring. When the engine speeds up above the normal, the spring is compressed and the weight lifted; and conversely, as the speed falls the weight or spring slides the shaft. Another form replaces the propeller by a paddle-wheel which turns in oil within a ribbed casing. The paddle-wheel tends to carry the oil around with it, and the oil catching on the ribs tends to revolve the casing. This tendency is resisted by a weight acting upon an increasing leverage, so that equilibrium can only exist at a definite speed. In these two latter forms the position of the spindle and of the casing as determined by the speed adjusts the admission of energy to the cylinder. (Fig. 655.)

Resistance-governors are isochronous in principle, but lack sensitiveness to respond instantly to minor variations of speed. The objection to them is that they absorb continuously a certain amount of power, while in the balanced types, when no rearrangement of forces occurs, nothing but friction has to be overcome. Resistance-governors will

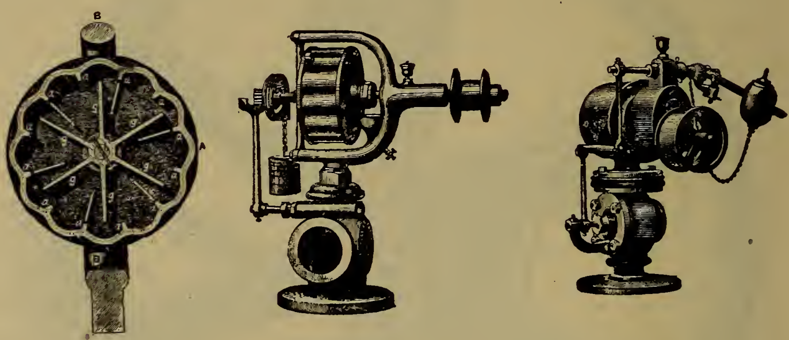


Fig. 655.

become large in proportion as the density of the fluid decreases. This has stood in the way of attempts to make fan-governors which would revolve in the air. The superior viscosity of oil makes is a better resistance than water.

492. Electromagnetic Governors. Governing devices of this sort have been applied with success in central-station work, both with steam and water as a source of motor energy, where the resistance is the generation of electric current by dynamos. In this case the speed and power of the engine are controlled directly by the resistance by simple devices. A governor of such sort consists of an electromagnet or solenoid to which current is supplied from the line wire. When the electromotive force rises beyond the normal, a motion of the armature towards the magnet takes place against the force of the weight or spring. The latter is so adjusted as to hold the armature in a fixed position at a normal speed and intensity of current. It is only necessary to connect the armature to the valve-gearing by convenient means. When the spring is in excess there is too little current, and more energy should be admitted. When the magnet is in excess there is too much current, and the energy of the engine should be cut down. Governors of this sort will vibrate on each side of the mean intensity of the current and keep up a perpetual approach to isochronism.

493. Dynamometric Governors. Designers of governor appliances for their engines have sought to make the resistance control the effort in the cylinder directly, and without having to make use of variation of speed indirectly to control the effort. While the electromagnetic governors just discussed (paragraf 492) belong to this class in one sense, they are indirect methods except where the work of the engine is the generation of electric energy. The best known attempt to solve this problem directly was to make the belt-wheel a sort of transmission-dynamometer. The belt-wheel was not keyed to the shaft, but was driven by the latter through a second wheel whose arms were connected to the arms of the belt-wheel by means of springs. It is obvious that with a given resistance in pounds on the belt-wheel the two sets of arms would separate until the stress in the springs balanced the resistance. From that time on the two wheels would remain in the same relative position until there was a change in the resistance, to which the springs would instantly respond and produce a new relation of position. The change in the angle between the driving arm on the shaft and the driven arm of the belt-wheel was made to vary the admission, so that the energy of the cylinder varied directly as the load. Such a governor was properly called a "weigh the load" governor. The difficulty connected with it and with the other governors by which the same object has been sought has been that the adjustment of the valves could not be controlled within sufficiently narrow limits. Even with dash-pots to deaden the oscillation it has not been convenient to secure isochronism of the engine. It was hypersensitive, and adjusted the valve-gear through a wider range than actual variation in the load required.

494. Safety-stops. It will have been noticed from the preceding that in the case of fly-ball governors the fall or drop of balls in gravity types and the drawing in of the balls in spring types are the motions by which the valves are opened wide. This fall or drop of balls will happen in belted governors when the belt runs off and breaks. As soon as the engine is released from the control of the governor, and the latter from its position admits the maximum energy to the cylinder, the engine runs away, with probable disaster in its train. To diminish this danger many forms of governors have attachments which are called safety-stops. Their object is to close the valve controlled by the governor when the latter shall have lost its normal action by some breakage so that the balls fall. They are of two kinds, mechanical and electrical. In the mechanical safety-stops the usual underlying principle is to have a detent or trip which the governor in its normal position does not touch, but which will be released should the drop of the balls permit the descent of a rod or lever to its lowest point. Such drop of the balls will release the detent, which shall permit the action of a spring or weight powerful enough to close the valve when thus released. In many constructions the setting of the weight or spring and its catching by the detent will be done by hand after the engine has reached its normal speed and the rotation of the balls has lifted the tripping-rod out of the way. In another form the spring is set by a ratchet motion, so that it sets itself after the normal speed is reached. To protect the engine from the danger of racing when a governor belt breaks, a tightener pulley which rests on the belt in normal working can be so connected to a throttling lever or quick moving valve (paragraf 241), that the fall of the tightener pulley and the arm which carries it will close the throttle if the belt parts, allowing the tightener to drop.

The electrical safety-stops usually act in essentially the same way, but the convenience for the transmission of power which is offered by electric methods permits their functions to be extended. A very practical form of electrical safety-stop has a weight or spring powerful enough when released to force the balls to the top of their range, and close off admission to the cylinder. This weight or spring is held out of action by a detent attached to the armature of an electromagnet. The armature may be held away from the magnet with a spring of graduated force, so that the normal current in the coil shall not be able to draw the armature to the magnet and thus release the weight. Overspeed, exciting the magnet beyond the equilibrium-point, will release the detent, releasing the weight and throwing the governor-balls up. This same result can be attained by differential currents. A convenient and useful extension of this idea has been to connect the releasing detent by buttons or switches to different rooms or departments. In case of accident in such department, by pressing the button or throwing the switch, the weight controlling the governor-ball would be at once released and the driving-engine would be stopped (Fig. 656).

Automatism with instantaneous action is a prime requisite of such devices, and it is very desirable that they should not have to depend upon the setting or memory of the engine-runner to be made ready.

495. Marine-engine Governors. The locomotive and the traction engine commonly use no governor. Their resistance does not vary suddenly, and a human intelligence must always be at hand to control them in any case. In marine engines, however, while in smooth waters the same condition prevails, in rough weather the pitching of the vessel may release the screw from its resisting medium and suddenly take the load off the engine. Obviously this is a source of danger both to the long and flexible shaft and to the screw itself when it suddenly re-enters the water while moving at too great velocity. Many marine engineers prefer to meet this difficulty by keeping one of their staff continually at the throttle-valve in bad weather, and no form of revolving governor

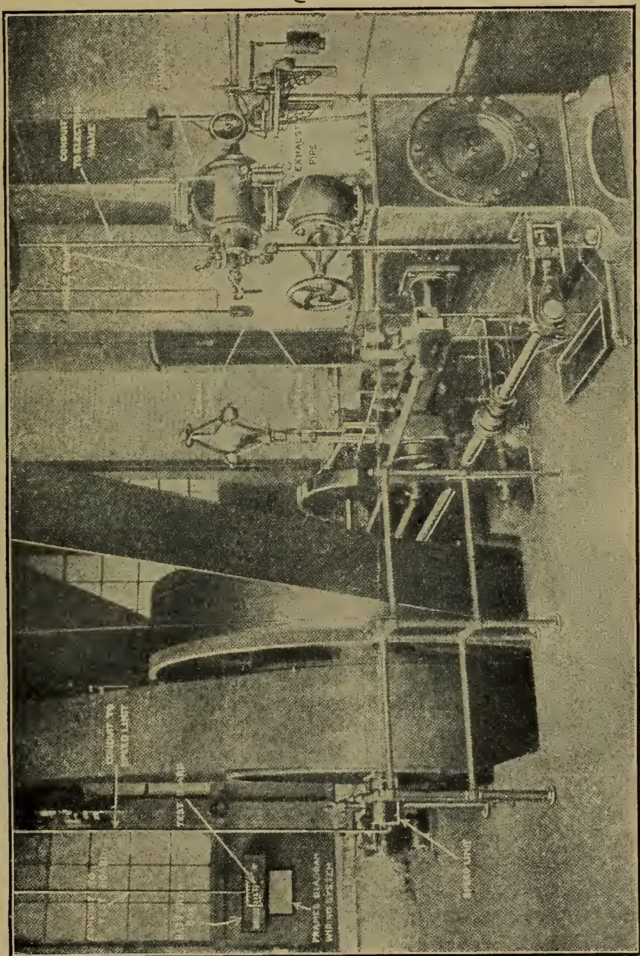


Fig. 656.

exactly meets the case. Some of the shaft-governors operating by springs independently of gravity would meet the case most nearly but for the size of the engines in question and the increased complications and weight which would be introduced. A form of marine governor which has been introduced in many marine engines is a species of pendulum arrangement operating a valve in the steam-pipe. Fig. 657

shows a general detail of such a device. When the vessel is on an even keel, the pendulum attached to a spherical casing hangs vertically, and all steam-openings coincide so as to leave free passage from boiler to engine. As the ship pitches it changes the angle of the steam-pipe, to which a fixed casing is attached, while the pendulum-ball remains vertical. The effect of this pitch or send of the ship slides the openings

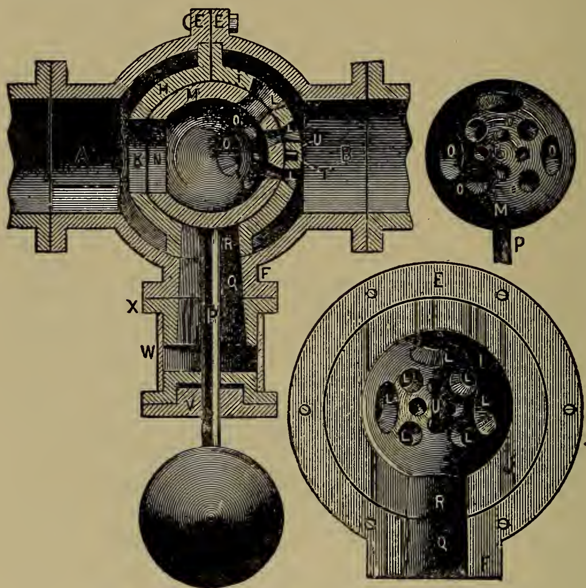


FIG. 657.

over each other, and throttles the passage for steam to the engine. If the pitch is enough to send the openings past each other, no steam can get through. The pendulum swings steam-tight by means of flexible or spherical joints at the opening through which it protrudes. Most engineers even with such a governor attached to their engines do not relax their vigilance at the throttle.

496. Connections of the Governor to Control the Engine. The student is referred to the discussion in Chapter XXIX for the methods which may be used to make the governor in any of its forms control the speed and energy of the engine. The number of combinations possible is very great, since almost any kind of governor can be applied to produce variation in the point of cut-off by the methods discussed in Chapter XXIX. The methods for hand-adjustment of such variation are usually made automatic by properly gearing the governor mechanism to the mechanism which operates the valves.

PART VI.

CHAPTER XXXI.

ENGINE AUXILIARIES. THE CONDENSER AND ATTACHMENTS.

500. Introductory. In the discussion of Power Plant Auxiliaries in Chapter XIII a somewhat arbitrary division was made into the auxiliaries primarily related to the boiler or pressure generator plant, and those attaching to the engine-room or pressure converting plant. The feed-pumps and feed-heaters for examples may be called engine-room auxiliaries with almost equal logic to that used in allotting them to the boiler-plant. In the condensing engine particularly the pumps are rather elements of the condensing apparatus, and may be treated under this head. The condenser is, however, so manifestly an engine attachment that with its attachments it falls into the class now to be considered.

In paragrafs 299, 306, and 307 of Chapter XVII the argument for selecting the condensing principle for securing low terminal and back pressure in expansive working was very completely reviewed, provided this was economically defensible. What is now to be done is to examine the apparatus required to realize the condensation desired. Reference should be specially made to objections 3, 4, 5, and 6 of paragraf 307.

501. The Principles of the Steam Condenser. The principles underlying the condenser or vacuum chamber of the condensing steam-engine, whether turbine or reciprocating, are derived from the properties of steam. If the steam exhausts into the atmosphere or into any vessel open to the atmosphere, the pressure therein cannot fall below 14.7 pounds absolute, which is the pressure above a zero of pressure belonging to one atmosphere, or at the pressure at which the air stands by reason of gravity, or the weight of air surrounding the earth. If the steam is to be reduced in pressure below this at exhaust, it must be done by leading it into a closed chamber, and tight enough to prevent leakage of atmospheric air into it from the atmospheric pressure without. Secondly, the steam must be cooled below 212° F. or 100° C. if its pressure is to be less than 14.7 pounds absolute, or one atmosphere, within this chamber. This cooling can be done by any means which shall be continuous. The vacuum chamber must be kept cool, so that its metal does not get hot from the incoming heat of the steam. Hence both the steam and the air-tight vessel must be cooled. With what? The two cooling media which are cheapest to buy and to apply are water and air. Water will be used where possible, because it has a greater cooling effect per pound than anything available known. Its specific heat is unity; or it takes less weight of water to cool a given weight of steam, and smaller pumps to handle it than any other cooling medium. Air is used where water is hard to get or carry or is costly for any reason; but it is much less effective, requires to be kept moving by a fan or rapid motion, and the surface for equal cooling effect has to be much larger. Water is therefore the standard.

When the steam meets the cool surface of metal or water and drops down temperature, a third effect is produced. The volume hitherto occupied by steam is now filled with water, resulting from the condensation due to the temperature drop; with some water vapor of the lower tension due to the lower temperature; and with leakage air, coming in with the steam, or with the cooling water, or through cracks or porosity of the metal. The change to water is responsible for much of the lowered tension, but some water has heat enough to boil at the lowered pressure and the vacuum is not theoretically perfect. The air is not effected by the temperature drop: to eliminate all watery vapor so great an excess of cooling water would be required that it is not economically practicable.

In the fourth place, the condenser or vacuum chamber, supposed air-tight, will begin to fill with the water resulting from the condensation of the incoming exhaust steam. As soon as it fills completely, the vacuum disappears and is lost, and if there is no relief valve, the exhaust steam keeps coming in and backing up pressure until it balances the boiler pressure, and the engine stops. Hence this water from condensation must be gotten out of the condenser, and the air which also came in. If this is removed by a pump, it will be called an air-pump. The cooling water for lowering temperature and therefore pressure will be called the "injection" water. The supply of injection water will come from a "cold-well." The delivery of water warmed by the cooling of the steam will be to a "hot-well." The hot water coming from the condensed steam will be returned to the boiler, and this will be done by the feed pump.

502. The Vacuum Gage. The pressure in the condenser being less than 14.7 pounds, or one atmosphere, cannot be registered by a common steam gage, whose zero point or reading is at the pressure of such one atmosphere. But a gage is easily made on exactly the same principle, except that it is designed to record the pressure outside of its spring (paragraf 171) or its tube, so that the needle reading will be the difference between the atmospheric pressure outside and the pressure within.

such tube. Such gage reads zero as it lies exposed to the air both outside and inside. When connected to a vacuum chamber it reads negatively as it were, or registers the pressure below the atmosphere in the vacuum chamber. To get total pressure then, the steam gage reading above zero or atmosphere is added to the vacuum gage reading below that same zero. The vacuum gage in pounds reads to 15; it sometimes is graduated to read inches of mercury, or up to 30 or 31.

503. The Weight of Injection-water. The injection-water must come into the system in sufficient mass per unit of time or per pound of steam to be condensed to condense first the steam to water by absorbing the latent heat of vaporization; and secondly to cool that hot condensed water to its own final temperature. Experience shows that to try and get a lower final temperature than 110° − 130° F. with an average of 120° costs more than is gained in smaller plants, without elaborate equipment. Hence if

H = total heat units in one pound of steam at the temperature of the steam as it enters the condenser from the cylinder. This is the total heat at that pressure in steam tables.

h = the heat per pound of the hot condensed water leaving the condenser. This is the temperature of that water, when the unit is one pound.

Then one pound of entering steam must dispose of a quantity of heat in heat units $= H - h$.

If now

Q = the weight of injection required:
t = the temperature at which it enters the condenser.
T = − the temperature at which it leaves the condenser.

Then if the metal of the condenser keeps at a constant temperature the equality must exist that

$$Q(T - t) = 1\,(H - h);$$

or

$$Q = \frac{H - h}{T - t}.$$

If the temperature of the air pump discharge as hot water be called t', then $h = t'$. The prevailing practice in design is to call H the total heat of steam at 30 pounds pressure, or 1190. This makes

$$Q = \frac{1190 - t'}{T - t}.$$

If the injection be put at 60° and the discharge at 110, this will make $Q = 21.6$ times the weight of steam in the minimum case. To allow for warmer injections, it is usually called 30 to 35 with jet condensers and 60 to 75 with surface types.

504. The Volume of the Air Pump. The air pump, which handles both the injection-water and the condensed water, requires to dispose of $Q + 1$ pounds per pound of steam condensed. If the water at the usual temperatures of the condenser be taken as weighing 61.7 pounds to the cubic foot, then $\frac{1728}{61.7} = 28$ inches will be the volume occupied by a pound. The pump volume per minute will therefore be for weight of steam W,

$$V = (Q + 1) \times \frac{28}{60} \times W.$$

But the actual volume to be displaced must be greater than this by an allowance for the inefficiency of the pump, and for the fact that it must handle air besides the water, and the quantity of air is not always predictable. The actual displacement will therefore become

$$V = (Q + 1)\left(\frac{28}{60} R\right)W .$$

For the air-pump for the surface condenser, see paragraf 510. The results of experience make $\frac{28}{60} R = 1$ for horizontal double-acting air-pumps and equal to 0.75 for vertical single-acting pumps.

505. The Jet Condenser. The simplest type of condenser brings the injection-water directly into contact with the steam to be condensed and the resultant water to be cooled. The most effective method of bringing them together is to divide the injection into a series of fine streams entering in jets; hence such condensers are called jet condensers. Fig. 660 shows the direct or jet condenser as used in river-boat practice where the injection comes from the water outside of the hull. It will be seen that the steam escaping from the cylinder enters the condenser at the side and near the top below a partition which runs across the condenser. This partition is perforated with a great number of holes about one-half inch in diameter. The pipe entering the side of the condenser and working upward through the perforated partition is the injection-pipe, which comes from a suitable opening in the hull through the skin of the vessel. The injection-pipe has a valve in it, operated by a lever or by a hand-wheel (see Fig. 389), whereby the flow of injection-water can be cut off and controlled. It will be apparent that if there is a vacuum in the condenser, and the opening of the injection-pipe is below the surface of the water outside of the hull, atmospheric pressure will force the injection into the condenser with considerable energy, so that the injection-valve is usually only partly open. In such river-boat engines as are presented in Fig. 389 there are usually three entrances to the injection-pipe. The usual one used will be the bottom inlet, opening through the hull near the keel

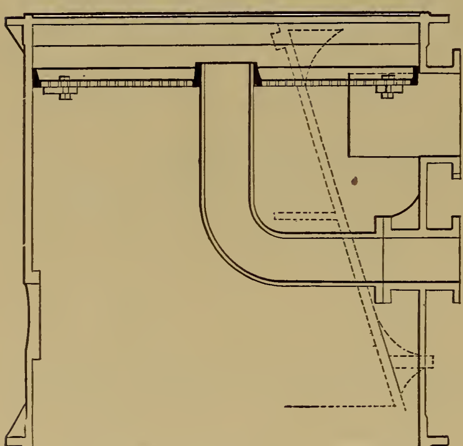


Fig. 660.

and of course always under water. The second one will be the side inlet, which will be used only when such shallow water is to be feared that there would be danger that the bottom inlet would draw in mud or become stopped with solid matter. The third inlet will be from the bilge of the boat, and will be called the bilge-injection. It will be used only when from a leak or an accident an excess of water has come within the skin of the vessel, so that the propelling engine can be used to empty the bilges and lighten the duty of the bilge-pumps proper. It will be seen that the injection descends across the exhaust-steam in a finely divided shower, whereby the least weight of water need be used. Sometimes the injection is sprayed into the steam through a simple nozzle like the rose-nozzle of a flower watering-pot (Fig. 668). The openings through the hull are protected on the outside by gratings or strainers, and in the inside are valves close to the skin which are called "sea-cocks" (Fig. 663). If the injection-pipe is broken inside the hull and there are no sea-cocks nor any means of closing them from the decks the engine-room becomes flooded with water from without.

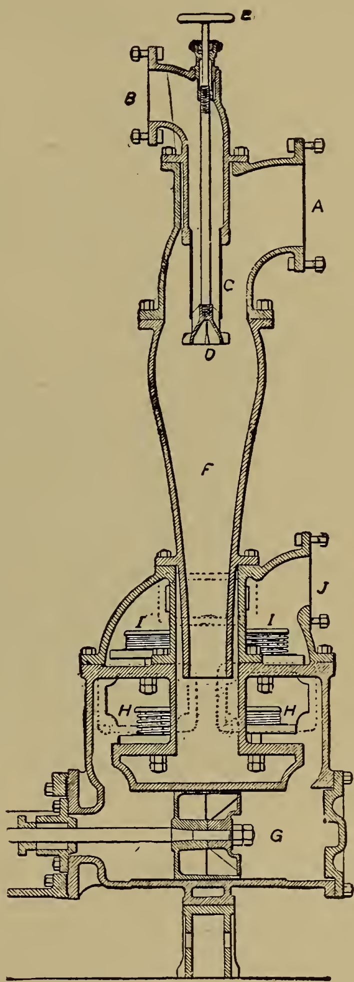


Fig. 661.

The more modern and usual form of the jet condenser for land practice abandons the idea of the larger volume of the condenser — perhaps one-half or two-fifths of the cylinder volume — and utilizes the advantages of rapid flow of the injection-water to entrain with it the disengaged air. Hence the cross-section is much less, and is made pear-shaped in form with the smaller area at the bottom to compel the more rapid flow of water and stop any tendency of air-bubbles to ascend and break the vacuum. In Fig. 661 the steam comes in at *A* and meets the injection water entering at B. The latter is sprayed by the inverted cone at D and intimate mixture is effected. The hand-wheel at E controls the thickness of the water film for best effect. The pump at G keeps up the vacuum by continuously exhausting the water from the space F. This type is more liable to quick flooding with water than the previous type of larger volume and cross-section if the pump fails

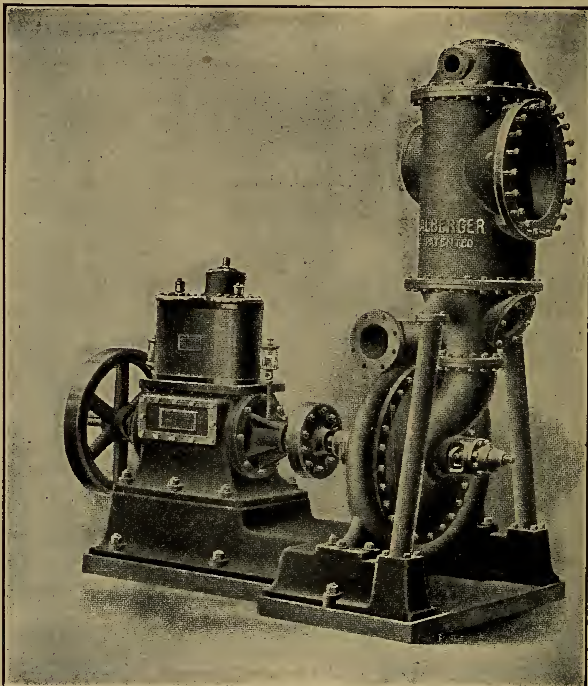


Fig. 662.

to operate from any cause. The injection backs up into A and over into the engine, where it makes trouble unless a relief valve is supplied. (Fig. 675.) The combined injection and condensed steam water pass out at J to the hot well. The left-hand portion of Fig. 663 shows such a condenser on a fresh-water vessel, with the injection entering at E through the sea-cock, and past the controlling valve D to the condenser N; thence outward and overboard through T. The pump L acts as a feed pump taking such water as is required from the pipe G under this set of conditions. The suction or exhausting pump may be also a volute or centrifugal pump, either turbine or motor or engine driven (Fig. 662).

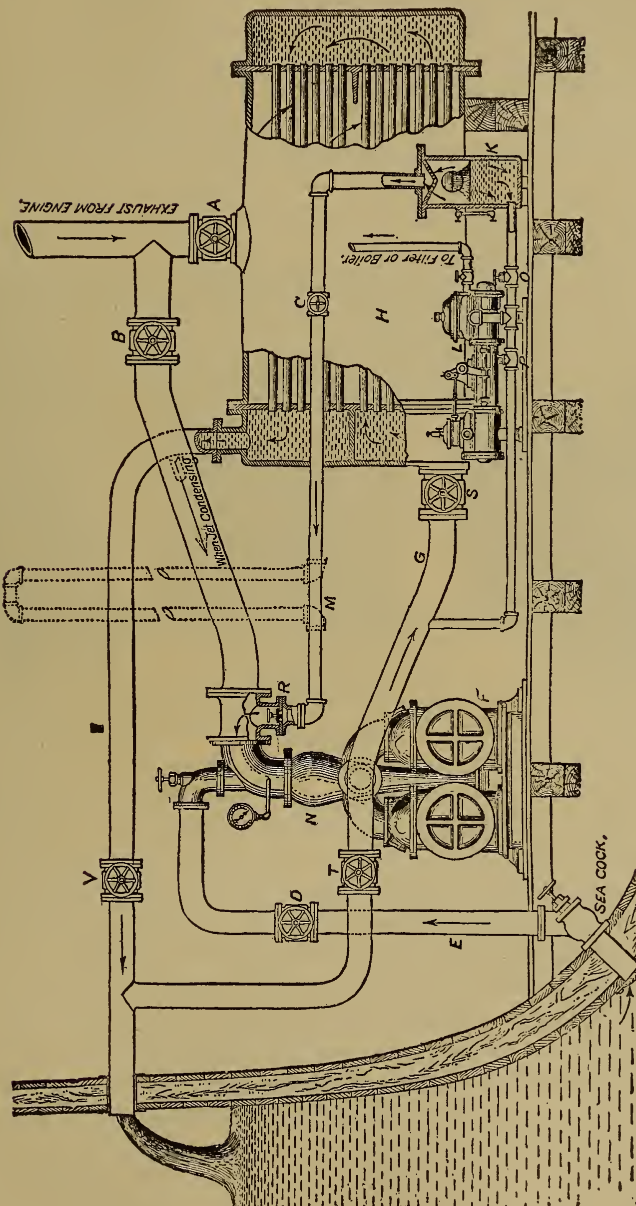

EXHAUST FROM ENGINE.
To Filter or Boiler.
When Jet Condensing
SEA COCK.
A
B
C
E
G
H
K
L
M
N
O
R
S
T
V

FIG. 663.

