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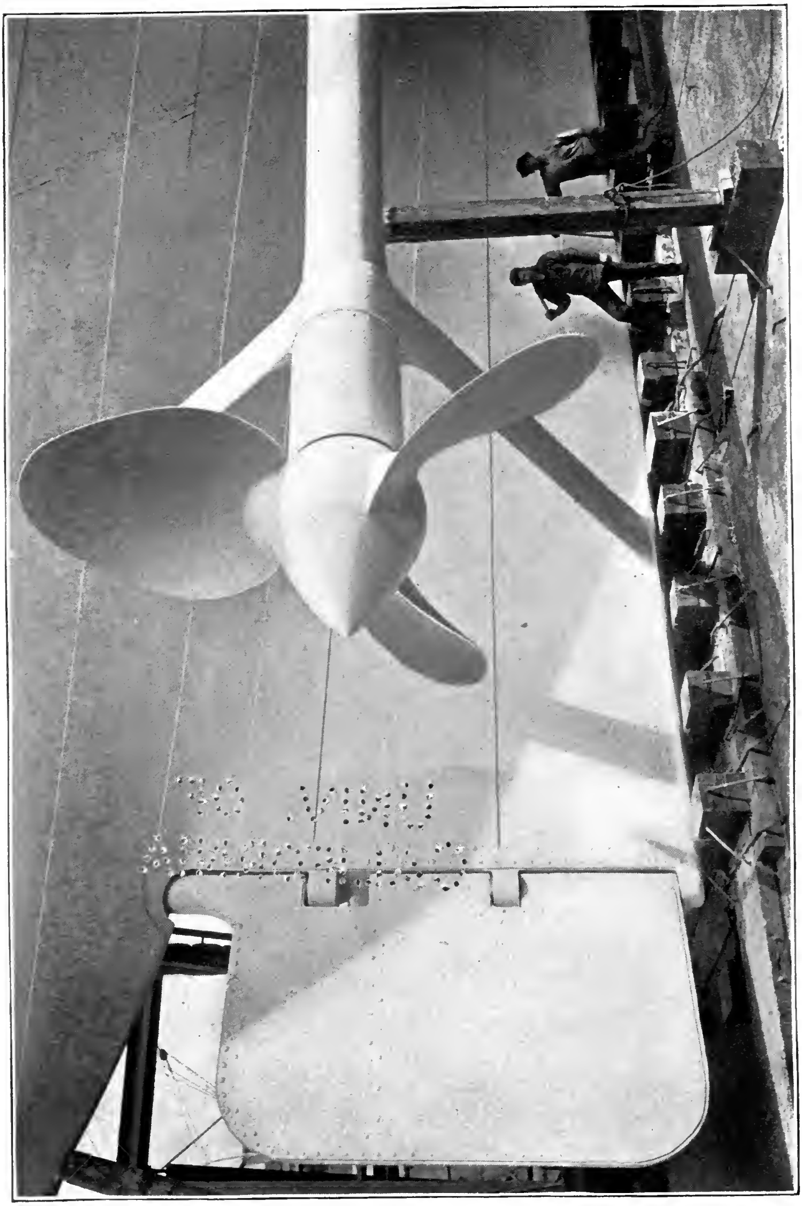

Practical Marine Engineering

FOR

MARINE ENGINEERS AND STUDENTS

WITH

Aids for Applicants for Marine Engineers' Licenses

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Preface.

THE purpose of the author in the preparation of this work has been to provide help for the operative or practical marine engineer, either for the man who has already entered the profession, but who may wish to perfect himself more fully in many branches of the subject, or for the applicant for the lowest round of the ladder, or for the young man whose attention is first turning to this field, and who may wish some simple and fairly complete presentation of the subject from the practical standpoint.

The treatment of the subject throughout has thus been with a view to simplicity, but without undue sacrifice of generality or exactness of statement. It has been the desire of the author to bring the subject, so far as treated in the present work, within the grasp of those who have not had the advantages of higher mathematical and engineering education, but who may wish, nevertheless, to fit themselves for positions of honor and responsibility in the field of operative marine engineering.

With this end in view only such parts of the general field of engineering have been included as are of special interest to the practical marine engineer. On these topics, however, the attempt has been made to give the largest amount of useful information in the simplest and most compact form. In the marine field itself likewise, selection has been necessary, and many interesting parts of the subject have been omitted or briefly referred to in order to give more room for the practical side of the subject. Thus the book does not treat of the designing of marine machinery except in an incidental way. For the operative engineer the topics of greater importance are construction, operation, management and care. The simpler parts of the subject of design are, however, represented by the U. S.

rules regarding the design and construction of marine boilers, and by many hints regarding proportions and relations scattered throughout the work.

In the chapters dealing descriptively with engines, boilers and auxiliaries, it has been impossible of course to describe exhaustively every form of design or appliance to be met with in marine practice. The purpose has been rather to describe typical or standard forms, and to give the general conditions which the various parts must fulfil. The illustrations have been specially chosen with a view to supplement the text in these various particulars, and it is hoped that they will form not the least instructive and acceptable feature of the work.

The subject of operation, management and repair has been given special attention, and it is hoped that this part of the work will be of value, especially to the young engineer lacking in practical experience.

In Chapter VIII is gathered a collection of miscellaneous problems and discussions, many of which, it is hoped, will be of value to the professional engineer in connection with the various questions likely to arise in his experience. The chapters on valve gears and on indicator cards while necessarily brief are intended to present the fundamental features of the subject in such manner as to aid the novice and instruct and stimulate the professional engineer to a better understanding of these important branches of the subject. The chapter on propulsion and powering is necessarily brief, but the fundamental principles are given, with a few simple rules and the discussion of most of the problems commonly arising in practical engineering work.

The chapters on refrigeration and on electricity on shipboard are added in order to give the marine engineer some notion of the fundamental principles controlling the operation of refrigerating and electric machinery, these two important auxiliaries of modern marine engineering practice. They are of necessity quite incomplete, especially Chapter XII, but it is hoped that nevertheless they may be of aid to the marine engineer in understanding the mode of operation of such machinery, and in giving to it the proper care.

In Part II is given an elementary discussion of computations for engineers, or rather of the mathematics upon which such computations depend. A general knowledge of the subject is pre-supposed, but the more essential features of the elementary mathematics usually required are given and illustrated with many problems. It is hoped that this feature of the work may be of aid to those who wish to drill themselves in such computations as the marine engineer is commonly called upon to make.

Attention may also be called to the appendix containing a list of questions, each with page reference to the part of the book where the answer may be found. The answer, of course, will not usually be found in the direct form suggested by the question, but a discussion of the subject will be found giving the information needed for the answer, which may be put into form by the reader for himself. It is believed that such an exercise will be of far greater value than the perusal of a series of questions and answers in the usual catechism form.

Throughout the work numerous problems have been scattered, accompanied usually by illustrative examples, showing the method of working. Numerous cross references have also been given, to aid in the more complete explanation of any given topic, and in finding these use should be made of the table of contents, giving the page location of each section and bracket subdivision.

A collection of miscellaneous problems is also added at the end of the book, as well as a set of steam tables for use in the solution of the various problems requiring a knowledge of its various mechanical and physical properties.

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Practical Marine Engineering

CHAPTER I.

Principal Materials of Engineering Construction.

Sec. 1. ALUMINUM.

The commercially pure metal, i. e., with less than 1 per cent impurity, is white in color, soft, ductile and malleable. It melts at about 1,160° F., has a tensile strength of about 15,000 lb. per square inch of section, but lacks in stiffness and *resilience*, or the power to withstand shocks.

Aluminum does not oxidize readily under the influence of ordinary air, but when in contact with sea water, or in air charged with sea water, the corrosion is often serious in extent. Aluminum cannot be welded except electrically, is not suitable for forging or rolling when hot, and cannot be tempered or hardened. It is, however, suitable for casting, and when cold can be rolled into sheets and drawn into wire, and in thin sheets or small pieces may be spun or flanged, or worked under the hammer in various ways.

Aluminum unalloyed is of comparatively small value to the engineer, but it enters into several valuable alloys, as described in Section 9, and its use in this way has increased to a considerable extent within the past few years.

Sec. 2. ANTIMONY.

The pure metal is whitish in color, quite brittle and crystalline or laminated in structure, and has a melting point of about 840° F. It is useless in the pure state for ordinary engineering purposes, but is a valuable ingredient of various alloys used for bearing metals, etc., as described in Section 9.

Sec. 3. BISMUTH.

The pure metal is light red in color, very brittle and highly crystalline in structure, with a melting point of about 510° F. It is useless in the pure state for engineering purposes, but forms a part of various alloys used for bearing metals, etc., as described in Section 9.

Sec. 4. COPPER.

In the pure state the metal is red in color, soft, ductile and malleable, with a melting point of about 2,000° F., and a tensile strength of from 20,000 to 30,000 lb. per square inch of section. Copper is not readily welded except electrically, but, on the other hand, is readily joined by the operation of brazing. Attempts have been made to temper or harden it, but the operation has not been made a practical success. It is readily forged and cast, and when cold may be rolled into sheets or drawn into wire, and in sheets or small pieces may be spun or flanged or worked under the hammer in various ways.

The tensile strength of copper rapidly falls off as the temperature rises above about 400° F., so that at from 800° to 900° the strength is only about one-half what it is at ordinary temperatures. This peculiarity of copper should be borne in mind when it is used in places where the temperature is liable to rise to these figures. Again, if copper is raised nearly to its melting point in contact with the air it readily unites with oxygen and loses its strength in large degree, becoming, when cool, crumbly and brittle. Copper in this condition is said to have been *burned*. The possibility of thus injuring the tenacity of copper is of the highest importance in connection with the use of brazed joints in steam pipes.

In the operation of brazing a joint, the surfaces to be joined are cleaned, bound together with wire or otherwise, then supplied with brazing solder in small bits, mixed with borax as a flux, and placed in a clear fire until the solder melts and forms the joint. The brazing solder, or *hard solder*, as it is often called, is usually a brass or alloy of copper and zinc. The melting point of all such alloys is below that of copper, and when copper is joined to brass, or two pieces of brass are joined together, the solder used must have a melting point lower than either of these metals. In the operation of brazing a copper joint, therefore, the greatest care must be taken in the selection of a solder and in attention to the fire, so that there may be no danger of *burning* the copper, and thus endangering the quality of the metal in the joint.

Copper unalloyed is used chiefly for pipes and fittings, especially for junctions, elbows, bends, etc. For large sizes the material is made in sheets, bent and formed to the desired shape and brazed at the seams. Small sizes are either made by the same general process or from solid drawn pipe, which may be bent as desired after drawing. Copper is also largely used as the chief ingredient of the various brasses and bronzes, as described in Section 9.

Sec. 5. IRON AND STEEL.

Classification.

It will be convenient to give here a general classification of iron and steel products based on the methods of manufacture. The following is the classification used by Prof. J. B. Johnson in his text book on the *Materials of Construction.*

MALLEABLE.

Wrought Iron.—Rolled or forged from a puddle ball; it contains slag and other impurities and cannot be hardened by sudden cooling.

Steel.—Rolled or forged from a cast ingot and free from slag and similar matter.

Soft Steel.—Will weld (with care), and cannot be hardened by sudden cooling. It is sometimes called *ingot iron,* and has the same uses as wrought iron.

Medium Steel.—Welds imperfectly except by electricity. Will not harden by sudden cooling.

Hard Steel.—Will not weld. Hardens by sudden cooling. Tool steel, etc.

SEMI-MALLEABLE.

Steel Castings.—Malleable metal cast into forms.

Malleable Cast Iron.—Non-malleable metal cast into forms and then brought to a semi-malleable condition.

NON-MALLEABLE.

Cast Iron; Hard Cast Steel.—Non-malleable metal cast into forms.

In describing these products at length we shall find it convenient to begin with cast iron.

[1] Cast Iron.

This material consists of a mixture and combination of iron and carbon, with other substances in varying proportions.

(1) *Influence of Carbon.* In the molten condition the carbon is dissolved by the iron and held in solution just as ordinary salt is dissolved by water. The mixture or combination of the two elements is thus entirely uniform. The proportion of carbon which pure melted iron can thus dissolve and hold in solution is about 3½ per cent. If *chromium* or *manganese* is present also, the capacity for carbon is much increased, while with silicon, on the other hand, the capacity for carbon is decreased. In the various grades of cast iron the proportion of carbon is usually found between 2 per cent and 4.5 per cent.

Now, when such a molten mixture cools and becomes solid, there is a tendency for a part of the carbon to be separated out and no longer remain in intimate combination with the iron. The carbon thus separated or precipitated out from the iron takes that form known as *graphite*, and collects together in very small flakes or scales. The carbon which remains in intimate combination with the iron is said to be *combined*, while that which is separated out is usually called *graphitic*.

The qualities of cast iron depend chiefly on the proportion of total carbon and on the relative proportions of combined and graphitic carbon.

With a high proportion of graphitic carbon the iron is soft and tough, with low tensile strength, and breaks with a coarse grained dark or grayish colored fracture. In fact the substance in this condition may be considered as nearly pure iron with fine flakes of graphite entangled and distributed through it, thus giving to the iron a spongy structure. The iron thus forms a kind of continuous mesh about the graphite, which decreases the strength by reason of the decrease of cross-sectional area actually occupied by the iron itself. Such irons are termed *gray*.

As the relative proportion of graphitic carbon decreases and that of combined carbon increases, the iron takes on new properties, becoming harder and more brittle. Its tensile strength also increases to a certain extent, and the fracture becomes fine grained or smooth and whiter in color. When these characteristics are pronounced the iron is said to be *white*. When about half the carbon is combined and half separates out as graphite, the effect is to produce a distribution of dark spots or points scattered over a whitish field. Such irons are said to be *mottled.*

In a general way with a large proportion of total carbon there is likely to be formed a considerable amount of graphitic carbon, and hence such irons are usually gray and soft. With a large proportion of carbon also the iron melts more readily and its fluidity is more pronounced. As the proportion of total carbon decreases the cast iron approaches gradually the condition of steel, whose properties will be discussed in a later paragraph.

Of the special ingredients in cast iron the *combined carbon* is one of greatest importance. It is that chiefly which by uniting with the iron gives it new qualities, and the principal influence of other substances lies in the effect which they may have on the proportion of this ingredient. As between graphitic and combined carbon, the former does not affect the quality of the iron itself, but acts *physically* by affecting the structure of the casting; while the latter, by entering into combination with the iron, acts *chemically* and produces a new substance with different qualities. The following percentages of combined carbon are recommended for qualities of iron as indicated:

	PROPORTION OF COMBINED CARBON.
Soft cast iron	.10 to .15 of one per cent.
Greatest tensile strength.	about .45 of one per cent.
Greatest transverse strength	about .70 of one per cent.
Greatest crushing strength	one per cent or over.

The proportions of combined and graphitic carbon are influenced by the rate of cooling, and by the presence or absence of various other ingredients. Slow cooling allows time for the separation of the carbon and thus tends to form graphitic carbon and soft gray irons. Quick cooling, or *chilling* in the extreme case, prevents the formation of graphitic carbon and thus tends to form hard, white irons.

In addition to *carbon*, small quantities of *silicon, sulphur, phosphorus, manganese* and *chromium* may be found in cast iron.

(2) *Influence of Silicon.* The fundamental influences of silicon are two. (a) It tends to expel the carbon from the combined state and thus to decrease the relative proportion of combined carbon and increase that of graphitic carbon. (b) Of itself silicon tends to harden cast iron and to make it brittle.

These two influences are opposite in character, since an increase in graphitic carbon softens the iron. In usual cases the net result is a softening of the iron, an increase in fluidity, and a general change toward those qualities possessed by iron with a high proportion of graphitic carbon. This applies with a proportion of silicon from 2 per cent to 4 per cent. With more than this the influence on the carbon is but slight and the result on the iron is to decrease the strength and toughness, giving a hard but brittle and weak grade of iron.

A *chilled* cast iron is an iron which if cooled slowly would be gray and soft, but, as explained in (1), by sudden cooling, from contact with a metal mould or other means, becomes white and hard, especially at and near the surface. Certain grades of cast iron tend to chill when cast in sand moulds. This property is usually undesirable. In such cases the tendency can be prevented by the addition of silicon, which, by forcing the carbon into the graphitic state on cooling, prevents the formation of the hard, chilled surface. In all cases the actual effect of adding silicon will depend much on the character of the iron used as a base, and only a statement of the general tendencies can here be given.

To sum up, a white iron which would give hard, brittle and porous castings can be made solid, softer and tougher by the addition of silicon to the extent of perhaps 2 or 3 per cent. As the silicon is increased the iron will become softer and grayer and the tensile strength will decrease. At the same time the shrinkage will decrease, at least for a time, though it may increase again with large excess of silicon. The softening and toughening influence, however, will only continue so long as additional graphite is formed, and when most of the carbon is brought into this state the maximum effect has been produced, and any further addition of silicon will decrease both strength and toughness.

(3) *Influence of Sulphur.* Authorities are not in entire agreement as to the influence of sulphur on cast iron, some believing that it tends to increase the proportion of combined carbon, while others maintain that it tends to decrease both the combined carbon and silicon. It is generally agreed, however, that in proportions greater than about .15 to .20 of 1 per cent it increases the shrinkage and the tendency to chill, and decreases the strength. Sulphur does not, however, readily enter cast iron under ordinary conditions, and its influence is not especially feared. An increase in the proportion of sulphur in cast iron

is most likely to result from an absorption of sulphur in the coke during the operation of melting in the cupola.

(4) *Influence of Manganese.* This element by itself decreases fluidity, increases shrinkage, and makes the iron harder and more brittle. It combines with iron in all proportions. With manganese less than one-half, the combination is usually called *spiegeleisen.* With manganese more than one-half it is called *ferro-manganese.* One of the most important properties of manganese in combination with iron is that it increases the capacity of the iron for carbon. Pure iron will only take about 3½ per cent of carbon, while with the addition of manganese the proportion may rise to 6 per cent or 7 per cent. Manganese is also believed to decrease the capacity of iron for sulphur, and to this extent may be a desirable ingredient in proportions not exceeding 1 per cent to 1½ per cent.

(5) *Influence of Chromium.* This substance is rarely found in cast iron, but it has the property, when present in large proportion, of raising the capacity of the iron for carbon from about 3½ per cent up to about 12 per cent.

(6) *Shrinkage of Cast Iron.* At the moment of hardening, cast iron expands and takes a good impression of the mould. In the gradual cooling after setting, however, the metal contracts, so that on the whole there is a shrinkage of about ⅛ in. per foot in all directions, though this amount varies somewhat with the quality of the iron and with the form and dimensions of the pattern. In a general way hardness and shrinkage increase and decrease together.

(7) *Strength and Hardness of Cast Iron.* The hardness of cast iron is chiefly dependent on the amount of combined carbon, as noted above in (1).

The strength is also chiefly dependent on the same ingredient. As shown in (1), the greatest crushing strength is obtained with sufficient combined carbon to make a rather hard, white iron, while for the maximum transverse or bending strength the combined carbon is somewhat less and the iron only moderately hard, and for the greatest tensile strength the combined carbon is still less and the iron rather soft. Metal still softer than this grade works with the greatest facility, but is deficient in strength.

Numerical values for the strength will be given at a later point.

(8) *Uses of Cast Iron in Marine Engineering.* Cast iron is used for cylinders, cylinder heads, liners, slide valves, valve chests and connections, and generally for all parts having considerable complexity of form. It is also used for columns, bed plates, bearing pedestals, caps, etc., though cast and forged steel are to some extent displacing cast iron for some of these items. It is also used for grate bars, furnace door frames, and minor boiler fittings, and for a great variety of special purposes usually connected with the stationary or supporting parts of machines.

(9) *Inspection of Castings.* In the inspection of castings care must be had to note the texture of the surface, and to this end the outer scale and burnt sand should be carefully removed by the use of brushes or chipping hammer, or, if necessary, by pickling in dilute muriatic acid. The flaws most liable to occur are blow holes and shrinkage cracks. The latter, however, are not often met with when the moulding and casting are properly carried out. The parts of the casting most liable to be affected by blow holes are those on the upper side or near the top. On this account a sinking head or extra piece is often cast on top, into which the gases and impurities may collect. This is afterward cut off, leaving the sounder metal below.

The presence of blow holes, if large in size or in great number and near the surface, may often be determined by tapping with a hand hammer. The sound given out will serve to indicate to an experienced ear the probable character of the metal underneath.

(10) *Special Operations on Cast Iron.* Cast iron may be softened and toughened by the process of malleablizing, as described in (2). It may be somewhat hardened on the surface by arresting the usual process of malleablizing at a suitable point and then hardening as for steel. This operation arrested before completion results in the formation of a surface layer of material having essentially the properties of steel.

Cast iron may be brazed to itself or to most of the common structural metals by the use of a brazing solder of suitable melting point, and with proper care in the operation. Cast iron may also be united to itself or to wrought iron or steel by the operation of burning. This consists in placing in position the two pieces to be united, and then allowing a stream of melted cast iron to flow over the surfaces to be joined, the adjacent parts being protected by fire clay or other suitable material. The result is to soften or partially melt the surfaces of the pieces, and by arresting the operation at the right moment they may be securely joined together.

[2] Malleable Iron.

(1) *Composition and Manufacture.* If a casting of hard, white iron, or one with a large proportion of combined carbon, be packed in some material which will not fuse at a red heat, which will exclude the air, support the piece and prevent deformation when hot, and if it be then subjected to continuous red heat for some days, the combined carbon will be separated from the iron, but will not be able to collect together in flakes or scales or to form the same structure as in soft, gray cast iron. In consequence the iron crystals remain in more intimate contact, much as in steel, and the tensile strength and toughness are greatly increased.

It has long been supposed that this operation involved an actual withdrawal of the carbon from the iron, and to this end the substances usually employed are either the common red oxide of iron in the form of hematite iron ore, or the black oxide in the form of mill scale, or the corresponding oxide of manganese. These have a decarbonizing effect; that is, under the conditions existing they will to some extent withdraw the carbon from the surface layer of iron. Analyses of malleable iron show, however, that only to a slight extent is the carbon actually withdrawn as a whole, and that the principal change is in the condition of the carbon, as above explained. The surface effect, however, extending in, as it does, for perhaps 1-16 in., is undoubtedly a valuable feature, and while a good quality of malleable iron has been made by the use of river sand as a packing medium, the use of the substances mentioned above is rather to be preferred.

In order that the process may be successful, the iron must have nearly all the carbon in the combined state, and must be low in sulphur, as the latter substance is found to greatly increase the time necessary. It has been customary to use only good charcoal-melted iron in which the sulphur is very low, though a coke-melted iron is quite as suitable, provided the proportion of sulphur is correspondingly small. The process can rarely be applied to very large castings, because such, cooling slowly, usually show a considerable proportion of graphitic carbon.

To carry out the process the castings are embedded in the material selected. The whole is then inclosed in a cast-iron box or pot and is subjected to a full red heat for from two or three days to as many weeks, depending on the size of the piece.

(2) *Physical and Mechanical Properties.* Outside of the numerical information, to be given later, attention may be called to the ductility of malleable iron, which is from four to six times that of cast iron, though only about one-tenth that of wrought iron. Nevertheless good malleable iron can be bent and twisted to a very considerable extent before breaking, and its ability to withstand blows or shocks is very much greater than cast iron. Malleable iron may with care be forged and welded, and it may be case hardened much as with wrought iron.

(3) *Uses in Marine Engineering.* Malleable iron is used for junction boxes and for pipe fittings in certain varieties of water-tube boilers, and to some extent for general pipe fittings on board ship. It would seem that the use of this material might with advantage be extended to many parts in which more strength and toughness are required than can be provided by cast iron of the ordinary type.

[3] Wrought Iron.

(1) *Composition and Manufacture.* Wrought iron is nearly pure iron mixed with more or less slag. Nearly all the wrought iron used in modern times is made by the *puddling* process. For the details of this process reference may be had to text-books on metallurgy. We can only note here that in a furnace somewhat similar to the *open-hearth* referred to in [5] (4) most of the carbon, silicon and other special ingredients of cast iron are removed by the combined action of the flame and of a molten bath of slag or fluxing material consisting chiefly of black oxide of iron. As this process approaches completion small bits of nearly pure iron begin to separate out from the bath of melted slag and unite together. This is helped along by the puddling bar, and after the iron has thus become separated from the liquid slag it is taken out, hammered or squeezed, and rolled down into bars or plates. Some of the slag is necessarily retained in the iron and by the process of manufacture is drawn out into fine threads,

giving to the iron a stringy or fibrous appearance when nicked and bent over or when pulled apart.

The proportion of carbon in wrought iron is very small, ranging from .02 to .20 of one per cent. In addition, small amounts of sulphur, phosphorus, silicon and manganese are usually present.

The proportion of sulphur should not exceed .01 of one per cent. Excess of sulphur makes the iron *red-short*, that is, brittle when red hot.

The proportion of phosphorus may vary from .05 to .25 of one per cent. Excess of phosphorus makes the metal *cold-short*, that is, brittle when cold.

The proportion of silicon may vary from .05 to .30 of one per cent.

The proportion of manganese may vary from .005 to .05 of one per cent. The influence of the silicon and manganese is usually slight and unimportant.

(2) *Special Properties.* Wrought iron is malleable and ductile, and may be rolled, forged, flanged and welded. It cannot be hardened as steel, though by the process of case-hardening a surface layer of steel is formed and may be hardened. Wrought iron may be welded, because for a considerable range of temperature below melting (which takes place only at a very high temperature indeed) the iron becomes soft and plastic, and two pieces pressed together in this condition unite and form on cooling a junction nearly as strong as the solid metal. In order to be thus successful, however, the iron must be heated sufficiently to bring it to the plastic condition, yet not overheated, and there must be employed a flux (usually borax) which will unite with the iron oxide and other impurities at the joint, and form a thin liquid slag, which may be readily pressed out in the operation, thus allowing the clean metal faces of the iron to effect a union as desired.

(3) *Uses in Marine Engineering.* In modern practice the place of wrought iron in marine engineering has been almost entirely taken by steel. Its former office was for all moving parts requiring strength and toughness. It is still used to some extent for the stay bolts and braces of boilers, and for boiler tubes.

[4] Blisters and Laminations.

With modern boiler material these defects are happily rare.

In older practice, however, when wrought iron was the material employed for boiler plates, such defects were quite frequently met with. A lamination consisted in the separation of the material of the plate into layers not welded together and therefore lacking the strength and solidity of the plate proper. The formation of such places was usually due to the presence of slag in the iron which in the operation of rolling the plates would become thinned out into a sheet or layer separating the two parts of the iron and preventing them from becoming welded together and thus forming a solid homogeneous plate. Such places may vary in size from a trifling amount up to patches of several square feet in area. Now when a plate with such laminations is worked into the structure of a boiler with the continual fluctuations of temperature and the consequent expansions and contractions, it very frequently happens that the two parts of the laminations become separated from each other, and in particular the thinner of the two will become raised often to a considerable extent, thus forming a so-called blister. The chief danger from such a blister on the heating surface arises from the non-conductivity of the plate for heat at this point, and the consequent danger of its overheating and giving rise to a serious rupture. Further reference to this point will be found in Sec. 38 [3].

In examining a plate for laminations or small blisters not plainly shown to the eye, the hammer test is usually considered the most reliable. The plate is tapped over its surface, and judging by the sound an experienced ear can usually detect the locality and approximate extent of trouble of this character.

[5] Steel.

(1) *General Composition.* The properties of steel depend partly on the proportions of carbon and other ingredients which it may contain, and partly on the process of manufacture. The proportion of carbon is intermediate between that for wrought iron and for cast iron. In the so-called mild or structural steel the carbon is usually from 1-10 to 1-4 or 1-3 of one per cent. In spring steel the carbon proportion is somewhat greater, and in high carbon grades such as are used for tool steel, etc., the carbon is from .6 to 1.2 per cent. In addition to the carbon there may be sulphur, phosphorus, silicon and manganese in varying but very small amounts.

From the proportion of carbon it follows that steel may be made either by increasing the proportion in wrought iron or decreasing the proportion in cast iron. The earlier processes followed the first method, and high-grade steels are still made in this way by the *crucible* process.

(2) *Crucible Steel.* In this process a pure grade of wrought iron is rolled out into flat bars. These are then cut and piled and packed with intermediate layers of charcoal and subjected to a high temperature for several days. This recarbonizes or adds carbon to the wrought iron, and thus makes what is then called *cement* or *blister* steel. These bars are then broken into pieces of convenient size, placed in small crucibles, melted, and cast into bars or into such forms as are desired.

NOTE.—Mild or structural steel is made wholly by the second general method—the reduction of the proportion of carbon in cast iron. There are two general processes, known as the *Bessemer* and the *Siemens Martin* or *Open-hearth.*

(3) *Bessemer Process.* In this process the carbon and silicon are burned almost entirely out of the cast iron by forcing an air blast through the molten iron in a vessel known as a converter. A small amount of *spiegel eisen* or iron rich in carbon and manganese is then added in such weight as to make the proportion of carbon and manganese suitable for the charge as a whole. The steel thus formed is then cast into ingots or into such forms as may be desired.

In this process no sulphur or phosphorus is removed, so that it is necessary to use a cast iron very nearly free from these ingredients in order that the steel may have the properties desired. A modification by means of which the phosphorus is removed, and known as the *basic* Bessemer process, is used to some extent. In this, calcined or burnt lime is added to the charge just before pouring. This unites with the phosphorus, removes it from the steel, and brings it into the slag. In the basic process the lining of the converter is made of gannister or a calcined magnesia limestone, in order that it may not also be attacked by the added limestone and the resulting slag.

In that form of Bessemer process first noted, and often known as the *acid* process in distinction from the latter or basic process, the lining of the converter is of ordinary fire clay or like material.

The removal of the phosphorus by the basic process makes possible the use of an inferior grade of cast iron. At the same time, engineers are not altogether agreed as to the relative values of the two products, and many prefer steel made by the acid process from an iron nearly free from phosphorus at the start.

(4) *The Open-hearth Process.* In this process a charge of material consisting of wrought iron, cast iron, steel scrap, and sometimes certain ores, is melted on the hearth of a reverberatory furnace heated by gas fuel on the Siemens Martin or regenerative system. The carbon is thus partially burned out in much the same manner as for wrought iron, and the proportion of carbon is brought down to the desired point or slightly below. A charge of spiegel eisen or ferro-manganese is then added in order that the manganese may act on any oxide of iron slag which remains in the bath, and which would make the steel red-short if allowed to form a part of the charge. The manganese separates the iron out from the oxide, returns it to the bath, while the carbon joins in with that already present, and thus produces the desired proportions.

Here as with the similar operation with the Bessemer converter there is no removal of either sulphur or phosphorus, and only materials nearly free from these ingredients can be used for steel of satisfactory quality. With very low carbon, however, a little phosphorus seems to be desirable to add strength to the metal. This limitation of the available materials has led, as with the Bessemer process, to the use of calcined limestone, which unites with most of the phosphorus and holds it in the slag. Here, as in the Bessemer process also, it is necessary to use a basic lining for the furnace, and it is known as the *basic* open-hearth process. By distinction the method without the use of the limestone has come to be known as the *acid* open-hearth process.

As between the products of these two kinds of open-hearth process, there is much difference of opinion among engineers. *Either* will produce good steel with proper care, and *neither* will without it. It is usually considered sufficient to specify the allowable limits for the proportions of phosphorus and sulphur and leave the choice of the acid or basic processes to the maker.

(5) *Open-hearth and Bessemer Steels Compared.* Open-hearth steel is usually preferred for structural material in marine engineering. This is because:

(a) It seems to be more reliable and less subject to unexpected or unexplainable failure than the Bessemer product.

(b) Analysis shows that it is much more homogeneous in composition than Bessemer steel, and experience shows that it is much more uniform in physical quality. This is due to the process of manufacture, which is much more favorable to a thorough mixing of the charge than in the Bessemer process.

(c) The open-hearth steel may be tested from time to time during the operation, so that its composition may be determined and adjusted to fulfil specified conditions. This is not possible with the Bessemer process, and the latter product is therefore not under so good control as is the open-hearth.

(6) *Influence of Sulphur on Steel.* Sulphur makes steel red-short or brittle when hot, and interferes with its forging and welding properties. Manganese tends to counteract the bad effects of sulphur. Good crucible steel has rarely more than .01 of one per cent. In structural steel the proportion may vary from .02 to .08 or .10 of one per cent. When possible it should be reduced to not more than .03 or .04 of one per cent.

(7) *Influence of Phosphorus on Steel.* Phosphorus increases the tensile strength and raises the elastic limit of low carbon or structural steel, but at the expense of its ductility and toughness or ability to withstand shocks and irregularly applied loads. It is thus considered as a dangerous ingredient, and the amount allowable should be carefully specified. This is usually placed from .02 to .10 of one per cent.

(8) *Influence of Silicon on Steel.* Silicon tends to increase the tensile strength and to reduce the ductility of steel. It also increases the soundness of ingots and castings, and by reducing the iron oxide tends to prevent red-shortness. The process of manufacture usually removes nearly all of the silicon, so that it is not an element likely to give trouble to the steel maker. The proportion allowed should not be more than from .10 to .20 of one per cent.

(9) *Influence of Manganese on Steel.* This element is believed to increase hardness and fluidity, and to raise the elastic limit and increase the tensile strength. It also removes iron oxide and sulphur, and tends to counteract the influence of such amounts of sulphur and phosphorus as may remain. It is thus an important factor in preventing red-shortness. The proportion needed to obtain these valuable effects is usually found between .20 and .50 of one per cent.

(10) *Semi-steel.* A metal bearing this trade name has in recent years attracted favorable attention among engineers and has come into considerable use where somewhat greater strength and toughness are required than can be provided by cast iron.

Semi-steel is made by melting up mild steel scrap, such as punchings and clippings of boiler plate, with cast iron pig, in the proportion of about 25 or 30 per cent of the former to 75 or 70 per cent of the latter. The presence of manganese and other special fluxes in small proportions is also found to add essentially to the strength, toughness and good machining qualities of the product. In this way is obtained a material having a tensile strength of 35,000 lb. or over, and a toughness and ability to withstand shocks decidedly greater than for cast iron, and with fairly good machining qualities. Semi-steel casts as readily as most grades of cast iron, and its shrinkage and general manipulation are about the same. The chief drawback seems to lie in the danger of hardness under the lathe, planer or boring tool, but with the proper mixtures this is avoided, and a material very satisfactory for many purposes in marine engineering is thus produced.

(11) *Mechanical Properties of Steel.* The tensile strength of the lowest carbon steel, say about .10 of one per cent carbon, is usually not above from 50,000 to 55,000 lb. per square inch of section. The strength increases with the increase of carbon, and with not above the usual proportions of sulphur and phosphorus, quite uniformly. Experiment shows that under these circumstances the strength will increase up to 75,000 lb. per square inch, or higher, at the rate of from 1,200 to 1,500 lb. per .01 of one per cent of carbon added. At the same time, with the increase in strength the ductility decreases, so that a proper choice must be made according to the particular uses for which the steel is intended. With the best grades of tool steel with carbon ranging from ½ to 1 per cent and over, the strength ranges from 80,000 lb. upward to 120,000 lb., and even higher in exceptional cases.

Flange and *rivet steel* must be tough and ductile in the highest degree. Such steel has usually a tensile strength between 50,000 and 60,000 lb. and an elastic limit of 30,000 to 40,000 lb. Its elongation in 8 in. is from 30 to 35 per cent, and reduction of area at the ruptured section from 50 to 60 per cent. It will bend cold on itself and close down flat under hammer or press, up to a thickness of ¾ in. to 1 in. without a sign of fracture.

Shell steel, used for boiler shells, etc., has usually a strength between 55,000 and 65,000 lb.; elastic limit from 33,000 to 45,000 lb.; elongation in 8 in. of 25 to 30 per cent, and reduction of area of 50 to 60 per cent.

For shafting the quality of the steel is about the same as for shell plates. For piston and connecting rods the strength is rather higher, and ductility somewhat lower.

For steel castings the strength required is usually from 60,000 to 65,000 lb., with an elongation in 8 in. of from 10 to 15 per cent.

(12) *Various Specifications for Structural Steel.*

U. S. Navy.

COMPOSITION OF BOILER PLATES.

Phosphorus: Not over .035 of one per cent.
Sulphur: Not over .040 of one per cent.

STRENGTH OF SHELL PLATES.

Tensile Strength: Between 65,000 and 73,000 lb. per square inch.

Elongation (transverse): Not less than 22 per cent in 8 in.
Elongation (longitudinal): Not less than 25 per cent in 8 in.
Elastic Limit: Not less than 35,000 lb. per square inch.

Cold Bending Test.—One piece cut from each shell and curved head plate, as finished at the rolls for cold-bending test, must bend over flat on itself without sign of fracture.

STRENGTH OF FURNACE AND FLANGE PLATES.

Tensile Strength: Between 52,000 and 60,000 lb. per square inch.

Elongation: Not less than 26 per cent in 8 in.

Quenching Test.—One piece shall be cut from each furnace or flange plate as finished at the rolls for quenching test, and after heating to a dark cherry red plunged into water at a temperature of 82 deg. F. The piece thus prepared must be bent double round a curve of which the diameter is not more than the thickness of the piece tested, without showing any cracks. The ends of the pieces must be parallel after bending.

BOILER RIVETS.

Kind of Material.—Steel for boiler rivets must be made by the open-hearth process and must not show more than .035 of one per cent of phosphorus, nor more than .04 of one per cent of sulphur, and must be of the best composition in other respects.