506. The Surface Condenser. In the other type of condenser, the injection does not meet the exhaust steam for direct contact and mixture, but the two are kept in separate circuits. The steam meets a cooling metallic surface of sufficient extent, usually of tubes of brass or copper coated usually with tin on both sides to prevent corrosion and lessen galvanic action. These tubes are one half or three quarters of an inch in diameter. Fig. 664 shows the usual arrangement of such a surface condenser in the older marine designs, which use sea-water for cooling, and in Fig. 662 at the right hand is the more modern cylindrical type, with the course of the circulating cooling water made obvious. The pump *F* will be now a circulating pump, to take in the water from the sea-cock and overcome the resistances of the tubes

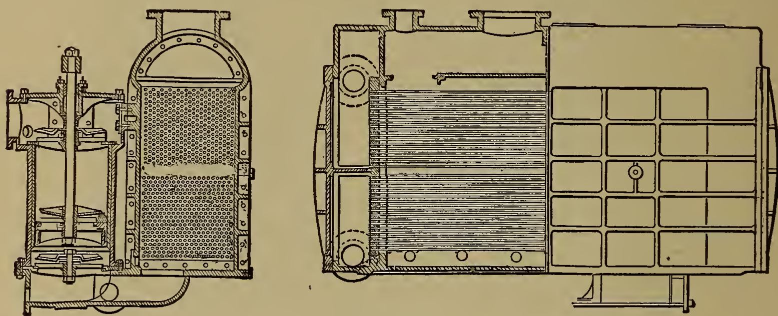


FIG. 664.

and lift the water the few feet necessary to send it overboard again through the valve *V*. In Fig. 664 the air pump is vertical and single-acting at the side of the condenser. In Fig. 663 it is horizontal and double-acting, drawing out the condensed water into the receiver *K* and pumping it from *L*, which is the air pump, back to the boiler. The air which separates in *K* is drawn off by the suction of the pump *F*. There is often no cogent reason other than convenience determining the question whether the injection-water should circulate within the battery of pipes while the steam is on the outside, or whether this plan should be reversed. English naval practice adopts the latter. It is most usual in the merchant marine to have the steam on the outside, because less difficulty is met from the clogging of the condensing surfaces by the condensation of the lubricating material on the cool surfaces of the condenser, and it is easier to clean the outside of the tubes than the inside, and the tubes can be drawn through the tube-plate more easily for cleansing. The scale from sea-water used in circulation is removable without taking out the tube; tubes can stand internal pressure better than external; the water circulates better; a large surface meets the steam; the design is simple and compact; and a packing can be used which contains organic matter. For the English plan it may be said that most of the lubricant is caught at the first tube-plate; the flat surfaces of the condenser have only upon them the light pressure of the water in circulation, and not the larger pressure of the atmosphere against the absence of pressure within; the metal of the condenser radiates less heat in the engine-room. On the other hand, packing of

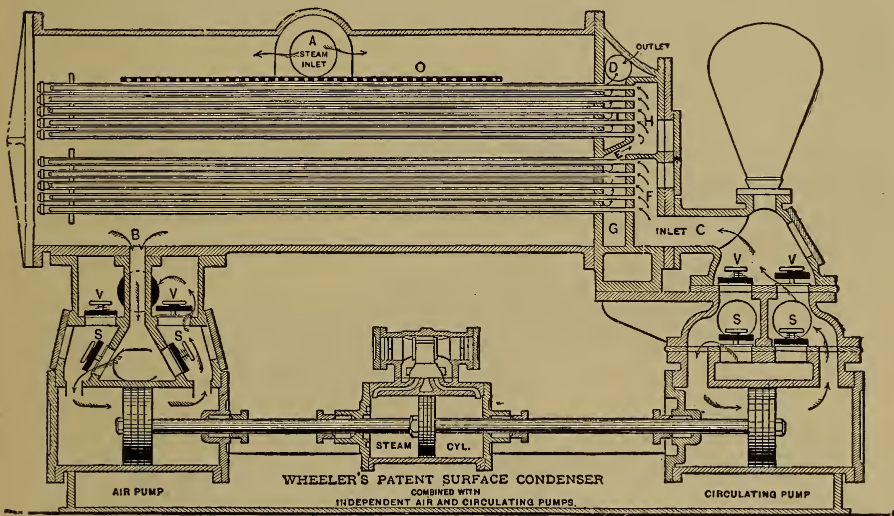


Fig. 665.

the tube-joints must be done by some device which will not be affected by the steam. The steam enters the surface condenser usually at the top, and the cold injection-water enters it at the bottom and as it becomes warm in cooling the tubes it is forced upwards so as to meet the hottest steam when it is itself warmest. This plan of having the injection travel against the steam secures the greatest difference of temperatures in all parts of the condenser as a whole, and transfer of heat is most rapid with greatest difference of temperature between the body to be cooled and the absorbent material. The condensed steam gathers in the bottom. Fig. 665 shows a form of a surface condenser designed to avoid one or two main difficulties of surface condensers. By reason of the conditions to which they are exposed the tubes are subject to changes of temperature which cause them to expand and contract, and makes it difficult to keep the tubes tight where they enter the two heads shown in the previous sketch. This has been sought in the prevalent designs by making an expanded or fixed joint at one end, and at the other fixing a species of stuffing-box kept tight with compressible packing and permitting the tube to slide.

Fig. 666A in the Appendix presents a grouping of such methods of flexible joints.

A is Howden's wick or hemp joint.

B and *C* are Lighthall's, packed with papier-maché.

D is Winton's hard-rubber ring.

E is Spencer's rubber washers.

F is Marshall's moulded rubber joint.

G is Stimer's tube.

H is Hall's stuffing-box.

I is Chapman's joint with Babbitt-metal calking.

J is a rubber washer with lock-nut.

K is Sewell's joint, compressing rubber by a cover-plate gland.

L is Archbold's, with brazed brass wire to prevent creeping.

M is Wilson's, similar to *K* except that each tube is packed separately.

N is Horatio Allen's soft wood packing.

Q is Todd's method.

The joints from *A* to *I* do not permit the removal of the tube without having to be themselves renewed. The cover-plate plan *K* packs all tubes at once.

The Wheeler condenser, shown in Fig. 665 and in detail at *P* in Fig. 666A, secures the tube tight in one end only by screwing, and the circulating water, instead of passing through the tube completely, is made to flow through the closed tubes by means of the smaller inner tube which is not attached directly to the outer. The difficulty from the tube-joints was a very serious obstacle to their first introduction on sea-going vessels. Their use now is universal, since this difficulty has been overcome.

The accepted rule for computing the number of square feet of tube surface in the surface condenser is that of Jay M. Whitham from experiments by Shock and others. In this:

S = square feet of condensing surface.
W = weight of steam to be condensed per hour.
L = latent heat of steam at the condenser temperature, from steam tables.
T = temperature of the condenser, taken the same as that of the water discharged from air-pump.
t = average temperature of injection or circulating water, taken as the half sum of its initial and final temperatures.

Then:

$$S = W \times \frac{L}{180\,(T - t)}$$

In average conditions $\frac{L}{T - t} = 17$, so that there will be required one square foot heating surface to 10.6 to 11 pounds of steam delivered per hour; or three square feet per average horsepower; or four square feet per kilowatt.

In the tropics, where the circulating water is hotter than in cooler climates, more surface will be required. In special conditions where the cooling water is unlimited as in surfaces of coils under the hull of a boat used as a condenser, 50 pounds of water will be condensed per square foot per hour. In mine or water-works pumps, where the discharge can go through the condenser without perceptible increase in temperature, 20 to 40 pounds per hour per square foot can be counted on. When reduced to gallons, about one gallon per minute per horsepower is a value much used for the amount of circulating water. It is more usual to provide for excess of circulating water over the lower quantity computed in paragraf 503 for the jet condenser, and this explains the larger figure of 60 to 70 pounds of cooling water per pound of steam condensed.

507. Jet and Surface Condensers Compared. In the jet condenser, the injection and condensed steam water are mixed; the air-pump must handle both, plus the air from leakage and entrainment. Hence the

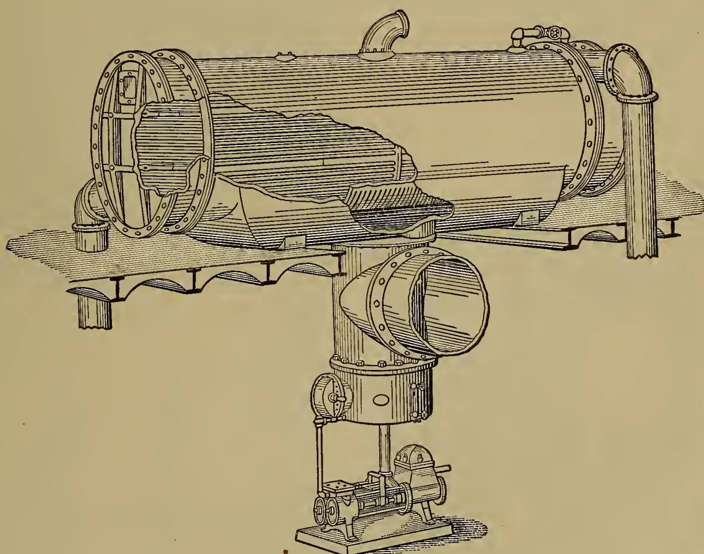


Fig. 667.

injection must be good enough to use in the boilers, and only a part of the mixture of injection and condensation is required for the boiler-feed. The rest must be wasted, or the method of feeding used which is discussed in paragraf 146.

The surface condenser, while more heavy and bulky to handle and cool a given weight of steam discharged as exhaust, can be used with any water of reasonable quality. The condensed steam leaves the surface condenser as distilled water with no impurities in it except the lubricating oil, and is therefore a most excellent material to pump back into the boiler if the oil can be extracted from it. The surface condenser has for this reason occupied the field with vessels traversing salt water, and has furthermore a wide scope on land in places where the available water contains solid matter or salts or acids which would be injurious to boilers. The same water is used over and over again, and the only addition of bad water which has to be made to that which filled the boilers in the first place is that which is lost by leakage at safety-valves, whistles, and joints. The steam-circuit is practically a closed one. The surface condenser for sea-vessels adds from ten to eighteen per cent to the first cost of the engines, but is more economical of fuel for them than jet condensers would be. In Fig. 667 is a type in which the exhaust enters at the bottom and the injection at the top. The circulation passes three times through the length of the cylinder.

508. The Cold-well Cooling Towers. In stationary practice on land the water for condensation and the injection must be supplied from a reservoir. In cities having a water-supply the city water can be used for this purpose, but ordinarily the quantity needed for a plant of considerable size will compel the engineer to consider other means. In the older designs it was very common to immerse the condenser in the tank from which the injection was to be drawn (Fig. 668); and even where city supplies under pressure are to be had, it is preferred not to connect the condenser to the mains, but to take the injection from a tank in which the supply of pressure-water shall be controlled by float-valves or ball-cocks. The expense of city water has compelled many proprietors to sink artesian or other private wells for the purpose of controlling the necessary quantity of injection-water, but even this is expensive and not always practicable. The tank from which the injection is taken was called by the early designers the cold-well, and latterly considerable pains have been taken to make it possible to use the same injection-water over and over again without making the cold-well of unmanageable size.

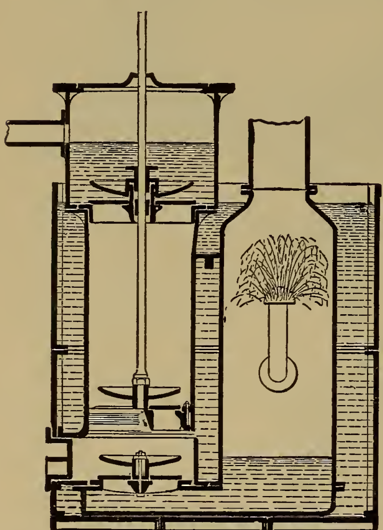


Fig. 668.

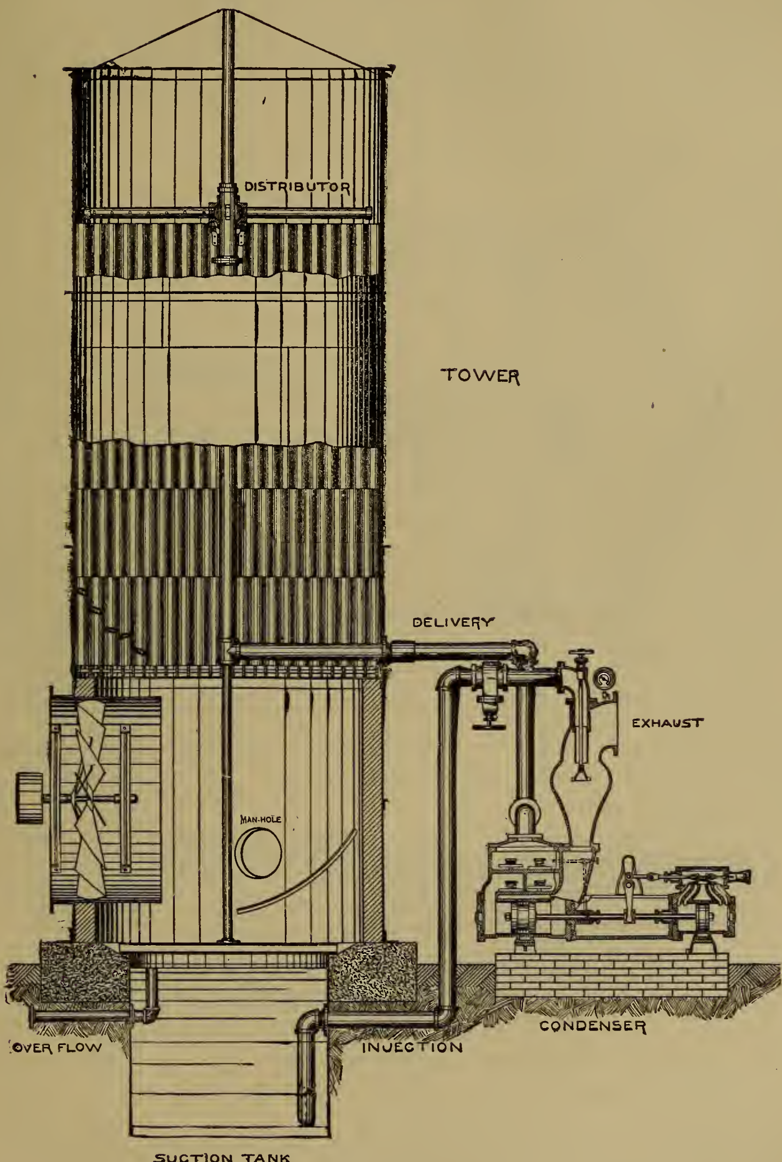

DISTRIBUTOR
TOWER
DELIVERY
EXHAUST
MAN-HOLE
CONDENSER
OVER FLOW
INJECTION
SUCTION TANK

FIG. 669.

Two general methods have been followed in the solutions which have been sought for this problem. The first has been to construct a series of shallow troughs in which the warm injection-water flowed in the open air exposed to the action of the natural winds. These troughs were arranged one over the other with a slight grade, so that the water flowed

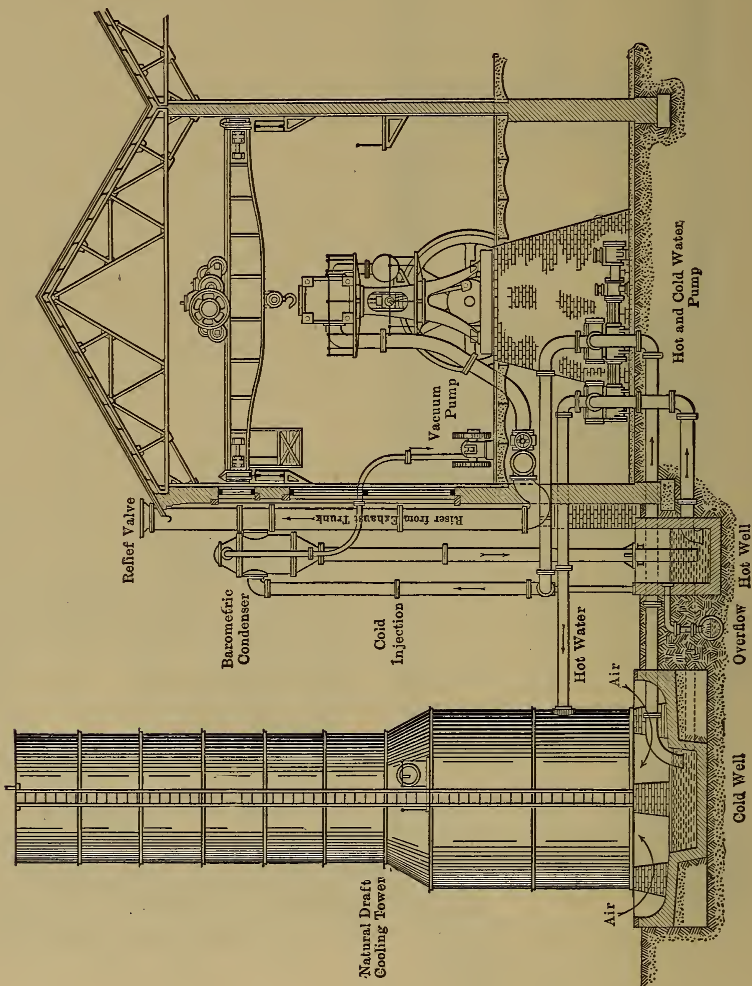


FIG. 670.

zigzag fashion from the top to the bottom, and after leaving the lower end of the latter flowed back to the well. The prolonged exposure in thin films to the vaporizing action of the open air and the cooling caused by such vaporization resulted in a considerable lowering of the temperature of the injection. The other plan, of modern introduction, is to cause the injection to descend in a closed tower in a fine state of division over tile or cypress wood or wire gauze arranged corn-cob or grating fashion with maximum surface and interstices between. A current of air forced by a fan causes a vaporization of the film of warm water pouring over the exposed surfaces, and the air-cooling and vaporization combined withdraws the heat from the injection, so that as it falls into the cold-well at the bottom of the tower it is in condition to be used again. This appliance has been in successful operation for some years, and is warranted wherever the cost of condensing water per annum without such device would exceed the interest upon the cost of the plant and the expense connected with operating it. Fig. 669 illustrates this arrangement using a fan and jet condenser and tile surfaces: Fig. 670 shows the tower at work in natural draft of air as a chimney without fan, and some fuller details of the complete installation. The condenser is here of a type to be later discussed (paragraf 513).

509. The Air-pump and Foot-valve. The vacuum created by condensing the steam by the injection must be maintained in the condenser. The condensed steam and injection would rapidly fill the volume of the condenser if no means were taken to empty it. Furthermore, all natural water and the steam contain a certain quantity of air which undergoes no condensation or reduction of volume, and whose presence in the condenser would soon destroy the vacuum created by condensation. Some means must therefore be provided to draw from the condenser the condensed steam, the injection, and the air. While the pump handles water mainly, the handling of the air is its most exacting requirement, and in starting, it pumps air. Hence it gets its name.

From Figs. 664 and 668 it will appear that single-acting vertical pumps have been used for this purpose; and from Figs. 661 and 669 that double-acting horizontal pumps can be used. The occasion for these differences will be discussed in paragraf 511.

The air-pump must meet the difficult condition of withdrawing water and air from a vessel within which the pressure is less than the atmosphere, and therefore atmospheric pressure cannot be counted on to fill its barrel as is the case with the ordinary lifting-pump. If the piston in the air-pump in Fig. 668 be supposed to be rising in its barrel, and the bottom of that barrel opens into the condenser through the valve which separates them, it will be apparent that as long as the pressure in the air-pump is greater than the pressure prevailing in the condenser the valve cannot open. The rise of the air-pump piston must create below it a rarefaction or vacuum greater than that in the condenser before the valve will open and any equalization of pressure occur. Furthermore, the water in the bottom of the condenser will only flow through the foot-valve of the pump by gravity, unless there is enough of it to seal the connection between the two volumes, and will then only rise in the air-pump sufficiently to counterbalance the differences in pressure. The bottom of the air-pump and the foot-valve must therefore be below the bottom of the condenser, so that the water may fall out of the condenser by gravity. When the air-pump reverses and begins to descend, the foot-valve closes and remains closed during the descent of the bucket. As the bucket goes farther down it strikes the water which has flowed from the condenser, and therefore the bucket-valve opens by excess of pressure below, and the bucket descends through the water to the bottom of its stroke. In its ascent the water above the bucket closes the valves and seals the piston while the cycle of the first stroke is repeated.

The foot-valve in most American marine vertical single-acting air-pumps direct-driven from their engines is a flat rubber flap of the necessary thickness (inch to inch and a half) which seats upon a grated inclined partition. Access to this valve for renewal and repair is had through a bonnet or cap formed in the casting just over the grating. The air-pump bucket-valves are also usually circular rubber disks, which are prevented from rising too far by brass guards. They also seat upon grated openings, and are easily accessible from the top of the air-pump. From the top of the air-pump the combined injection and condensed steam are discharged to the organ of the condensing engine which is called the hot-well (Fig. 668).

With the surface condenser the air-pump is still required, but its function is slightly different. As the injection does not meet the condensed steam, the air-pump does not have to handle the former, but has only to draw out the condensed steam and the air which gets into the condenser with the steam and by leakage. This latter however is a much more important factor in percentage of material handled in the surface-condenser system than in the other. Hence a very common practice is to make the factor ($R\frac{28}{60}$) in paragraf 504 to be five times as large as for the combined system for a vertical single-acting pump, and 9 times for the double-acting pump. That is, the air pump displacement in cubic inches per minute is $D = 5W$ and $D = 9W$ for these two cases.

510. The Wet and Dry Air-Pump. Advanced designing has presented many examples of separating the two functions of the air-pump and using a separate pump for each. One will be connected to the bottom or lower part of the condenser to handle the water, while at a point above the level which water will ever reach will be a second pump working on relatively dry air, or a mixture of uncondensed vapor and air. This dry-air pump principle enables each pump to be of smaller volume than if one cylinder had both functions to perform, and also secures a better vacuum, since the dry-air pump is more effective, both by location and functions, for the removal of air than the single wet pump can be. This system will be found in all modern stationary power plants which operate with condensation. The opening at the top of Fig. 667 will be for such dry-air pumps. These will often be of most careful design, with minimum clearances and mechanically operated pump-valves. With them a vacuum within a half a pound of the absolute limit of pressure for that day has been attained.

Fig. 671 shows a form of combined wet and dry-air pump known as the Edwardes design where the usual foot-valve is dispensed with and the water flows by gravity to the bottom of the air-pump through

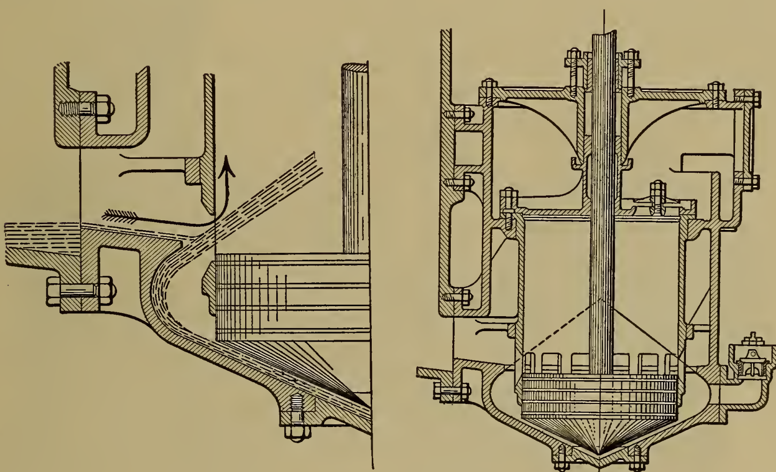


Fig. 671.

ports in its barrel. The top of the barrel is closed so that on the descent of the piston the pressure is much reduced above it by the enforced expansion of what air it contains. The bottom of the piston is conical, and as it enters the water in the bottom of the pump, the liquid is forcibly and positively displaced through the ports landing on top of the piston, at the same time that the air above the water rushes as shown by the arrow. The piston rising closes the ports communicating to the condenser, and the increase of volume below it causes water and air to flow out together into the pump as before. Any excess of water to be provided for while the ports are covered passes out by displacement through the valve at the right.

511. The Independent Air-pump. The beam-engine of most types drives its own air-pump. The inverted vertical reciprocating-engine for marine practice has also driven a special beam or lever from its cross-head to operate its air-pump (Fig. 664). See also Figs. 384 and 402.

Many advantages follow from abandoning the principle of the attached pump (paragraf 147) and driving such air-pump independently of the main engine, either by a steam cylinder, an electric motor, or by a transmission from the shafting which the main engine drives. In turbine plants the air-pump must be independent.

The principle offers these advantages:

1. The pump can be located anywhere. The attached principle compels the air-pump to work in the plane of the main engine if directly attached, and this may make the location for the air-pump either cramped or inaccessible or inconvenient.

2. The air-pump being independent of the main engine can be run without it. This is of advantage in starting the main engine, since the vacuum in the condenser can be created before steam is turned on to the main engine.

3. The air-pump can be run at varying speeds while the main engine is run at a constant speed. This enables the designer and runner to provide for varying temperatures of the injection-water according to the season of the year, and in marine practice according to the latitude and corresponding temperature of the ocean-water. The air-pump can further be run faster than the normal rate in case of leakage into the condenser which it may not be convenient to arrest. The vacuum will be maintained as it cannot be with the attached pump.

4. Since the air-pump can be run faster, and can usually be double-acting, it will be much smaller than the attached pump. This is a saving in bulk, a saving in weight, a saving in friction, from the lessened weight, and the small-diameter cylinder has a less clearance-volume, which is always troublesome when air is to be rarefied.

5. By detaching the air-pump, which is a water-pump as well, the speed of the main engine is not limited by the limitations of satisfactory working of the air-pump. The high rotative-speed engine can thus be conveniently condensing.

The only objection to be urged against the independent air-pump is that the small steam-cylinder which drives it uses steam less economically than the large cylinder of the main engine. This is not true when the air-pump is motor-driven. The necessary clearance-volume, although smaller in the independent engine, is filled and emptied more often.

The superior convenience of the independent principle has made it a feature of much recent designing, particularly for high speeds and of necessity for turbines. It will be seen from Figs. 665 and 672 that it is very simple to combine the air-pump and the circulating-pump for surface-condenser practice so that one steam-cylinder shall drive both pumps. This makes an arrangement which is both convenient and very economical of space. The inertia effect of the levers, piston, and water of the attached system are not negligible at even moderate speeds (paragraf 260) and must be computed in a complete treatment. This plan is practically limited to slow and massive engines.

512. The Circulating Pump. If the surface condensing system is used, the injection-water of the jet system becomes the circulating

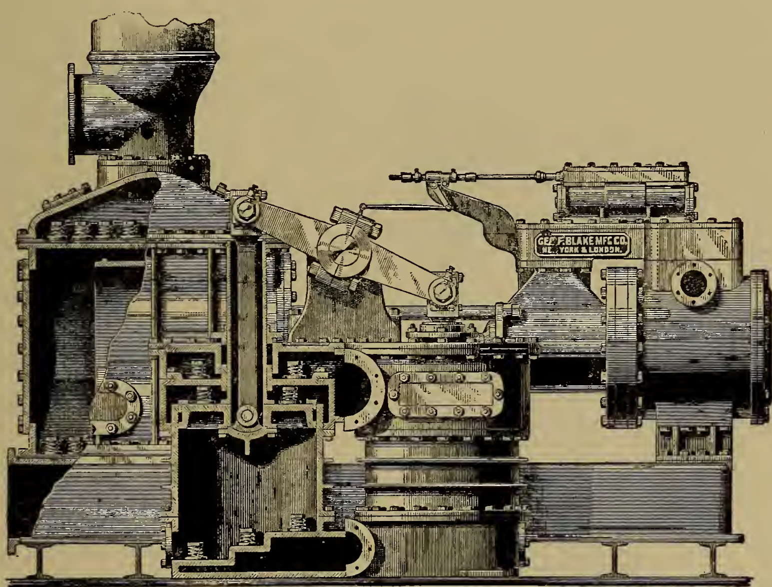


Fig. 672.

water through the tubes. Fig. 663 shows the general requirement of overcoming a few feet of head and the resistance to motion through the tubes from friction, bends, and valves, friction of the condensing tubes, and to lift the water through the few inches of difference of level between the water outside and the discharge-level of the overboard-valve. By reason of the small resistance and the large volume of water which are the conditions of such circulation (seventy times the volume of feed-water required by the engine), centrifugal pumps have been the very prevalent type of circulating-pumps. They are driven by their own independent engines or by motors. More recent American practice introduces reciprocating-pumps for this kind of work with satisfaction (see Figs. 665 and 672). When single-acting reciprocating-pumps are used, the volume of the cylinder is from one twentieth to one thirtieth

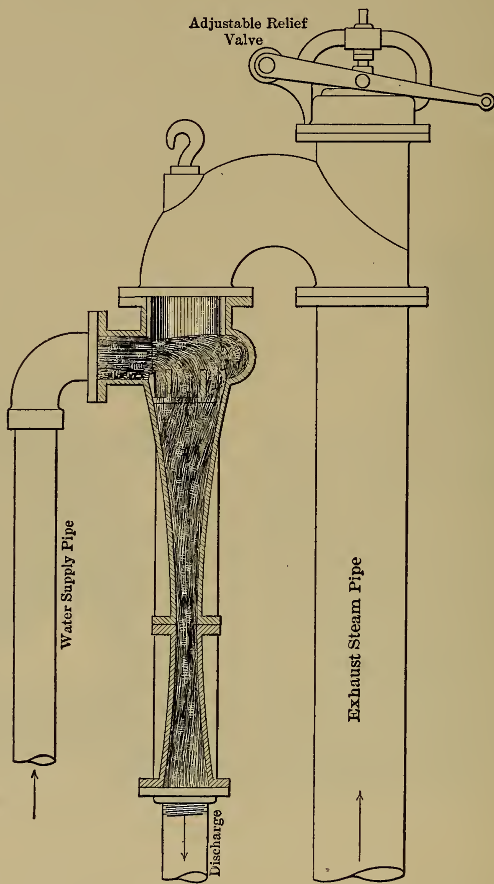


Fig. 673.

of the steam-cylinder volume. For small launches where the quantity of steam to be condensed makes such an arrangement practical, a form of surface condenser has been used which consists of a coil of pipe zigzagging on the outside of the hull on both sides of the keel. The steam passes through the inside of this coil of pipes, and the motion of the boat causes a continual impact of cool water against the coil and produces the same phenomena as by the circulating-pump. It has been found, as might be expected, that the more rapid the motion of the vessel through the water the more efficient are such condensing coils (paragraf 506). The extra resistance offered by such coils is the compensation for the avoiding of the circulating-pump, but in small boats it is a distinct gain to get the bulky condenser outside of the hull.

513. The Barometric, Injector, Siphon, or Gravity Condenser. The wet-air pump is to draw out water of condensation from the bottom of the condenser (paragraf 510) and the dry-air pump to exhaust the air and watery vapor uncondensed. If the condenser can be put over 32 feet up in the air, with an air-tight pipe descending from it and water-sealed at its foot in the hot-well (Fig. 670), it is plain that atmospheric pressure on the water in the well will balance a column in that air-tight pipe only 32 feet upward, and that above that level will be practically a Torricellian vacuum if the air can be gotten rid of. The later and better system is to use a dry-air or vacuum pump as shown in Fig. 670; the other plan is to entrain the air in the rapid flow of the water of injection in carefully moulded nozzles (Fig. 673). The Torricellian principle, with the vacuum pump, makes the condenser a barometric tube; the jet or siphon in action gives the name of siphon to the type; in both the withdrawal of excess water above the 32-foot limit is by gravity and not by any pump.

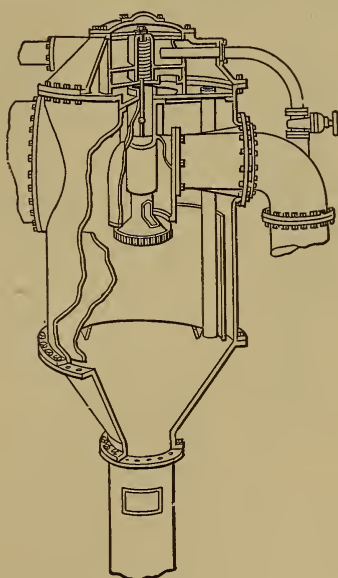


Fig. 674.

Instead of the air-pump, however, the injection-water has to be lifted to the height of the condenser; but power is not required all the way. Atmospheric pressure is behind the pump suction, and it need only lift through the difference in height and overcome resistances. The early forms were an elevated pot (Ransom's) but they did not adequately get rid of the entrained and leakage air.

Fig. 674 shows detail of the head of the barometric type, with the vacuum chamber on top, and special by-pass injection to condense vapor therein. The spraying cone is constructed to secure fine division of the jet, and is held in place by an adjustable spring which thins the

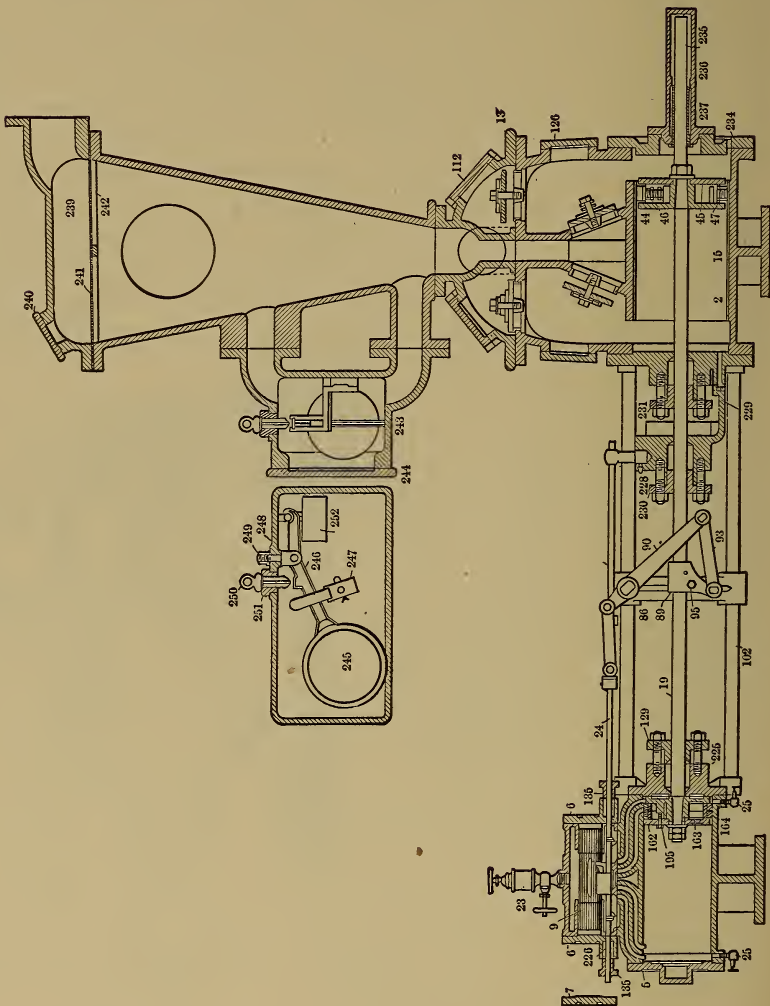


FIG. 675.

jet as the injection pressure diminishes. The water used is thus made altogether effective. It is desirable both in the barometric and the siphon types to provide a relief valve and connection, so that from a backing up of the water or other cause the pressure in the exhaust-pipe can not exceed the determined limit, and that water from the condenser can not work back into the exhaust-pipe at pressures so nearly atmospheric as to give trouble. Fig. 675 shows a float-valve in an appendage to the condenser, so arranged that if the condenser floods, the rise of the float in the water will open the valve and break the vacuum. The engine must thereon operate non-condensing through a similar relief valve as in Fig. 670. The spray-cone in Fig. 661 acts similarly, since when submerged by water, the water surface alone is inadequate.

514. The Exhaust-steam Ejector Condenser. It early suggested itself to apply the principle of induced currents as used in the steam-injector to draw up the injection-water and to make use of the living force of the water thus set in motion to oppose the balancing effect of the pressure of the air. The first design of this sort is identified with the name of Morton (Fig. 676) in England, and the more usual forms with the name of Schutte in Germany and America (Fig. 677). The philosophy of the Schutte injector condenser depends on such an enlargement of the discharging end of the condenser that when the condensed steam and injection leave the outlet they have such a velocity as just to overbalance the tendency of the water in the hot-well to flow through that outlet back into the space where the vacuum due to condensation is maintained. As will appear from the sectional cut (Fig. 677), the steam enters through the side into an annular chamber and passes through a series of inclined orifices or nozzles. The steam moves with considerable velocity, and draws in water from a cold-well A (Fig. 679), and when the steam and water meet, the steam is condensed and flows with the rapidly moving water out through the discharge into the hot-well B. The discharge is sealed as shown in the general view, and the velocity of flow overbalances atmospheric pressure on the well. The small steam-connection enables water to be drawn into the condenser on the injector principle, in order to start it when water does not flow naturally to this level. The by-pass controlled by the valve permits the exhaust to be carried to the open air when for any reason it is desirable to run the engine non-condensing.

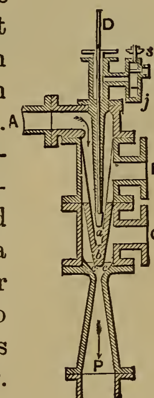


Fig. 676.

It will be observed that this arrangement also, like that in paragraf 505, enables the condenser to be operated without the 34 feet of elevation. The series of gravity siphon and ejector condensers are all jet or direct-contact condensers.

To start the condenser the starting-jet connected to the small inlet marked "Steam" in Fig. 677 is opened and the auxiliary water-valve *D*. When the vacuum has been created, and the main engine is started, steam and water meet in the combining-chamber and after condensation flow together obliquely downward through the multiple outlets, creating and maintaining a vacuum behind them. If the condition of a transverse film of water be considered at the orifice of the discharge-pipe, it has on its outer face the pressure of the atmosphere transmitted from the surface of the hot-well. On its inner face is the pressure due to the impact of the moving mass of injection and condensed steam discharging from the nozzles behind in the injector-tube. So long as this latter is greater than the former, atmospheric pressure cannot exert its effort to force the water in the well back into the condenser.

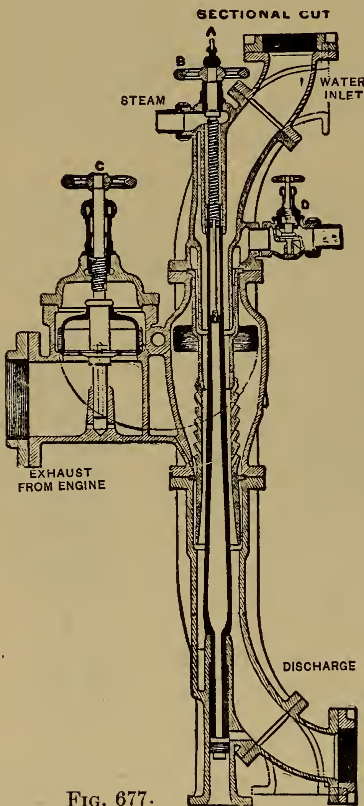


Fig. 677.

515. The Hot-well. The delivery of warmed water from the air-pump in the jet-type of condensers will be into some reservoir, from which the feed-pump can take what is desired for supplying the boiler. In engines with surface condensers the hot-well becomes a much less significant organ, because only the condensed steam is delivered by the air-pump while the injection or cooling water is passing upon another circuit.

Referring to Figs. 389 and 359A as a typical river-boat engine with attached air-pump, it will be observed that the top of the air-pump, which is the discharging end, is enlarged into a cylinder of nearly twice the diameter, fitted with a loose cover and a valve. This enlargement of the air-pump is the hot-well. In engines operating with independent air-pumps the hot-well will be any convenient tank or reservoir. It need not be tightly closed, as there is no pressure in it, and it simply has to take care of warm water and serve as a cistern from which the boiler feed-pumps may draw their suction. In land practice it is usually arranged with an overflow whereby the excess of water not needed by

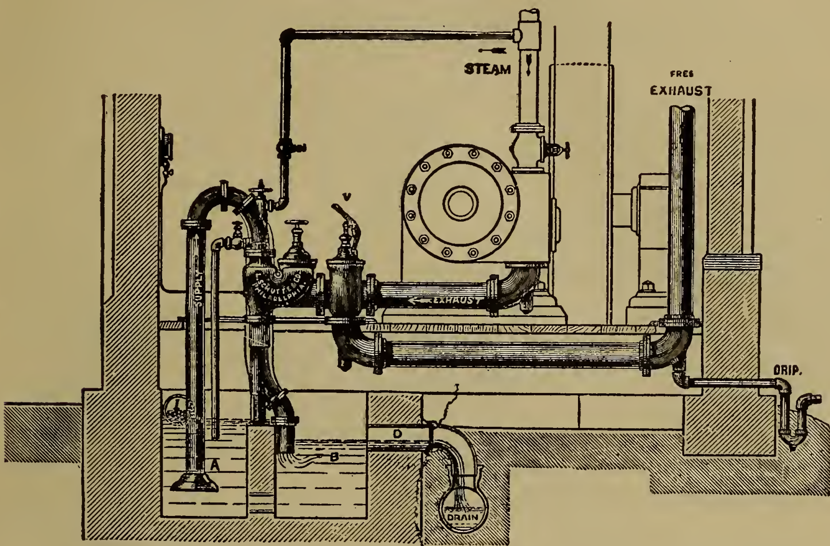
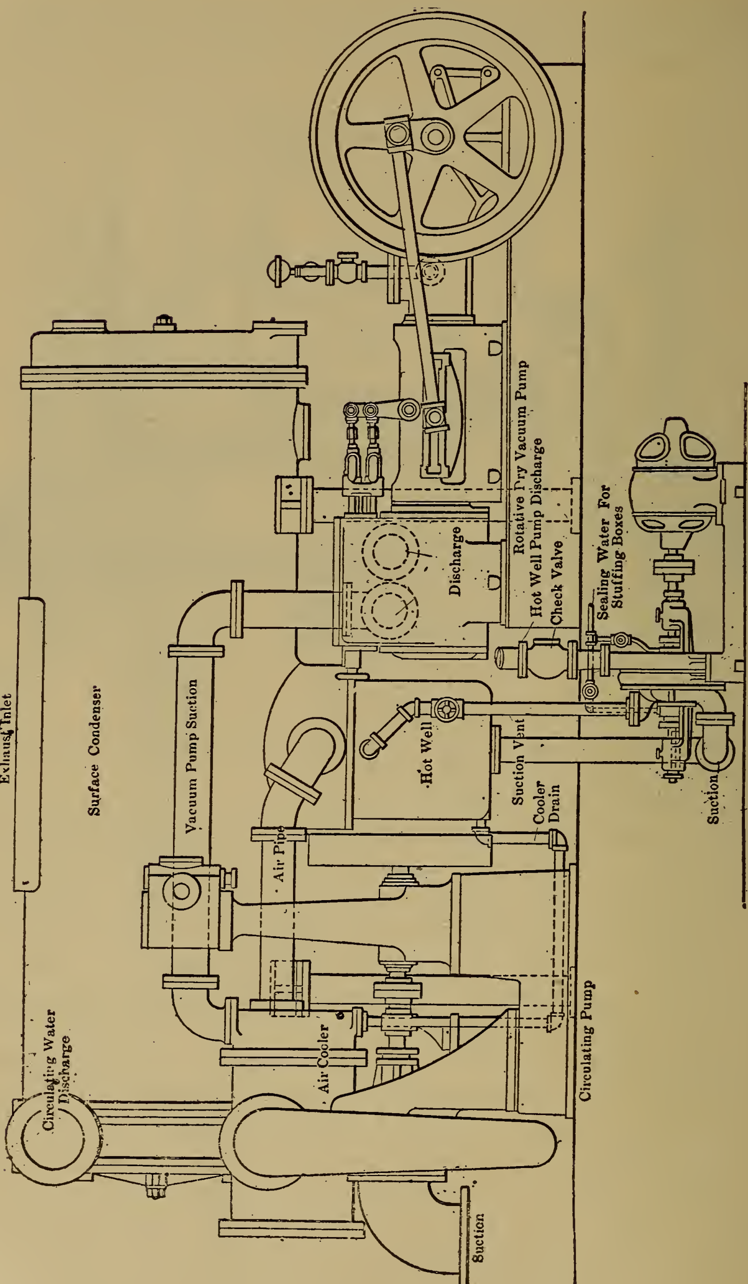


FIG. 679.

the boilers may escape to waste. In river-boat practice the excess is usually taken care of by pumps.

516. The Feed-pump. The feed-pump of a condensing engine draws its suction from the hot-well. It may be attached as in the older forms (paragraf 147), or independent as in the newer. In the attached system it is usual in river-boat engines to attach to the rod of the air-pump one or more brackets or half cross-heads, whereby the rod from the beam shall operate the pump or pumps which take care of the water discharged into the hot-well. In Fig. 359A this smaller pump is operated from the small beam which drives the air-pump. In engines of this class these pumps must have capacity sufficient to empty the hot-well continuously. Where the hot-well can overflow as on land, such pump need have only a capacity sufficient to feed the boilers with the water which they require. In the former case, where the pumps are handling an excess of water, it is common to arrange the discharge from the pumps to branch into two outlets. One of these outlets goes to the boiler, and the other goes overboard through the hull. The valve in the overboard branch will control the proportion which goes through each branch, since if that valve be wide open the entire delivery of the



Exhaust Inlet
Surface Condenser
Vacuum Pump Suction
Air Pipe
Air Cooler
Circulating Water Discharge
Circulating Pump
Suction
Hot Well
Suction Vent
Cooler Drain
Discharge
Rotative Dry Vacuum Pump
Hot Well Pump Discharge
Check Valve
Sealing Water For Stuffing Boxes
Turbine Hot Well Pump
Suction

FIG. 680.

pumps must go overboard, because the pressure in the boilers must be overcome before the water will flow through the branch connected to them. On the other hand, if the valve is shut in the overboard branch, the entire discharge of the pumps goes into the boilers. At intermediate degrees of closure, part will go overboard and part into the boilers. In the independent systems any form of power driven pump may be used. (See paragraf 147.)

517. Condensers for Steam Turbines. The condensers for a turbine plant may be either surface or jet. The surface is probably more usual as giving a higher feed-water temperature with a given vacuum when the dry-air pump system is used. The turbine is particularly susceptible to the advantages of low terminal pressures and temperatures, and the better the vacuum the greater the economy. Fig. 680 shows a typical auxiliary equipment for a turbine with surface condenser with centrifugal circulating pump and reciprocating dry vacuum pump.

Fig. 481 may also be studied in detail with advantage. All condenser auxiliaries in the turbine plant must be independent, with the possible exception of turbo or centrifugal pumps, which may be driven by belting from the rotor shaft if they can be conveniently alined. Independence of location is usually worth the slight loss from driving these auxiliaries by electric motor.

518. Sundry Special Condensers. The pumping engine may have a surface condenser installed in its suction or water delivery pipe, using its own pumped water as circulation water (Fig. 675A in Appendix). Mine pumps may also discharge their exhaust directly into the water they are pumping.

In motor vehicles, the motion of the car through the air may be used to cool the pipes of a surface condenser with the steam within them. Here it is usual to supply a fan in addition to secure rapid air circulation and the exterior surface of the pipes is increased by metallic fins or supplementary areas.

519. Concluding Comment. The power to drive the independent auxiliaries (Fig. 3) should be charged to the main engine to equalize the alternative condition where the main engine drives them. Such consumption of power is the offset to the theoretical advantage of the lowered back-pressure from condensing.

CHAPTER XXXII.

ENGINE AUXILIARIES. LUBRICATORS AND LUBRICATION.

525. Objects of Lubrication. The process of lubrication is to introduce between two surfaces which have a relative motion of one upon another some suitable material to prevent a metallic contact. This medium should be one which will further have a tendency to reduce the coefficient of friction which would prevail if there were no film of lubricant present.

The object or purpose of lubrication is therefore two-fold. The first is to keep these contact surfaces from wearing each other; the second is to reduce the resistance to motion, or to the transmisson of effort from power to resistance. The better the lubrication, the less the difference between indicated horsepower in the cylinder and effective horsepower at the shaft (paragraf 3).

526. Lubricants, or Lubricating Materials. The materials to use for lubrication require to be such as have a low coefficient of friction in themselves, and which shall be capable of introduction in their films between the rubbing surfaces, and shall also not be too inclined to leave these surfaces by pressure, by the rubbing or scraping action of the surfaces, and by heat. These requirements seem best met by the oils, the greases and by graphite. The greases and graphite once introduced are difficult to displace by pressure and rubbing; the oils are most easily and copiously fed to the desired point

An oil must not be prone to oxidize or gum by heat and exposure to the air; it must not be easily inflammable by heat. It must be free from acid as a residual of its refining process. It must not be so viscous as to have a high coefficient of internal friction of its own; it must not be so limpid or thin as to be squeezed out easily from its contact area.

The greases should be capable of being gotten into the desired bearing surface by pressure, or fusible enough to flow there if melted by a gentle heat originating from a friction rise of temperature in the bearing. Graphite in fine flake form and free from grit can be introduced either as an addition to grease or to an oil. It must not be allowed to segregate or settle in the oil which carries it as a vehicle. Greases are specially convenient in moving elements where an oil would be easily thrown out, as in the connecting-rods of locomotives.

There will be two sets of conditions to be met in lubricating an engine, and hence two types of lubricant. The one is the lubrication of the cylinder and its valves, which are under pressure and the heat of the steam. The condition is particularly exacting when the steam is superheated, free from moisture and at high temperature. The other set is met in the turning and sliding surfaces of shaft, connecting-rod and guides.

527. Tests of Lubricants. The subject of the various lubricants is too broad a one to receive full treatment under present conditions, but brief reference may be made to three important tests. An oil is liable to fail of its purpose when for any reason it is prone to oxidize from heat or use and to become gummy as the result of that chemical change. Gumminess is a relative quality, and consequently the test to determine this is a relative test between the most limpid and the most readily oxidizable of the oils. The test for the gumming quality of the oil is to drop a certain weight or volume of the oil to be tested in the middle groove of three, made upon a surface of cast iron which is inclined to the horizon at a slight angle.* In one of the other grooves is dropped an equal weight or volume of sperm oil, which has no tendency to gum, and in the third an equal weight or volume of linseed-oil, whose gumming qualities are so great that it is used as a drying oil. The three oils slowly run down the grooves, undergoing oxidation and becoming more and more sluggish as they flow. The distance covered by the oil to be tested, as compared with the distance covered by an oil having the greatest and least quality of gumming as represented in the other two, measures its excellence in this respect.

The test for acid in an oil is made by putting a small quantity of the oil in a test-tube with a little copper scale of the suboxide of copper, Cu_2O. If there are fatty acids present, after some hours' exposure and with gentle heat the reactions with the copper turn the solution green. If the oil has a vegetable acid, it will turn blue. Further qualities of oils for lubricating purposes are determined by their low fire-point. If they give off an inflammable vapor by heat, they are of course a dangerous element.

The third test is for its friction coefficient. This is made in laboratories by a lubricant testing-machine, where the oil is exposed to determined load in a test-bearing, and the resistance necessary to keep the bearing from turning with the shaft measures the friction existing in the film of oil which keep them apart.

* One foot elevation in six of length if the oil is to be tested in ordinary air. If the slab is heated, the slope may be steeper, and the test will require less time.

The mineral oils withstand heat better than most of the animal oils and than nearly all the vegetable oils.

528. Lubrication of Cylinder and Valves. The lubricant required within the valve-chest and cylinder must be introduced against the pressure prevailing therein. This can be done in one of several ways.

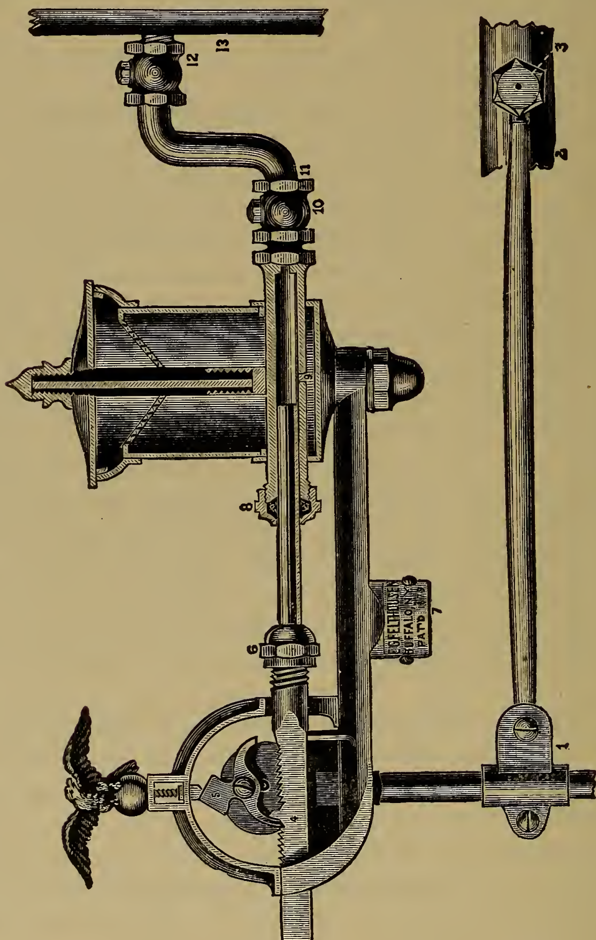


Fig. 685.

(1) In condensing engines a simple open cup, closed at the bottom with a valve or cock, can be screwed into the clearance of the cylinder, and opened when the pressure in the cylinder is less than atmospheric. This is particularly applicable to vertical cylinders, but will not lubricate the valve.

(2) The oil can be forced into a cylinder by a pump either operated by hand or driven by the engine, or in large engines by a small steam-

cylinder. If driven continuously by the engine or by an independent oil-pump, the feeding of oil is continuous and economical (Fig. 685). If driven by hand, the supply is intermittent. In compound engines this is often used as supplementary to other methods.

(3) A popular method of lubrication which prevails most widely for simple engines is the delivery of oil by drops continuously into the steam-pipe, using a column of water as a source of power. The oil is contained in a closed cup from which two connections enter the steam-pipe (Fig. 686). Upon the short one, K, close to the cup, steam-pressure in the pipe is acting, while upon the other, F, connected to the steam-pipe at some distance above the cup, both the same steam-pressure and

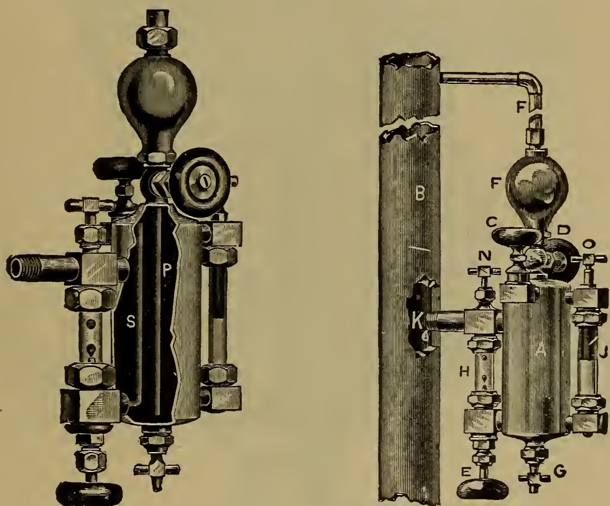


FIG. 686.

the weight of a column of water condensed in that longer connection are acting. This column of water displaces the oil at a controlled rate into the surfaces to be lubricated. Its action is continuous.

(4) What is called the oil-cup or cylinder-cup is a brass vessel with a pipe-connection from its bottom, in which is a valve. The cup has a lid at the top which is screwed on and steam-tight. When the valve in the bottom is closed the oil-cup is cut off from the cylinder and the lid can be lifted off and the cup filled. When the lid is in place the lower valve can be opened, and the pressure, equalizing, will permit the lubricating material to descend into the cylinder either slowly or fast according as the valve-opening may permit. This is the air-lock principle, but the feeding by it is intermittent.

529. Graphite as a Lubricant. The objection to oiling cylinders with fluid oils in condensing engines is the difficulty from the oil in the exhaust and in the boiler (paragraf 197). Graphite possesses a lubri-

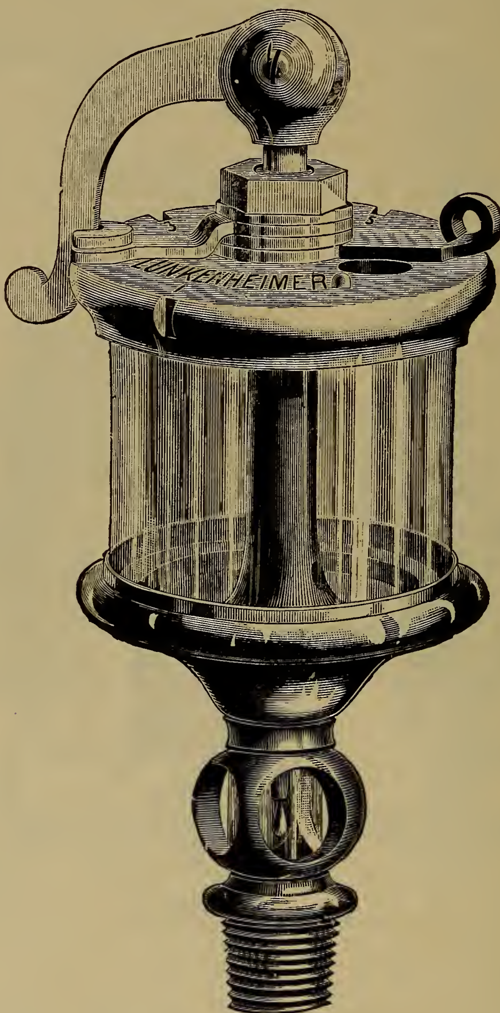


FIG. 687.

cating quality, has a low coefficient of friction, a body which prevents it from being forced out of the surfaces where it should act, and furthermore seems to fill the pores of the surfaces so that they acquire a singularly smooth and mirror-like surface where it has been used. It is introduced either as a powder or in combination with some other lubricating material as a vehicle.

530. Lubrication of Bearings. The bearings in a steam-engine which require to be lubricated are those of the shaft, eccentric-straps, the

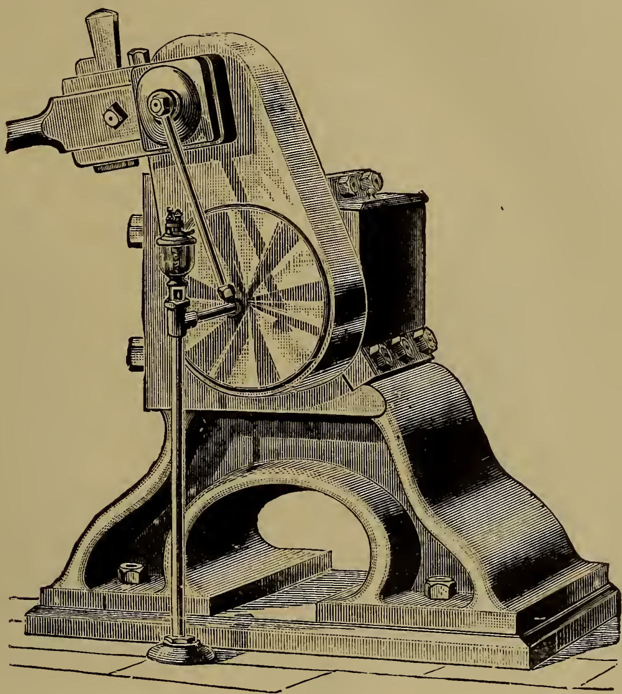


Fig. 688.

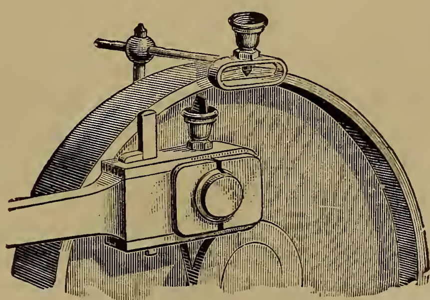


Fig. 689.

crank-pin, and the cross-head pin and guides. The main-shaft bearings are the only ones which are stationary so as to be reached by the ordinary hand methods, and the convenient and automatic lubrication of all bearings has brought the application of many devices to maintain a continuous and abundant supply of oil. What are called sight-feed oil-cups are those in which a supply of oil is held in the cup and is allowed to drip from its bottom through a valve which controls the opening in such a way that the size and number of drops can be seen and regulated (Fig. 687). The oil from such cups falls through pipes which conduct it to the fixed bearings, where they are distributed by proper grooves cut in the bearing by which the motion of the shaft makes the oil spread where required. The connecting-rod bearings are the most difficult to provide for, because it is a secondary piece supported upon two other moving pieces. The crank-pin is lubricated either by centrifugal force as shown in Fig. 688, whereby oil received near the center of motion is carried outward through a pipe to the center of the pin, and is there distributed through a radial hole outwards upon the bearing-surface, or else a flat piece of metal is brought against a webbing by the oscillation of the connecting-rod, and wipes the excess of oil off the webbing so that it is delivered downwards upon the pin (Fig. 689). For the cross-head pin a similar fixed cup is placed over the path of the cross-head, having on its under side a piece of webbing or similar textile material upon which the oil drops and is spread. Illustrations of these methods of lubricating will be seen in the various types of engines hitherto presented.