Tensile Tests.—These specimens for rivets for use in the longitudinal seams of boiler shells shall have from 62,000 to 70,000 lb. per square inch tensile strength, with an elongation of not less than 25 per cent in 8 in.; and all others shall have a tensile strength of from 54,000 to 62,000 lb. per square inch, with an elongation of not less than 28 per cent in 8 in.

Shearing Tests.—From each heat, rivets must show a shearing strength of at least 51,000 lb. per square inch for rivets to be used in longitudinal seams of boiler shells, and at least 44,000 lb. per square inch for all other boiler rivets. Rivets to be driven at the same heat used for working.

Hammer Test.—From each lot six rivets are to be taken at random and submitted to the following tests:

(a) Two rivets to be flattened out cold under the hammer to a thickness of one-half the diameter of the part flattened without showing cracks or flaws.

(b) Two rivets to be flattened out hot under the hammer to a thickness of one-third the diameter of the part flattened without showing cracks or flaws—the heat to be the working heat when driven.

(c) Two rivets to be bent cold into the form of a hook with parallel sides without showing cracks or flaws.

RODS, SHAPES AND FORGINGS FOR BOILER BRACING.

Kind of Material.—Steel for stay rods and braces must be made by the open-hearth process, and must not show more than .035 of one per cent of phosphorus, nor more than .04 of one per cent of sulphur, and must be of the best composition in other respects.

Treatment.—All material for boiler bracing must be annealed after working.

Tensile Test.—Bracing coming into contact with the fire must have a tensile strength of from 50,000 to 58,000 lb., and an elongation of not less than 28 per cent in 8 in., or of 33 per cent in 2 in. in case 8-in. specimens can not be secured. Other bracing must have a tensile strength of not less than 65,000 lb., and an elongation of not less than 24 per cent in 8 in., or of 30 per cent in 2 in. in case 8-in. specimens can not be secured.

Bending Test.—One bar ½ in. thick, cut from each lot of the bracing coming in contact with the fire, must stand bending double to an inner diameter of 1 in. after quenching in water at a temperature of 82 deg. F., from a dark cherry-red heat without showing cracks or flaws. A similar piece cut from each lot of the other bracing must stand cold bending double to an inner diameter of 1 in. without showing cracks or flaws.

Opening and Closing Tests.—Angles, T bars, etc., are to be subjected to the following additional tests: A piece cut from one bar in twenty to be opened out flat while cold; a piece cut from another bar in the same lot shall be closed down on itself until the two sides touch without showing cracks or flaws.

CONNECTING AND PISTON RODS AND VALVE STEMS.

Tensile Strength: Not less than 80,000 lb. per square inch.

Elongation: Not less than 26 per cent in 2 in.

Elastic Limit: Not less than 50,000 lb. per square inch.

Bending Test.—One longitudinal bar ½ in. thick, cut from each forging, must stand bending double, when cold, to an inner diameter of 1 in. without showing cracks or flaws.

THRUST LINE AND PROPELLER SHAFTING.

Tensile Strength: Not less than 80,000 lb. per square inch.

Elongation: Not less than 25 per cent in 2 in.

Elastic Limit: Not less than 50,000 lb. per square inch.

CRANK SHAFTS.

Tensile Strength: Not less than 58,000 lb. per square inch.

Elongation: Not less than 30 per cent in 2 in.

Bending Test.—Bars ½ in. thick, cut from each length of shaft, must stand bending double to an inner diameter of 1 in. without showing cracks or flaws.

STEEL CASTINGS.

Phosphorus: Not more than .06 of one per cent.

Tensile Strength: Not less than 60,000 lb. per square inch.

Elongation (for moving parts): Not less than 15 per cent in 8 in.

Elongation (other castings): Not less than 10 per cent in 8 in.

Bending Test.—A bar 1 in. square shall bend cold without showing cracks or flaws, through an angle of 120 deg. for castings for moving parts of machinery, and 90 deg. for other casting, over a radius not greater than 1½ in.

U. S. Inspection Requirements for Boiler Plate.

Phosphorus: Not more than .06 of one per cent.

Sulphur: Not more than .04 of one per cent.

Elongation (¼ in. and under): 25 per cent in 2 in.

Elongation (¼ in. to 7-16 in. inc.): 25 per cent in 4 in.

Elongation (7-16 in. to 1 in. inc.): 25 per cent in 8 in.

Elongation (1 in. and over): 25 per cent in 6 in.

Reduction of Area at Rupture (½ in. and under): Not less than 50 per cent.

Reduction of Area at Rupture (½ in. to ¾ in.): Not less than 45 per cent.

Reduction of Area at Rupture (¾ in. and over): Not less than 40 per cent.

American Boilermakers' Association Requirements.

Phosphorus: Not over .04 per cent.

Sulphur: Not over .03 per cent.

Tensile Strength: 55,000 to 65,000 lb.

Elongation (⅜ in. and under): 20 per cent in 8 in.

Elongation (⅜ in. to ¾ in.): 22 per cent in 8 in.

Elongation (¾ in. and over): 25 per cent in 8 in.

Cold Bending.—For plates ½ in. thick and under, specimen must bend back on itself without fracture. For plates over ½ in. thick, specimen must bend 180 deg. around a mandril one and one-half times thickness of plate without fracture.

British Board of Trade Requirements.

Tensile Strength of Plates Not Exposed to Flame: 60,480 to 71,680 lb. per square in.

Tensile Strength of Plates Exposed to Flame: 58,240 to 67,200 lb. per square inch.

Elongation: From 18 to 25 per cent in 10 in.

Standard Specifications Adopted by the Association of American Steel Manufacturers.

Special Open-hearth Plate and Rivet Steel.

Steel shall be of four grades, as follows: *Extra Soft, Fire-box, Flange* or *Boiler*, and *Boiler Rivet Steel.*

Extra Soft, Fire-box and Boiler Rivet Steel: Maximum phosphorus, .04 per cent; maximum sulphur, .04 per cent.

Flange or Boiler Steel: Maximum phosphorus, .06 per cent; maximum sulphur, .04 per cent.

PHYSICAL PROPERTIES.

Extra Soft and Boiler Rivet Steel.

Ultimate Strength: 45,000 to 55,000 lb. per square inch.

Elastic Limit: Not less than one-half the ultimate strength.

Elongation: 28 per cent.

Cold and Quench Test: Bends 180 deg. flat on itself without fracture on outside of bent portion.

Fire-box Steel.

Ultimate Strength: 52,000 to 62,000 lb. per square inch.

Elastic Limit: Not less than one-half the ultimate strength.

Elongation: 26 per cent.

Cold and Quench Test: Bends 180 deg. flat on itself without fracture on outside of bent portion.

Flange or Boiler Steel.

Ultimate Strength: 52,000 to 62,000 lb. per square inch.

Elastic Limit: Not less than one-half the ultimate strength.

Elongation: 25 per cent.

Cold and Quench Test: Bends 180 deg. flat on itself without fracture on outside of bent portion.

(13) *Special Properties of Steel.* Mild or low carbon steel may be welded, forged, flanged, rolled and cast. It can not be tempered or hardened with a proportion of carbon lower than about ¾ of one per cent. High carbon steel can be welded only imperfectly and if very high in carbon not at all. It can be forged with care, and cast into forms as desired. It can be tempered or hardened by heating to a full yellow and quenching in cold water or by other means, and then drawing the temper to the point desired.

Mild steel should not be worked under the hammer or flanging press at a low or "blue" heat, as such working is found in many cases to leave the metal brittle and unreliable. Steel in order to weld satisfactorily should have a low proportion of sulphur, and special care is required in the operation, because the range of temperature through which the metal is plastic and fit for welding is less than with wrought iron.

In the operation of tempering, the steel after quenching is very hard and brittle. In order to give to the metal the properties desired, the temper is drawn down by heating it up to a certain temperature, and then quenching again, or, better still, allowing it to cool gradually, provided the temperature does not rise above the limiting value suitable for the purpose desired. If the reheating is done in a bath of oil the conditions may be kept under good control and the final cooling may be slow. If the reheating is in or over a fire the control is lacking and the piece must be quenched as soon as the proper temperature is reached. This is usually determined by the color of the oxide or scale which forms on a brightened surface of the metal. The following table shows the temperatures, corresponding colors, and uses for which the various tempers are suited:

430°	Faint yellow.	Hardest and keenest cutting tools.
450°	Straw yellow.	
470°	Full yellow.	Cutting tools requiring less hardness and more toughness.
490°	Brown yellow or orange.	
510°	Purplish.	Tools for working softer materials, or those required to stand rough usage.
530°	Purple.	
550°	Light blue.	Spring temper. Used for tools requiring great elasticity or toughness, or for working very soft materials.
560°	Full blue.	
600°	Dark blue.	

(14) *Special Steels.* In the common grades of steel the valuable properties are due to the presence of carbon modified in some degree by other ingredients as already described. There are other substances which by uniting with iron in small proportions are able to give to the combination increased strength or hardness or other valuable properties. We have thus various special steels in which the properties may be due to the presence of both carbon and other ingredients, or due chiefly to special ingredients other than carbon. Of these special steels we may note the following:

Nickel steel, containing somewhere about 3 per cent of nickel and varying amounts of carbon, is found to have increased strength and toughness as compared with ordinary steel. Nickel steel is most extensively used for armor plate, though to some extent it has been employed in Government work for screw-shafts and for boiler plates. For the former purposes it has given excellent satisfaction, but for the latter use difficulty has been met with in obtaining plates free from surface defects.

Chrome steel, containing from .5 to 1.5 or 2 per cent of chromium may be made excessively hard, but it is not always reliable, and is not regarded with general favor.

Tungsten steel or *mushet steel* is a steel containing carbon and tungsten, the latter in proportions as high as 8 to 10 per cent. This steel must be forged with care and is excessively hard. The hardness is not increased by tempering, but is naturally acquired as the metal cools. Hence it is said to be self-hardening. Some specimens contain also small amounts of manganese and silver. Its chief use is for lathe and planer or other cutting and shearing tools where excessive hardness is required.

(15) *Uses of Steel in Marine Construction.* In modern practice mild or structural steel is used entirely in the construction of ships.

The same general class of material is used for all parts of boilers, though the tubes are still sometimes made of wrought iron.

Cast steel is used for various parts of engines such as pistons, crosshead blocks, columns, bed-plates, bearing pedestals and caps, propeller blades, and for many small pieces and fittings. Pistons are made almost exclusively of cast steel. For most of the other items mentioned cast iron is still used, probably to a larger extent than cast steel, especially where the castings are large and complicated in form, as with columns and bed-plates.

Forged steel is used for columns, piston-rods, connecting-rods, crank and line shafting, and for many other smaller and minor parts.

Sec. 6. LEAD.

Lead is a very soft, dense metal, grayish in color after exposure to the air, but of a bright silvery luster when freshly cut. Commercial lead often contains small amounts of iron, copper, silver and antimony, making it harder than the pure metal. It is very malleable and plastic. In engineering, lead is chiefly of value as an ingredient of bearing metals and other special alloys. Lead piping is also used to some extent for water suction and delivery pipes where the pressure is only moderate, and where the readiness with which it may be bent and fitted adapts it for use in contracted places.

Sec. 7. TIN.

Tin is a soft, white, lustrous metal with great malleability. Commercial tin usually contains small portions of many other substances, such as lead, iron, copper, arsenic, antimony and bismuth. It is largely used as an alloy in the various bronzes and other special metals. Tin resists corrosion well and in consequence is often used as a coating for condenser tubes. It is also used for coating iron plates, the product being the so-called "tin plate" of commerce. It melts at about 450 deg., which corresponds to a steam pressure of about 400 lbs. per square inch. Due to this low melting point tin is often used as the composition for *safety plugs* in boilers.

Sec. 8. ZINC.

Zinc, or "spelter," as it is often called commercially, is a brittle and moderately hard white metal with a very crystalline fracture. The impurities most commonly found in zinc are iron, lead and arsenic. It is used chiefly as an alloy in the various brasses, bronzes, etc., and as a coating for iron and steel plates, rods, etc. The process of applying zinc for such a coating is called "galvanizing," and the product "galvanized" iron or steel. Electricity, however, is not used in the process, the articles, after being well cleaned, being simply dipped in a tank of melted zinc and then withdrawn. Slabs of zinc are also used in marine boilers to prevent corrosion.

Sec. 9. ALLOYS.

A mixture of two or more metals is called an *alloy*. The properties of an alloy are often surprisingly different from those of its ingredients. The melting point is sometimes lower than that of any of the ingredients, while the strength, elastic limit and hardness are often higher than for any of them.

Mixtures of *copper* and *zinc* are called *brass*. Mixtures of *copper* and *tin*, or *copper*, *tin* and *zinc*, with sometimes other substances in small proportion, form *gun metals*, *compositions* and *bronzes*. These terms are, however, rather loosely employed. Various mixtures of two or more of the metals—*copper*, *tin*, *zinc*, *lead*, *antimony*—form the various bearing metals.

Brass and composition are used for piping and pipe-fitting; globe, gate, check and safety valves; condenser tubes and shells; sleeves for tail shafts, and for a great number of small fittings and attachments for which the metal may be suited. The bronzes are employed for many of the uses of brass where more hardness, strength or rigidity are required. They are used with especial success as a material for propeller blades.

The white metals, supported or backed by some other metal, such as brass, cast iron or cast steel, to give the necessary strength, are now very largely used for bearing surfaces.

PROPORTIONS OF INGREDIENTS FOR VARIOUS ALLOYS.

In the following proportions the numbers after the ingredients denote the number of parts in 100 of the mixture. They represent either the usual proportions, or the results of special analyses of samples, and have been collected from various sources. The alloys are arranged in the alphabetical order of their names to facilitate ready reference:

Admiralty Bronze.—Copper 87, tin 8, zinc 5.

Aluminum Brass.—Copper 63, zinc 34, aluminum 3.

Aluminum Bronze.—Copper 89 to 98, aluminum 11 to 2.

Anti-Friction, A.—Zinc 1, iron .65, lead 78.75, antimony 19.6.

Anti-Friction, B.—Copper 1.6, tin 98.13, iron trace.

Anti-Friction, C.—Copper 3.8, tin 78.4, lead 6, antimony 11.8.

Babbitt (Light).—Copper 1.8, tin 89.3, antimony 8.9.

Babbitt (Heavy).—Copper 3.7, tin 88.9, antimony 7.4.

Brass, Common Yellow.—Copper 65.3, zinc 32.7, lead 2.

Brazing Metal.—Copper 84, zinc 16.

Brazing Solder.—Copper 50, zinc 50.

Bush Metal.—Copper 80, tin 5, zinc 10, lead 5.

Delta Metal.—Copper 50 to 60, tin 1 to 2, zinc 34 to 44, iron 2 to 4.

Deoxidized Bronze.—Copper 82, tin 12.46, zinc 3,23, iron .10, lead 2.14, phosphorus trace, silver .07.

Gun Metal.—Copper 89, tin 8.25, zinc 2.75.

Magnolia.—Tin ?, zinc trace, iron trace, lead 83.55, antimony 16.45.

Manganese Bronze.—Copper 88.64, tin 8.7, zinc 1.57, iron .72, lead .30.

Muntz Metal.—Copper 60, zinc 40.

Navy Brass.—Copper 62, tin 1, zinc 37.

Navy Composition.—Copper 88, tin 10, zinc 2.

Navy Journal Boxes.—Copper 82.8, tin 13.8, zinc 3.4.

Parsons White Metal.—Copper 1.68, tin 72.9, zinc 22.9, lead 1.68, antimony .84.

Phosphor Bronze.—Copper 90 to 92, phosphide of tin 10 to 8.

Steam Metal.—Copper 85, tin 6.5, zinc 4.5, lead 4.25.

Tobin Bronze.—Copper 59 to 61, tin 1 to 2, zinc 37 to 38, iron .1 to .2, antimony .30 to .35.

White Metal.—Lead 88, antimony 12.

Sec. 10. THE TESTING OF METALS.

[1] Different Kinds of Tests.

Metals may be tested for strength in various ways—in *tension*, by pulling apart a test piece of specified pattern and size; in *compression*, by crushing a piece of suitable dimensions; in *cross breaking*, by supporting a bar at two points and breaking or bending it in the testing machine by a load applied at an intermediate point; in *torsion*, by twisting apart a bar in a machine especially designed for the purpose; in *direct shearing*, by breaking a riveted or pin joint connection in the usual machine; for *impact* or *shock*, by letting a weight drop through a certain height and by its blow develop suddenly the stress in the material.

[2] Explanation of Terms Used.

Ultimate Strength.—The ultimate strength of a test piece is the load required to produce fracture, reduced to a square inch of original section; or in other words, the ultimate or highest load divided by the original area. Thus if the area of the cross-section of a test piece is .42 sq. in. and the load producing fracture is 28,400 lb., the ultimate strength equals $28,400 \div .42 = 67,620$ lb. per square inch.

Elastic Limit.—The elastic limit is the smallest load, reduced to one square inch of area, which will produce a permanent set or distortion of the material. Thus in a tension test if the cross-section is .68 sq. in. and a permanent elongation or set is just produced by a load of 27,600 lb., the elastic limit is at $27,600 \div .68 = 40,600$.

Elongation.—A certain length being marked off on the test piece as described in [3], [4], the percentage of elongation is found by dividing the actual extension of the length just before rupture by the original length, and reducing to per cent. Thus if a length of 8 in. is marked off on the test piece and if the length between the same marks at fracture is 10.2 in., the actual elongation is 2.2 in. and the percentage elongation is $220 \div 8 = 27.5$ per cent. When a test piece is first put under load the

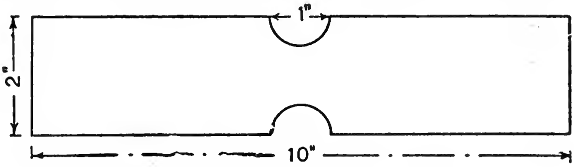


Fig. 1. Test Piece for Iron Plate.

elongation is distributed nearly uniformly over its length. This continues until the piece begins to neck down near the point of final fracture. Nearly all of the remaining elongation is restricted to the immediate vicinity of this point. Hence the percentage elongation with short length of test piece may be much

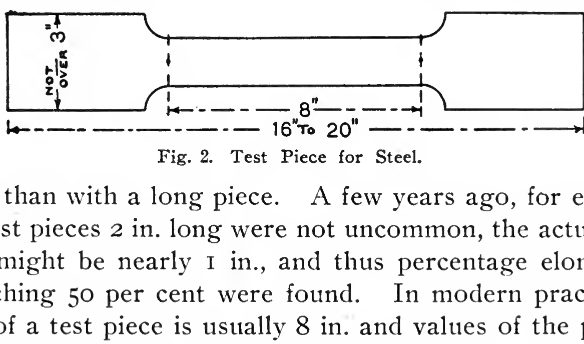


Fig. 2. Test Piece for Steel.

greater than with a long piece. A few years ago, for example, when test pieces 2 in. long were not uncommon, the actual elongation might be nearly 1 in., and thus percentage elongations approaching 50 per cent were found. In modern practice the length of a test piece is usually 8 in. and values of the percentage elongation over 30 per cent even with vastly superior material, are rarely met with. In reporting elongation the length used should always be stated.