In large engines with many cylinders and multiple mechanisms a practice has been followed of bringing all oil-cups to a few points and connecting these oil-cups by pipes to the various bearing-surfaces to be lubricated. In vertical engines of the marine type it is usual to lubricate the crank-pin by means of a pipe running along the connecting-rod, and ending near the cross-head pin in a flaring mouth, into which the sight-feed oil-cup shall deliver its oil and from which the pipe shall carry it to the pin. It will be apparent that, as the cross-head travels in a straight line, the mouthpiece will always be under the end of the oil-cup in all positions. Bearings have also been made self-lubricating by means of rings or chains (Fig. 549) which turn in a bath of oil below the bearing, and rest upon the shaft to which they are internally tangent. As these rings revolve with the shaft, the oil which adheres to them is continuously brought up from the reservoir and delivered at the top of the shaft from which it is distributed.

The certainty of mechanical or forced lubrication for crank-pins, cross-head pins and main bearings has given wide acceptance to the plan of making these elements with a central and radial holes, through which a power pump from the engine, or independently driven, forces the oil where required. The principle of splash lubrication of crank cases is also illustrated in many types chosen for presentation of detail (see Figs. 345). Cylinders are also oiled by positive pumps.

Siphons of lamp-wick have also been used as a means of securing a continuous slow feed from an oil-cup. The oil rises by capillary action in the wick, and when it has reached the bend in the tube within the cup in which the wick is placed it descends by gravity down the longer arm and is delivered in drops in the bearing below (see Fig. 527).

Greases are another form of lubricant whose delivery from a reservoir can be secured by the slight rise of temperature which will fuse the

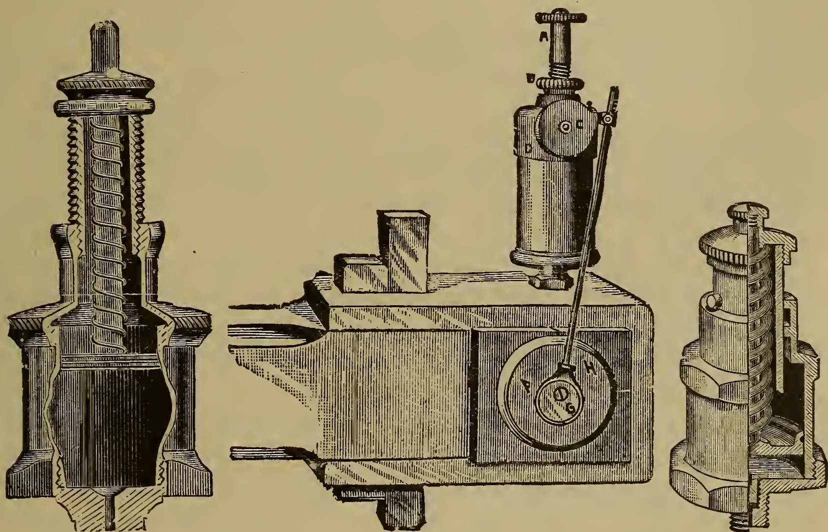


FIG. 690.

grease; or grease cups can be used in which a more resistant viscid grease is forced through the delivery-opening by the pressure of a spring controlled by a screw and nut (Fig. 690). This is particularly convenient for lubrication of locomotive-rods, where it is desirable that the oil-cup should be closed from grit and dirt, and where the methods of the stationary plant cannot be applied. Or, the grease-cup cap or piston can be screwed down by hand at intervals; or the descent of the displacing piston can be made positive by a mechanical action such as shown in the central cut of Fig. 690. Here the revolving crank-pin turns the little eccentric and by a ratchet and screw the grease is forced down.

It does no harm for bearings to be run at about 100° to 110° F., or so as to be pleasantly warm to the touch of the back of the human hand. Turbines require no cylinder lubrication, but careful supply of oil to the bearings. Here also it is usually forced by a pump.

It is an economy to catch used oil which has been through the engine and use it over again. Care must be taken however to free it from any dirt or grit or wearings from surfaces which it may have picked up; and after a while, oil loses its capacity to lubricate and must then be thrown away. Oils may be filtered for cleansing and successive use.

PART VII.

CHAPTER XXXIII.

CARE AND MANAGEMENT. ACCIDENTS.

535. Starting an Engine. So many and so widely different are the types of engine and the work which they are to do that anything like detailed instructions is impossible. The best that can be done is to lay down certain general principles applicable either widely or universally, and leave the application of them in detail to the judgment and skill of the operator in each case.

The most general case would be of an engine for a power house or similar plant, and the first distinction which will make a difference in procedure must be based on whether the engine is non-condensing or condensing.

536. To Start a Non-condensing Engine. The engine having been properly erected and all connections supplied (Chapter XXI), the fly-wheel should be turned in such a position that the valves uncover the ports, and that there is a turning leverage for the pressure of steam to turn the crank. This will be done either by the notches in the fly-wheel rim, or by a block and fall, or jack, or other power.

It is of the greatest advantage in starting to be able to work the valves by hand. This is possible in all gears having gab-hooks or their equivalent (paragraf 461) and is a feature of all link motions. For the first step in starting is to warm up the cylinder by admission of a little steam — not enough to turn the cold engine over, but enough to create a circulation in the cylinder and out of the open drip connections. With the drip valves all open and the steam-pipe previously drained of water through its own drips and separator (paragraf 243) hot steam is admitted through the throttle-valve by a very slight opening whereby it will be allowed to blow through and heat the piston and the walls of the cylinder to a temperature sufficient to prevent excessive condensation. This also rids the steam-pipe of the water which has accumulated within it. In positively connected valve-gears this will heat but one end of the cylinder, but where the valves can be operated by hand the steam can be admitted for warming to both ends, and the whole mass of metal brought up to the necessary temperature. It is desirable, however, to leave the drip-connections from the cylinder open until after the engine has started. The cylinder being fully warmed up, which will be recognized by touch and by the high temperature of the drip-pipes, a little more steam can be admitted either through the wider opening of the throttle-valve, or through the more ample movement of the distributing-valves, so that sufficient pressure comes on the piston to start the engine. It becomes a matter of importance to store sufficient energy in the fly-wheel to carry it past its first dead-center, on which otherwise it would be likely to catch, and particularly if there is water in the cylinder whereby its motion can be arrested just at the time of getting ready to pass the center. If this difficulty is not met, the engine will then take up its regular motion, slowly at first, until all danger from water shall be passed, and then gradually more steam is admitted until its regular rate is reached, at which its governor will take hold and control the supply of steam. The danger from the water of condensation is usually passed at the end of two or three complete strokes, but it may last longer, and it may occur from priming even under regular service. The drip-valves are therefore closed cautiously, to be sure that all water has been blown out. It will make its presence manifest by a characteristic snapping or cracking like the blows of a metallic substance within the cylinder, which once heard will always be recognized. The danger from water has already been alluded to. If the engine is one requiring its cut-off to be regulated by hand, and not by the governor, as in link-motion engines, adjustable cut-off engines, etc., the throttle-valve will usually be opened wide and the regulation effected by the use of such adjusting mechanism when the normal speed of the engine is attained.

537. To Start a Condensing Engine. Here again the variety of methods used to effect condensation makes it difficult to include all conditions (Chapter XXXI). If the engine is surface condensing, the circulating water will be started in motion before the main engine is started. If it is a jet-condensing engine driven by independent air-pump connections, the vacuum in the condenser will be created before the main engine is started by starting the independent air-pump. With the attached system or the gravity or siphon systems the vacuum must be created after the first steam is delivered to the condenser. With the attached system and large air-pump it is desirable to start the engine with the crank in such position that the first motion of the piston shall cause the working stroke of the pump to take place and create a partial vacuum in the condenser. Sometimes the vacuum is created in advance of starting the attached mechanism by permitting the condenser to fill with water, and attaching an independent pump to the condenser which shall draw out the water until its capacity for equalizing pressures in its own cylinder and the condenser are reached.

In many cases the cool metal of the condenser will serve to effect the first condensation and create a sufficient initial vacuum for the engine to get its air-pump to work without difficulty. The drip-connections of a condensing engine are different from those of a non-condensing engine, because as soon as the engine has started, the flow would be into the cylinder through them. For this reason they are either left off or are connected into the condenser piping. After the engine has turned its centres, the handling of its condensing appliance will involve the control of the injection-water in jet or direct-contact condensers and the speed of the circulating-pumps in the surface type. Since it may happen, in condensing arrangements where the air-pump and circulating-pump are driven from the same rod, that the full capacity of the circulating-pump is not required, while the air-pump must work full stroke, a by-pass valve is usually made on the circulating connections, so that it shall be able to pass its own water round and round in part, and not be compelled to handle an unnecessary weight to effect condensation. The injection of jet condensers enters them by atmospheric pressure, so that the valve which controls need not usually be wide open. The operation of the condenser is regulated by the reading of the vacuum-gauge, which is graduated either from zero to 15 pounds of vacuum, or from zero to 30 inches of mercury. The vacuum is satisfactory if it reads over 13 pounds or 27 inches. It will be less than this either if water is sufficiently in excess to overfill or drown the condenser, or if there is too little water to dispose of all the heat which the steam brings into the condenser.

538. To Start a Compound Engine. The compound engine, having both a non-condensing and a condensing cylinder, requires to be handled in starting according to the principles laid down in both the previous paragrafs. It is usual, however, to derive in the compound engine an advantage in starting which is not present in the single engine. If the two cranks are at an angle with each other, which is usual in power-house practice, it becomes possible to start the engine, even if the first or high-pressure cylinder stands, with its crank on the dead-centre. A valve connecting the receiver of the low-pressure cylinder with the steam-pipe will be controlled by a valve which will be called the by-pass valve. By opening it, boiler-steam is admitted directly to the low-pressure cylinder, which will be at its best mechanical advantage if the high-pressure crank is at its dead-center, and by these means the engine can always be started either as a low-pressure or as a high-pressure according to the position of the cranks. The complication of the steam-heated receiver and steam-jackets adds nothing of difficulty to an engine of this sort. The jackets make it unnecessary to pay special attention to the heating of the cylinder, since to open steam on the jackets will accomplish this purpose. Some recent practice has attached the steam-pipe of the independent air-pump to the jackets and receiver-circuits, so that the steam must be turned on to the jackets for warming the main cylinder before the engine is started. This is desirable to guard against the possibility of difficulty caused by unequal expansion of parts which are hot and cold.

The compound locomotive with intercepting-valve operating automatically starts just like the simple engine non-condensing.

In large vertical engines such as are used in marine practice, where the engine is on different levels, it is often necessary to have several hands for proper starting. Usually one level is the working platform in such cases. The common practice on board ship is that the chief in starting is at the throttle-valve, the first assistant in charge of the valve-gear, and while the second is in charge of the fire-room, the third or junior will take the bell or signalling apparatus from the bridge. In large engines fitted with a barring engine (paragraf 410) the starting is particularly simple, after the cylinders are warmed. It is a matter of many minutes to warm up and start a large multiple-cylinder engine by reason of the danger from water in the cylinders and it cannot be hurried. Twenty minutes or so should be allowed, a fact which has a bearing upon emergency calls upon such engines for fire or other similar uses.

539. To Start a Steam Turbine. In the steam turbine the danger from starting with the casings too cool is not from the condensed water, but from unequal expansions, and hence a lack of balance when the rotor gets to speed. Steam is first turned on and blown through with the rotor at rest. The auxiliaries are started while the heating is in progress, first the circulating pump, then the hot-well and dry-air pump. It will take 10 to 15 minutes to start a large Curtis or Parsons turbine, or two or three minutes after it has become thoroughly warm. With a superheating apparatus, which usually becomes more effective as more steam flows through it after starting, it is necessary to prevent vibration due to unequal heating after the work comes on. The exciter set for the generator can be started after the rotor and generator are up to speed and running in balance.

In stopping a turbine, after the steam is shut off, the condenser should be also, either by its valve or by stopping the auxiliaries. A turbine will run in a vacuum for 30 to 60 minutes after the load is off, but by admitting air through the drips this acts as a brake; or the load may be left on.

In Curtis vertical turbine plants, the foot-step pump must be started and its accumulator put in service before the rotor is allowed to turn.

In electric stations which operate alternators in parallel the process of synchronizing the independent engines must also be a feature of the starting process, so as to get them in step and phase. Where there are glands on the rotor shaft for centrifugal water packing, it is sometimes difficult to start condensing by reason of the cold air leakage which takes place until the rotor is up to speed. This is not a serious trouble in any case, and can be avoided by starting non-condensing and then changing over. Or the glands may be allowed to leak until the load comes on, and have the gland water turned on then. With a turbine plant, the auxiliaries call for more attentive care than the turbine itself.

540. Management of Engines. The operation of an engine which has been properly designed in the light of experience and application of scientific principles should consist only in keeping up the functions of each detail. The oiling system and all channels and ducts for oil should be kept open and free; the packings should be just tight enough to prevent leakage, and should be renewed when worn. The piston rings must be kept tight, and all lost motion in the joints taken up as it occurs from wear. The whole previous part of this treatise has been directed to make this service an intelligent one.

The control of the operation of the steam in the cylinder under the action of valve-gear and governor is the point where economy is most effected. The valves should be tight, and the governor and gear in high efficiency. Occasional tests are most effective to see that this result is obtained, and computations made of the cost of the horsepower per hour from such test, as a check upon increasing losses.

The loss of income from a shut-down or stoppage of the engine may so far exceed the cost of the power-plant or its annual operation charge in large undertakings that almost any expense is justified if its result is to keep the engine running during every working hour. It should be the object of careful management to prevent or forestall any shut-down.

541. Shut-downs in the Engine Room. Major Accidents. A shut-down in the engine room may be the result of a disaster or accident, or of a mishap. The disaster is the breaking or wrecking of some element, and is only to be met by a replacement. Such are:

1. The engine runs away and bursts the fly-wheel.

2. Water in the cylinder cracks or breaks the cylinder or some part of the mechanism.

3. A steam-pipe breaks or bursts, or a valve or some attachment on the pipe-system.

4. Some element breaks from a defect in the material developed in use, or present from the start and hidden.

When such a major accident occurs, a controversy nearly always follows as to the responsibility for it, or the proportion assignable to the designer who computed the strains, the builder who embodied the design in metal, the contractor or erecter who installed the machinery and the operator in charge. The second one is usually the only one for which the operator is exclusively responsible, and even here defective designing can help greatly in bringing it about. The operator is responsible only for the first when the governing apparatus was inoperative from his fault; and for the third when it is due to water-hammer from careless handling of valves.

In addition to these disabling accidents shut-downs may also be made necessary for short periods from minor accidents, more properly called mishaps.

542. Minor Accidents or Mishaps in the Engine Room. The most usual thing which goes wrong is the overheating of a bearing, either from too tight fitting of the bearings (Chapters XXIII and XXIV), or by defective alinement (Chapter XXI), or from the use of poor oil or too little of it. The hot bearing is first annoying from the excess of friction which it indicates, but after a short period of heating the parts expand, increasing the friction or hold which they have upon each other, whereupon the contact-surfaces begin to cut each other and the presence of the abraded material caused by such cutting occasions greater heating and finally destroys the contact-surfaces so that until refitted they will never run cool. For a revolving bearing which heats only moderately a wick or mat of some fibrous material which dips into a bucket of water can be used upon the heating shaft. Most marine engines (where alinement is troublesome by reason of the flexibility of the hull which carries the bearings) are fitted with special arrangements for carrying a current of water to be discharged upon the bearings and keep them cool. When cutting has begun it can sometimes be arrested by using a lubricant of heavier body, or by compounding a lubricant by mixing tallow and graphite. If the bearing is very large and the cutting very serious, a mixture of tallow with lead-filings and powdered sulphur makes a compound which fills up the abraded surface in part, and often has prevented the cutting from going further until the bearings can be permanently refitted. Mercury may take the place of the lead.

In an engine otherwise well designed heating may be due also to the concentration of the load upon too small an area. This is incurable as a fault in design, but the heating which has been referred to above is the type which is preventable.

An engine may give trouble by a knock or pound at some part of its stroke. Probable causes for this knock or pound are:

(1) The main shaft out of line, so that the crank-pin is not perpendicular to the cylinder-axis (Chap. XXI).

(2) Lost motion in the pin-joints (Chap. XXIII).

(3) Lost motion of the piston-follower, or of the entire piston on its rod by reason of the slacking of the nuts or keys (Chap. XXII).

(4) The valve loose on its rod or within its yoke.

(5) A shoulder in the cylinder, worn in the bore, which some change in the length of the mechanism causes the piston to strike.

(6) A side motion of the piston forced against the side of the bore when the steam comes on a piston which overlaps the port.

(7) An up-and-down motion of the piston toward the middle of its stroke by a deflection of the guides under the oblique pressure from the connecting-rod.

(8) A loose guide, or the cross-head, does not have full contact against the guide at all points.

(9) Defective proportion of the steam-pressure to the weight of the reciprocating parts, so that the effort of the steam does not reach the crank-pin until after the latter have been accelerated. Delayed admission of steam produces the same effect.

(10) Improper compression, so that the lost motion necessary in the bearings for lubrication is taken up upon the crank-pin instead of upon a steam-cushion in the cylinder. Excessive compression may lift the valve, whereby a knock occurs when the valve returns to its seat.

The renewing of packing in stuffing-boxes of rods and stems is scarcely to be considered under the head of an accident, but belongs rather to the general maintenance of an engine in its proper working condition. In a special class also are accidents to the employees, due to their own negligence, or unavoidable by them. These of course do not belong to the class of accident or mishap to the machinery.

CHAPTER XXXIV.

TESTING THE POWER PLANT FOR ECONOMY AND EFFICIENCY.

545. General. The testing of the power plant belongs to a department which has been called experimental engineering and whose practitioners have been called steam experts. It forms a field too large to receive more than general allusion in a treatise such as this. The object in any power plant will be to ascertain whether the energy supplied in the form of fuel and liberated as heat in the furnace is being utilized as well as it might be, and with as great economy as possible; and further, to find, if such is not the case, at what points improvement and elevation of standard are to be sought. In a plant consisting of engine and boiler or a number of both it is obvious that there is an efficiency of the plant as a whole, and there is an efficiency of the boiler and efficiency of the engine separately. Such questions also come to the manager in control of a power plant when new appliances which are called improvements are presented for adoption. It is undoubtedly a stimulus to the operators of a power plant to know that at certain convenient intervals the efficiency of the plant is to be observed by the conduct of proper tests.

Power-plant tests for economy and efficiency will involve tests of the boiler plant, and of the engine plant either together or separately. There is usually also an acceptance test to ascertain if the guarantees and the specifications have been complied with and met.

546. The Boiler Test. The boiler test is made to secure the data for the heat balance (paragraf 144). In general terms this involves weighing the water supplied to the boiler through the feed-pump in a given time, and the coal charged during that same time. The ash and incombustible matter withdrawn from the ash-pits are to be subtracted from the coal burned as a means of finding out the percentage of ash and crediting the boiler with the actual combustible supplied to it, and the steam passing off through the steam-pipe should be sampled at frequent intervals during the test to see whether it is delivering evaporated water in the form of steam, or is entraining water through the pipe of the engine without forming steam. It is obvious that to refrain from this check upon the quality of steam is to credit the boiler with evaporating more water and disposing of more heat than it actually did, and therefore to increase in the result the amount of water really and effectively handled by the boiler in a given time. The weight of coal charged into the furnace is determined by scales of any reliable structure which will read to a quarter of a pound, and the weight of water by having two tanks similarly mounted on scales into which the suction-pipe of the pump can be placed alternately, and the weight of water fed determined by the difference between the initial and final readings as each tank is alternately filled and emptied. It is usual to have an observer specially detailed for the coal and the water scales, with blanks upon which he makes the entries as observed and which form the log of the test. Meters may be used to check the weighings.

547. Flue-gases. It is desirable in a boiler-test to know whether the products of combustion escaping from the setting are carrying an unnecessary amount of heat to waste, and whether the furnace-gases are of proper constitution with respect to waste of fuel in them or excess of oxygen reducing the temperature in the furnace. The temperature of the flue-gases can be observed by a standard pyrometer, if such are at hand; or a very close result can be obtained by the method with a ball or mass of iron inserted in the flue until it acquires the temperature of the gases, and then cooled in a known weight of water whose rise of temperature in cooling the mass of iron is observed. The volume of the flue-gases or the weight of the products of combustion can be ascertained from the readings of a gauge introduced so as to determine the difference of pressure within the flue and outside of it. The composition of the chimney-gases is determined by gas-analysis methods, the best known apparatus being that of Orsat. Such appliances are specially directed to determine the amount of carbonic oxide, carbonic acid, and oxygen.

Coal calorimeters for observing the calorific power of the fuel have already been referred to in paragraf 287.

548. The Calorimeter. There are several forms of calorimeter which are used to determine the quality of the steam or the percentage of moisture which it contains, in order to correct the record of the scales which weigh the feed-water. These instruments withdraw from the main steam-pipe a sample of the steam which is passing through it by means of a nipple which crosses the pipe, and suitable perforations in it withdraw the material which is passing through the pipe at all its sections. The material drawn out through such a nipple is then analyzed by the calorimeter proper of which there are many forms.

The most accepted of current practice is a combined separating and throttling calorimeter, in which the water in the sample taken from the pipe is first separated, and then the steam analyzed by passing it through a throttle orifice whereby it becomes superheated. The heat necessary for evaporating the water which it contains is measured by the difference in reading of thermometers inserted into the instrument. Other types of calorimeters are the coil-calorimeter, in which the determination of the percentage of moisture is based upon the amount of heat necessary to condense the mixture which passes through a coil, and the barrel-calorimeter, in which a sample of the mixture from the nipple is taken into the barrel of water through a flexible hose for similar condensation. The determinations are made by observing the difference in the heat-units required to condense the mixture as compared with what would be required if it was altogether steam.

549. Report of a Boiler-test. The importance of a reasonably close agreement in methods for the conduct of a boiler-test have induced engineers to attempt to agree upon such standard methods, and a uniform method of tabulating and reporting them, together with the calculations which are involved in making the deductions from a boiler-test. The headings of such a standard form of report will be found in the Transactions of the American Society of Mechanical Engineers, to which those interested are referred.* The points covered are the dimensions and proportions of the boilers and furnaces, pressures, temperatures, analysis of fuel, ash quantities, calorific power of fuel and quantities, steam quality, water weights fed, horsepower developed, economic results, efficiency, cost of evaporation, smoke data, methods of firing, gas analysis, and include eighty-eight separate headings for a complete treatment. In the Appendices of the Committees' Report are many valuable data and suggestions.

550. The Engine-test. It has already been made apparent (paragrafs 6 and 7) that for many engines the resistance appears in a form in which it can be directly measured, so as to determine the net or effective work received from the engine. Such cases would be where the work of the engine is pumping or hoisting, or the generation of electric energy. In many cases, however, where the resistance of the engine consists in driving the transmissive machinery of large establishments, the net resistance is not directly measurable, and the only method of determining the power and work of the engine is by means of measurements made upon the effort in the cylinder. Moreover, under many circumstances the insertion of the measuring apparatus between the motor engine and the net resistance would be inconvenient or impossible. This limitation of direct measurement is often imposed by the magnitude of the units involved, if for no other reason. If

* Transactions Am. Soc. Mech. Engrs., Vol. 21, 1899–1900, Paper No. 827. Address for separate copies No. 29 West 39th St., New York, N. Y.

the power is small enough to be conveniently determined by direct measurement, the apparatus used for this purpose will be called a dynamometer. If the work is to be measured in the engine-cylinder, the instrument used will be called an indicator.

For detailed discussion of dynamometers of the absorption and transmitting types, and for the discussion of the indicator, its forms and calibration, the reader is referred to admirable special treatises. The treatments in paragrafs 3, 295, 302, 304, 305 should have made clear what deductions can be made from the indicator diagram.

The American Society of Mechanical Engineers in 1902–03 formulated also a standard code for the Testing of Engines, along lines similar to those for the conduct of tests of boilers.* It covers dimensions of the engine, boilers and auxiliaries, quantities of water, steam, and coal, pressures and temperatures, heat measurements, data of the indicator diagrams, speed, power, efficiency results, and ratios, and special data needed for tests of special types of engine or for particular services. The full report covers 121 headings. The preamble and appendices have also much important scientific and practical information, carrying its scope beyond the limits of the present treatment.

* Transactions Amer. Soc. Mech. Engrs., Vol. 24, 1902–03, Paper No. 973.

PART VIII.

CHAPTER XXXV.

GENERAL REMARKS UPON THE POWER PLANT.

555. Concentrated or Subdivided Steam-power. There are two policies possible in the design of a power plant where the resistance to be overcome is extended over a large number of units, tools, machines, or whatever. The power may be liberated from the fuel in a central location and transformed into motor energy in a large engine near the boiler plant, and from this large engine power may be transmitted by shafting and belting all over the plant for use as required. The other plan is to carry the power in the form of steam or other energy to a large number of small steam-engines or motors located at convenient points and each of which drives its own section or group of machines.

There is no question as to the wisdom of concentrating the generating or power-furnace plant, whichever of the other two systems be considered advisable. The reason for this is that the handling of fuel and of ashes and superintendence of the boiler plant is made economical in proportion as the number of these units is large when they are concentrated under one superintendence and in one place. The fire-risk and insurance problem is also diminished by the scheme of concentration. It becomes of advantage to use mechanical methods for handling fuel where large numbers of horsepower are concentrated and where one mechanical plant can serve for them all. The cost of stack or artificial-blast appliance is less per unit when they are together.

Much the same arguments are to be urged for the principle of driving the plant from one central engine. The concentration of supplies, repairs, and superintendence, which will vary with the number of engines and not with their size, all point to the advantage of this system, as in the case of the boiler plant. There is the further advantage that the large engine will be more economical in proportion than the individual small ones, furnishing in the aggregate the same amount of horsepower. This is one of the arguments for the central-station method of furnishing power for street-railway propulsion rather than by individual motors. With the central engine the loss in transmitting its power by shafting, belting, or similar means to the individual and subdivided machines is a loss in friction; and furthermore, with some exceptions it will be necessary to drive the whole plant of transmissive machinery in order to run a small section of it for work overtime or where it must not be intermitted, as in the boring of cylinders and such work. Moreover, the failure of the central engine or any part of the transmission machinery makes it necessary to stop the entire establishment. With subdivided power only the part affected need be isolated for repair, while the rest runs on without interruption.

With the system of subdivided power among small engines the transmission loss is from condensation of steam in the pipes which connect the boiler plant to the various engines, which is probably, with an efficient system of non-conducting coverings (Chapter XIV), less than the loss by friction expressed in percentage of the whole power furnished to the piping system. This plan furthermore has the advantage that only the section of the plant which is desired need be run for overtime or special work, and the system is further flexible if it is desired to run one engine with its attached machinery at higher speed or slower than the normal. The aggregate first cost of the number of engines, if of the same character as to workmanship as the single large engine, when the cost of foundations and pipe and of drip and exhaust connections is added, is likely to exceed the first cost of the large engine. On the other hand, the whole power from the plant does not have to pass through the first set of transmissive shafting, but the principle of subdivision enables each section of shafting and its corresponding pulleys to be lighter in proportion, diminishing the friction which is caused by weight, and failure of one engine or main belt does not arrest the whole plant.

556. Distribution of Power by Electricity, Gas, or Air. In addition to the methods of transmitting power by steam or shafting discussed above, the methods of distributing by other transmission systems should be considered. The first plan is that of using an electric generator in connection with one or more central engines, from which the power will be distributed by wires carrying the current to the sections driven each by its own independent motor, or to separate machines each with its own motor. The cleanliness, convenience, compactness, and easy control of the electric transmission makes it very attractive, and the loss in the conversion of the steam energy into electric energy and its transmission and reconversion into motion are apt to be about the same as the losses in friction in high-grade plants, and will be less than such losses where settling or careless management has permitted the transmissive machinery to deteriorate in quality. If but one generator is used, there is the same difficulty as with the central engine in the previous paragraph, that a breakdown of that central engine stops the entire plant; but this can be met by either duplicate engines, or by the principle of subdivision in the power house, where the aggregate of several units makes up the entire source of energy, and they are not likely to fail all at once. The expense of multiplying motors must be considered in this system, although it must not be forgotten that with it the cost of shafting, hangers, and pulleys does not have to be incurred, and serves to offset the cost of motors. Moreover the whole shafting does not have to be run for a few tools; and no power is wasted for any tool or machine while not actually running. This may be a large proportion of the time of working days.