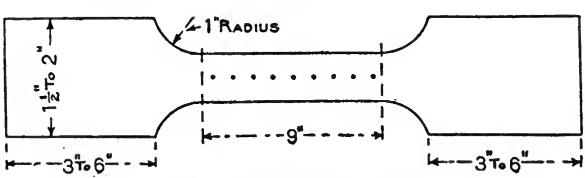


Fig. 3. Test Piece for Steel.

Reduction of Area.—The percentage reduction of area is found by substracting the final area of the section at the point of fracture from the original area at the same point, dividing the difference by the latter, and reducing to per cent. Thus if the original area is .68 sq. in. and the final area is .36 sq. in., the actual reduction is .68 — .36 = .32 sq. in., and the percentage reduction is 3200 ÷ 68 = 47.5 per cent.

[3] Test Pieces for Iron.

In modern practice the form of test pieces for iron is usually

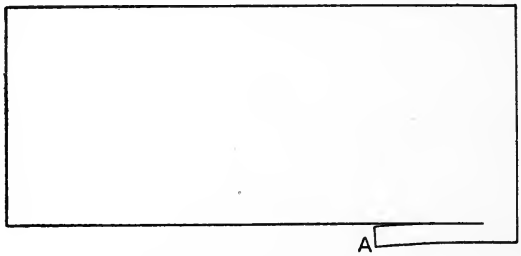


Fig. 4. Plate with Coupon.

the same as for steel, and as described in [4]. The form of test piece for wrought iron plate prescribed by the U. S. Board of Supervising Inspectors of Steam Vessels is, however, somewhat different, and is illustrated in Fig. 1. If the plate is 5-16 in. thick or less, the width at the reduced section must be one inch.

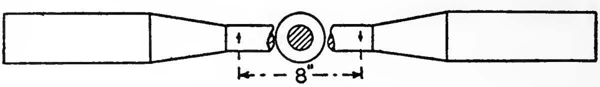


Fig. 5. Round Test Piece.

If the plate is over 5-16 in. in thickness, the width of the piece must be reduced so that the cross-sectional area at the reduced section shall be about .4 sq. in., but it must not be greater than .45 sq. in. nor less than .35 sq. in.

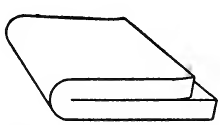


Fig. 6. Bending Test.

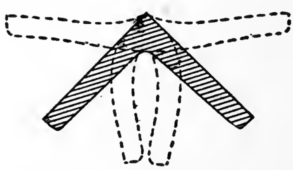


Fig. 8. Angle Test.

[4] Test Pieces for Steel and Other Materials.

Fig. 2 shows the form of test piece for tension prescribed by the Navy Department for tests of steel plate for naval uses.

Fig. 3 shows the form prescribed by the Association of American Steel Manufacturers, and adopted by the U. S. Board of Supervising Inspectors of Steam Vessels. The test piece for plates is cut from a "coupon," as it is called, left on one corner of the plate as shown at *A*, Fig. 4. The U. S. law requires further that:

"Every iron or steel plate intended for the construction of boilers to be used on steam vessels shall be stamped by the manufacturer in the following manner: At the diagonal corners, at a distance of about 4 in. from the edges and at or near the center of the plate, with the name of the manufacturer, the place where manufactured, and the number of pounds tensile strain it will bear to the sectional square inch."

Fig. 5 shows the usual round form of test piece for all material except plates.

[5] Bending, Quenching and Hammer Tests.

The nature of these tests has already been described in Sec. 5, [5], (11), (12).

Fig. 6 illustrates a cold bending test on a piece of steel plate. A *drift* test is also sometimes required. This is illustrated in Fig. 7, and consists in driving taper drifts of con-

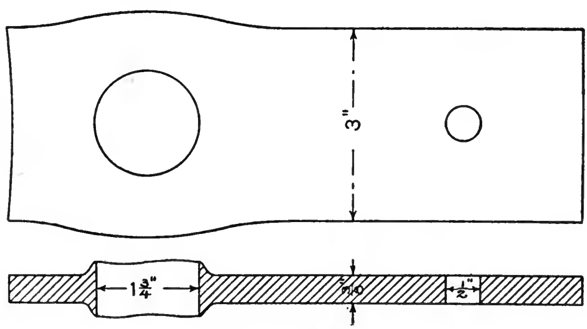


Fig. 7. Drift Test.

tinually increasing size into a punched or drilled hole until the diameter is increased to at least twice its original size. The metal must stand this test without sign of fracture about the edges of the hole.

Bending tests for angle and Tee irons, as referred to in Sec. 5, [5], (12), are also illustrated in Fig. 8.

TABLE I.—Various Properties of the Useful Metals.

Metal.	Tensile Strength.	Elastic Limit for Tension.	Specific Gravity.	Weight per Cubic Inch.	Weight per Cubic Foot.	Melting Point.
Gray Cast Iron	18,000– 25,000	....	7.22	.260	450	2,500
White Cast Iron	25,000– 35,000					2,300
Wrought Iron	45,000– 55,000	30,000–35,000	7.7	.278	480	....
Structural Steel	55,000– 80,000	60–70% of tensile strength	7.8	.283	488	2,800
Tool Steel	80,000–150,000					2,600
Cast Steel	60,000– 80,000	....	7.92	.284	490	2,800
Malleable Iron	40,000– 50,000	30,000–35,000	7.3	.264	456	2,600
Copper, Sheet	30,000– 35,000	hot rolled 7,000–8,000 cold hammered 20,000	8.6–8.9	.318	av. 550	2,000
Copper, Wire	35,000– 40,000	25,000	8.6–8.9	.318	av. 550	2,000
Brass	30,000– 45,000	about 50% of tensile strength	7.8–8.5	.283–.307	488–530	1,200–1,800
Bronze	50,000– 65,000	50–60% of tensile strength	8.0–8.5	.289–.307	500–530	1,500–1,800
Aluminum	18,000 cast (20,000–40,000 rolled and drawn)	about 50% of tensile strength	2.56–2.75	.0943	av. 163	1,200
Zinc	5,000–6,000	....	6.9–7.1	.253	437	800
Tin	4,000	....	7.3	.264	456	450
Lead	1,600–2,400	....	11.33	.410	708	625
Antimony	1,000	....	6.71	.243	420	842
Nickel	....	....	8.3–8.9	.311	538	3,000

CHAPTER II.

FUELS.

Sec. 11. COAL.

[1] Composition and General Properties.

The principal fuel for engineering purposes is coal. It consists of the following chief substances.

(A) *Uncrystallized Carbon.*

(B) *Volatile Hydrocarbons.* Hydrocarbons are chemical substances formed of carbon and hydrogen in certain proportions. They often become partially oxidized* by the union of part of their hydrogen with oxygen, in the same proportion as in water. Upon the application of heat to the coal they escape in the form of gas, and are hence said to be *volatile.*

(C) *Nitrogen and Oxygen.* These gases, the constituents of air, are also found, the latter in addition to this amount joined to the hydrogen as above referred to.

(D) *Sulphur.* This is found in small amounts, chiefly as a part of the mineral known as *iron pyrites*†. The proportion of sulphur is rarely above three per cent and usually much less.

(E) *Ash.* This consists of the earthy and incombustible substances present as impurities in the coal.

Coal may be roughly divided into two chief varieties, *Anthracite* and *Bituminous,* with intermediate grades, *Semi-anthracite* and *Semi-bituminous* occupying the general middle ground between the two. In the present chapter we shall frequently use the terms *anthracite* and *bituminous* as denoting the general division into the two chief varieties, as above noted.

Anthracite coal is sold commercially in hard, compact lumps, showing a shiny, smooth surface when first broken.

* Oxidized means united with oxygen.

† *Iron pyrites* is a mineral formed of iron and sulphur in the proportion of 46.7 parts of iron to 53.3 of sulphur.

Bituminous coal is relatively soft and is sold commercially in lumps or irregular size. It crumbles easily, showing often a rather dull surface when broken.

In the anthracite coal the proportion of volatile matter varies from 3 to 10 per cent; in semi-anthracite and semi-bituminous, from 10 to 20 per cent, and in bituminous, from 20 to 50 per cent. The amount of ash in good coal should not exceed from 8 to 10 per cent, while occasionally it falls as low as 5 per cent. Anthracite coal is graded commercially, according to size, the chief terms being the following, in the order of increasing size: *Buckwheat, pea, chestnut, stone, egg, broken* and *lump.*

[2]. Combustion.

Combustion means simply the chemical union of a substance with oxygen. The oxygen is furnished by the air, which contains oxygen and nitrogen. These in air are not in chemical union, but simply as a mixture, in the proportion by weight of twenty-three parts oxygen to seventy-seven parts nitrogen in one hundred parts of air. When bodies enter into combustion, or into combustion with oxygen, heat is set free, and the products formed by the combustion are very much hotter than the original fuel and oxygen.

The manner in which coal burns, or enters into combustion, depends upon its composition and upon the nature of the fire and the supply of air. The elements available for the liberation of heat are the carbon and the hydrogen. Small quantities of sulphur are frequently present, but the amount is so small and the heating power so feeble that its influence may be neglected. A pound of pure carbon requires for its complete combustion, 2 2-3 pounds of oxygen, and the result is 3 2-3 pounds of *carbonic acid* or *carbon dioxide* in the form of gas. The total amount of heat set free in this operation is about 14,500 heat units. Now, since the proportion of oxygen in the air is about 23 per cent, the number of pounds of air required per pound of carbon will be 2 2-3 $\div$ .23, or 2.66 $\div$ .23, or about 12. Similarly a pound of pure hydrogen requires for its complete combustion, 8 pounds of oxygen, and the result is 9 pounds of water vapor. The total amount of heat set free in this operation is about 62,000 heat units. In the same way as above, it follows that the combustion of a pound of hydrogen will require the presence of 8 $\div$ .23, or about 35 pounds of air. The amount of hydrogen, however, is usually small, and allowing for the ash the amount of air necessary to barely furnish the oxygen required for one pound of fuel is about 12, or substantially the same as for one pound of carbon. In practice, however, it is found that this would be insufficient to maintain the draft, nor could we expect that the air would be so distributed as to give exactly the right amount of oxygen at the right place. It is, therefore, necessary practically, to provide a large excess of air and the amount actually passing into the furnaces is usually not less than 18 or 20 pounds per pound of coal, and may even considerably exceed this amount. At 12.5 cu. ft. per pound this will give for the volume of air required per pound of coal from 225 to 250 cu. ft. and upward.

Let us now consider the process of combustion with bituminous or semi-bituminous coal. When such coal is put on the fire the first result is not a combustion of the carbon, but a distillation or driving off of the hydrocarbons in the form of gas, and until this operation is nearly completed there will be little or no combustion of the carbon. During this first operation of distillation, heat is absorbed by the fresh coal from the remainder of the fire for the liberation of these gases which are substantially the same as those forming ordinary illuminating gas. After these gases are liberated from the coal they rise into the furnace and combustion chamber. Here, if they meet with a suitable supply of air at a proper temperature, they will be burned, both carbon and hydrogen, and will thus set free all the heat which is obtainable from them. If the air is insufficient in amount the gases will be only partly consumed, the oxygen uniting most readily with the hydrogen and leaving the carbon in fine particles to form *smoke* or *soot*, according as they float away with the products of combustion, or become closely packed together on some of the surfaces of the boiler. If the air is not sufficiently hot likewise, we may have a partial combustion resulting in burning the hydrogen into water vapor and in setting the carbon free as smoke or soot as before. If, however, the temperature is too low the gases may become chilled and pass off as a whole unburnt, thus carrying away not only their own heat of combustion, but also the heat which was absorbed for their liberation. If, on the other hand, hydrocarbon gases are subjected to a very high temperature before being mixed with the air, they will become more or less broken up into free hydrogen and carbon in fine particles. If these are kept at a temperature high enough for ignition and are supplied with oxygen, they will burn; but if they fall below the proper temperature they pass off unburnt, the carbon constituting smoke or soot, as before. Smoke is, therefore, the sign of a fuel containing hydrocarbons, and of a more or less imperfect combustion. The actual amount of fuel lost in ordinary smoke is, however, quite small; so small that it is often considered as having no significant influence on the question of economy. Hence, smoke prevention is often considered as hardly worth special effort, so far as the saving of fuel alone is concerned. There may be other losses, however, in connection with the general condition of which smoke is an indication, and any mode of design and of general operation which reduces the smoke formation will usually tend toward economy of combustion.

We have already seen that the conditions for burning the hydrocarbon gas are high temperature and an air supply above the grates and in the combustion chamber. We have here, then, one of the reasons for providing openings for the proper admission of air above the grates as well as underneath.

Let us now return to the residue left on the grates after the escape of the hydrocarbon gases. During this part of the operation certain kinds of bituminous coal swell up and cake more or less firmly together on the grate. Such are called *caking* coals. The swelling up is due to the formation of gas in the midst of the coal and to its efforts to escape, while the caking is due to a partial softening or melting of the substance under heat as the hydrocarbons are set free. Other kinds of bituminous coal undergo little change in their external form, while still others break up into small particles or grains. Those latter varieties are called *non-caking* or *free-burning* coals. In any case the residue, after the hydrocarbons are set free, is called coke and consists of nearly pure carbon with ash.

As we have already seen the carbon burns by uniting with oxygen, and this must take place at the burning lump itself. Hence, it is necessary that the air should penetrate thoroughly all parts of the fire, and to this end it is brought in, in part at least, under the grate and by the draft pressure is forced upward through the mass of burning coal. If the fire is rather thick the operation proceeds in the following way. The carbon and oxygen first unite in complete combustion, 1 pound of carbon to 2 2-3 pounds of oxygen, and the product, carbon dioxide, proceeds upward through the fire. As this gas comes in contact, however, with the cooler coal in the midst or near the top of the bed of fuel it absorbs some of the carbon and becomes changed to a combination in the proportion of 1 pound of carbon to 1 1-3 pounds of oxygen. This gas is called *carbon monoxide.* In this operation also is absorbed back again more than two-thirds of the heat which the first combustion had liberated. If the gas should escape unburnt, a serious loss would result, as only about 4,450 heat units or less than one-third of the heat available in the carbon would have been liberated. If, however, the gas finds air above the grate and a suitable temperature, the carbon which was absorbed is burnt out again and the corresponding heat is given back, so that the final result is the complete combustion of the carbon and the liberation of all the heat possible. The formation of carbon monoxide in this way shows again the need of admitting air above the grate, as well as underneath. This gas burns with the peculiar blue flame so often seen, especially after a fresh firing with anthracite coal, and the presence of this flame thus indicates the formation and recombustion of carbon monoxide in the way described. After the coal has all been thoroughly ignited and raised to a bright glowing heat, the combustion into carbon dioxide is completed at once, and there is little or no formation of carbon monoxide to be burned as a gas above the grate. The thinner the fire the more quickly is this condition reached.

The combustion of semi-anthracite and of anthracite coal proceeds in the same general manner as for bituminous coal, except that the period of distillation becomes shorter and less important as the proportion of hydrocarbon is decreased. It thus results that an ordinary anthracite coal burns almost entirely in the manner described for the coke residue of bituminous coal, except that in consequence of the lower temperature of the fuel during the early stages of combustion, there is apt to be a more pronounced formation of carbon monoxide for the combustion of which there must be a supply of air above the grate as already noted.

[3]. Impurities in Coal. Clinker Formation.

The chief impurities in coal may be divided as follows: (A) Nearly infusible slate, stone, and earthy matter either in separate lumps or distributed through the coal as a whole, thus giving it a low carbon value. (B) Mineral materials more or less fusible, and thus capable of melting and forming a slag which uniting with the ash and slate forms clinker. Substances liable to be present in coal and which are more or less fusible at the high temperatures in the furnaces are: *Potash, soda, lime* and *silica.* The melting point of these substances is also considerably lowered by mixing with iron oxide, which is always formed by the oxidation or combustion of iron pyrites. The presence of this substance in the coal will thus result in lowering the melting point of the other mineral earths and impurities, and in the greater liability to form clinker. This formation of clinker may be so considerable as to seriously interfere with the combustion of the coal, and in such cases its removal must be carefully attended to from time to time in order to keep the fires in good condition.

Iron oxide, or common iron rust as we call it more familiarly, will give to the ashes a reddish tinge so that such a color noted in the ash may usually be accepted as an indication of the presence of iron pyrites in the coal, with the various results which have been already noted. Its presence in any considerable amount is also usually shown by a yellowish or brassy appearance of the coal. For the formation of little or no clinker a coal should have little or no alkali, lime, or pyrites. Such coal in burning gives a nearly white, soft and friable ash.

[4]. Weathering of Coal.

When coal is exposed to the air and weather for a considerable period of time there is a slow absorption of oxygen, and thus a real combustion and wasting of the fuel value of the coal. It thus results that the coal during this operation is really burning up, though at a rate so slow that the heat developed is hardly appreciable and the change in the outward appearance of the coal is so gradual as to escape ordinary notice. The hydrocarbons are much more readily subject to this operation of gradual oxidation or combustion than pure carbon, the latter entering only with great difficulty into union with oxygen at ordinary temperatures. It thus follows that bituminous coals are much more subject to waste and change by weathering than anthracite coals. In addition to the loss due to this slow combustion there is often a gradual escape of gaseous hydrocarbons imprisoned within the lumps, or a gradual vaporization of liquid hydrocarbons and their escape as vapor. Such losses also are, of course, more marked with bituminous than with anthracite coals. A bituminous caking coal often becomes changed to a non-caking coal after exposure to the air and weather for a considerable period of time.

The chief external conditions which may influence *weathering* are *moisture* and *heat*. If the coal contains no iron pyrites, moisture is believed to slightly retard the operation of slow combustion, and thus to act beneficially rather than the reverse. If iron pyrites is present in the coal, the conditions are changed. Iron pyrites oxidize with comparative readiness at ordinary temperatures, both the sulphur and iron uniting with the oxygen. It thus tends to set up the operation of oxidation and to break up the lump into small bits, while the heat developed is a further aid to the continuance of the process. The oxidation of iron pyrites is, moreover, much aided by moisture, which, therefore, with such coals, becomes a distinct disadvantage. In any event a coal with iron pyrites may be expected to suffer more seriously by weathering than one free from this substance. In extreme cases the oxidation of the pyrites has caused the crumbling of the coal into such small bits that it has become nearly worthless for its original purposes.

Heat in general always increases the activity of this slow combustion, and hence tends to increase the loss due to weathering. The heat developed by slow oxidation in the interior of large piles or masses of coal escapes with great difficulty and thus accumulates and raises the temperature, thus making the conditions still more favorable for the continuance of the process. So far as this effect goes, therefore, the loss would be more serious in large piles than in small. This is, however, offset by the greater difficulty which the oxygen has in penetrating to the interior of the pile as it is larger in size. It results that with other things equal there is no great difference in the loss due to weathering with coal either in large or in small bulk.

[5]. Spontaneous Combustion.

We have already seen under the head of *weathering* that coal at ordinary temperatures is subject to a very slow oxidation, or combustion, which gradually wastes away its fuel value. When the coal is freshly mined this oxidation seems to be especially active due to the property which carbon has of absorbing or condensing gases upon its surface. The volume of oxygen that different coals are capable of absorbing varies from 1¼ to 3 times the volume of the coal. The oxygen thus absorbed is very active chemically, due to the fact that coming from the air it is absorbed more readily than the nitrogen, and is thus less diluted than in the air. This absorption is itself attended by the production of heat, and this heat, in conjunction with other conditions favorable to chemical action, brings about an oxidation of the hydrocarbons of the coal, thus generating still more heat.

Now, if the coal is in small bulk and well ventilated, there will be little chance for the gradual accumulation of the heat and a consequent rise of temperature. A few lumps of coal exposed to the open air may lose much by weathering in the course of six months or a year, but the heat set free will readily escape and the rise of temperature will be unnoticeable. If, on the contrary, the coal is in large bulk, or is confined in bunkers with little or no ventilation, the heat developed by slow oxidation will be imprisoned and the temperature may gradually rise to the point where active combustion will proceed according to the supply of air available.

It thus appears that there may be danger from no ventilation or from insufficient ventilation. Opinions differ on these points, but it may probably be accepted that unless the ventilation can be made thorough, the compartment should be kept tight and the air excluded as much as possible. At the same time before such closed compartments or bunkers are entered with a light they should be thoroughly ventilated, especially if the coal is of a quality likely to freely disengage hydrocarbon gases.

A further important point to be noted relates to the influence which the initial temperature has on the rapidity of chemical actions of this kind. Below a temperature of 100 F. the action will go on slowly with little chance of undue heating taking place, but as soon as the temperature rises much above 100 F., especially with certain coals, spontaneous ignition is only a question of time.

It appears, therefore, that the true index of the danger of spontaneous combustion must be taken as the capacity of the coal for absorbing or condensing gases in its outer layers or near its surface. This in turn will be shown by the amount of moisture which it can absorb from the air. A coal which absorbs a large amount of moisture from the air will at the same time absorb a large amount of oxygen, and will, therefore, be relatively a dangerous coal as regards spontaneous combustion, while on the other hand a coal which absorbs but a small amount of moisture from the air will likewise absorb but little oxygen, and will be comparatively safe as regards spontaneous combustion. The percentage of moisture which can be absorbed from the air by coal is found to vary from about 2.5 to 10 per cent, and experience has shown that the liability to spontaneous combustion varies closely with this percentage.

In general then the liability to spontaneous combustion depends on,

(1) The size of the cargo or compartment, increasing as the bulk increases.

(2) The size of the coal, increasing as the lumps are smaller, and thus present relatively more surface.

(3) The presence of iron pyrites with moisture. Iron pyrites has sometimes been thought a possible direct cause of spontaneous combustion, but the proportion of this substance is small, rarely rising above 3 or 4 per cent, and the heat developed by the combustion of sulphur and iron is very much less per pound than for carbon and hydrogen. The heat developed by the oxidation of iron pyrites is, therefore, hardly sufficient to do more than help along the general condition of slow combustion as referred to above. In another way, however, the presence of the pyrites may have an important influence on the result. The presence of moisture favors the oxidation of the pyrites and as a result of this it will swell and tend to split up the coal, thus decreasing the size of the lump and increasing its absorbing surface. It is presumably in this way that the presence of iron pyrites tends to aid spontaneous combustion.

(4) The quality of the coal. Bituminous and semi-bituminous coals are more liable to spontaneous combustion than anthracite coal, because they are more porous and friable and present more absorbing surface, and, furthermore, are rich, in easily oxidisable hydrocarbons as already noted in [1]. In fact, under usual conditions anthracite coal may be considered as beyond danger of this character.

(5) The amount of weathering the coal has had. If it has

been exposed to considerable weathering and has since been subjected to but little breakage, then additional oxygen will be absorbed, but very slowly, and the danger of spontaneous combustion will be very small indeed.

(6) The temperature of the bunkers or compartments as affected by the nearness to boilers, funnel, etc.

(7) Ventilation of cargo. For ventilation to be thoroughly effective cold air would have to sweep continuously and freely through every part, a condition hardly possible to attain with a coal cargo. Anything short of this may possibly increase the danger by supplying just about the right amount of air to create the maximum heating.

The gases imprisoned within the coal, reference to which was made in [4], may also escape and collect in a closed and unventilated compartment or bunker, and upon the later introduction of a light an explosion may result. This is exactly similar to the way in which firedamp explosions in mines may occur.

[6]. Corrosion.

As a further possible result of the presence of iron pyrites and the sulphur which is one of its constituents, the corrosion of the metal surfaces of the boiler may be mentioned. Sulphur in burning in the presence of moisture may produce sulphuric acid, and this may seriously corrode such surfaces as it comes in contact with, especially if there is opportunity for its gradual action during periods when the boiler is not in active use. The conditions necessary for the formation of sulphuric acid are, however, not commonly present in marine boilers, and the danger of corrosion from the combustion of iron pyrites is not usually considered as serious.

[7]. Transportation and Stowage.

In general anthracite coal bears transportation and handling better than bituminous on account of its greater hardness. It also stows more evenly on account of the greater uniformity in size of lump.

The weight of coal in the solid lump is from 70 to 80 lb. per cubic foot for bituminous grades, and from 85, or 90 to 100 lb. per cubic foot for anthracite grades. When broken up in ordinary commercial sizes, however, its weight in bulk is usually from 50 to 54 lb. per cubic foot for bituminous, and from 53 to 58 lb. per cubic foot for anthracite. These weights correspond to an allowance of from 42 to 45 cu. ft. per ton of 2,240 lb. for bituminous grades and from 39 to 42 cu. ft. per ton for anthracite grades.

[8]. General Comparison Between Bituminous and Anthracite Coal.

As between the two kinds of coal, bituminous burns more readily than anthracite, and requires a somewhat lower temperature for the process. This is because with the former the hydrocarbon gases are first driven off and burnt, while the coke residue is left in a light and porous condition, and thus well suited for intimate contact with the oxygen, and for rapid elevation to the temperature of ignition. On the other hand, with anthracite coal the hydrocarbon gases are so small in amount as to have little influence on the process, and the coal has, therefore, to burn as compact, solid lumps of carbon, with little opportunity for contact with the oxygen except at the outer surface.

It results that under the same conditions of draft, firing, etc., considerably more coal can be burned per square foot of grate surface with bituminous than with anthracite. The excess of the former will depend, of course, on the special conditions, but will usually reach from 20 per cent to 40 per cent, or even more.

From the explanation given in [2] it will be clearly seen that smoke and soot are chiefly the products of bituminous coal. With anthracite coal, immediately after firing, a slight show of smoke may be formed, but with neither the volume nor density of the smoke formed by bituminous coal, while the soot formed with anthracite is too small in quantity to be of any significance.

Bituminous coal, as we have seen, is more liable to spontaneous combustion than anthracite, and the loss by weathering is usually more serious.

A good quality of free-burning semi-bituminous coal is usually considered as the best variety for all around steaming purposes.

Sec. 12. BRIQUETTES AND ARTIFICIAL FUEL.

This fuel is made from small bits of coal, coal dust, or from certain grades of coal which are so soft or crumbling that they cannot be readily used in their natural state. The material after selection and removal of impurities, so far as practicable, is reduced to powder by grinding, and is then mixed with some binding material and pressed into cubes or blocks weighing from one to three or four pounds each. The binding material is usually coal-tar, asphalt, crude oil refuse, or some similar substance. In some cases a caking coal has been used by heating until softening occurs and then pressing into moulds while hot.

The character of such fuels will, of course, depend on the nature of the materials of which they are made. By a proper choice of the ingredients or by a suitable enriching with hydrocarbons in the form of pitch or crude oil, a fuel of most excellent quality may be made. The pressure to which the blocks are subjected is so great that the materials become closely compacted together and hold their form with no more breakage through handling than with a good quality of semi-bituminous or even semi-anthracite coal. The best grades of artificial fuel ring when struck and absorb little or no water, thus showing a compact and firm structure throughout. They ignite readily and burn freely without an excessive formation of smoke, holding their shape without crumbling too rapidly on the grate. In evaporative power the best briquettes are the equivalent of good coal, from which indeed they differ chemically in no essential character.

The weight per cubic foot and the number of cubic feet per ton when stowed loosely are about the same as for good semi-bituminous coal of like quality. If packed regularly the waste space is much decreased and the cubic feet per ton will be reduced to from 25 to 35.

Sec. 13. LIQUID FUEL.

[1]. Composition.

The only liquid fuel of importance to the engineer is either crude petroleum oil, or the residue left after removing from the crude oil by distillation the lighter constituents, consisting of naphtha, illuminating oil, etc. Crude petroleum oil is a liquid of brownish tint varying from light straw to almost black. It consists of a very complex mixture of many hydrocarbons. Some of these vaporize very easily and escape rapidly, even at ordinary temperatures. Such constitute the naphthas and gasoline. Next in order come the constituents which form common illuminating oil or kerosene.

Then still heavier and denser come the lubricating oils of various kinds. After the removal of these there still remains a residue capable of yielding paraffine and vaseline, and last of all a certain amount of gas tar and coke.

When the process of refining or distillation is arrested after the removal of the naphthas and illuminating oils, and perhaps some of the lubricating oil, the residue consists of a rather thick viscid liquid, not readily ignited as compared with the crude oil, but under proper conditions burning readily and with great heating power. Such residue in the Russian oil wells on the Caspian is called *astatki*. It constitutes more than one-half of the crude oil. On the other hand, with American oils the similar residue is much less in amount, rarely rising to one-third of the crude oil. Furthermore, the processes of refining are so much superior in the United States that of final residue after the removal of all marketable products, there is almost nothing left. With the best modern methods of treatment, therefore, the use of the residue after partial refinement does not mean the utilization of a waste or by-product, but the use of a substance having a definite market value for other and long established uses. For the direct use of crude oil the same is true in still higher degree, so that under modern conditions the use of liquid fuel means simply a competition with the various other industries involving the use of the various products of crude petroleum oil.

[2]. Combustion.

For the combustion of crude oil or liquid refuse, two methods are in use. In the earlier and better known the chief essential is that the liquid must be "pulverized" or "atomized"—that is, broken up into a very fine spray and thus brought into intimate contact with the oxygen of the air. This is accomplished by special devices fitted to the furnace and called "pulverizers" or "atomizers." They are of two chief varieties according to the means used—either compressed air or steam. In each the oil is fed by pump or allowed to flow by gravity to the nozzle of the device. Here it is caught by a jet of air or steam, as the case may be, issuing near or through it, and by this means is thoroughly broken into a fine spray and blown into the furnace in this condition. Once the fire started, the spray is ignited as it issues from the nozzle so that the result is a long, fiercely burning jet of flame directed into the furnace. In order to produce the conditions best for complete combustion, it is usually found advantageous to have fire brick so disposed as to take the direct action of the flame. These bricks become heated to a high temperature and by their radiating action help to produce and maintain a temperature suitable for the complete combustion of all gaseous products formed from the liquid spray. In a later and on the whole more efficient method the oil is first vaporized and then introduced into the furnace as a vapor and there burned as such.

Under proper conditions, and especially by the latter method, oil may thus be burned with little or no formation of smoke or soot. The absence of all soot is especially favorable to the maintenance of a high efficiency of operation. There is furthermore no clinker or ash, no cleaning of fires, no opening of furnace doors either for firing or cleaning, and no handling of ashes.

[3]. Danger of Explosion.

The danger of explosion or of the formation of an explosive mixture by the slow distillation of the lighter hydrocarbons is considerable with crude oil unless due attention is given to the airing and ventilation of the spaces where such gases can collect. So long as the spaces containing oil are full there is no danger of any such trouble, but when they are partially empty the gases may collect in the vacant parts, forming with the air an explosive mixture which needs only a spark or other source of fire to explode with violence. Crude oil residue does not contain these lighter substances and is, therefore, safe from danger of this character, though in all cases a due attention to the matter of ventilation may be recommended. It may also be noted that the more dangerous parts of the oil evaporate with readiness under ordinary temperatures, and that crude oil exposed to the open air rapidly loses these constituents and becomes thereby the safer for use. This operation corresponds to the *weathering* of coal, and entails, of course, some loss, but a loss the more permissible as it makes the fuel the safer to use.

[4]. Evaporative Power.

Liquid fuel has a much higher evaporative power pound for pound than coal. According to chemical analysis the combustion of one pound of liquid fuel should liberate from 20,000 to 22,000 heat units, or about one and one-half times as much as good coal. This, combined with better efficiency of operation than with coal, has given experimental results showing an evaporative power twice that of coal or even higher. A ratio of 1.7 : 1 or 1.6 : 1, however, is more commonly considered as representing the average relation under ordinary conditions, though in some cases the advantage has been still less marked.

[5]. Stowage and Handling.

The best oil residue is of about the same density as sea water or slightly heavier. It will thus run from 34 to 35 cu. ft. per ton. While, therefore, its specific gravity is less than that of a lump of coal, it stows much better, so that a given space is capable of holding from 15 to 20 per cent more fuel in the shape of oil than in the shape of coal. Combining this advantage with that in evaporative power per pound, it follows that the final result is to very nearly double the capacity for steam generation per cubic foot of space occupied by fuel. As a further point it may be mentioned that oil or liquid fuel may be stowed in many places on board ship not available for coal or for cargo in general. Such are ballast tanks, double bottoms, etc. Again, the ease with which oil may be handled, flowing as it does by gravity and stowing itself, is a further point in its favor. With proper facilities a ship may be provided with oil much more rapidly than with coal. It should be added, however, that oil refuse at a low temperature may become quite stiff, flowing only sluggishly, especially in small pipes. This condition may require the provision of special means for heating the oil so as to insure the necessary degree of fluidity. Oil refuse is, therefore, not a fuel suitable for arctic exploration.

A further advantage for liquid fuel lies in the great reduction in the fireroom force which is possible with its use. The handling and firing being practically automatic, only a few men are required to look after the oil tanks and supply pipes in a general way, and one fireman can give to a large number of furnaces the slight general attention which they require. The chief work in the fireroom is, therefore, reduced to water tending, which remains as with coal.

[6]. Use of Oil and Coal Combined.

In some cases the use of oil in conjunction with coal has given promise of good results, especially on war ships, where it may be of the utmost importance to be able to very rapidly increase the power developed. In such cases the coal would be used alone under ordinary conditions, and oil added when the increase of power is desired. Experiments in the Italian Navy show that for the most complete combustion and best efficiency the proportion of oil to coal should be about one of oil to five of coal. In the same manner as above described the oil is pulverized and blown as a spray into the furnaces, where it burns with the gases given off by the coal. In this way the oil furnishes a powerful resource for suddenly forcing the fires and increasing the I. H. P. developed, while the amount of oil carried or consumed is quite small compared with the supply necessary if oil were the only fuel used.

[7]. Cost.

The great drawback regarding oil fuel, and one that is apparently too serious to be overcome, is that relating to its price. At ordinary figures a pound of steam would now cost at least as much if generated by means of oil as by means of coal, and if the use of oil were undertaken by several large steamship companies or other large consumers, its price would rise to a point impossible at present to foresee. The use of liquid fuel is quite within the reach of present engineering means, and may be considered as a mechanical success. Due, however, to the limited supply, and to the uncertainties regarding its price, its use will probably, under present conditions, be quite limited as a fuel for the generation of steam.

CHAPTER III.

BOILERS.

Sec. 14. TYPES OF BOILERS.

In the general sense, any receptacle in which steam is generated by the application of heat is a boiler. A boiler must, therefore, contain three fundamental features: a place for the fire, a place for the water, and a division or partition between them. The great variety of boilers arises from the different

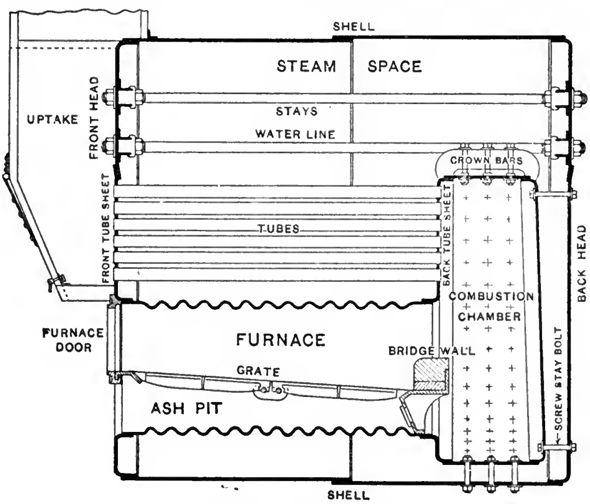


Fig. 9. Scotch Boiler.

forms which these features take, and the different manner in which they are arranged. The keynote of the development of steam boilers from the earliest forms is contained in the word *sub-division;* sub-division of the hot gases and of the water so that no particle of either shall be very far from the partition or *heating surface*, as it is called. If in addition to this sub-division provision is made for a definite flow of the hot gas along one side of the heating surface and of the water along the other in the opposite direction, the conditions for the most efficient transfer of the heat of the gas through the surface into the water will be fulfilled. In modern boilers the principle of sub-division has been carried to a high degree of development, but the conditions for proper circulation are but imperfectly fulfilled. The

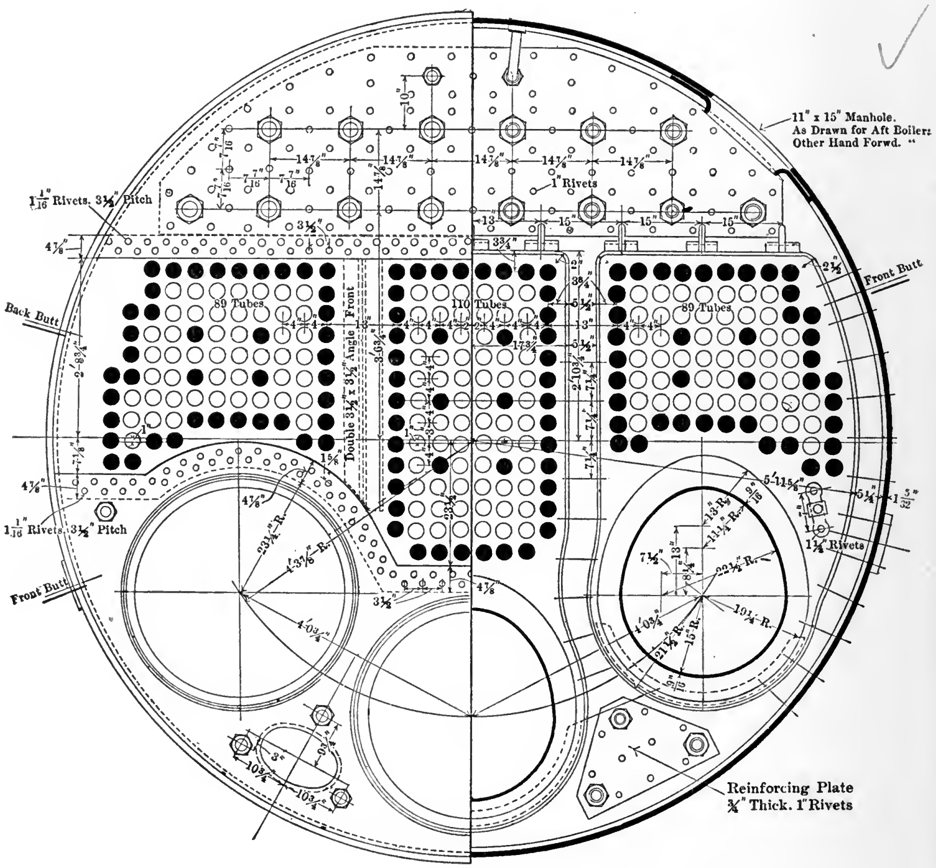


Fig. 10. Scotch Boiler, End View.

sub-division is obtained by the use of a large number of tubes or tubular elements surrounded by a shell or casing. The chief classification of boilers is made according to the relation of the water and hot gas to these tubular elements. If the gas is led through the inside and the water is on the outside, the arrangement is known as a *fire-tube* boiler. If, on the contrary, the gas is on the outside and the water circulates through the inside, the arrangement constitutes a *water-tube* boiler.