Only since the commercial problem of the storage of electrical energy has been successfully solved, have electrical transmission systems ceased to offer the same objections which belong to the preceding plans, that there is no storage of energy when not wanted, to be given out when it is called for. This is a great advantage which is offered by the use of gas-engines operated by gas made in a producer and stored in a holder. The gas-engine operates only when wanted and when gas is shut off from it all expense connected with it stops except interest. Gas can be made at maximum efficiency for a short period, and then the expense connected with its generation stops until the supply is exhausted. The system possesses all the other advantages of subdivided power.

The distributing of power by compressed air for motors has not been widely extended by reason of the usual absence of any conditions which make the use of the exhaust-air convenient or desirable at the place where it is discharged. In mining practice and similar places this is an immense advantage for compressed-air machinery, which is furthermore clean and convenient. There is a loss in the double conversion at the air-compressor in the power house, and the reconversion at the air-engine, which is only to be offset by the use of extra heating appliances at the motor whose cost must be charged to the method of transmission. This in no way is to be considered as an argument against the convenience of compressed air for many machines of the portable or detached character.

557. Location of a Power Plant. The choice of a location for a power plant is often fixed by considerations over which the engineer has no control. When such control is possible the considerations directing the choice of a location are mainly those of good sense and experience with respect to some of the following points:

(1) It must be accessible for the delivery of the fuel-supply and for the removal of ashes. In cities with a waterfront so that coal can be carried directly into the storage-bins from boats a considerable saving in cost per ton is to be effected, and this points to the selection of such

water-front when otherwise convenient and possible. In the absence of water-transportation, the railway and the possibility of use of sidings from it are important features. It has already been discussed (Chapter XIII) that the delivery of coal into a boiler-room by gravity diminishes the cost of a plant, but the fuel can as well be elevated within the power plant as without it. In cities where the transportation within the streets may be interrupted by the winter snows it is important that a sufficient storage capacity should be supplied in the plant to prevent possibilities of stoppage if there should be any interruption of regular transportation. In Figs. 691 and 692 are presented two typical large water-front power plants from the practice of New York City, which well deserve critical and careful study in details of arrangement and planning to meet the obstacles of their enforced location.

(2) The water supplied to a power plant is a vital question, and a disregard of it in advance has often increased the operating expenses considerably. In most cities the water for a power plant is metered from the city or water company's mains, so that a fixed charge per annum for water is an element which must be considered. If condensation is to be effected, a supply of water for this purpose is also required, and in a large plant it becomes a very considerable quantity. It is quite usual to obtain this water of condensation from wells sunk within the grounds of a power plant, and a nearness to large bodies of water in streams or rivers is of manifest advantage in this respect. It is often found that well-water either from deep artesian wells or the driven-well sources is apt to contain matter deleterious to boilers, rendering such water unfit for steam-making. References to methods for saving water used in condensation have been given in Chapter XXXI.

(3) Proximity to the water-front or the railway often favors the third element in selecting a location, which is to find a place where the smoke from the furnaces discharged through the chimneys shall not make the power plant a nuisance in the view of the neighborhood. The large chimney-stacks, if that method of draft is chosen, are useful rather than ornamental outside of the industrial district of cities; and if by the use of artificial draft or from the nature of the power plant (paragrafs 284 to 286) there is noise within it or an unpleasant vibration caused by the engine exhausts or other reciprocating motion, it may give rise to obstacles, legal and otherwise, to the satisfactory operation of the plant.

(4) The securing of draft from the chimney-stack, if natural draft is used, must be sought by locating the stacks in such a relation that surrounding conditions shall not prevent their satisfactory working.

High buildings either in the line of prevailing winds to windward or to leeward of a stack will make conditions unfavorable to it.

(5) The cost of the ground is also likely to be affected by the location chosen with regard to the previous conditions, but in their absence it cannot be disregarded. It is usually to be foreseen that the power-plant will grow with the increased demands which are likely to be put upon it, and the obtaining of the necessary land for such growth is a matter to be considered to some extent in location.

558. Construction of a Power House. The construction of a power house will be conditioned very largely by the price of land and the ground which it may be allowed to cover. If ground is not expensive there are great advantages in making the power plant all on the ground-level, both engines and boilers. It is desirable not to put them in the same room with no separating partition, by reason of the heat and dust which the fires cause and the moisture in the air which comes from leakage and evaporation. If the boilers and engines must be under one roof, a separating partition with as few door-openings as possible is necessary. It is desirable on account of fire-danger to keep the engines and boilers within separate fire-walls.

If ground is too costly to permit this arrangement, the boilers and engines must be arranged vertically in successive stories, and it becomes a question whether to put the great weight of the stationary boilers on the ground or in a basement, and the lighter weight, but moving masses, in the upper layers, or to reverse this arrangement. The older plan was to put the boilers and coal below, and the engines above. It is interesting to observe the reversal of this system in some modern power plants in the larger cities where ground must be economized. The revolving machinery is put in the basement on the ground, where its vibration can produce the least effect upon the walls and floors. The dead load of the boilers and their contents is borne upon the next tier of floors, and the coal-bins are put at the top of all with elevators or hoists to fill them conveniently from the street-levels. This offers manifest advantages in economy of handling material, and the only objection to be urged is the slightly diminished effective height of chimney which is caused by elevating the boilers.

The construction of the power house will be conditioned somewhat by the foregoing principles. If it is a single-story building, the ordinary construction of brick walls with proper foundations and a light iron roof is the typical and approved design. It is, however, exceedingly convenient in the power plant to have it commanded by a travelling crane spanning from wall to wall, so that the rapid handling of machinery in case of repair or substitution or extension is possible with such

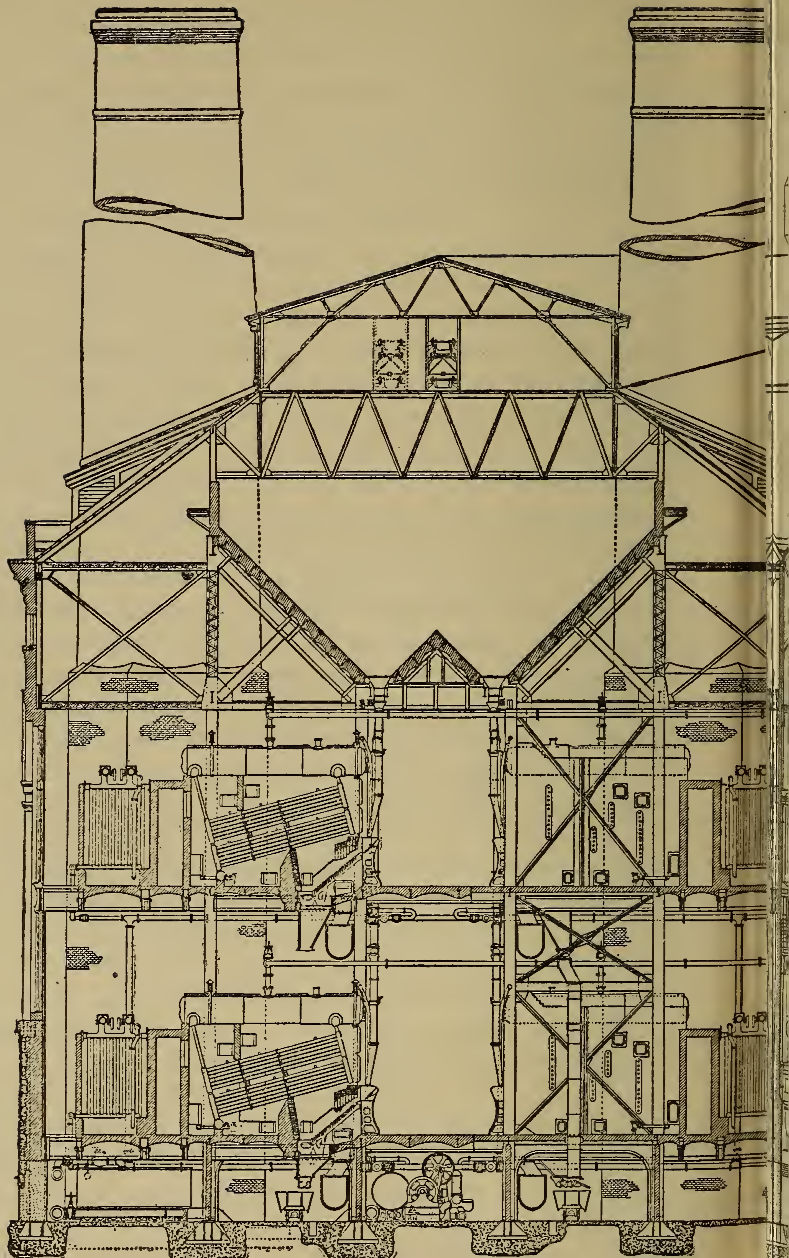
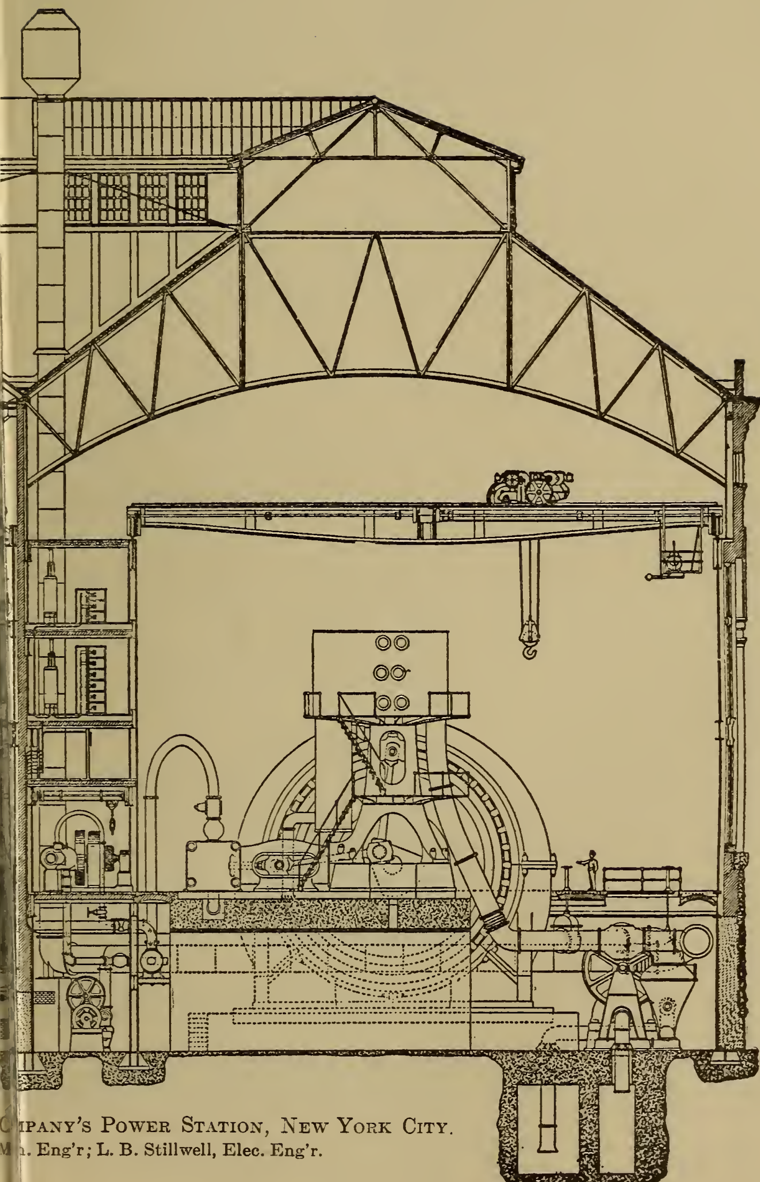
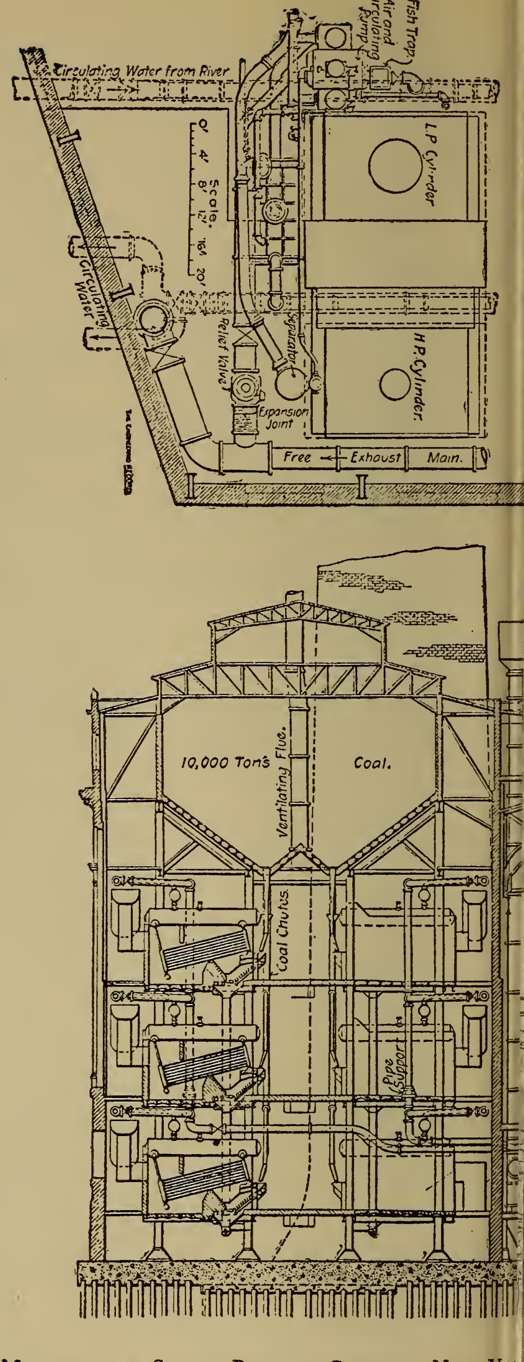


Fig. 691. Cross-Section, Manhattan Railway
George H. Pegram, Ch. Eng'r; W. E. Baker

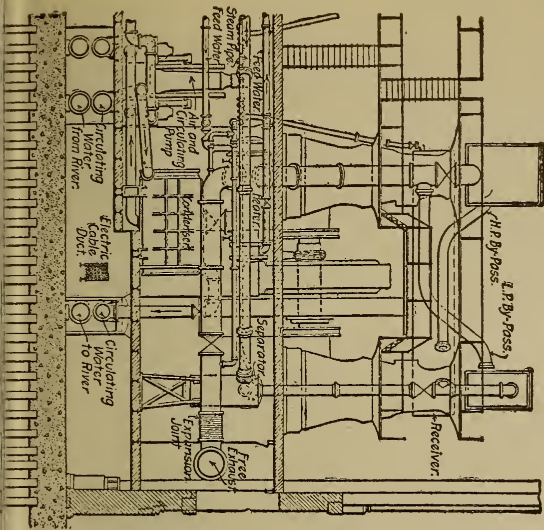


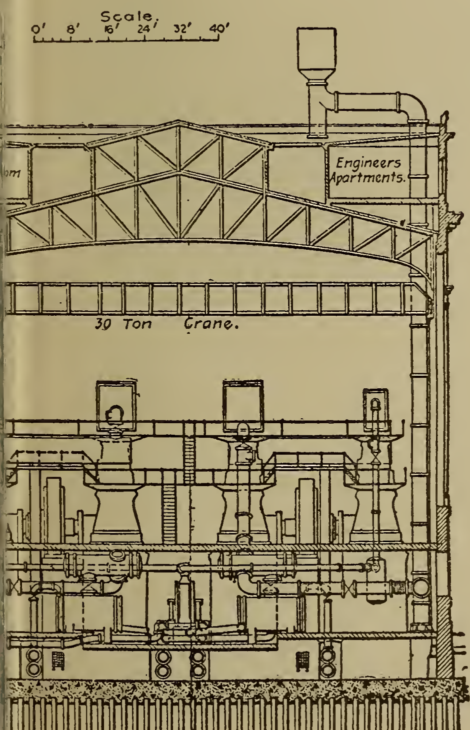
C[?]IPANY'S POWER STATION, NEW YORK CITY.
M[?]. Eng'r; L. B. Stillwell, Elec. Eng'r.



METROPOLITAN STREET RAILWAY COMPANY, NEW YO

F. S. PIERSON, CONSULTING ENGINEER.


Steam Pipe
Feed Water
Feed Water
Air and Circulating Pump
Circulating Water from River
Electric Cable Duct
Condenser
Heater
Separator
Circulating Water to River
Expansion Joint
Free Exhaust
Receiver
H.P. By-Pass
L.P. By-Pass


Scale.
0' 8' 16' 24' 32' 40'
Engineers Apartments.
39 Ton Crane.

facilities. In the two-story or many-storied power plant the construction becomes the more costly, by reason of the weight to be provided for per square foot of floor-space, and of the necessity for fire-proof construction and of the weights which come upon the walls. This opens a department of the subject with which the present limits of subject and space make it impossible to cope, and which belong to the department of the structural engineer. In such a building the provision for growth by addition of engine-units is to be foreseen and provided for, since the limitations imposed by the wall-construction are positive and fixed.

559. Arrangement of the Power Plant. It is a conceded principle of power-house practice for public use that the machines must be in whole or in part in duplicate, so that the failure of any part shall not necessitate an entire stoppage of the supply of power to users. If, therefore, the duplication is only partial, the failure of some detail in those departments of which there is but one example may cripple the whole plant. It is usual to have spare boilers and spare engines in a large plant, but there are many in which there is but one steam-pipe, and in which a failure or accident to the pipe would be as fatal as to the motive machinery itself. Some of the best and newest power plants have everything duplicated, so that there are practically two plants in one. It is an advantage if the principle of subdivision in the plant itself is carried out to make the power units of different capacity, so that when the demand for power varies it may be made by running units of different size to their full capacity, which is their most economical working. This is better than to have large units but partly loaded and running at a disadvantage for most of the time.

Where the plant is to be driven from a single engine by belts and shafting, the engine should be at or near the center of the length of the main line of shafting. This diminishes the weight of shafting and friction, because only half as much power has to be transmitted by the torsion of each half-length if the resistance is wisely distributed. This plan gives rise to a ground-plan which develops into a capital letter H, the power plant being in the cross-bar between the two buildings.

There are special details of construction belonging to power plants which drive electric generators, as to the use of iron nails in floors and walls, which belong specifically to that department.

560. Fire Protection of the Power Plant. The structural methods to be observed in power plants with respect to danger from fire have been a special study by the insurance companies. The conditions in general are that whatever woodwork there is in the power house should be massive, and the least possible space left concealed where the fire might start and lurk undetected until it had acquired headway. The construction of fire-doors and shutters to prevent the passage of fire and flame through walls is also a matter of some importance.

561. Floors of the Power Plant. The floors of a machine-shop or engine-room are very important features of the building. A concrete floor either of the ordinary construction or made of some of the proprietary materials is the suitable arrangement for the fire-room space where not exposed to cracking from heat of ashes and similar condition. It is usual to lay down a fire-brick pavement close to the boilers, and the hot ashes and clinkers should be kept upon it. Ordinary brick or flagstone paving-stones will be found in many places where cement or artificial stone is costly or inconvenient. It is desirable that the floor should be one on which an abundance of water can be used for washing, and which should be arranged to drain itself into suitable catchbasins and drains by the grades used. A brick or concrete floor is not desirable for a room containing machinery, by reason of the continual grit which is worn from the floor by treading on it, and which currents of air carry into the revolving bearings.

For the engine-room a wooden floor in two thicknesses is quite usual. The standard basement floor of the fire-insurance companies is a two-inch plank tongued and grooved, and laid on asphaltic concrete, while above that the floor-boards proper, $1\frac{1}{4}$ inches thick, are blind-nailed. For upper floors the plank is 3-inch. The upper surface is the part subject to injury from weights upon it, and can be removed when worn without disturbing the main floor-surfaces. The floor of an engine-room should be of a structure which shall not be slippery by reason of oil which may get upon it. In electric-power stations wooden floors are of special significance, because a brick or cement flooring makes a sufficiently good electrical connection with the ground to make accidental contact with a dynamo dangerous to a man standing on such floor, while with wood he is adequately insulated.

The subject of construction of industrial buildings is too broad a one to be more than hinted at in such a connection as the foregoing, and the interested reader is referred to more extended discussions for exhaustive treatment.

APPENDIX I.

HISTORICAL SUMMARY.

B.C. 120. Hero of Alexandria describes a steam reaction-wheel in his *Spiritalia seu Pneumatica.*

A.D. 1601. Giovanni Battista della Porta in his *Pneumatics* describes condensation of steam in a closed vessel as a means of lifting water.

1615. Solomon de Caus (*Les Raisons des forces Mouvantes*) describes raising water by steam-pressure above it in a closed vessel.

1629. Branca describes turning a wheel by jet of steam against vanes.

1663. Edward Somerset, second Marquis of Worcester, describes in his *Century of Inventions* a separate boiler whose pressure was admitted upon water in a closed vessel.

1680. Denis Papin invents the digester for boiling at high pressure.

1680. Huyghens proposes a true cylinder with piston traversing it.

1681. Denis Papin invents the lever safety-valve.

1690. Denis Papin applies the piston to receive motor-pressure, the cylinder being also the boiler.

1697. Thos. Savery pumps by forcing water up by pressure, and lifts water into the chambers by the vacuum caused by condensing.

1698. Savery's first patent for a pumping-engine.

1705. Papin applies lifted water to turn a rotating wheel. Uses internal fire-box boiler.

1705–07. Thos. Newcomen and John Cawley, with Savery, combine separate boiler, cylinder and piston, and surface condensation. The Atmospheric Engine.

1716–18. Dr. Desaguliers improves Savery engine by using jet condensation.

1713. Automatic valve-gear attributed to Humphrey Potter.

1718. Plug-tree valve-gear for pumps designed by Henry Beighton.

1725. Leupold designs a high-pressure non-condensing engine.

1730–58. Smeaton improves Newcomen engines.

1763–64. James Watt repairs model of Newcomen engine at Glasgow University.

1766. William Blakey proposes a water-tube boiler.

1769. Watt's patent of separate condenser.

1781. Jonathan Hornblower invents double-cylinder or compound engine.

1782. Watt's patent of expansive working of steam and double-acting engine.

1784. Watt patents parallel motion, governor and indicator.

1799. Murdock invents the three-ported slide-valve.

1800. Trevithick in England and Oliver Evans in America introduce high-pressure non-condensing engines.

1804. Arthur Woolf combines two-cylinder engine of Hornblower type with higher steam-pressure and Watt's condenser.

1804. John Stevens designs a sectional boiler.
1821. Julius Griffith designs a sectional water-tube boiler.
1838. S. Hall uses a surface condenser on S. S. Wilberforce.
1840–42. Stephenson link-motion introduced.
1840–45. Shepard & Co. of Buffalo introduce the plug-valve with loose spindle for steam-distribution.
1841. F. E. Sickles patents a drop or trip cut-off.
1841–44. Henry R. Worthington invents and introduces direct-acting pumps without fly-wheel and with valve thrown by stored energy in springs and by steam-pressure on auxiliary engine-piston: also later the duplex pump.
1849. Geo. H. Corliss introduces a trip-gear, combined with wrist-plate, and plug-valves.
1849. B-valve designed by Henry R. Worthington for pumps.
1854. Randolph & Elder introduce compound engine for vessels.
1855. Greene trip-valve gear introduced.
1856. Stephen Wilcox uses inclined water-tubes for a boiler.
1857. Charles T. Porter invents the central-weighted or Porter governor.
1859. Radial valve-gear proposed by Hackworth.
1859. Independent circulating-pump used for condenser of S. S. Moulton.
1859. John F. Allen invents a valve-gear having a variable cut-off with positive movement, and introduces the multiport principle, and in 1863 makes it a balanced valve.
1860. Chas. T. Porter employs these inventions of Mr. Allen to make a high-speed engine with automatic cut-off.
1860. Charles T. Porter invents the isochronous spring-governor, using a spring with initial tension so as to exert a resistance which varies directly as the diameter of the circle described by the balls. This underlies the shaft-governor.
1860. Chas. B. Richards invents the first indicator in which the motion of the piston was multiplied.
1868. Hartnell of England patents control of throw of eccentric by revolving weights balanced by springs, in plane of rotation of the engine-shaft, but moves eccentric from forward towards backward position, and not towards the center of the shaft.
1872–73. John C. Hoadley applies the balanced-spring shaft-governor to control the throw of a single piston-valve in an automatic cut-off engine by moving eccentric across the shaft and giving invariable lead but variable cut-off release and compression
1874. A. C. Kirk introduces triple-expansion in S. S. Propontis.
1883. Gustav DeLaval of Sweden introduces the nozzle steam turbine.
1885. C. A. Parsons of New Castle, England, introduces the compound steam turbine.
1892–94 Inertia-principle of governing developed by Francis M. Rites.
1896. Chas. G. Curtis of New York introduces the stage compound nozzle steam turbine.

APPENDIX II.

STEAM TABLES.

566. For convenient reference in computations involving the properties of steam, the following tables are reproduced.

Note. The data have been taken from the published work of Prof. C. H. Peabody except where otherwise stated. The data above 300 pounds absolute are not reliable where the specific heat has entered as a factor, since its value at these higher pressures has not been fixed by general acceptance.

The columns after No. 5 have been calculated as follows:

$$\lambda = 1091.7 + 0.305(t - 32);$$

$q = 1 + 0.00004t + 0.0000009t_2$ in centigrade units, reduced to Fahr.;

$$r = \lambda - q = \text{Col. } 6 - \text{Col. } 7;$$

$$r_1 = r - \frac{pu}{778} = \text{Col. } 8 - \text{Col. } 10;$$

$$\text{Col. } 10 = (\text{Col. } 1 \times 144) \times \text{Col. } 4 \div 778;$$

$$\text{Col. } 11 = \text{Spec. Ht.} \times \text{Hyp Log} \frac{T}{T_0} \quad \text{(see table, paragraf 567)};$$

$$\text{Col. } 12 = \frac{\text{Column } 8}{\text{Column } 3} + \text{Column } 11; \text{ or } \phi_s = \frac{L}{T} + \phi_w.$$

APPENDIX II. STEAM TABLES.

Pressure above Vacuum in Pounds per Sq. Inch.	Temperature, Degrees Fahr.	Temperature, Degrees Absolute.	Volume of 1 Pound in Cu. Feet.	Weight of 1 Cu. Ft. in Pounds.	Total Heat. λ	Heat of Liquid. q	Heat of Vaporization. r B.T.U.	Heat Equiv. of Internal Work. r_1	Heat Equiv. of External Work. $\frac{pu}{778}$	Entropy of Liquid. $\phi_w = \theta$	Entropy of Vapor. ϕ_s	Pressure in Pounds by Gauge.
1	2	3	4	5	6	7	8	9	10	11	12	1
0.085	32 0	492.70	3336.0	.00030	1091.7	0	1091.7	....	52.4	0	2.215	−14.615
0.5	80.0	540.70	640.8	.00158	1106.3	48.04	1058.3	....	58.8	.095	2.052	−14.2
1	101.99	562.69	334.6	.00299	1113.1	70.0	1043.0	981.1	61.9	.1329	1.9845	−13.7
2	126.27	586.97	173.6	.00576	1120.5	94.4	1026.1	961.9	64.2	.1754	1.9235	−12.7
3	141.62	602.32	118.4	.00844	1125.1	109.8	1015.3	949.5	65.8	.2013	1.8869	−11.7
4	153.09	613.79	90.31	.01107	1128.6	121.4	1007.2	940.4	66.8	.2203	1.8612	−10.7
5	162.34	623.04	73.22	.01366	1131.5	130.7	1000.8	933.1	67.7	.2353	1.8416	− 9.7
10	193.25	653.95	38.16	.02621	1140.9	161.9	979.0	908.4	70.6	.2842	1.7812	− 4.7
11	197.78	658.48	34.88	.02866	1142.3	166.5	975.8	904.8	71.0	.2912	1.7730	− 3.7
12	201.98	662.68	32.14	.03111	1143.6	170.7	972.9	901.5	71.4	.2976	1.7657	− 2.7
13	205.89	666.59	29.82	.03355	1144.7	174.6	970.1	898.4	71.7	.3035	1.7588	− 1.7
14	209.57	670.27	27.79	.03600	1145.8	178.3	967.5	895.5	72.0	.3091	1.7525	− 0.7
14.7	212.00	672.70	26.42	.03794	1146.6	180.9	965.7	893.3	72.4	.3122	1.7477	± 0
15	213.03	673.73	26.15	.03826	1146.9	181.8	965.1	892.6	72.5	.3143	1.7467	+ 0.3
16	216.32	677.02	24.59	.04067	1147.9	185.1	962.8	890.0	72.8	.3192	1.7412	1.3
17	219.44	680.14	23.22	.04307	1148.9	188.3	960.6	887.6	73.0	.3228	1.7351	2.3
18	222.40	683.10	22.00	.05547	1149.8	191.3	958.5	885.3	73.2	.3282	1.7313	3.3
19	225.24	685.94	20.90	.04786	1150.7	194.1	956.6	883.2	73.4	.3324	1.7269	4.3
20	227.95	688.65	19.91	.05023	1151.5	196.9	954.6	881.0	73.6	.3363	1.7224	5.3
25	240.04	700.74	16.13	.06199	1155.1	209.1	946.0	871.5	74.5	.3539	1.7039	10.3
30	250.27	710.97	13.59	.07360	1158.3	219.4	938.9	863.6	75.3	.3685	1.6890	15.3
35	259.19	719.89	11.75	.08508	1161.0	228.4	932.6	856.6	76.0	.3811	1.6765	20.3

* From "Hutton's Heat and Heat Engines."