Fire-tube boilers may be divided into the *Return tubular* or *Scotch boiler*, the *Direct tubular* or *gunboat boiler*, the *Locomotive boiler*, the *Flue and return tubular* or *leg boiler*, and the *Flue boiler*

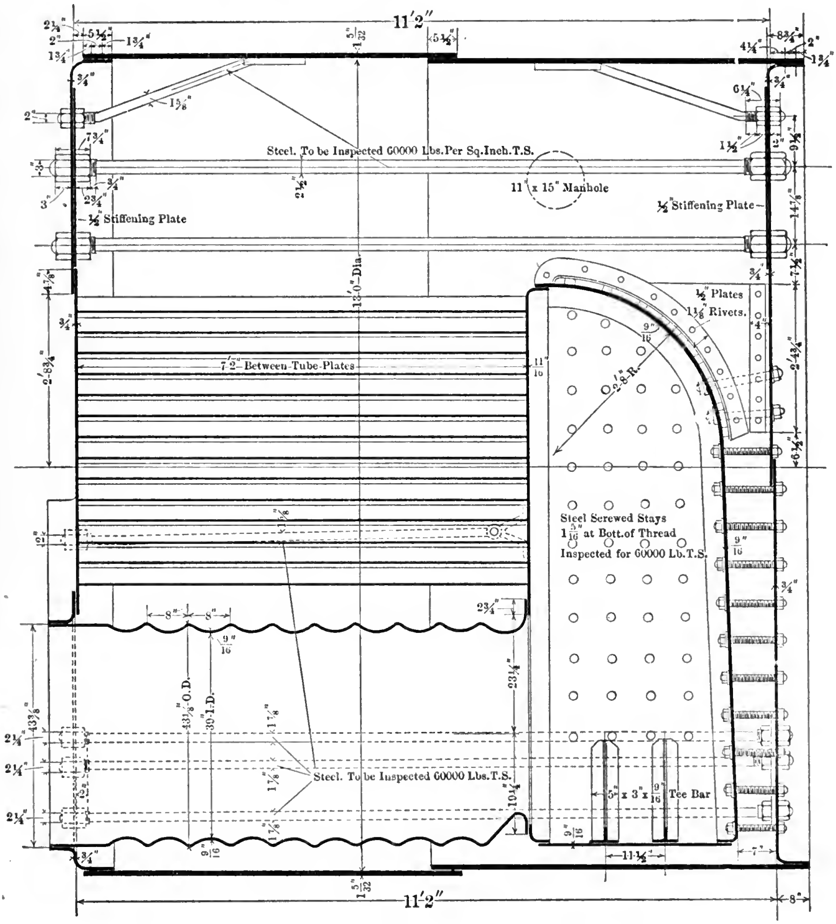


Fig. 11. Scotch Boiler, Longitudinal Section.

as used on western river steamers. These are illustrated in Figs. 9-14.

Water-tube boilers are found in great variety, depending on the details of arrangement of the tubes, and drums or headers of which they are composed. A few representative types are shown in Figs. 15-26. We will first give brief descriptions of the important features of these boilers, and then take up at a later point the subjects of their design and construction.

[1] The Scotch Boiler.

In present practice, for marine purposes, the Scotch boiler is used more than any other one type, and, in fact, more than all other types combined. This boiler, as illustrated in Figs. 9, 10, 11 consists essentially of a cylindrical *shell* containing one or more cylindrical *furnaces*, usually corrugated circumferentially for strength, opening into *combustion chambers* at the back end, from which a large number of small *tubes* lead again to the front end or *head* of the boiler. The *grates* are placed at about the center of the height of the furnace, and the fire and hot gases occupy the upper part of the furnaces, the combustion chambers, and the inside of the tubes, while the water and steam fill all the remaining parts of the shell, the water level being usually some 6 in. to 8 in. above the highest part of the tubes or combustion chambers. The hot gases pass from the fire on the grate-bars into the combustion chamber, thence forward through the tubes and out through the *uptake* or *front-connection* to the *smoke-stack* or *funnel*. Several varieties of this boiler are in common use. Thus the number of furnaces may be one, two, three, or four. They may be fitted with separate combustion chambers, or there may be one combustion chamber for all furnaces, or, as is common with four furnaces, there may be two combustion chambers—one for the two furnaces on either side. Again, the boilers may be *single-end* or *double-end*. Fig. 11 is an example of the first. A double-end boiler consists of two sets of furnaces opening from either end of a shell of double length. It is evidently equivalent to a pair of single-end boilers placed back to back with the back heads removed and the shells joined. Such boilers may also have either separate combustion chambers for each end, or a common combustion chamber for both ends. The former arrangement is to be preferred, and becomes necessary where forced draft is used.

[2] Direct Tubular Boiler, Gunboat Type.

This boiler is rarely used except in war ship practice, where with low head room it has been occasionally employed. It consists of a shell with furnaces and combustion chamber somewhat as in the Scotch boiler, but the tubes, instead of returning to the front, lead on to the farther head. To this head is fitted a smoke-box or uptake leading to the funnel. In such cases the boiler for the same power is of smaller diameter and greater length than the Scotch type, and it is readily seen that the whole arrangement is simply a mode of exchanging diameter for length.

[3] Direct Tubular Boiler, Locomotive Type.

The locomotive type of marine boiler as illustrated in Fig. 12 consists of a cylindrical shell extended to the front and modi-

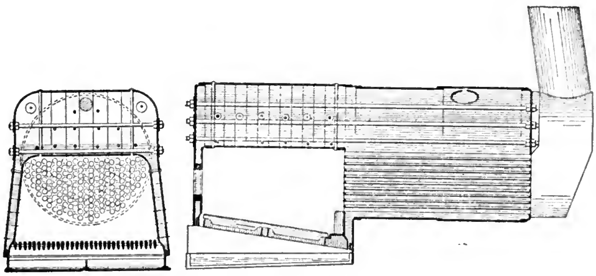


Fig. 12. Locomotive Type Boiler.

fied in form with flat sides and bottom, and flat or rounded top. The furnace is of rectangular cross-section, and is surrounded by the shell at the front, leaving on the sides a narrow space known as the *water-leg* and sometimes a like space underneath known as the *water bottom*. The gases take the same general course as in the gunboat type, the chief difference in the two being in the form of the furnaces and in the absence of the combustion chamber in the locomotive type.

[4] The Flue and Return Tubular or Leg Boiler.

In this boiler, as illustrated in Fig. 13, the hot gases pass from the furnace through large tubes or *flues*, as they are termed, to a combustion chamber at the farther end. They then return to the front through small tubes, and are led by an appropriate uptake to the funnel. The furnace is of rectangular cross-section, and the front end of the boiler is modified on the sides and bottom to correspond to this form, as in the locomotive type. Water legs are also formed in the same way on the sides of the furnace, and from this feature the boiler receives its common name. This form of the front end of the boiler with flat sides and rounded top is sometimes known as a *wagon-top*. Very commonly, as shown in Fig. 13, an attachment to the shell, known as a *steam chimney*, surrounds the lower part of the funnel, the office of which is to subject the steam to the drying and superheating effects of the gases on their way

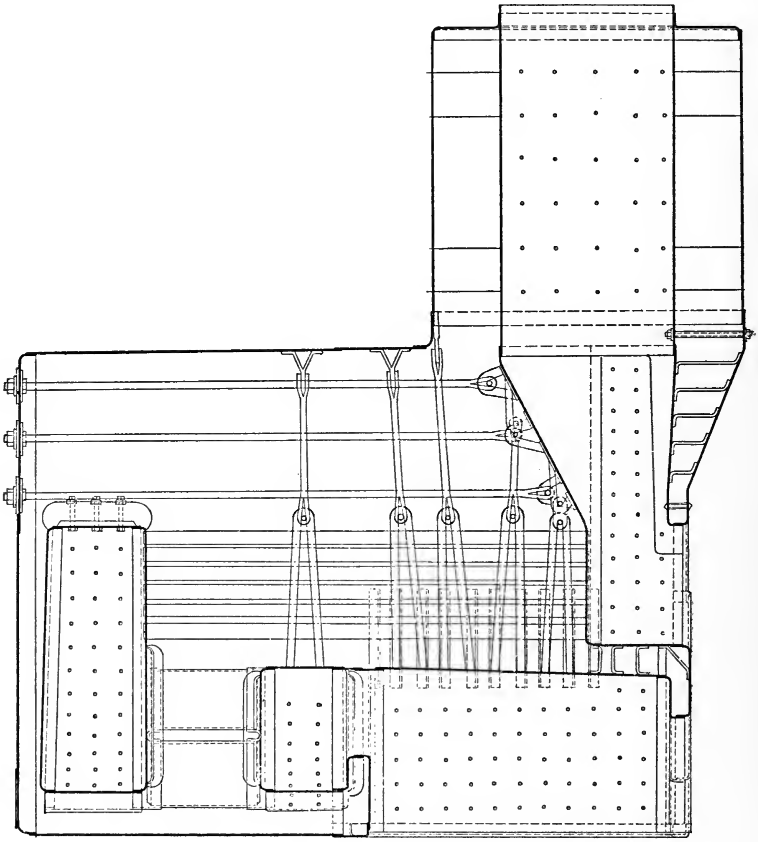


Fig. 13. Leg Boiler.

through the funnel. Boilers of this type have been used to a considerable extent on tug and river boats.

[5] The Flue Boiler.

In Western River practice use is quite commonly made of the return flue boiler as illustrated in Fig. 14. This boiler is externally fired. The flames and hot gases pass back along the outside of the boiler to a back connection and then enter the flues and return through them to the front, and thence to the uptake and funnel. Boilers of the locomotive type, or tubular fire-box boilers as they are often called, are also used to a considerable extent in western river practice.

[6] Water Tube Boilers.

Turning now to this type, a brief description will be given of the leading features, which may be combined in the greatest possible variety, thus giving the vast number of forms of such boilers on the market at the present time. To aid in the descrip-

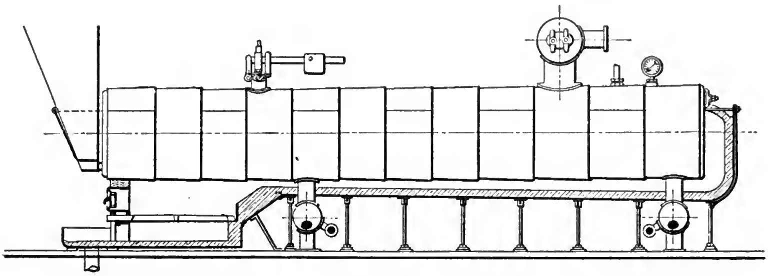


Fig. 14. Return Flue Boiler.

tion a few typical forms of such boilers are shown in Figs. 15-26.

Most boilers of this type have one or more cylindrical drums or chambers on top and one or more similar drums below, the two sets of drums being connected by sets of tubes. The feed usually enters first the upper drum, frequently passing on its way through a coil heater in the base of the stack or top of the boiler. It then flows down certain of the tubes to the lower drums. If these tubes are of extra large size and specially intended for down flow, the boiler is said to have special *down flow* or *down cast tubes* or pipes, as shown in Figs. 15, 16, 17, 18, 20. In some cases such tubes are omitted, and the feed must descend through part of the small inner tubes. In any case, after finding its way to the lower drum it enters the *up flow* or steam forming tubes, which are surrounded by the hot gases coming from the furnace below them. During the passage of the water upward it is partly converted into steam, and the mixture issues from the upper end of the tubes into the upper drum. There the steam is separated and led to the engine, while the water joins that already in this drum, and thus begins another round. In some cases the upper ends of the steam forming or delivery tubes are below the level of the water in the upper drum, and they are then said to be *drowned* or *wet*. In other cases they are above the water level and are said to be *dry*. In still other forms they enter at about the middle of the drum or about the water level, and may be *wet* or *dry* as the level varies. See the various cuts for examples. Water tube boilers are often divided into two general classes: *large tube* and

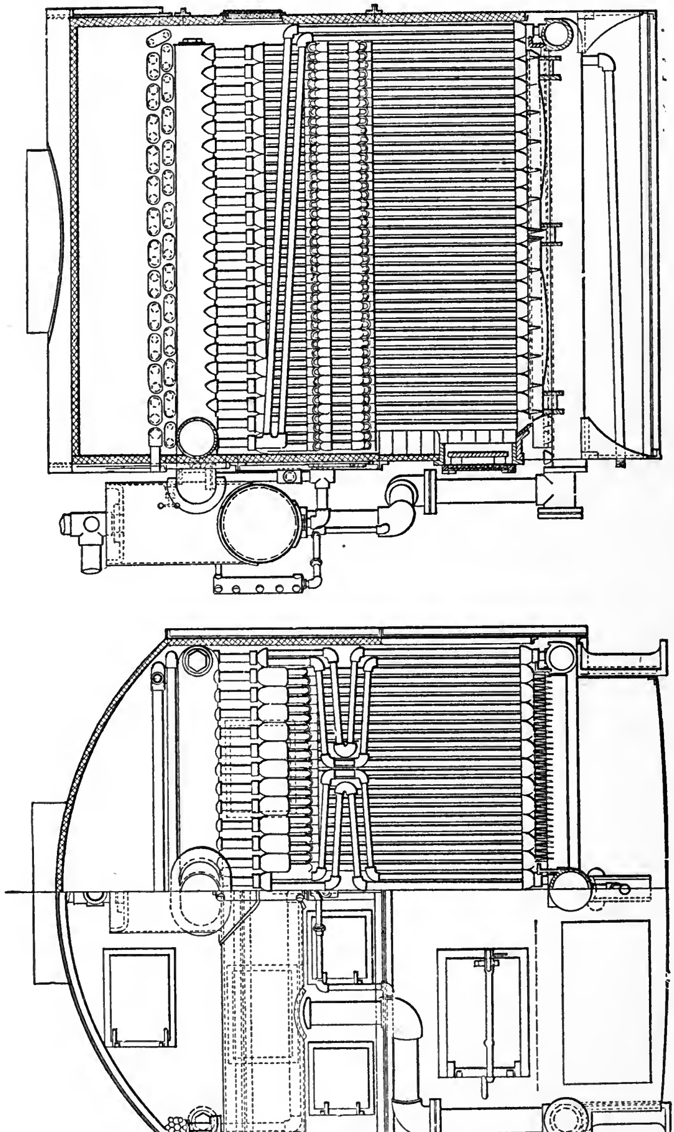


Fig. 15. Almy Boiler, Front and Side Views.

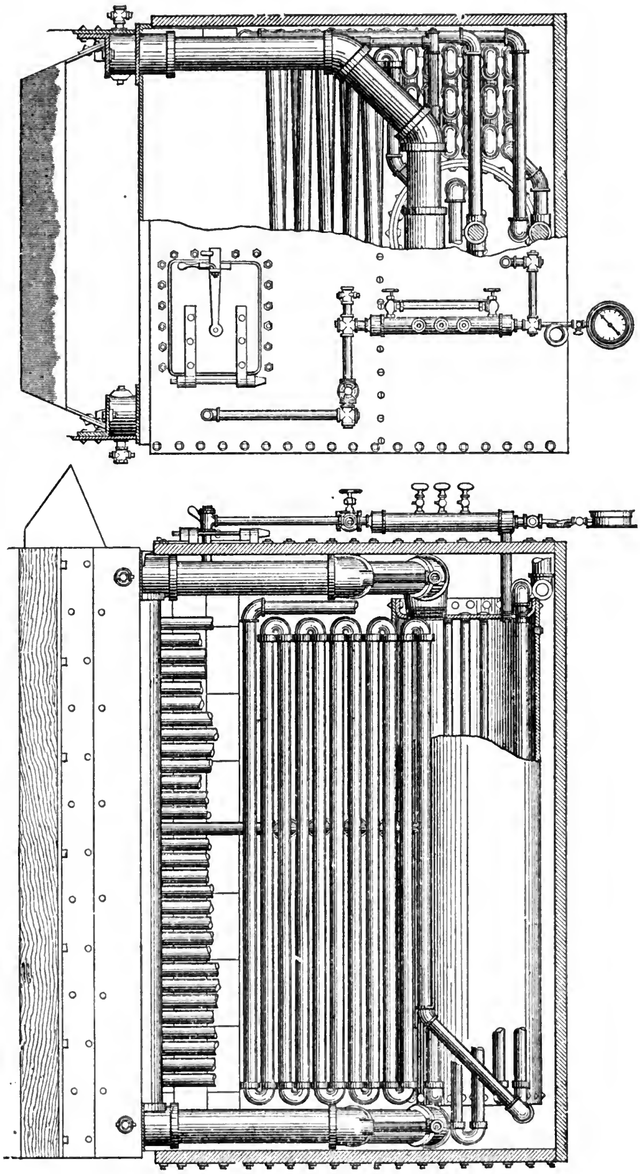


Fig. 16. Roberts Boiler, Front and Side Views.

small tube boilers. In the former they are usually 3 or 4 or even 5 in. dia., while in the latter they are usually from $1\frac{1}{4}$ to $1\frac{1}{2}$ or 2 in. dia.