APPENDIX II. STEAM TABLES — *continued.*

Pressure above Vacuum in Pounds per Sq. Inch.	Temperature, Degrees Fahr.	Temperature, Degrees Absolute.	Volume of 1 Pound in Cu. Feet.	Weight of 1 Cu. Ft. in Pounds.	Total Heat. λ	Heat of Liquid. q	Heat of Vaporization. r B.T.U.	Heat Equiv. of Internal Work. r_1	Heat Equiv. of External Work. $\frac{pu}{778}$	Entropy of Liquid. $\phi_w = \theta$	Entropy of Vapor. ϕ_s	Pressure in Pounds by Gauge.
1	2	3	4	5	6	7	8	9	10	11	12	1
40..........	267.13	727.83	10.37	.09644	1163.4	236.4	927.0	850.3	76.7	.3921	1.6657	25.3
45..........	274.29	734.99	9.297	.1077	1165.6	243.6	922.0	844.8	77.2	.4020	1.6564	30.3
50..........	280.85	741.55	8.414	.1188	1167.6	250.2	917.4	839.7	77.7	.4109	1.6480	35.3
55..........	286.89	747.59	7.696	.1299	1169.4	256.3	913.1	834.9	78.2	.4191	1.6404	40.3
60..........	292.51	753.21	7.096	.1409	1171.2	261.9	909.3	830.7	78.6	.4267	1.6339	45.3
65..........	297.77	758.47	6.583	.1519	1172.7	267.2	905.5	826.5	79.0	.4337	1.6275	50.3
70..........	302.71	763.41	6.144	.1628	1174.3	272.2	902.1	822.7	79.4	.4402	1.6218	55.3
75..........	307.38	768.08	5.762	.1736	1175.7	276.9	898.8	819.1	79.7	.4464	1.6165	60.3
76..........	308.28	768.98	5.691	.1757	1176.0	277.8	898.2	818.4	79.8	.4476	1.6156	61.3
77..........	309.18	769.88	5.621	.1779	1176.2	278.7	897.5	817.6	79.9	.4487	1.6144	62.3
78..........	310.06	770.76	5.554	.1801	1176.5	279.6	896.9	817.0	79.9	.4499	1.6135	63.3
79..........	310.94	771.64	5.488	.1822	1176.8	280.5	896.3	816.3	80.0	.4511	1.6126	64.3
80..........	311.80	772.50	5.425	.1843	1177.0	281.4	895.6	815.5	80.1	.4522	1.6116	65.3
81..........	312.66	773.36	5.362	.1865	1177.3	282.3	895.0	814.9	80.1	.4534	1.6106	66.3
82..........	313.51	774.21	5.301	.1886	1177.6	283.2	894.4	814.2	80.2	.4545	1.6097	67.3
83..........	314.36	775.06	5.240	.1908	1177.8	284.1	893.7	813.4	80.3	.4557	1.6087	68.3
84..........	315.19	775.89	5.182	.1930	1178.1	285.0	893.1	812.8	80.3	.4568	1.6078	69.3
85..........	316.02	776.72	5.125	.1951	1178.3	285.8	892.5	812.1	80.4	.4579	1.6069	70.3
86..........	316.84	777.54	5.069	.1973	1178.6	286.7	891.9	811.5	80.4	.4590	1.6067	71.3
87..........	317.65	778.35	5.014	.1994	1178.8	287.5	891.3	810.8	80.5	.4601	1.6052	72.3
88..........	318.45	779.15	4.961	.2016	1179.1	288.4	890.7	810.2	80.5	.4612	1.6043	73.3
89..........	319.25	779.95	4.909	.2037	1179.3	289.2	890.1	809.5	80.6	.4622	1.6033	74.3

APPENDIX II. STEAM TABLES — *continued.*

Pressure above Vacuum in Pounds per Sq. Inch.	Temperature, Degrees Fahr.	Temperature, Degrees Absolute.	Volume of 1 Pound in Cu. Feet.	Weight of 1 Cu. Ft. in Pounds.	Total Heat. λ	Heat of Liquid. q	Heat of Vaporization. r B.T.U.	Heat Equiv. of Internal Work. r_1	Heat Equiv. of External Work. $\frac{pu}{778}$	Entropy of Liquid. $\phi_w = \theta$	Entropy of Vapor. ϕ_s	Pressure in Pounds by Gauge.
1	2	3	4	5	6	7	8	9	10	11	12	1
90.........	320.04	780.74	4.858	.2058	1179.6	290.0	889.6	808.9	80.7	.4633	1.6027	75.3
91.........	320.83	781.53	4.808	.2080	1179.8	290.8	889.0	808.3	80.7	.4643	1.6018	76.3
92.........	321.60	782.30	4.760	.2101	1180.0	291.6	888.4	807.6	80.8	.4653	1.6009	77.3
93.........	322.37	783.07	4.712	.2122	1180.3	292.4	887.9	807.1	80.8	.4663	1.6001	78.3
94.........	323.14	783.84	4.665	.2144	1180.5	293.2	887.3	806.4	80.9	.4673	1.5992	79.3
95.........	323.89	784.59	4.619	.2165	1180.7	294.0	886.7	805.8	80.9	.4683	1.5984	80.3
100.........	327.58	788.28	4.403	.2271	1181.9	297.9	884.0	802.8	81.2	.4733	1.5947	85.0
105.........	331.13	791.83	4.206	.2378	1182.9	301.6	881.3	799.9	81.4	.4780	1.5909	90.0
115.........	337.86	798.56	3.862	.2589	1185.0	308.7	876.3	794.4	81.9	.4869	1.5842	100.0
120.........	341.05	801.75	3.711	.2695	1186.0	312.0	874.0	791.9	82.1	.4911	1.5812	105.0
135.........	350.03	810.73	3.323	.3009	1188.7	321.4	867.3	784.7	82.6	.5027	1.5724	120.0
140.........	352.85	813.55	3.212	.3113	1189.5	324.4	865.1	782.3	82.8	.5064	1.5697	125.0
150.........	358.26	818.96	3.011	.3221	1191.2	320.4	861.2	778.1	83.1	.5133	1.5648	135.0
165.........	365.88	826.58	2.751	.3635	1193.6	338.0	855.6	772.0	83.6	.5230	1.5581	150.0
175.........	370.65	831.35	2.603	.3841	1195.0	343.0	852.0	768.2	83.8	.5290	1.5538	160.0
180.........	372.97	833.67	2.535	.3945	1195.7	345.4	850.3	766.4	83.9	.5319	1.5518	165.0
185.........	375.23	835.93	2.470	.4049	1196.4	348.2	848.6	764.6	84.0	.5347	1.5498	170.0
190.........	377.44	838.14	2.408	.4153	1197.1	350.1	847.0	762.9	84.1	.5375	1.5492	175.0
195.........	379.61	840.31	2.349	.4257	1197.7	352.4	845.3	761.1	84.2	.5402	1.5461	180.0
200.........	381.73	842.43	2.294	.4359	1198.4	354.6	843.8	759.5	84.3	.5429	1.5435	185.0
205.........	383.82	844.52	2.241	.4461	1199.0	356.8	842.2	757.8	84.4	.5454	1.5426	190.0
210.........	385.87	846.57	2.190	.4565	1199.6	358.9	840.7	756.2	84.5	.5480	1.5410	195.0

APPENDIX II. STEAM TABLES — *continued.*

Pressure above Vacuum in Pounds per Sq. Inch.	Temperature, Degrees Fahr.	Temperature, Degrees Absolute.	Volume of 1 Pound in Cu. Feet.	Weight of 1 Cu. Ft. in Pounds.	Total Heat. λ	Heat of Liquid. q	Heat of Vaporization. r B.T.U.	Heat Equiv. of Internal Work. r_1	Heat Equiv. of External Work. $\frac{pu}{778}$	Entropy of Liquid. $\phi_w = \theta$	Entropy of Vapor. ϕ_s	Pressure in Pounds by Gauge.
1	2	3	4	5	6	7	8	9	10	11	12	1
215.......	387.88	848.58	2.142	.4669	1200.2	361.0	839.2	754.6	84.6	.5504	1.5393	200
225.......	391.79	852.49	2.051	.4876	1201.4	365.1	836.3	751.6	84.7	.5553	1.5363	210
240.......	397.41	858.11	1.928	.5186	1203.2	371.0	832.2	747.3	84.9	.5621	1.5319	225
250.......	400.99	861.69	1.854	.5393	1204.2	374.7	829.5	744.5	85.0	.5665	1.5292	235
260.......	404.47	865.17	1.785	.5601	1205.3	378.4	826.9	741.7	85.2	.5707	1.5264	245
270.......	407.85	868.55	1.722	.5809	1206.3	381.9	824.4	739.2	85.2	.5748	1.5239	255
280.......	411.12	871.82	1.662	.6020	1207.3	385.3	822.0	736.7	85.3	.5787	1.5215	265
290.......	414.32	875.02	1.607	.6220	1208.3	388.6	819.7	734.3	85.4	.5826	1.5193	275
300.......	417.42	878.12	1.554	.6440	1209.3	391.9	817.4	732.0	85.4	.5863	1.5171	285
350*......	432.0	892.70	1.325	.7545	1213.7	406.3	807.5	721.7	85.8	.5878†	1.4923	335*
400.......	444.9	905.60	1.167	.8572	1217.7	419.8	797.9	711.5	86.4	.6043	1.4853	385
450.......	456.6	917.30	1.042	.9595	1221.3	432.2	789.1	702.4	86.7	.6152	1.4754	435
500.......	467.4	928.10	0.942	1.0620	1224.5	443.5	781.0	693.9	87.1	.6313	1.4728	485
550.......	477.5	938.20	0.859	1.1640	1227.6	454.1	773.5	686.2	87.3	.6419	1.4663	535
600.......	486.9	947.60	0.790	1.2660	1230.5	464.2	766.3	678.6	87.7	.6523	1.4609	585
700.......	504.1	964.80	0.680	1.4700	1235.7	482.4	753.3	665.2	88.1	.6678	1.4485	685
800.......	519.6	980.30	0.597	1.6740	1240.3	498.9	741.4	653.2	88.2	.6831	1.4393	785
900.......	533.7	994.40	0.532	1.8780	1244.7	514.0	730.6	642.3	88.3	.6981	1.4328	885
1000.......	546.8	1007.50	0.480	2.0820	1248.7	528.3	720.3	631.5	88.8	.7129	1.4278	985

* From Buel's Tables above 350 pounds.   † Not reliable. Specific not known at these temperatures and above, and assumed to be unity.

LOGARITHMS

567. In arithmetical computations, the usual base of the system is 10, so that x, the logarithm for a number m, will be the exponent to which 10 is to be raised to give the quantity m, or $x = \log_{10} m$. In analytical mathematical work, the base generally employed is not 10, but is represented by e, whose value is $2.71828+$. To convert common or Briggs logarithms into Napierian logarithms, the former are to be multiplied by 2.3026.

The equation of the hyperbola in the form $xy =$ constant leads to the deduction that the area between the hyperbolic curve and its nearest asymptote cut-off by two ordinates parallel to the other asymptote and distant respectively from the origin by a and b will be proportional to $\log \frac{b}{a}$. Hence it will be true that the integral of $\frac{dx}{x}$ will be the hyperbolic logarithm of x. To save trouble of conversion, a table is appended covering the usual ranges required.

HYPERBOLIC LOGARITHMS

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.01	.0099	1.30	.2624	1.59	.4637	1.88	.6313	2.17	.7747
1.02	.0198	1.31	.2700	1.60	.4700	1.89	.6366	2.18	.7793
1.03	.0296	1.32	.2776	1.61	.4762	1.90	.6419	2.19	.7839
1.04	.0392	1.33	.2852	1.62	.4824	1.91	.6471	2.20	.7885
1.05	.0488	1.34	.2927	1.63	.4886	1.92	.6523	2.21	.7930
1.06	.0583	1.35	.3001	1.64	.4947	1.93	.6575	2.22	.7975
1.07	.0677	1.36	.3075	1.65	.5008	1.94	.6627	2.23	.8020
1.08	.0770	1.37	.3148	1.66	.5068	1.95	.6678	2.24	.8065
1.09	.0862	1.38	.3221	1.67	.5128	1.96	.6729	2.25	.8109
1.10	.0953	1.39	.3293	1.68	.5188	1.97	.6780	2.26	.8154
1.11	.1044	1.40	.3365	1.69	.5247	1.98	.6831	2.27	.8198
1.12	.1133	1.41	.3436	1.70	.5306	1.99	.6881	2.28	.8242
1.13	.1222	1.42	.3507	1.71	.5365	2.00	.6931	2.29	.8286
1.14	.1310	1.43	.3577	1.72	.5423	2.01	.6981	2.30	.8329
1.15	.1398	1.44	.3646	1.73	.5481	2.02	.7031	2.31	.8372
1.16	.1484	1.45	.3716	1.74	.5539	2.03	.7080	2.32	.8416
1.17	.1570	1.46	.3784	1.75	.5596	2.04	.7129	2.33	.8458
1.18	.1655	1.47	.3853	1.76	.5653	2.05	.7178	2.34	.8502
1.19	.1740	1.48	.3920	1.77	.5710	2.06	.7227	2.35	.8544
1.20	.1823	1.49	.3988	1.78	.5766	2.07	.7275	2.36	.8587
1.21	.1906	1.50	.4055	1.79	.5822	2.08	.7324	2.37	.8629
1.22	.1988	1.51	.4121	1.80	.5878	2.09	.7372	2.38	.8671
1.23	.2070	1.52	.4187	1.81	.5933	2.10	.7419	2.39	.8713
1.24	.2151	1.53	.4253	1.82	.5988	2.11	.7467	2.40	.8755
1.25	.2231	1.54	.4318	1.83	.6043	2.12	.7514	2.41	.8796
1.26	.2311	1.55	.4383	1.84	.6098	2.13	.7561	2.42	.8838
1.27	.2390	1.56	.4447	1.85	.6152	2.14	.7608	2.43	.8879
1.28	.2469	1.57	.4511	1.86	.6206	2.15	.7655	2.44	.8920
1.29	.2546	1.58	.4574	1.87	.6259	2.16	.7701	2.45	.8961

HYPERBOLIC LOGARITHMS

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
2.46	.9002	3.02	1.1053	3.58	1.2754	4.14	1.4207	4.70	1.5476
2.47	.9042	3.03	1.1086	3.59	1.2782	4.15	1.4231	4.71	1.5497
2.48	.9083	3.04	1.1119	3.60	1.2809	4.16	1.4255	4.72	1.5518
2.49	.9123	3.05	1.1151	3.61	1.2837	4.17	1.4279	4.73	1.5539
2.50	.9163	3.06	1.1184	3.62	1.2865	4.18	1.4303	4.74	1.5560
2.51	.9203	3.07	1.1217	3.63	1.2892	4.19	1.4327	4.75	1.5581
2.52	.9243	3.08	1.1249	3.64	1.2920	4.20	1.4351	4.76	1.5602
2.53	.9282	3.09	1.1282	3.65	1.2947	4.21	1.4375	4.77	1.5623
2.54	.9322	3.10	1.1314	3.66	1.2975	4.22	1.4398	4.78	1.5644
2.55	.9361	3.11	1.1346	3.67	1.3002	4.23	1.4422	4.79	1.5665
2.56	.9400	3.12	1.1378	3.68	1.3029	4.24	1.4446	4.80	1.5686
2.57	.9439	3.13	1.1410	3.69	1.3056	4.25	1.4469	4.81	1.5707
2.58	.9478	3.14	1.1442	3.70	1.3083	4.26	1.4493	4.82	1.5728
2.59	.9517	3.15	1.1474	3.71	1.3110	4.27	1.4516	4.83	1.5748
2.60	.9555	3.16	1.1506	3.72	1.3137	4.28	1.4540	4.84	1.5769
2.61	.9594	3.17	1.1537	3.73	1.3164	4.29	1.4563	4.85	1.5790
2.62	.9632	3.18	1.1569	3.74	1.3191	4.30	1.4586	4.86	1.5810
2.63	.9670	3.19	1.1600	3.75	1.3218	4.31	1.4609	4.87	1.5831
2.64	.9708	3.20	1.1632	3.76	1.3244	4.32	1.4633	4.88	1.5851
2.65	.9746	3.21	1.1663	3.77	1.3271	4.33	1.4656	4.89	1.5872
2.66	.9783	3.22	1.1694	3.78	1.3297	4.34	1.4679	4.90	1.5892
2.67	.9821	3.23	1.1725	3.79	1.3324	4.35	1.4702	4.91	1.5913
2.68	.9858	3.24	1.1756	3.80	1.3350	4.36	1.4725	4.92	1.5933
2.69	.9895	3.25	1.1787	3.81	1.3376	4.37	1.4748	4.93	1.5953
2.70	.9933	3.26	1.1817	3.82	1.3403	4.38	1.4770	4.94	1.5974
2.71	.9969	3.27	1.1848	3.83	1.3429	4.39	1.4793	4.95	1.5994
2.72	1.0006	3.28	1.1878	3.84	1.3455	4.40	1.4816	4.96	1.6014
2.73	1.0043	3.29	1.1909	3.85	1.3481	4.41	1.4839	4.97	1.6034
2.74	1.0080	3.30	1.1939	3.86	1.3507	4.42	1.4861	4.98	1.6054
2.75	1.0116	3.31	1.1969	3.87	1.3533	4.43	1.4884	4.99	1.6074
2.76	1.0152	3.32	1.1999	3.88	1.3558	4.44	1.4907	5.00	1.6094
2.77	1.0188	3.33	1.2030	3.89	1.3584	4.45	1.4929	5.01	1.6114
2.78	1.0225	3.34	1.2060	3.90	1.3610	4.46	1.4951	5.02	1.6134
2.79	1.0260	3.35	1.2090	3.91	1.3635	4.47	1.4974	5.03	1.6154
2.80	1.0296	3.36	1.2119	3.92	1.3661	4.48	1.4996	5.04	1.6174
2.81	1.0332	3.37	1.2149	3.93	1.3686	4.49	1.5019	5.05	1.6194
2.82	1.0367	3.38	1.2179	3.94	1.3712	4.50	1.5041	5.06	1.6214
2.83	1.0403	3.39	1.2208	3.95	1.3737	4.51	1.5063	5.07	1.6233
2.84	1.0438	3.40	1.2238	3.96	1.3762	4.52	1.5085	5.08	1.6253
2.85	1.0473	3.41	1.2267	3.97	1.3788	4.53	1.5107	5.09	1.6273
2.86	1.0508	3.42	1.2296	3.98	1.3813	4.54	1.5129	5.10	1.6292
2.87	1.0543	3.43	1.2326	3.99	1.3838	4.55	1.5151	5.11	1.6312
2.88	1.0578	3.44	1.2355	4.00	1.3863	4.56	1 5173	5.12	1.6332
2.89	1.0613	3.45	1.2384	4.01	1.3888	4.57	1.5195	5.13	1.6351
2.90	1.0647	3.46	1.2413	4.02	1.3913	4.58	1.5217	5.14	1.6371
2.91	1.0682	3.47	1.2442	4.03	1.3938	4.59	1.5239	5.15	1.6390
2.92	1.0716	3.48	1.2470	4.04	1.3962	4.60	1.5261	5.16	1.6409
2.93	1.0750	3.49	1.2499	4.05	1.3987	4.61	1.5282	5.17	1.6429
2.94	1.0784	3.50	1.2528	4.06	1.4012	4.62	1.5304	5.18	1.6448
2.95	1.0813	3.51	1.2556	4.07	1.4036	4.63	1.5326	5.19	1.6467
2.96	1.0852	3.52	1.2585	4.08	1.4061	4.64	1.5347	5.20	1.6487
2.97	1.0886	3.53	1.2613	4.09	1.4085	4.65	1.5369	5.21	1.6506
2.98	1.0919	3.54	1.2641	4.10	1.4110	4.66	1.5390	5.22	1.6525
2.99	1.0953	3.55	1.2669	4.11	1.4134	4.67	1.5412	5.23	1.6541
3.00	1.0986	3.56	1.2698	4.12	1.4159	4.68	1.5433	5.24	1.6563
3.01	1.1019	3.57	1.2726	4.13	1.4183	4.69	1.5454	5.25	1.6582

HYPERBOLIC LOGARITHMS

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
5.26	1.6601	5.82	1.7613	6.38	1.8532	6.94	1.9373	7.50	2.0149
5.27	1.6620	5.83	1.7630	6.39	1.8547	6.95	1.9387	7.51	2.0162
5.28	1.6639	5.84	1.7647	6.40	1.8563	6.96	1.9402	7.52	2.0176
5.29	1.6658	5.85	1.7664	6.41	1.8579	6.97	1.9416	7.53	2.0189
5.30	1.6677	5.86	1.7681	6.42	1.8594	6.98	1.9430	7.54	2.0202
5.31	1.6696	5.87	1.7699	6.43	1.8610	6.99	1.9445	7.55	2.0215
5.32	1.6715	5.88	1.7716	6.44	1.8625	7.00	1.9459	7.56	2.0229
5.33	1.6734	5.89	1.7733	6.45	1.8641	7.01	1.9473	7.57	2.0242
5.34	1.6752	5.90	1.7750	6.46	1.8656	7.02	1.9488	7.58	2.0255
5.35	1.6771	5.91	1.7766	6.47	1.8672	7.03	1.9502	7.59	2.0268
5.36	1.6790	5.92	1.7783	6.48	1.8687	7.04	1.9516	7.60	2.0281
5.37	1.6808	5.93	1.7800	6.49	1.8703	7.05	1.9530	7.61	2.0295
5.38	1.6827	5.94	1.7817	6.50	1.8718	7.06	1.9544	7.62	2.0308
5.39	1.6845	5.95	1.7834	6.51	1.8733	7.07	1.9559	7.63	2.0321
5.40	1.6864	5.96	1.7851	6.52	1.8749	7.08	1.9573	7.64	2.0334
5.41	1.6882	5.97	1.7867	6.53	1.8764	7.09	1.9587	7.65	2.0347
5.42	1.6901	5.98	1.7884	6.54	1.8779	7.10	1.9601	7.66	2.0360
5.43	1.6919	5.99	1.7901	6.55	1.8795	7.11	1.9615	7.67	2.0373
5.44	1.6938	6.00	1.7918	6.56	1.8810	7.12	1.9629	7.68	2.0386
5.45	1.6956	6.01	1.7934	6.57	1.8825	7.13	1.9643	7.69	2.0399
5.46	1.6974	6.02	1.7951	6.58	1.8840	7.14	1.9657	7.70	2.0412
5.47	1.6993	6.03	1.7967	6.59	1.8856	7.15	1.9671	7.71	2.0425
5.48	1.7011	6.04	1.7984	6.60	1.8871	7.16	1.9685	7.72	2.0438
5.49	1.7029	6.05	1.8001	6.61	1.8886	7.17	1.9699	7.73	2.0451
5.50	1.7047	6.06	1.8017	6.62	1.8901	7.18	1.9713	7.74	2.0464
5.51	1.7066	6.07	1.8034	6.63	1.8916	7.19	1.9727	7.75	2.0477
5.52	1.7084	6.08	1.8050	6.64	1.8931	7.20	1.9741	7.76	2.0490
5.53	1.7102	6.09	1.8066	6.65	1.8946	7.21	1.9754	7.77	2.0503
5.54	1.7120	6.10	1.8083	6.66	1.8961	7.22	1.9769	7.78	2.0516
5.55	1.7138	6.11	1.8099	6.67	1.8976	7.23	1.9782	7.79	2.0528
5.56	1.7156	6.12	1.8116	6.68	1.8991	7.24	1.9796	7.80	2.0541
5.57	1.7174	6.13	1.8132	6.69	1.9006	7.25	1.9810	7.81	2.0554
5.58	1.7192	6.14	1.8148	6.70	1.9021	7.26	1.9824	7.82	2.0567
5.59	1.7210	6.15	1.8165	6.71	1.9036	7.27	1.9838	7.83	2.0580
5.60	1.7228	6.16	1.8181	6.72	1.9051	7.28	1.9851	7.84	2.0592
5.61	1.7246	6.17	1.8197	6.73	1.9066	7.29	1.9865	7.85	2.0605
5.62	1.7263	6.18	1.8213	6.74	1.9081	7.30	1.9879	7.86	2.0618
5.63	1.7281	6.19	1.8229	6.75	1.9095	7.31	1.9892	7.87	2.0631
5.64	1.7299	6.20	1.8245	6.76	1.9110	7.32	1.9906	7.88	2.0643
5.65	1.7317	6.21	1.8262	6.77	1.9125	7.33	1.9920	7.89	2.0656
5.66	1.7334	6.22	1.8278	6.78	1.9140	7.34	1.9933	7.90	2.0669
5.67	1.7352	6.23	1.8294	6.79	1.9155	7.35	1.9947	7.91	2.0681
5.68	1.7370	6.24	1.8310	6.80	1.9169	7.36	1.9961	7.92	2.0694
5.69	1.7387	6.25	1.8326	6.81	1.9184	7.37	1.9974	7.93	2.0707
5.70	1.7405	6.26	1.8342	6.82	1.9199	7.38	1.9988	7.94	2.0719
5.71	1.7422	6.27	1.8358	6.83	1.9213	7.39	2.0001	7.95	2.0732
5.72	1.7440	6.28	1.8374	6.84	1.9228	7.40	2.0015	7.96	2.0744
5.73	1.7457	6.29	1.8390	6.85	1.9242	7.41	2.0028	7.97	2.0757
5.74	1.7475	6.30	1.8405	6.86	1.9257	7.42	2.0041	7.98	2.0769
5.75	1.7492	6.31	1.8421	6.87	1.9272	7.43	2.0055	7.99	2.0782
5.76	1.7509	6.32	1.8437	6.88	1.9286	7.44	2.0069	8.00	2.0794
5.77	1.7527	6.33	1.8453	6.89	1.9301	7.45	2.0082	8.01	2.0807
5.78	1.7544	6.34	1.8469	6.90	1.9315	7.46	2.0096	8.02	2.0819
5.79	1.7561	6.35	1.8485	6.91	1.9330	7.47	2.0108	8.03	2.0832
5.80	1.7579	6.36	1.8500	6.92	1.9344	7.48	2.0122	8.04	2.0844
5.81	1.7596	6.37	1.8516	6.93	1.9359	7.49	2.0136	8.05	2.0857

HYPERBOLIC LOGARITHMS

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
8.06	2.0869	8.58	2.1494	9.24	2.2235	9.90	2.2925	19.00	2.9444
8.07	2.0882	8.60	2.1518	9.26	2.2257	9.92	2.2946	19.50	2.9703
8.08	2.0894	8.62	2.1541	9.28	2.2279	9.94	2.2966	20.00	2.9957
8.09	2.0906	8.64	2.1564	9.30	2.2300	9.96	2.2986	21	3.0445
8.10	2.0919	8.66	2.1587	9.32	2.2322	9.98	2.3006	22	3.0910
8.11	2.0931	8.68	2.1610	9.34	2.2343	10.00	2.3026	23	3.1355
8.12	2.0943	8.70	2.1633	9.36	2.2364	10.25	2.3279	24	3.1781
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8.14	2.0968	8.74	2.1679	9.40	2.2407	10.75	2.3749	26	3.2581
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8.18	2.1017	8.82	2.1770	9.48	2.2492	11.75	2.4636	30	3.4012
8.19	2.1029	8.84	2.1793	9.50	2.2513	12.00	2.4849	31	3.4340
8.20	2.1041	8.86	2.1815	9.52	2.2534	12.25	2.5052	32	3.4657
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8.24	2.1090	8.90	2.1864	9.56	2.2576	12.75	2.5455	34	3.5263
8.26	2.1114	8.92	2.1883	9.58	2.2597	13.00	2.5649	35	3.5553
8.28	2.1138	8.94	2.1905	9.60	2.2618	13.25	2.5840	36	3.5835
8.30	2.1163	8.96	2.1928	9.62	2.2638	13.50	2.6027	37	3.6109
8.32	2.1187	8.98	2.1950	9.64	2.2659	13.75	2.6211	38	3.6376
8.34	2.1211	9.00	2.1972	9.66	2.2680	14.00	2.6391	39	3.6636
8.36	2.1235	9.02	2.1994	9.68	2.2701	14.25	2.6567	40	3.6889
8.38	2.1258	9.04	2.2017	9.70	2.2721	14.50	2.6740	41	3.7136
8.40	2.1282	9.06	2.2039	9.72	2.2742	14.75	2.6913	42	3.7377
8.42	2.1306	9.08	2.2061	9.74	2.2762	15.00	2.7081	43	3.7612
8.44	2.1330	9.10	2.2083	9.76	2.2783	15.50	2.7408	44	3.7842
8.46	2.1353	9.12	2.2105	9.78	2.2803	16.00	2.7726	45	3.8067
8.48	2.1377	9.14	2.2127	9.80	2.2824	16.50	2.8034	46	3.8286
8.50	2.1401	9.16	2.2148	9.82	2.2844	17.00	2.8332	47	3.8501
8.52	2.1424	9.18	2.2170	9.84	2.2865	17.50	2.8621	48	3.8712
8.54	2.1448	9.20	2.2192	9.86	2.2885	18.00	2.8904	49	3.8918
8.56	2.1471	9.22	2.2214	9.88	2.2905	18.50	2.9173	50	3.9120

APPENDIX III.

Certain illustrations presenting historical interest and incorporating features of design discussed in the body of the text will be of value to students in this field.

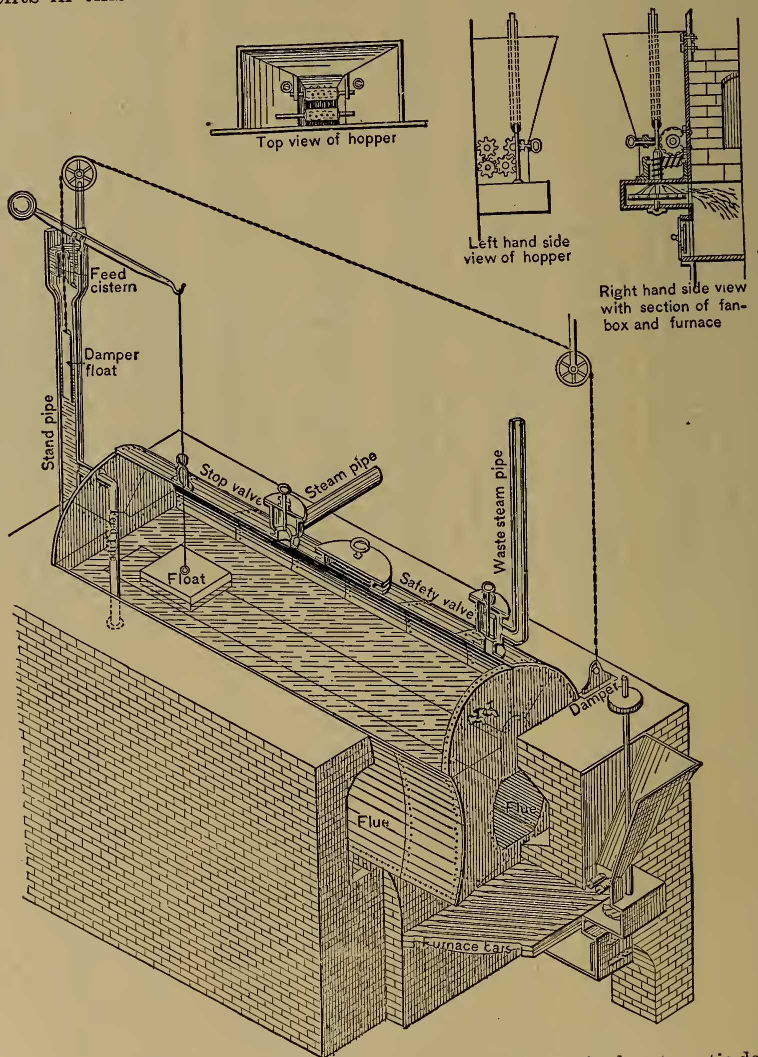


FIG. 259A.—Watt wagon boiler, showing automatic float water feed, automatic damper regulator by pressure and hopper-fed automatic mechanical stoker.

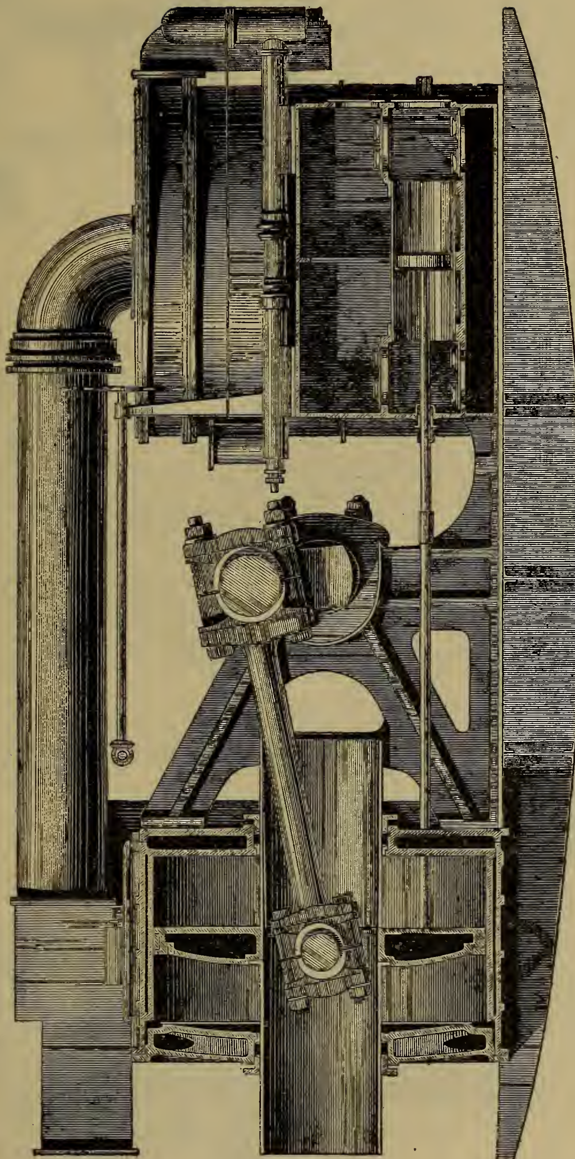


FIG. 355A.—Double-trunk engine of H.M.S. Bellerophon.

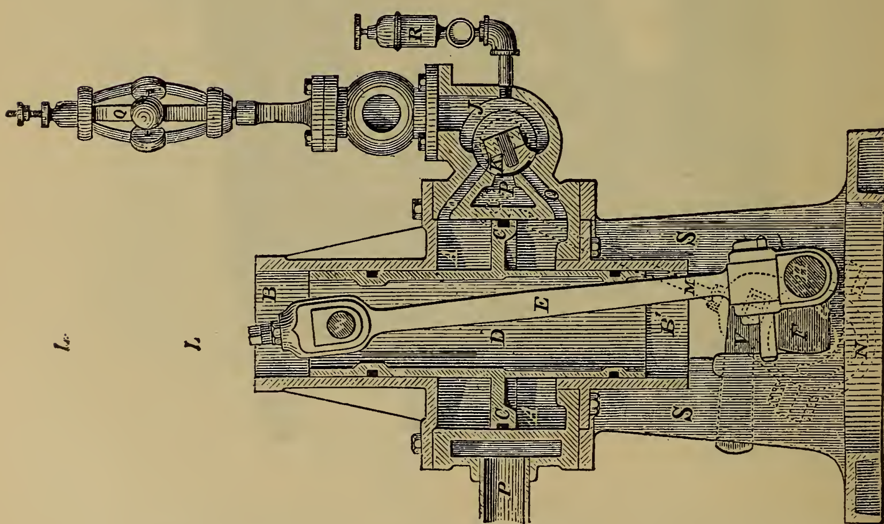


FIG. 357A.—Prolonged trunk type.

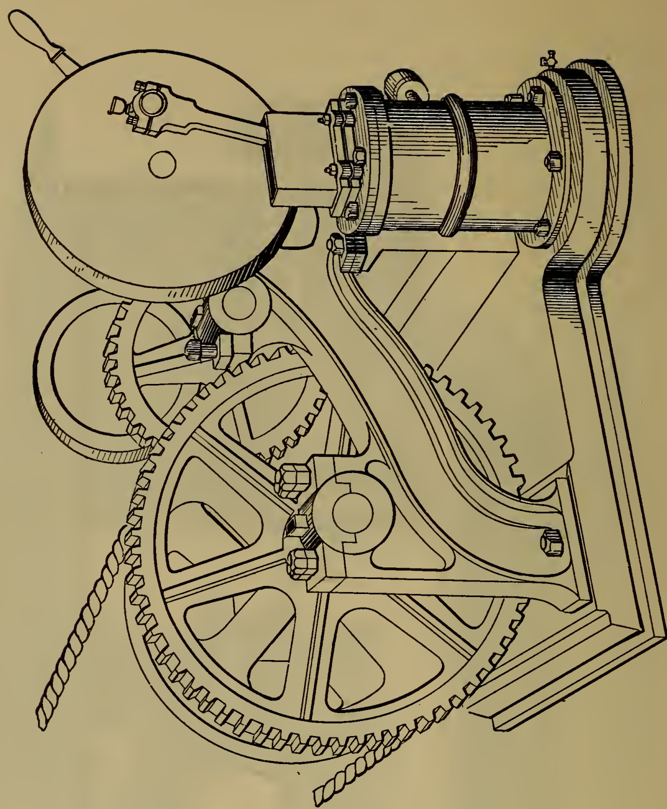


FIG. 358A.—Oval-trunk type. E. C. Bacon.

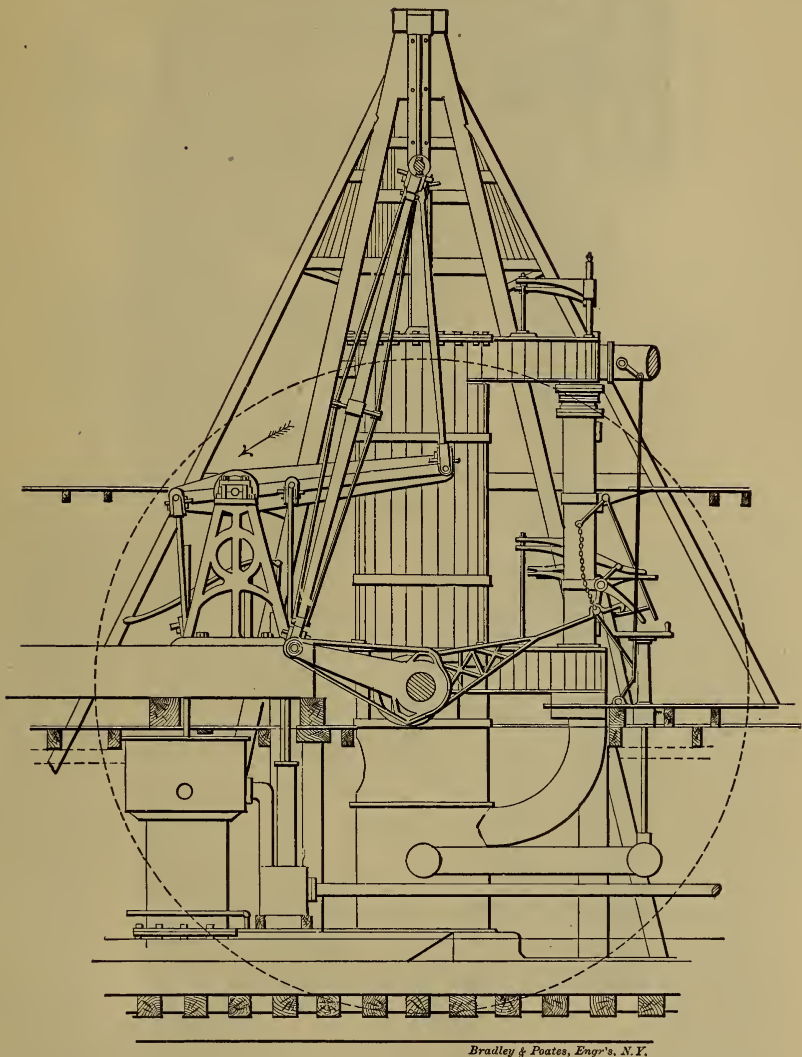


FIG. 359A.—Back-acting Hudson River paddle wheel tow-boat engine. S.S. Belle.

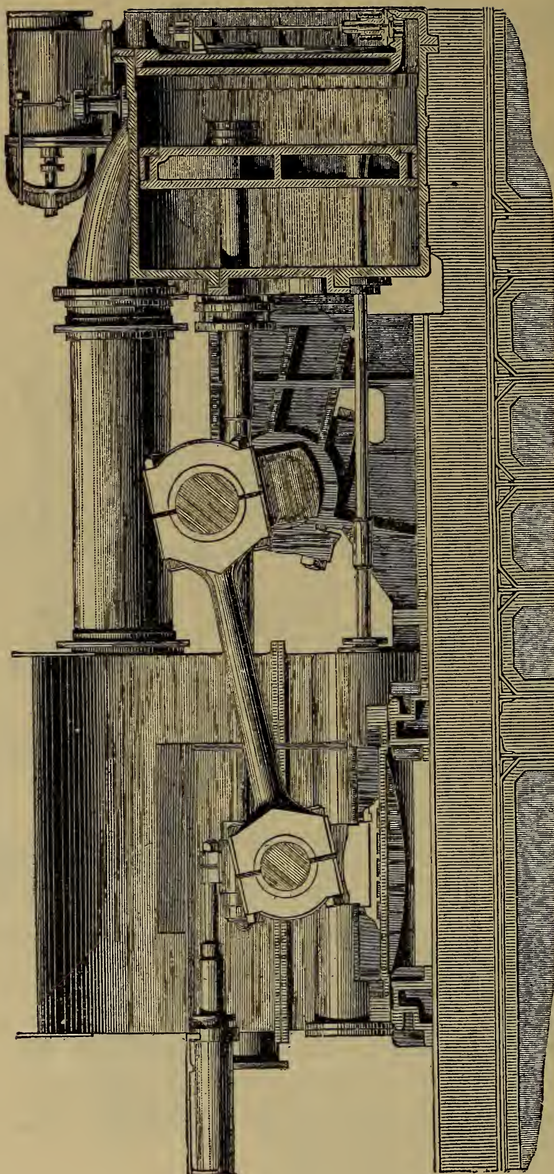


Fig. 360A.—Back-acting engine of H.M.S. Agincourt.

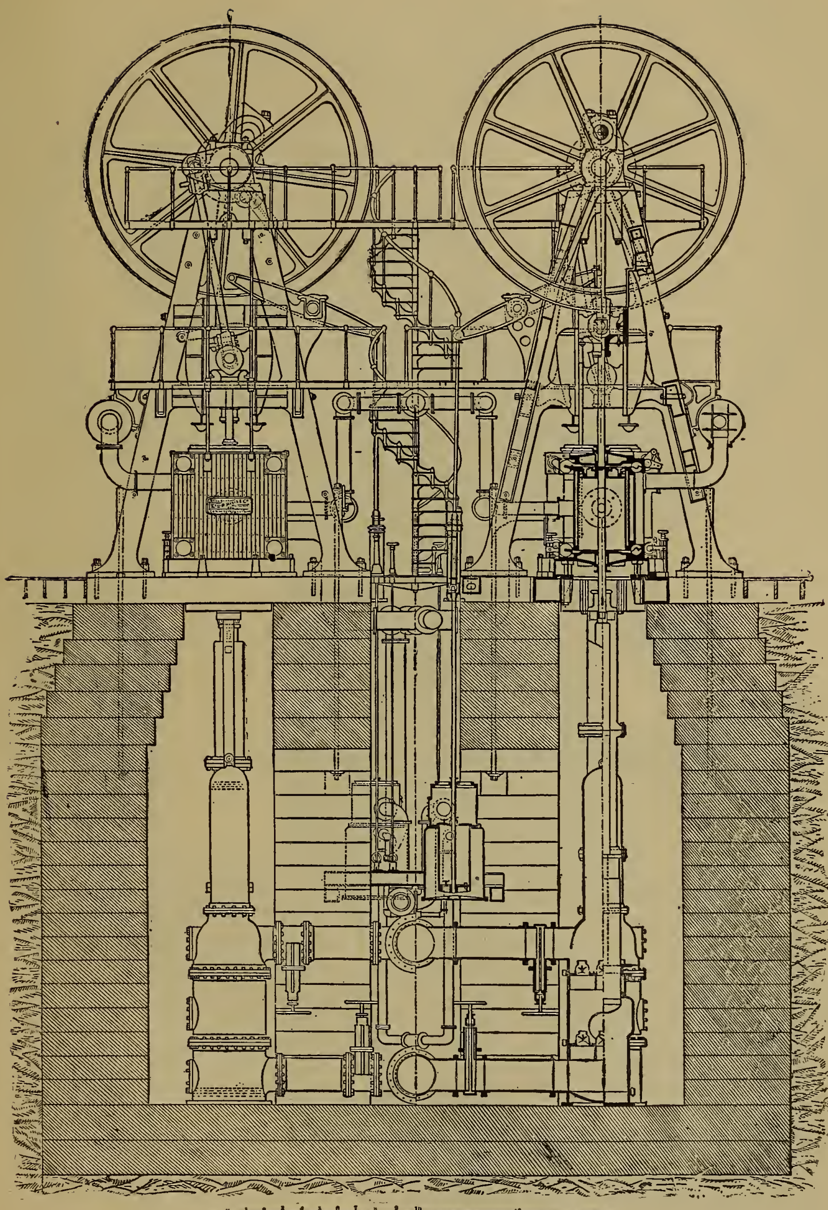


FIG. 383A.—Direct vertical pumping engine by Holly Mfg. Co.

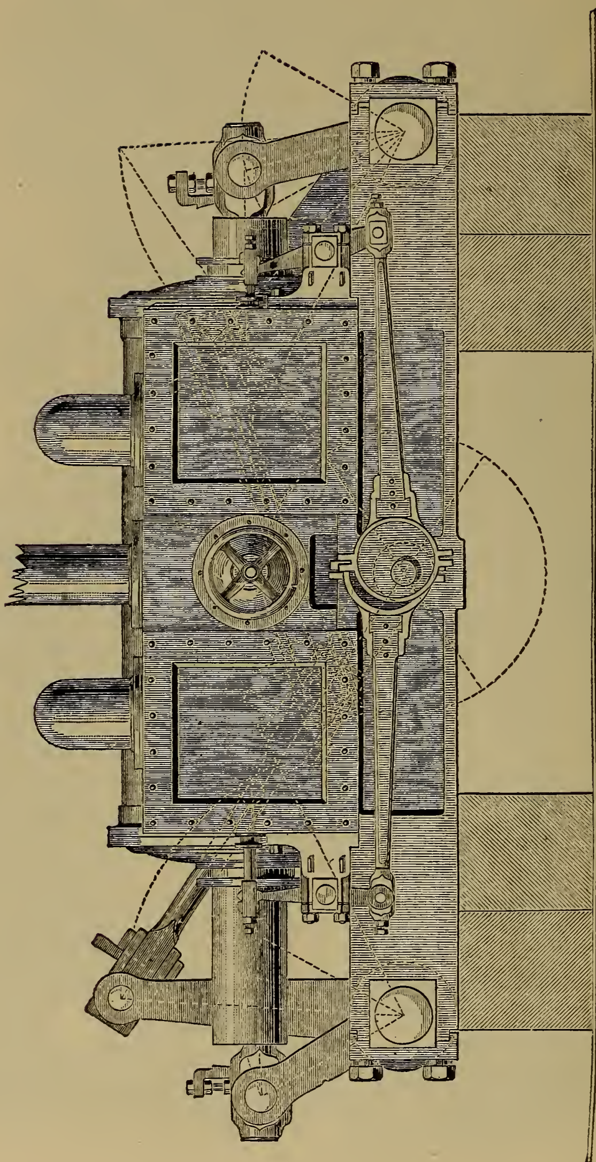


Fig. 396A.—Half-beam engine of U. S. Monitor Monadnock.

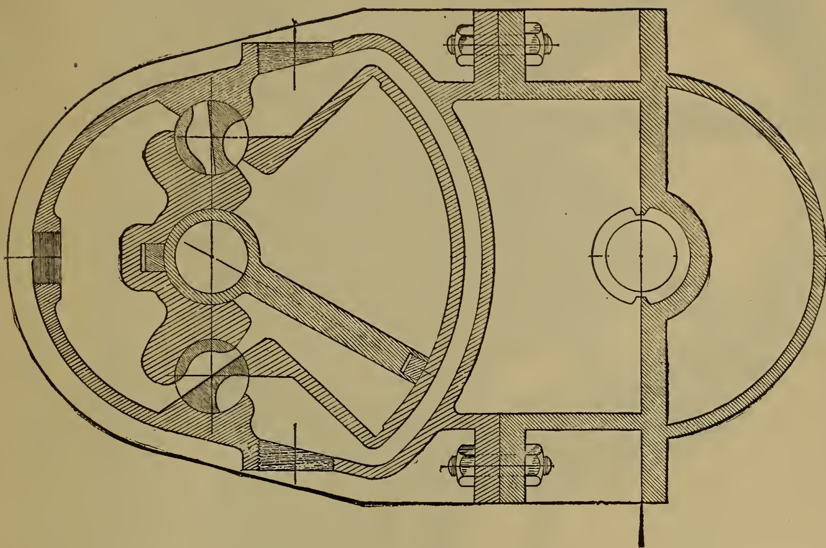


Fig. 457A.—John Ericsson's vibrating piston engine.

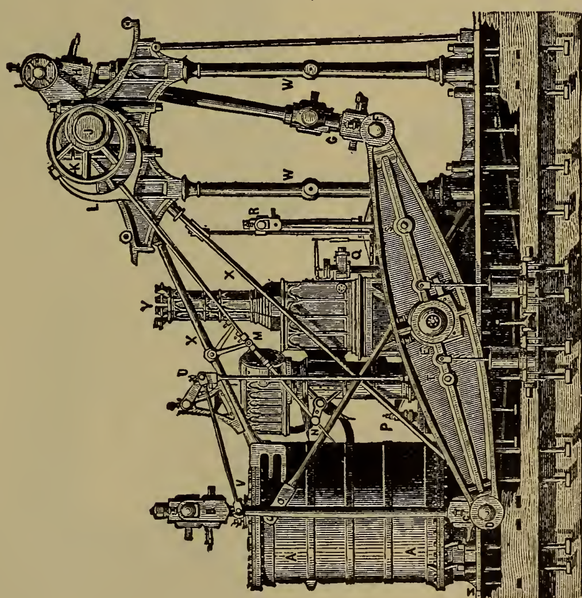


Fig. 397A.—Geo. W. Copeland's side-lever engine of 1849.

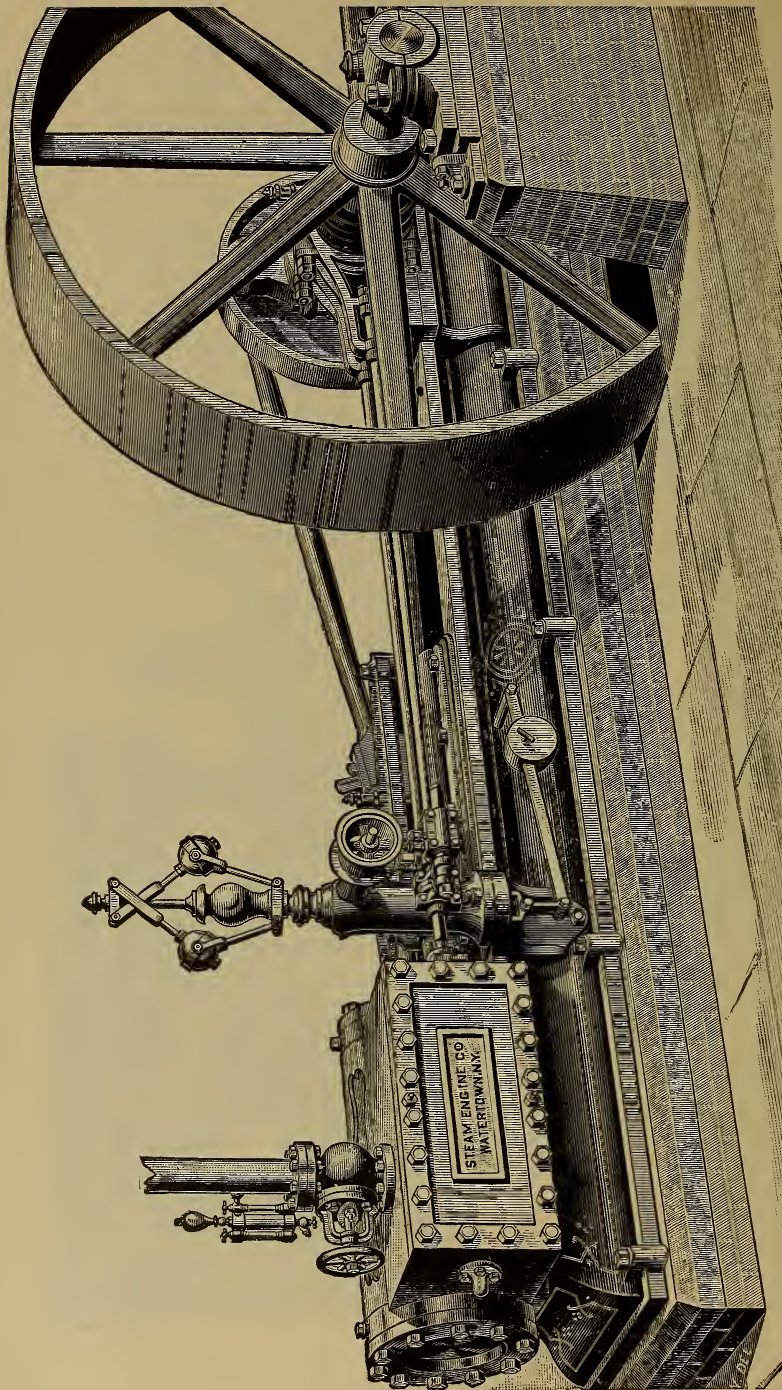


Fig. 507A.—Tank-type engine bed-plate, crossed-arm weighted governor.

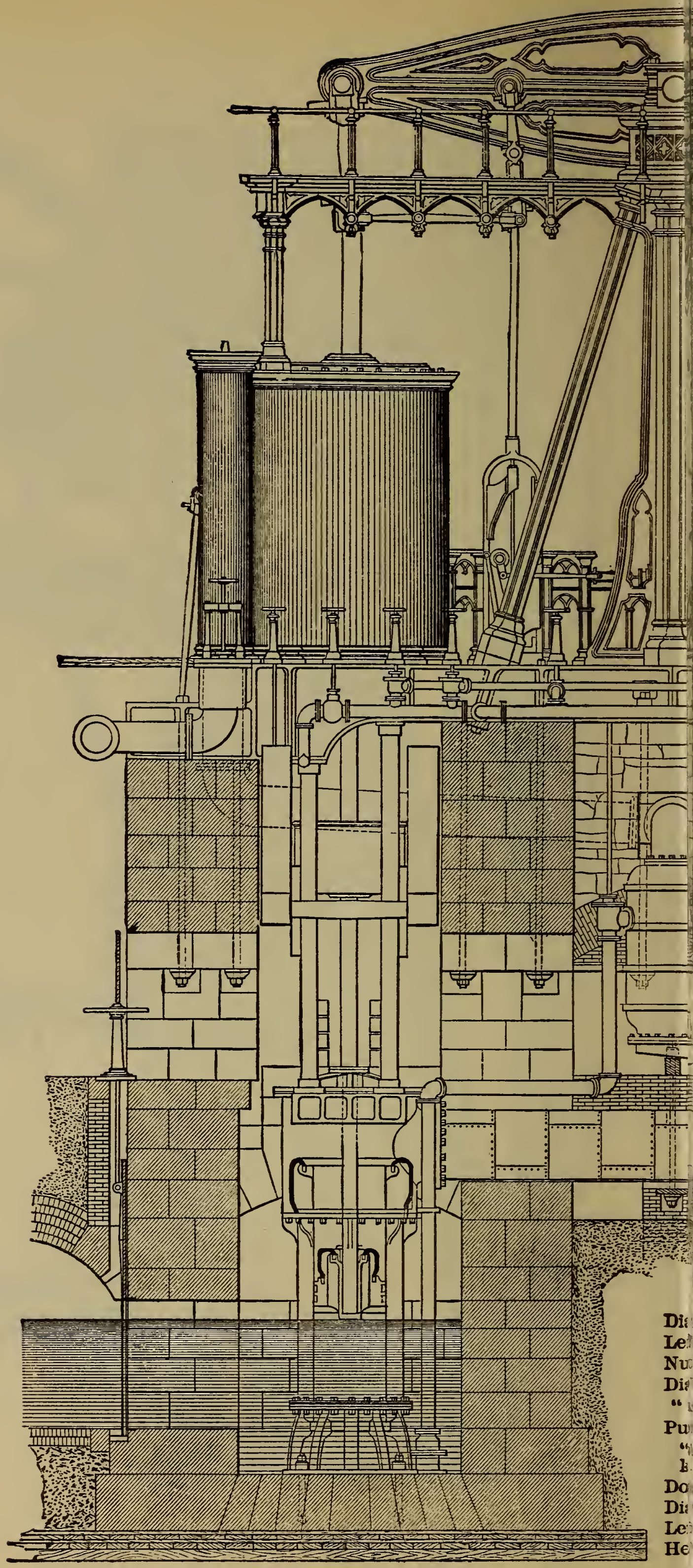
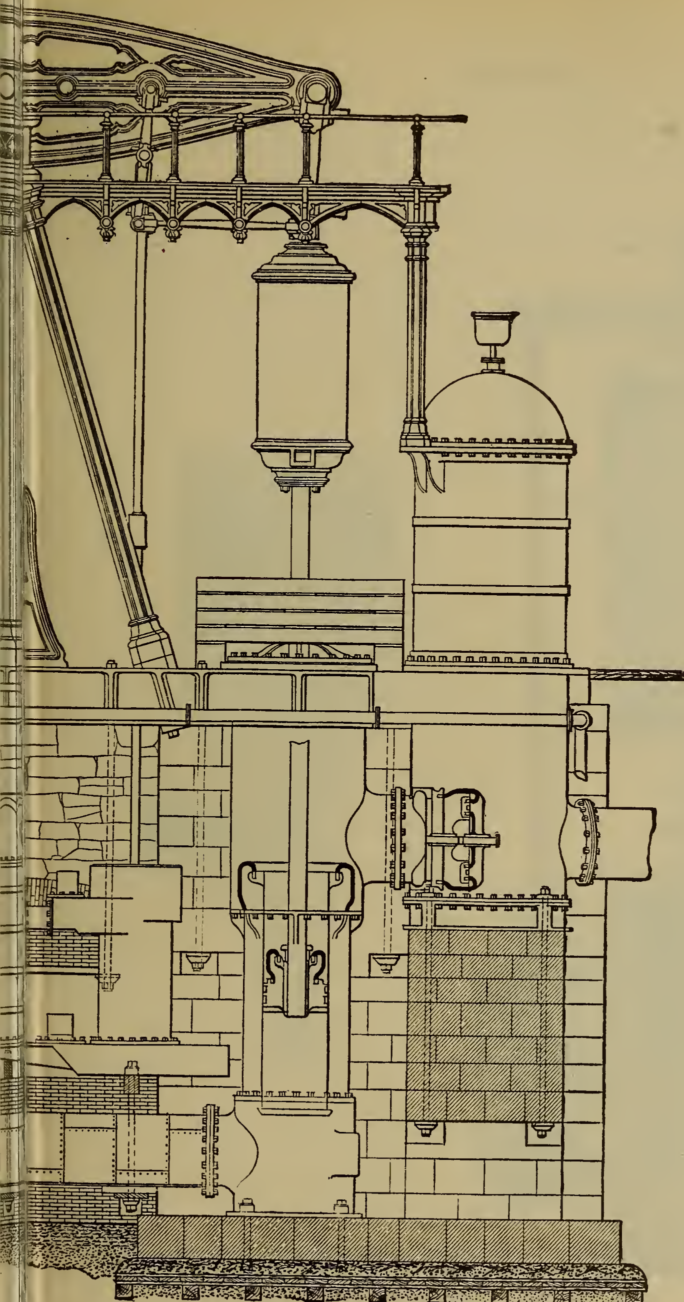


FIG. 402.—Brooklyn Water W



Di er of Cylinder............90"
Le of Stroke of Piston.....10'
Nu r of Pumps.............. 2
Di Working Barrel..........36"
" Auxilliary Barrel........54"
Pu Stroke....................10
" Valves of the double-beat
....
Do -acting Air Pump.......
Di er of Air Pump........... 3'
Le of Stroke................. 5'
He of centre of Beam above
level of Floor.................26'3"
Length of Beam between end centres.......................30'
Depth of Beam in middle.......7'2"
Average thickness of Web.....0'6"
Diameter of Air Chamber......6' 6¼"
Height of Air Chamber.........25'4"
" " " " above Floor..........................13'10"
Total weight of Engine, Boilers and Appurtenances.......440 Tons

W , Cornish Pumping Engine.