Again, the tubular elements may be made up in a great variety of forms. In some they are straight, in others curved, as shown in the various figures. In small tube boilers they are very commonly curved or bent, while in the large tube types they are straight. Also in some types the elements are continuous between drums or headers as in Figs. 17, 18, 19, 20, while in others, as in Figs. 15, 16, 26, they are made up of lengths or of different parts with screwed joints, elbows, returns, junction boxes, etc. In some they are expanded into the shells of the drums; in others screwed. In some all joints are carefully protected from the direct action of the flame; in others screwed joints are freely exposed to the flame. In some the general direction of the tubes is nearly horizontal; in others nearly vertical, and in others bent or curved in various forms. In some types, as illustrated in Figs. 24, 25, the lower drums are omitted, or consist merely of the lower portions of the tubes and *headers*, or members to which the tubes are connected. In all cases the grate lies below the tubes and frequently between the lower drums, as shown in the various figures, while the whole is surrounded by a casing intended to prevent, so far as possible, the loss of heat by radiation.

[7] Relative Advantages of Different Types of Boilers.

For large ships under ordinary conditions and where the extremes of lightness or of speed on a given displacement have not to be attained, the Scotch boiler seems at present to be considered as fulfilling most satisfactorily the all around requirements for a marine boiler, and in consequence it is found almost universally in the mercantile deep sea marine, as well as on the Great Lakes, and to a large extent on inland craft of all descriptions, except those of small size. It is also used to some extent in naval practice, though the use of water-tube boilers is at the present time extending quite rapidly into this field, where their special features become of marked value. The present is a time of change with regard to types of boiler. It is not too much to say that in most of the modern naval construction the water tube boiler is accepted as the standard type to the exclusion of the fire-tube boiler. The water-tube boiler is also making large advances in the mercantile field, and not a

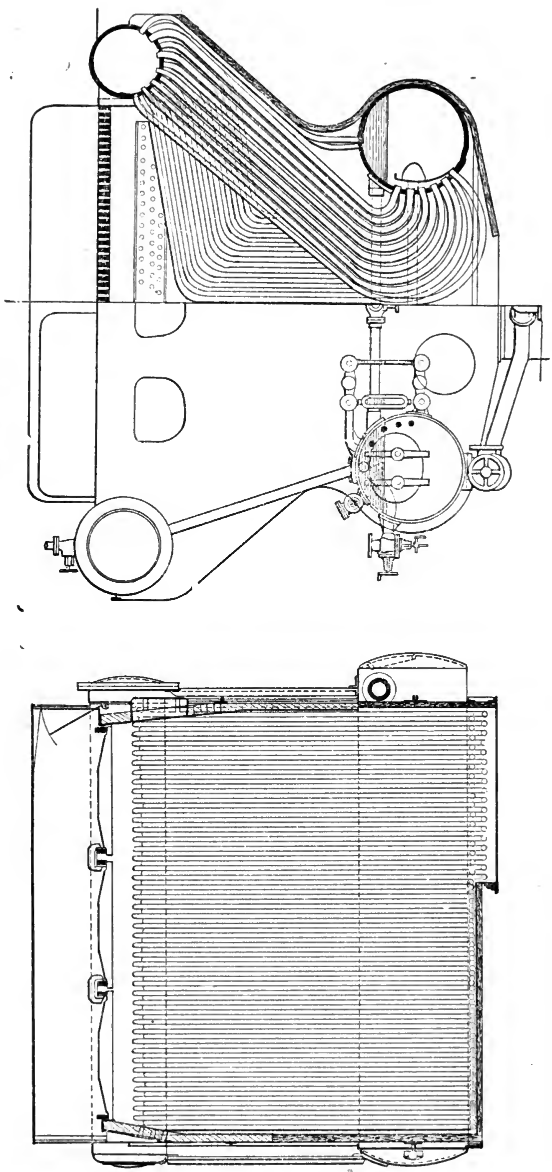


Fig. 17. Mosher Boiler, Front and Side Views.

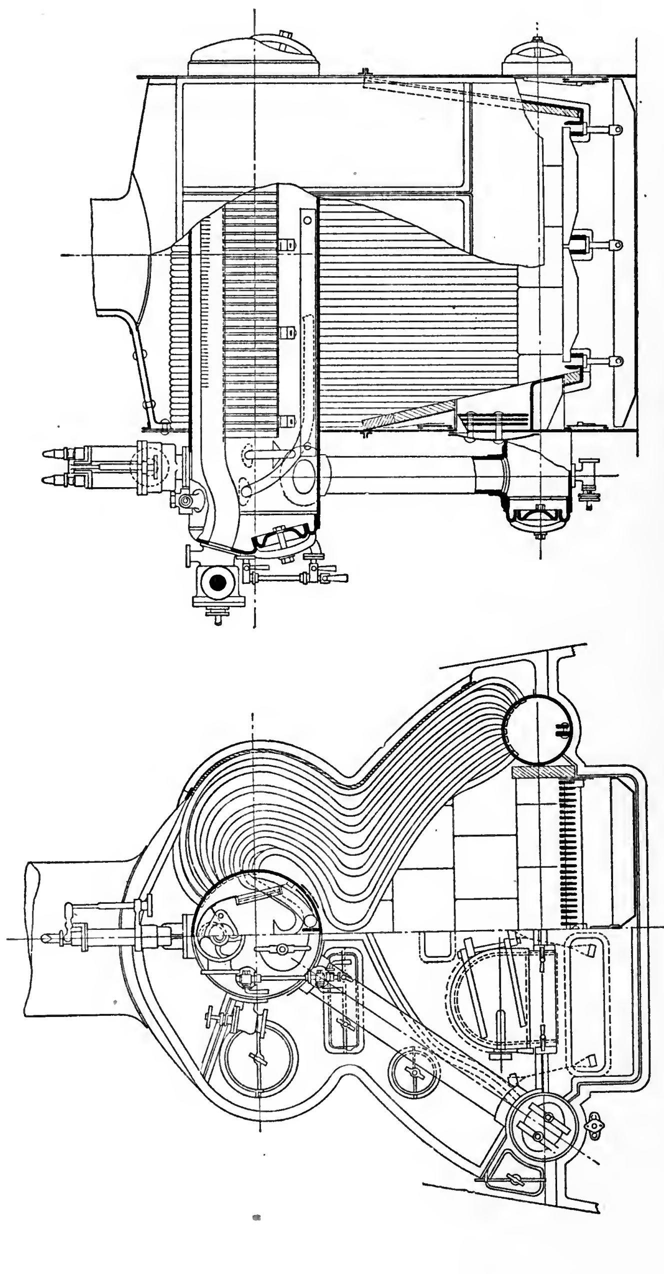


Fig. 18. Thornycroft Boiler, Front and Side Views.

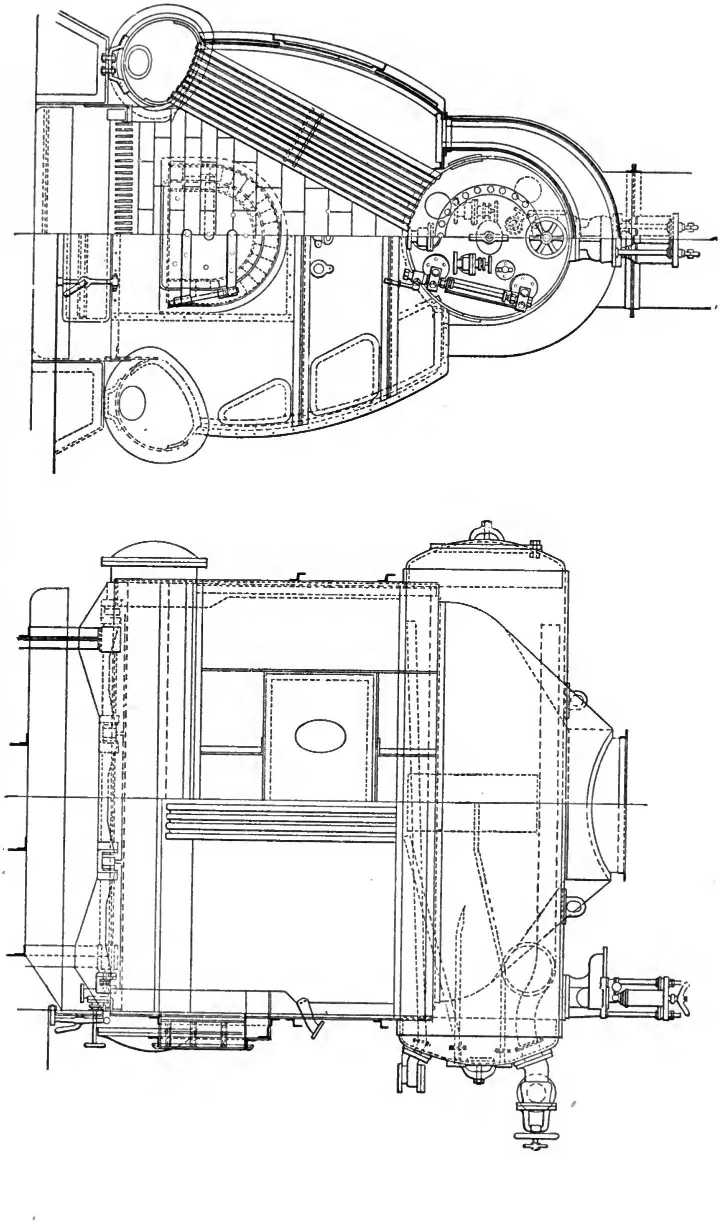


Fig. 19. Yarrow Boiler, Front and Side Views.

few modern ships of the mercantile marine are now equipped with this type of boiler.

For tugboats, river steamers, and a variety of small craft, the various types of direct fire-tube and flue boilers have been much used. These boilers are more readily adapted to a variety of demands regarding size, form and arrangement, and in small sizes are perhaps more cheaply built than Scotch boilers. In many cases, however, the preference for boilers of this type has doubtless depended on local and special conditions quite independently of their relative value from the engineering standpoint.

For fast yachts, launches, all craft of the torpedo-boat type, and in fact in all cases where the highest speed is to be attained on the least weight, the water-tube boiler has become a necessity, and in one or another of its many forms is universally employed.

The weight of Scotch boilers without water, per square foot of heating surface, is usually from about 25 to 30 lb.; of water-tube boilers of the lighter types from 12 to 20 lb. The weight of the contained water per square foot of heating surface is usually from 12 to 15 lb. for Scotch boilers, and from, say 1.5 to 3 lb. for water-tube boilers. It results that Scotch boilers with water will weigh from, say, 35 to 50 lb. per square foot of heating surface, while water-tube boilers will similarly weigh from 13.5 to 23 lb. These figures are not to be considered as giving absolute limits, but simply as representative values for average types. It should be noted, however, that a square foot of heating surface in a Scotch boiler seems to be somewhat more efficient than in a water-tube boiler. It is difficult to estimate the difference numerically, but other conditions being equal, it would probably be safe to give to the water-tube boiler additional heating surface to the extent of from ten to twenty per cent. On the other hand, it must be remembered that water-tube boilers can stand forcing to a much higher degree than fire-tube boilers. With the latter supplying steam to triple expansion engines the ratio of heating surface to I. H. P. can hardly be reduced below 2, while with the former this ratio has been reduced in many cases to less than one and one-half, and, as reported in certain extreme cases, to between one-half and one. Water-tube boilers have the further advantage that they are more readily constructed for the higher and higher steam

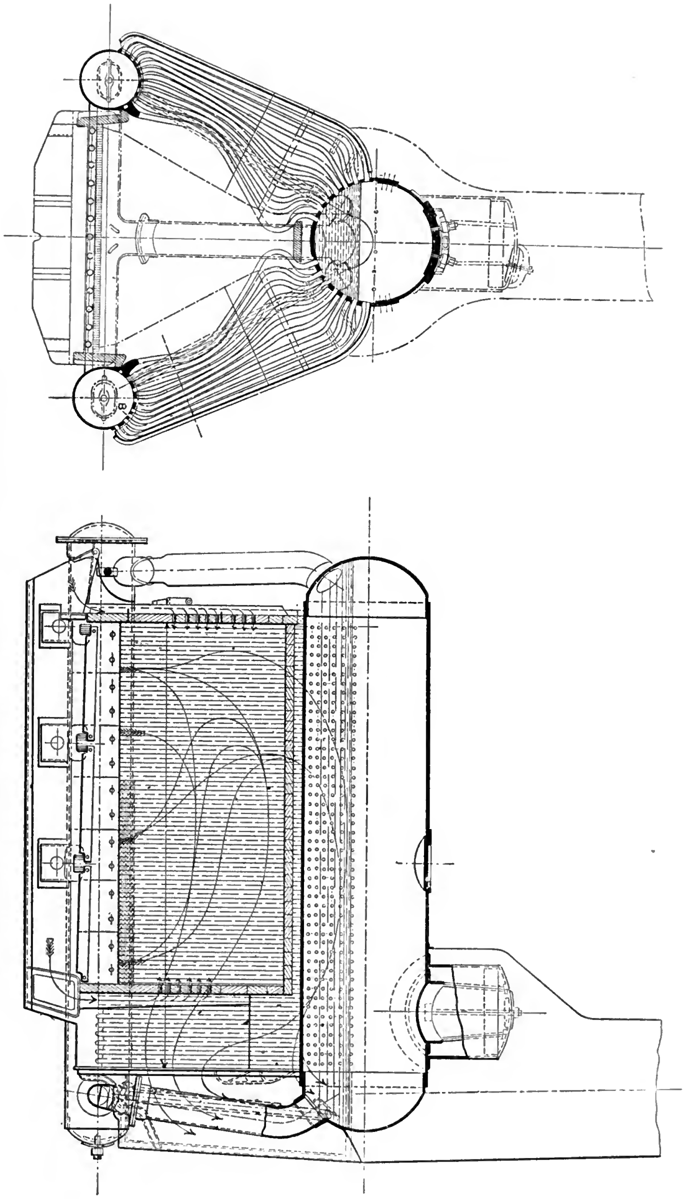


Fig. 20. Normand Boiler, Front and Side Views.

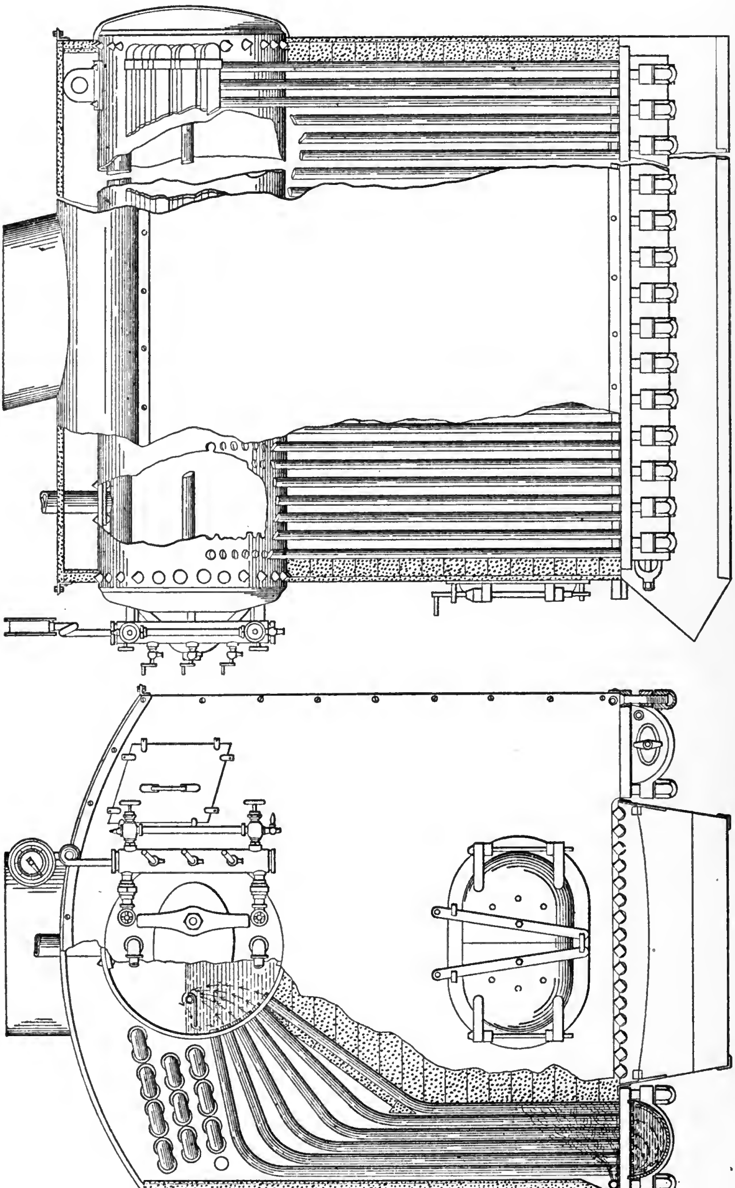


Fig. 21. Seabury Boiler, Front and Side Views.

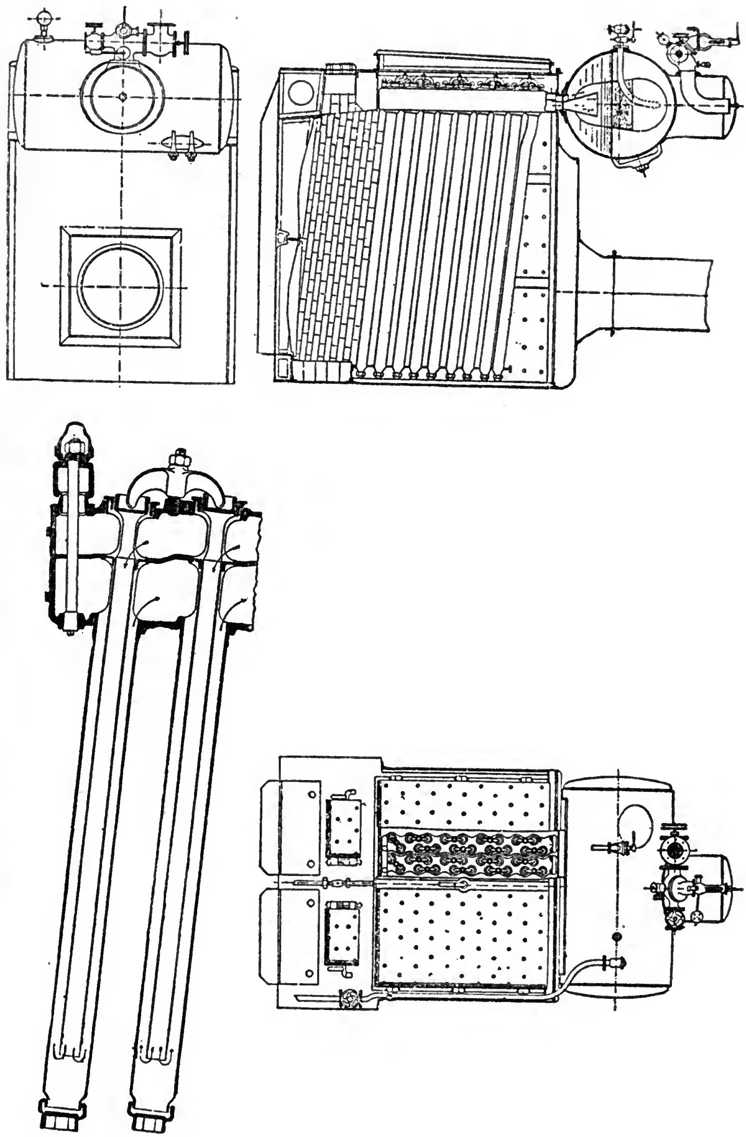


Fig. 22. Niclausse Boiler, Front and Side Views, with Detail of Tubes.

pressure which modern practice is continually demanding. With water-tube boilers, due to the construction and to the smaller amount of contained water, there is also less danger from disastrous explosion. With water-tube boilers steam may be raised much more quickly than with fire-tube boilers; from one-quarter to one-half hour is sufficient with the former, while from three to four hours should be taken with the latter.