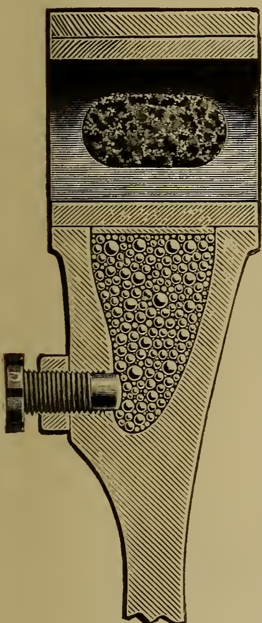


FIG. 536A.—C. W. Hunt's ball-compression stub end.

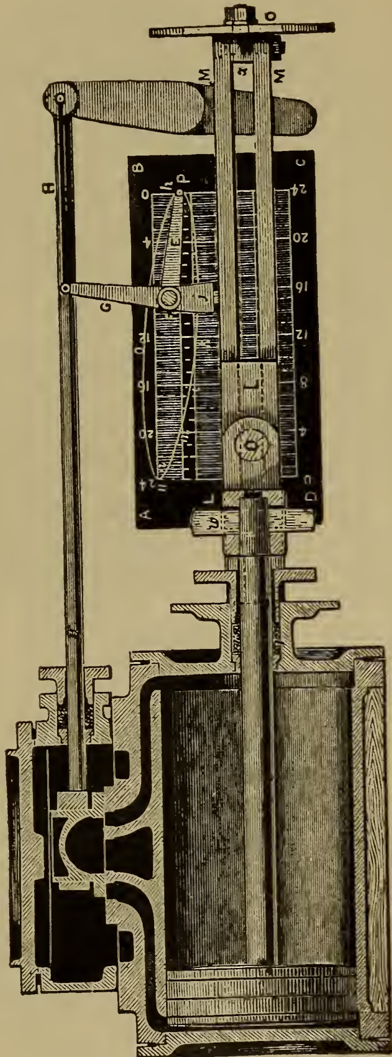


FIG. 577A.—M. N. Forney's mechanism for drawing motion-curves of slide-valve gears.

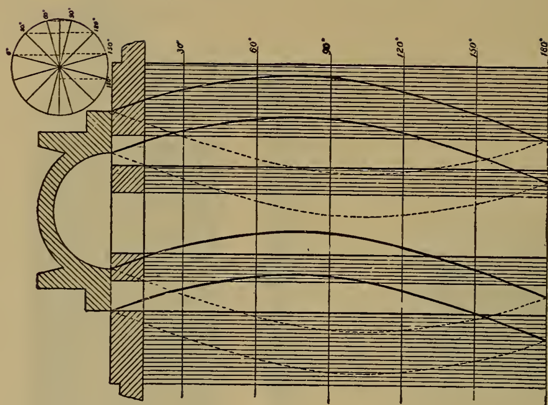


FIG 579A.—Valve motion curve with lap, no lead.

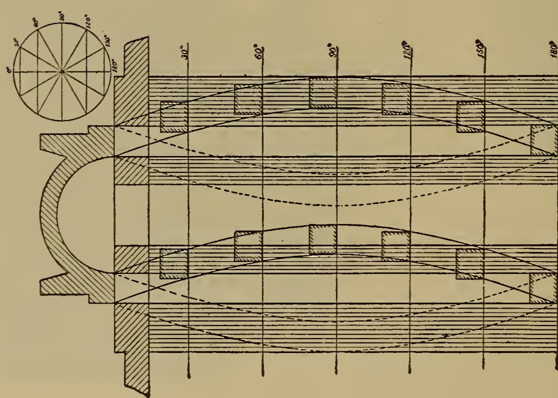


FIG. 578A.—Valve motion curve of plain slide valve, no lap nor lead.

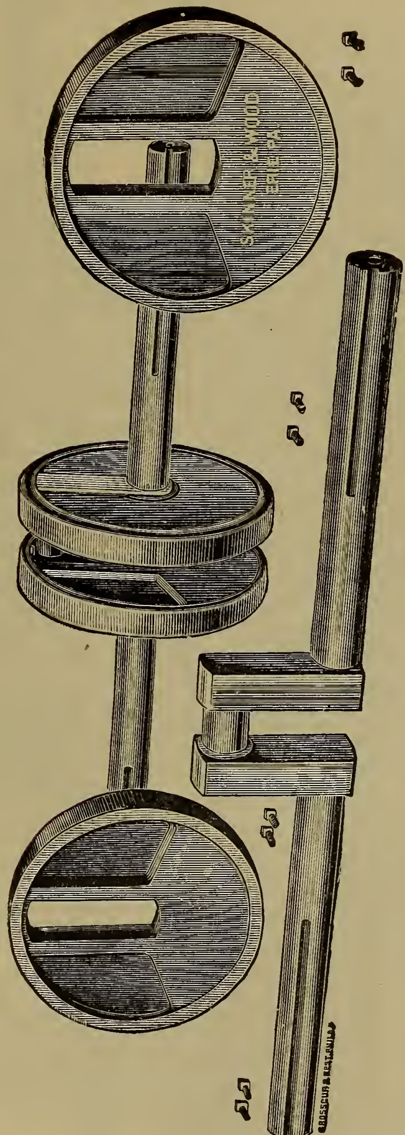


FIG. 544A.—Combination steel and cast-iron counterweighted crank.

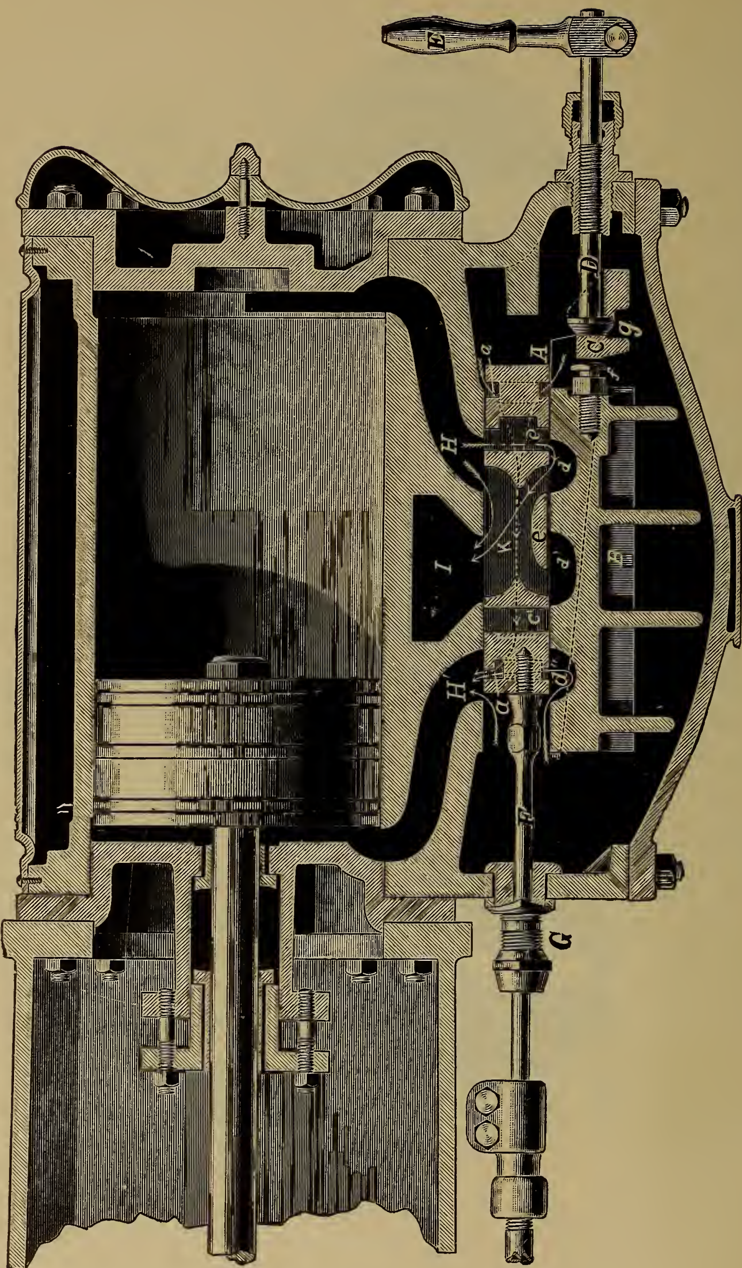


Fig. 598A.—Pressure plate adjustment of former Woodbury engine.

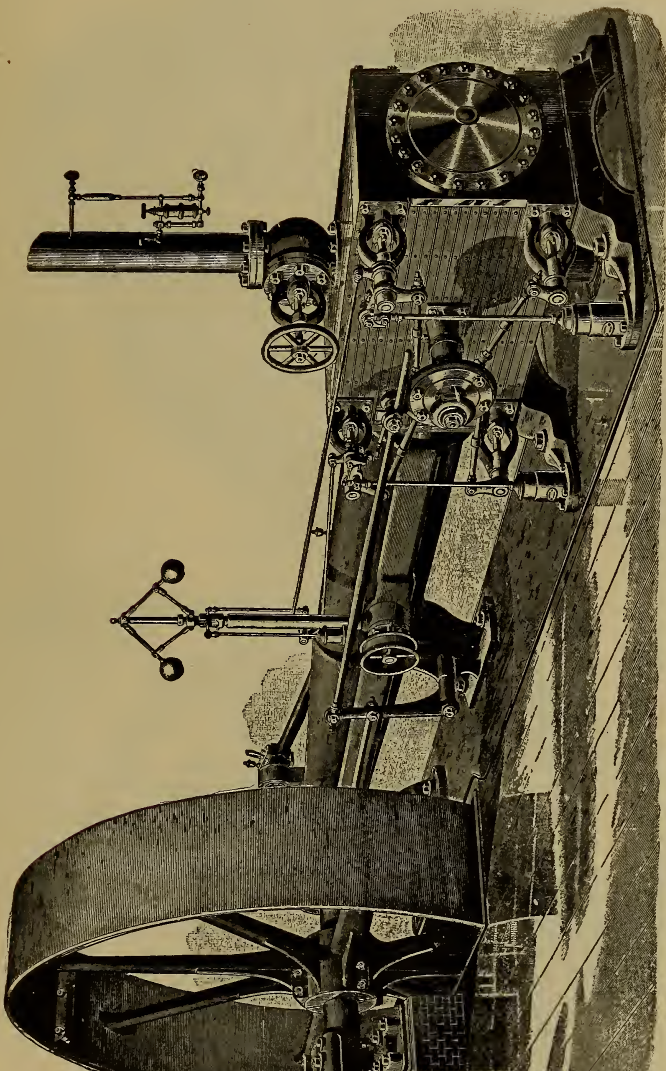


FIG. 635A.—Old Fishkill landing Corliss type.

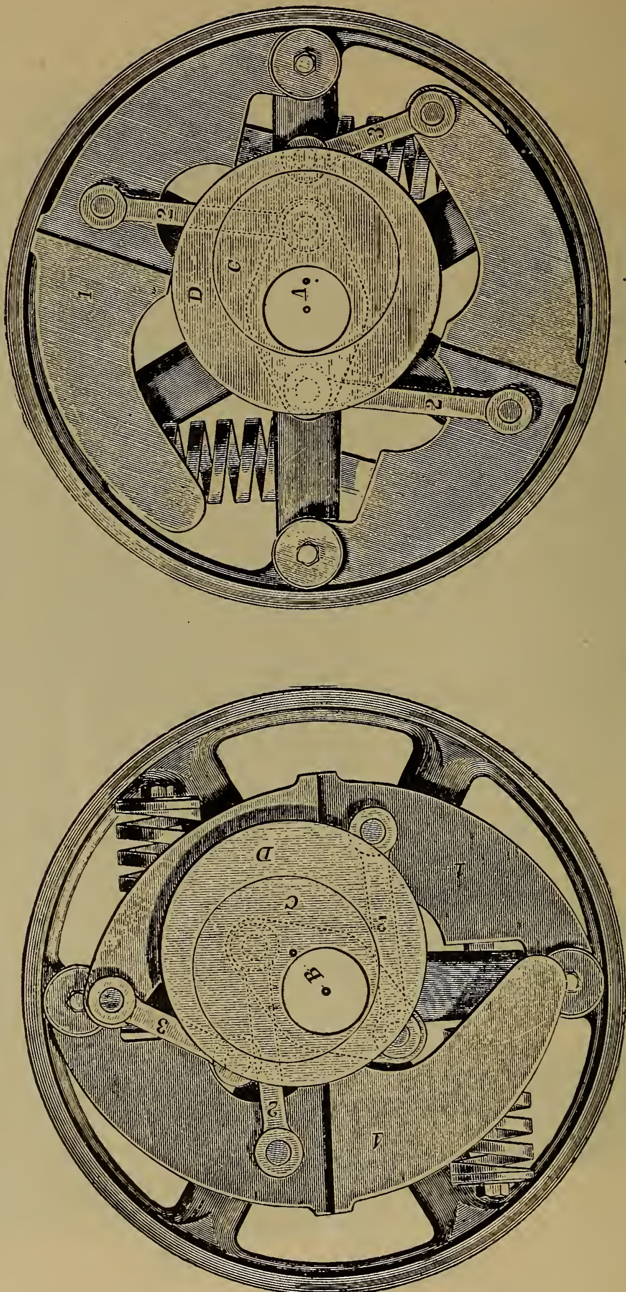


FIG. 653A.—Shaft-governor. Fly-weights change angular advance of eccentric.

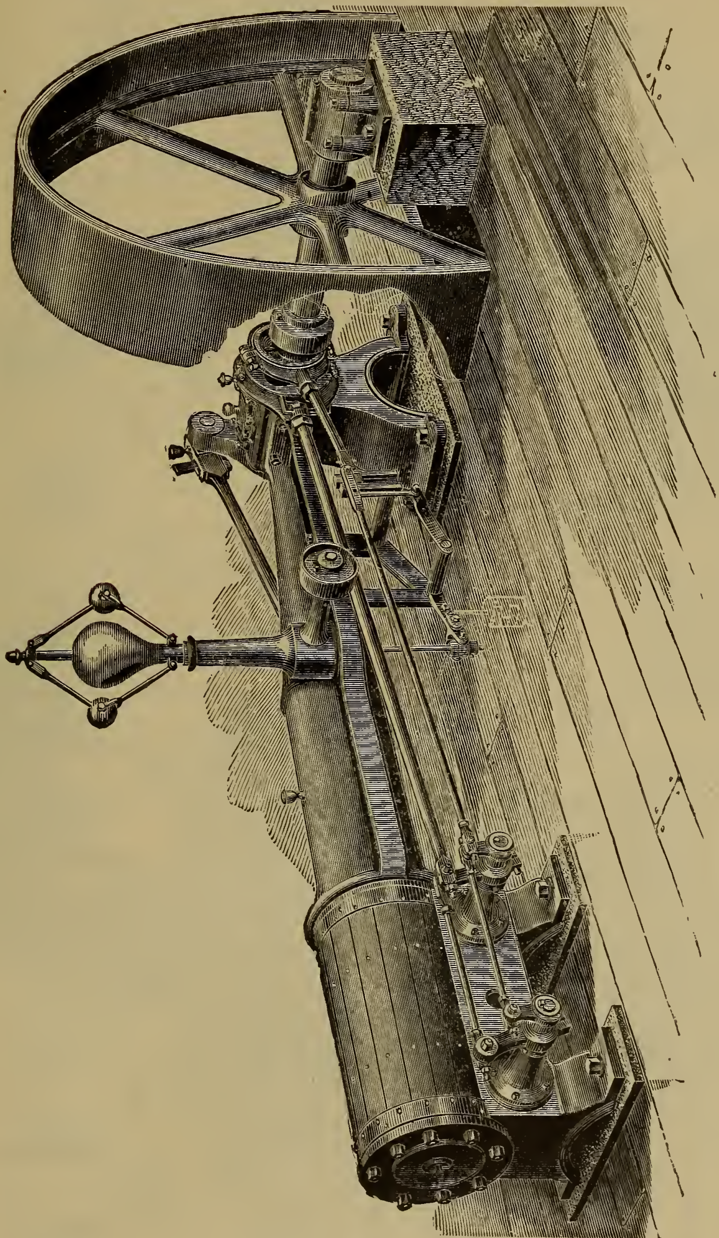


FIG. 654A.—Old Twiss engine with Porter loaded governors.

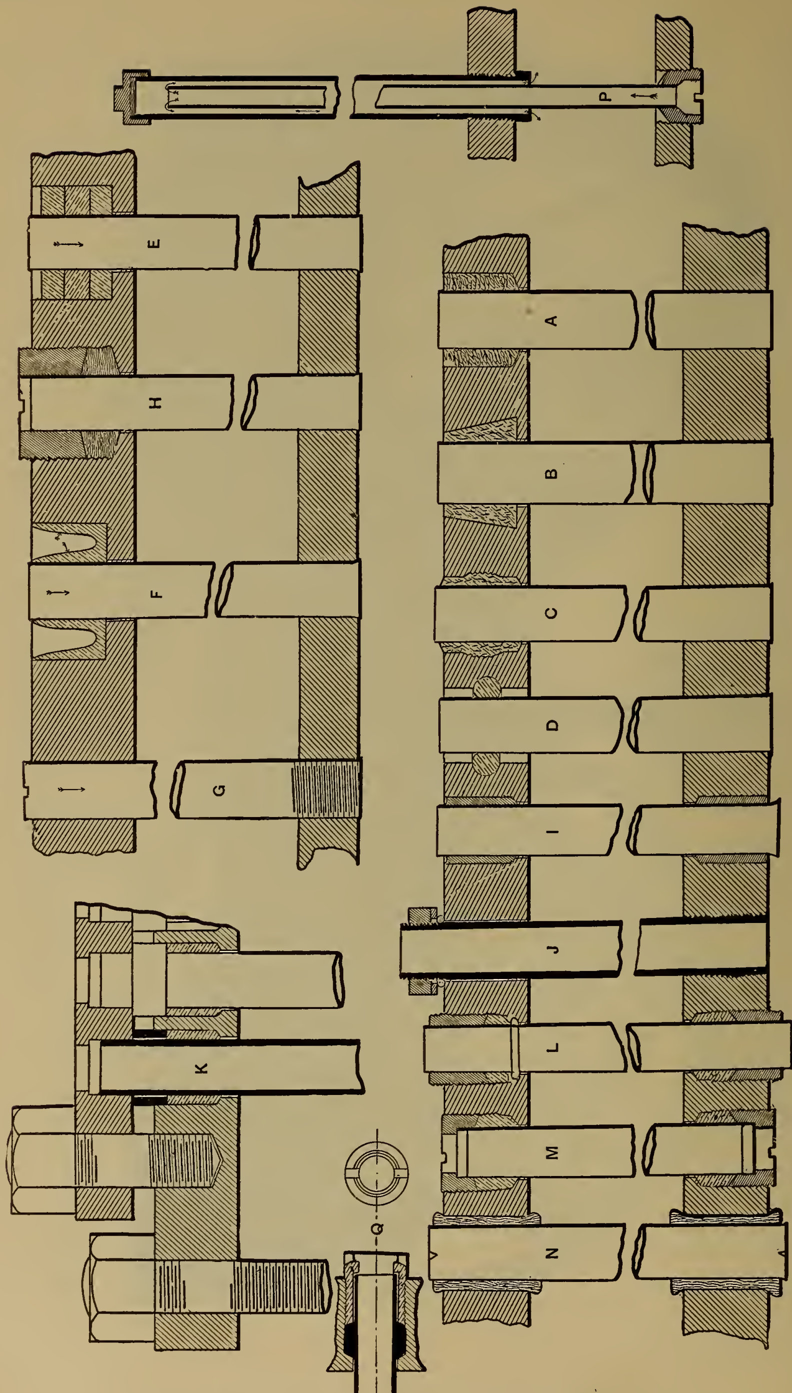


Fig. 666A.—Surface condenser packings and joints.

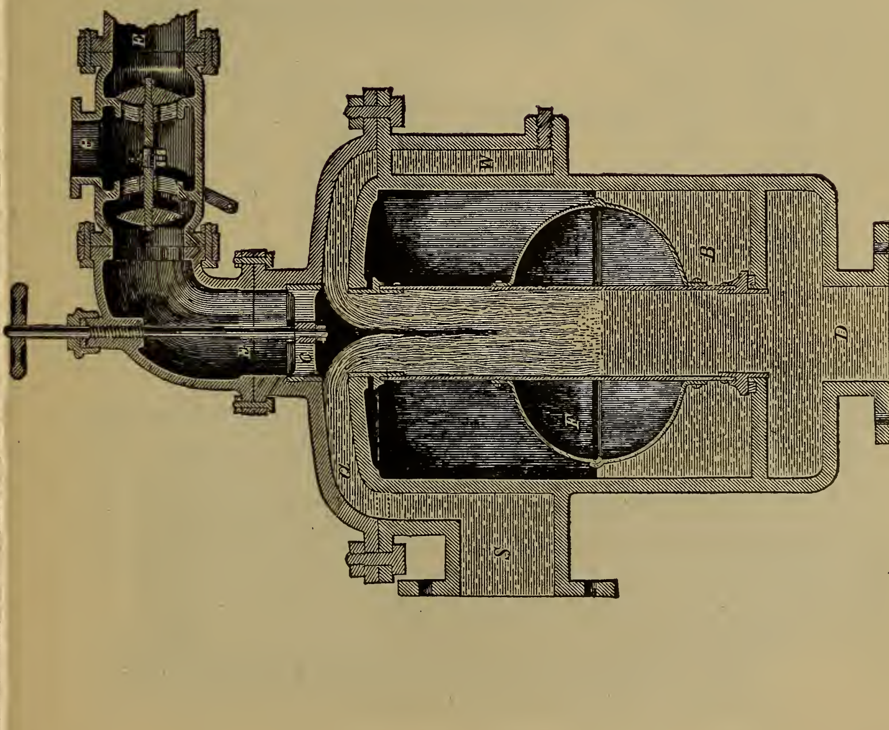


FIG. 675A.—Craig pump condenser with float control of injection.

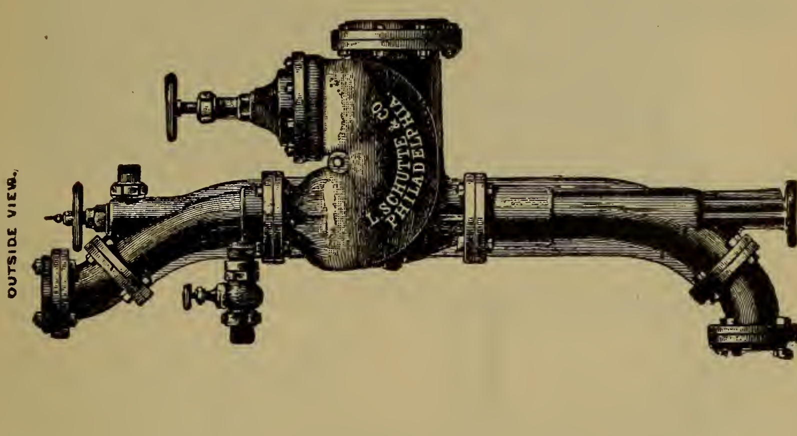


FIG. 678A.—Jet condenser, exhaust-steam type.

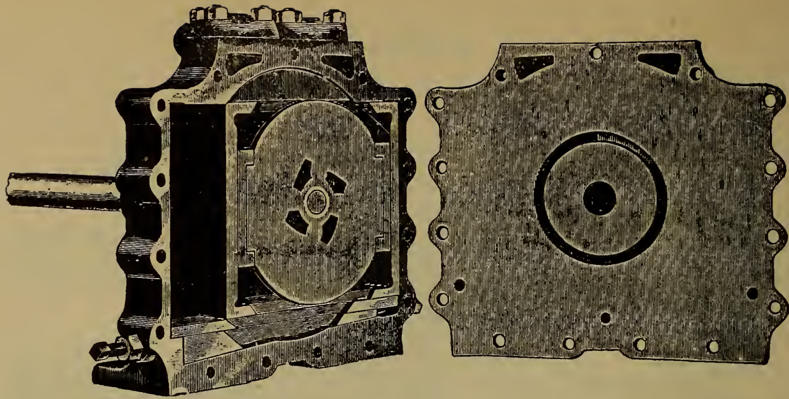


FIG. 695A.—Dake square piston engine.

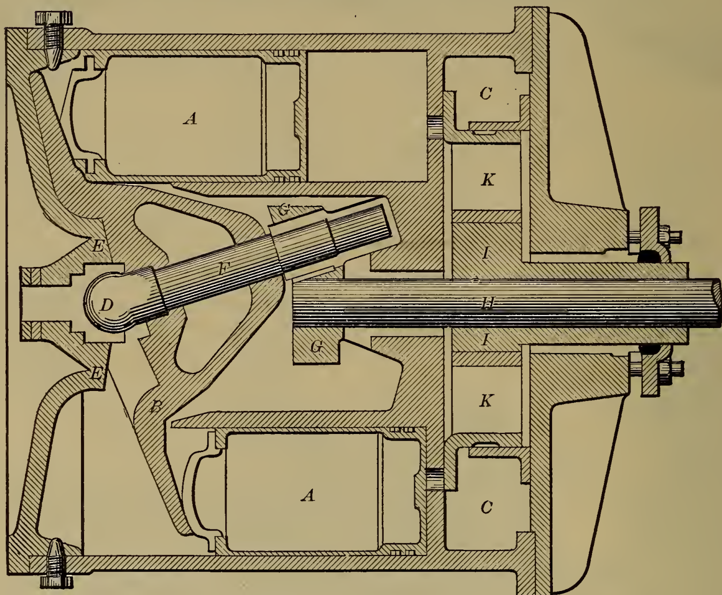


FIG. 669A.—Colt disk engine.

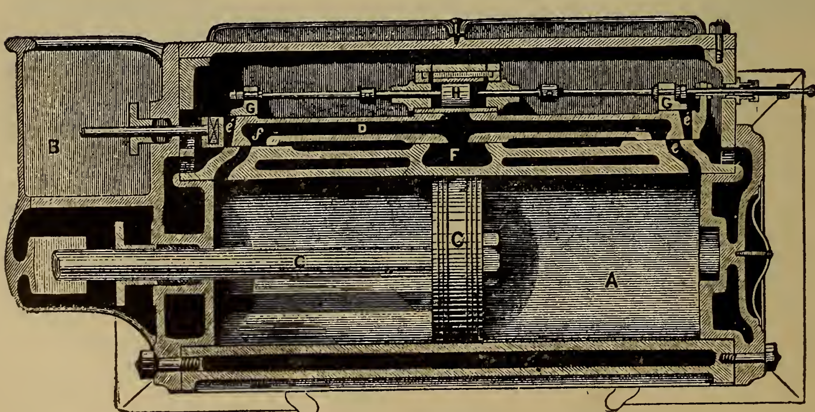


 FIG. 697A.—Babcock and Wilcox cut-off valve thrown by steam

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