Water-tube boilers are also much more portable than fire-tube. In many forms spare parts or even the whole boiler may be shipped in elements or sections across country by rail or to

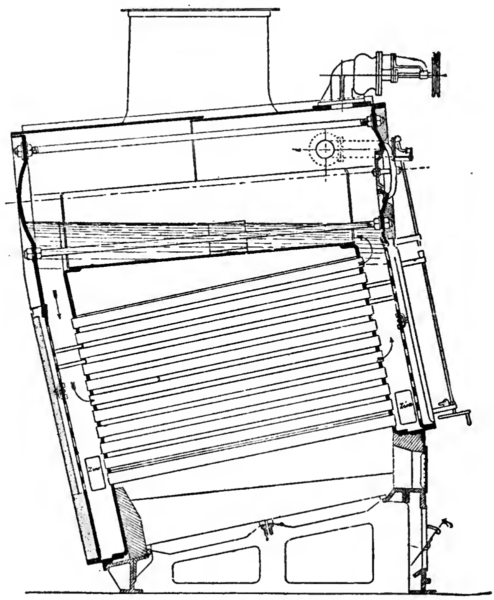


Fig. 23. Lagrafel and D'Allest Boiler.

foreign ports by ship transport, put on board the steamer for which they are intended, and erected in place without difficulty.

On the other hand, the water-tube boiler imperatively requires fresh water feed. Under modern conditions this should be provided no matter what the type of boiler in use, but if in emergency salt water must be used, the fire-tube boiler will receive the lesser injury. Again, from the small amount of water contained as a stock upon which to draw, the water-tube boiler requires a more uniform feed than the fire-tube boiler, and is generally more sensitive to variations in the conditions under which it works. Again, the rupture of a tube is a more serious

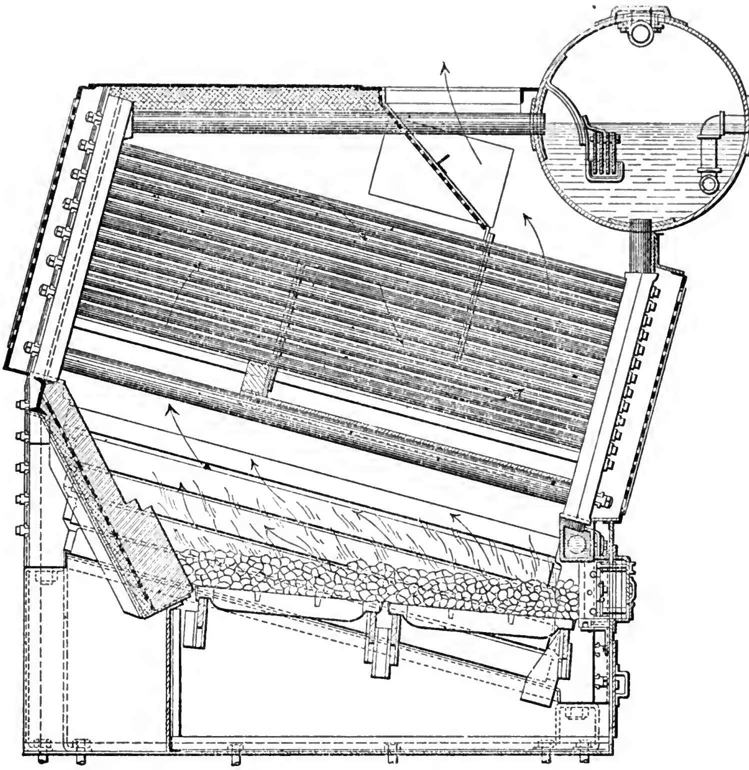


Fig. 24. Babcock and Wilcox Boiler.

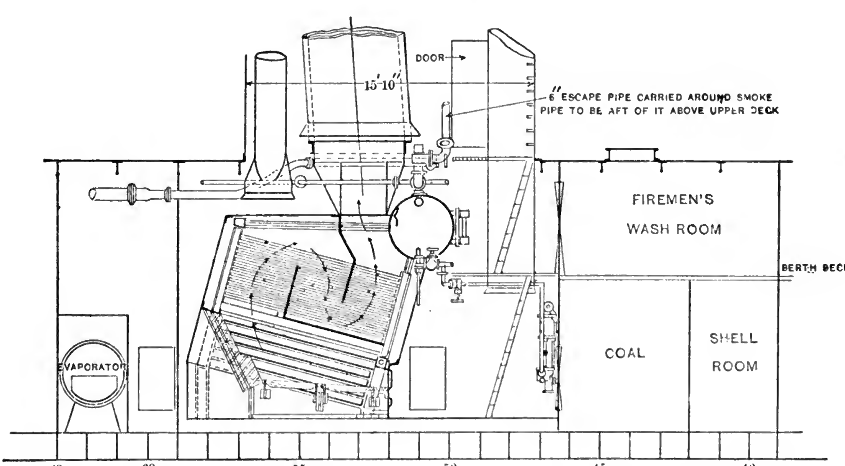


Fig. 25. Arrangement of Fireroom with Babcock and Wilcox Boilers.

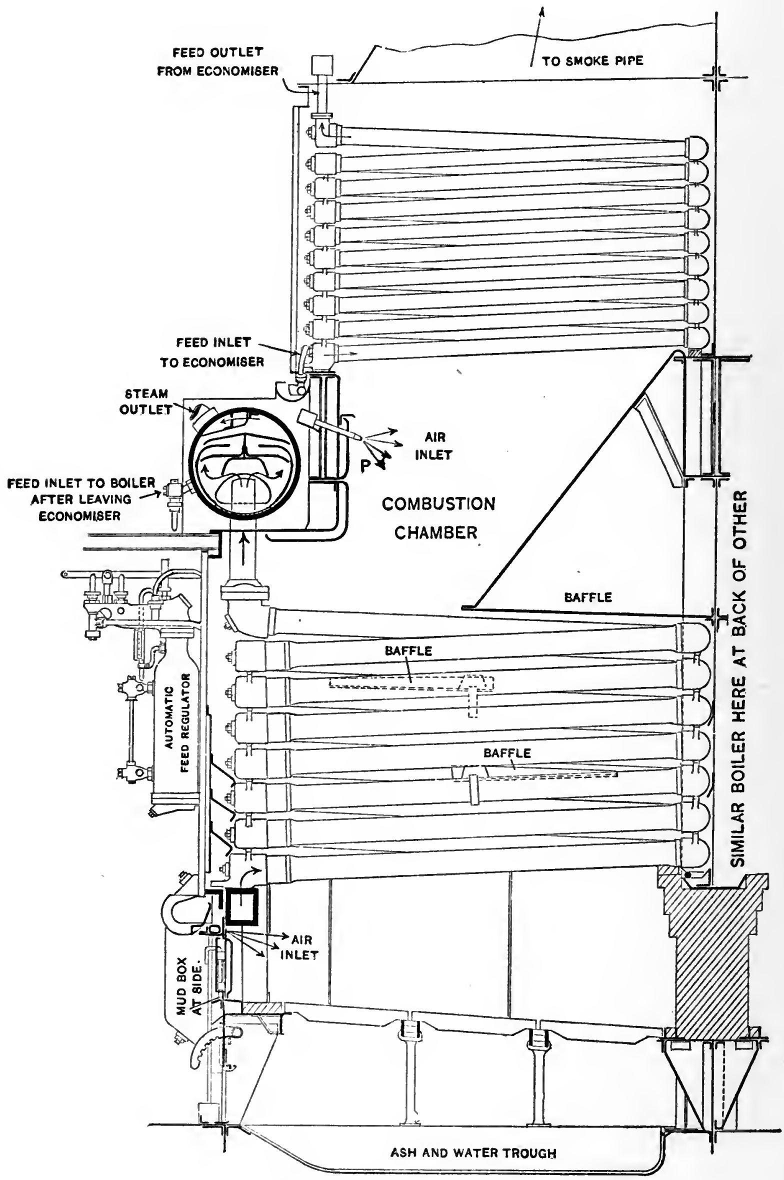


Fig. 26. Belleville Boiler.

matter in the water-tube than in the fire-tube boiler. In the latter it may be plugged without disturbing the water and steam in the boiler, and with only a momentary interruption to its operation. In the former it is usually necessary to disconnect the boiler, draw the fires, blow down the water, and plug or insert a new tube. Water-tube boilers are also not readily made in large sizes or units. Scotch boilers may be made in 2,000 horse power units or even larger, while half of this or less is about the maximum for the water-tube boiler. An outfit of the latter for large power requires, therefore, a large number of boilers with a corresponding increase in the fittings and attachments. On the other hand, the temporary removal of one boiler for repair is of less importance, as the size is decreased and number increased.

To summarize the general comparison between water-tube and fire-tube boilers, the former have relative advantages in the following chief points: Weight, ability to stand forcing, suitability for high pressures, greater safety from disastrous explosion, and quickness of raising steam. On the other hand they have relative disadvantages in these points: A more rigid restriction of the feed to fresh water, the necessity of greater regularity of feed, greater difficulty in dealing with leaky tubes, and general sensitiveness to variation in the conditions of use, to which may be added the present feeling of uncertainty as to their durability and efficiency under the conditions prevailing on deep water voyages.

Sec. 15. RIVETED JOINTS.

The various joints in a boiler are usually of the riveted form. The use of welded joints in various parts of boiler construction is increasing somewhat as greater skill is acquired in making them, but in ordinary practice the joints are riveted, and of various types, as follows:

Riveted joints are divided into *lap* joints and *butt* joints, according as the plates lap over each other (see Figs. 31-34), or butt together at the edges, and are covered by one or two butt straps (see Figs. 35-39). They are also divided according to the number of rows of rivets into *single*, *double* or *triple* riveted joints (see Figs. 31-33).

The rivets are usually *staggered* in arrangement, as shown in Figs. 32-39. Sometimes, though rarely, the *chain* arrangement, as shown in Fig. 27, is used. While chain riveting is as strong, or perhaps even slightly stronger, than staggered riveting, the latter gives a better disposition of rivets for making a steam or water-tight joint, and this fact leads to its more frequent use in boiler construction. In butt joints the arrangement of rivets is duplicated on each side of the joint, and the style of riveting is named according to the arrangement on one side. Thus, Fig. 35 shows a double-riveted and Fig. 36 a triple-riveted butt joint.

A riveted joint may fail: (1) In the plate by tearing out or across from hole to hole, see Figs, 28, 29; (2) In the rivet by shearing; (3) in the plate or rivet by a crushing of the material.

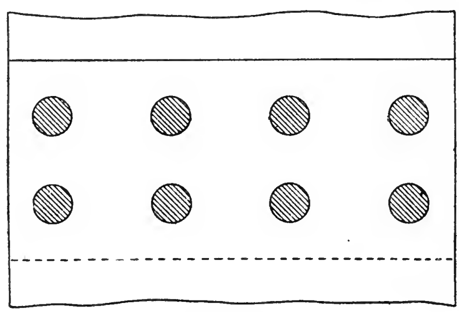


Fig. 27.

The failure of a joint by the tearing out of the plate in front of the rivet, as in Fig. 29, is safely guarded against by placing the row of rivets at a proper distance from the edge of the plate. This, by experience, is found to be about one diameter in the clear, or one and one-half diameters from edge of plate to center line of rivets. In lap joints and butt joints with one cover, as in Fig. 30, the rivets resist shearing at one section only. In butt joints with double covers, as in Figs. 35-38, the rivets resist shearing at two sections. The total shearing strength of a rivet in double shear is usually taken as somewhat less than twice the strength in single shear. The British Board of Trade rules give 1¾ as the ratio to be used.

With usual proportions the last mode of failure mentioned above is the least likely to occur, so that so long as the proper limits are not exceeded the resistance to crushing needs no especial examination. These limits will be given in detail at a later point.

The strength of a riveted joint is, of course, determined by whichever is the weaker of the two, the plate or the rivets. In a properly designed joint the strength of the plate and that of the rivets should be equal, so that there will be no more likelihood of failure in one way than the other. It may be remarked, however, that since corrosion usually affects the plate only, it is often considered good practice to give to the plate a slight ex-

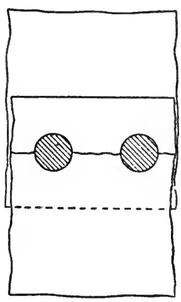


Fig. 28.

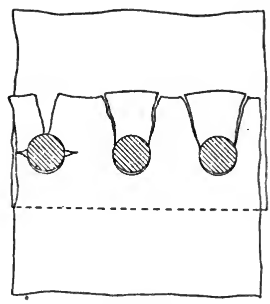


Fig. 29.

cess of strength, so that even after some wasting by corrosion the joint may still be in fair proportion as to the relative strength of plate and rivets. No exact directions can be given for this increase, as it is simply a matter of judgment.

The investigation of the strength of riveted joints by any simple theory is necessarily quite imperfect, because we do not know in just what way the stress is distributed through the remaining part of the plate, nor through the section of the rivet,

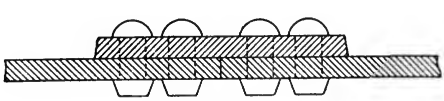


Fig. 30.

nor what allowance to make for the frictional grip of the joint. The proportions given by the following equations, however, are those which will give practically equal strength of plate and rivets, using the British Board of Trade rules. These rules represent standard and reliable practice, based on wide experience, and are substantially adopted by the United States inspection authorities. In thus considering a joint we take simply an element such as that between *AB* and *CD* in the following diagrams. It is clear in each case that the whole joint may be considered as made up of a series of such elements:

Let p denote the *pitch* of the rivets: that is, the distance from center to center. Where the rivets in one row are pitched twice as far apart as in another (see joints D, H, etc.), p denotes the larger of the two values.

" d denote the diameter of rivet.*

" $n = p \div d$ = number of rivet diameters in the pitch.

" t = thickness of plate.

" $a = t \div d$.

" T = tensile strength of plate per square inch of section.

" S = shearing strength of rivet per square inch of section.

The ratio of S to T is taken as 23 : 28 or $S = .821\ T$.

The *efficiency* of the joint is the ratio between the strength of the joint and the original strength of the plate. It will be seen by the formulæ given later that the efficiency of a joint is increased as d and p are made larger. There is, however, a practical limit to the increase in d, due to the difficulty of heading up very large rivets, and a limit to the increase in p, due to the necessity of guarding against leakage. If the general proportions between d and t, as indicated later in connection with the various joints, are observed, the result will be a pitch within safe limits, and a joint agreeing well with the best practice.

The largest permissible values of the pitch, according to the Board of Trade rules, are given by the following formula:

$$p = C\,t + 1\tfrac{5}{8}$$

where C is drawn from the following table:

Form of Joint as Shown Below.	C	Form of Joint as Shown Below	C
A	1.31	F..............	4.63
B..............	2.62	G..............	5.52
C	3.47	H..............	6.00
D..............	4.14	I..............	6.00
E	3.50		

In no case should the pitch exceed 10 inches.

We will now proceed with the equations and proportions for various forms of riveted joints.

* Strictly speaking the diameter of the rivet *hole* should be used, as it is about 1-16 inch larger than the rivet before heading up. In the Board of Trade Rules, however, the diameter of *rivet* is used. The difference in proportion of joint is quite small, and probably not of practical importance.

Joint A. *Lap Joint.* *Single Riveted.*

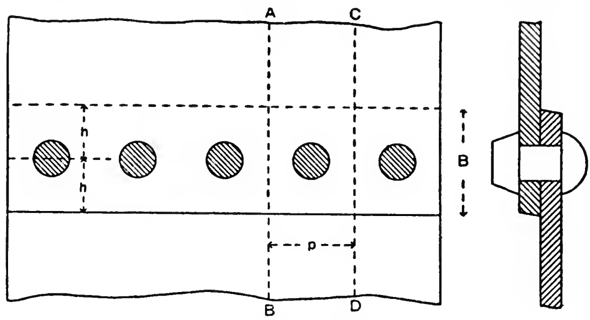


Fig. 21. Joint A.

$h = 1\tfrac{1}{2}\,d$

$B = 3\ \ d$

The element is $A\ B\ D\ C$, containing one rivet. We have in this case:

Strength of Plate $= t\,(p - d)\,T$

Strength of Rivet $= \frac{1}{4}\pi d^2 S$

For equal strength of plate and rivet,

$$\frac{p}{d} \text{ or } n = 1 + .645\,\frac{d}{t}$$

$$\text{Efficiency} = \frac{n-1}{n} = \frac{.645}{a + .645}$$

The ratio $d \div t$ may vary from 1.5 to 2.5, the lower values being more commonly employed with very thick plates on account of the difficulty of heading up excessively large rivets, and the necessity of a moderate pitch to insure against leakage. In order, furthermore, to guard against danger of rupture by crushing, the upper limit, 2.5, should not be exceeded.

The foregoing operations may be expressed also by the following:

Rule. (1) Select a diameter of rivet according to the thickness of the plate and the directions given.

(2) Multiply this diameter by .645 and divide by the thickness of plate.

(3) Add 1 to the result obtained in (2).

(4) Multiply the diameter of rivet by the result obtained in (3), and the result will be the pitch suited to the diameter chosen.

(5) Select the nearest working dimension, going usually above in order to give slight excess of strength to the plate.

(6) To find strength of plate in the joint, subtract the diameter of rivet from the pitch, multiply by the thickness and by the tensile strength per square inch of section.

(7) To find strength of rivet, find area of section, multiply by the same strength per square inch as in (6), and then by .821.

(8) To find original strength of plate multiply pitch by thickness of plate, and by the tensile strength per square inch, as in (6).

(9) To find the efficiency, divide the lower of the two results found in (6) and (7) by that found in (8).

Example. To lay out a single riveted lap joint for $\frac{1}{2}$ inch plates, using $\frac{7}{8}$ inch rivets.

$$\text{Then } \tfrac{7}{8} \times .645 \div \tfrac{1}{2} = \frac{7 \times .645 \times 2}{8} = 1.13$$

And $1.13 + 1.00 = 2.13$.
And $2.13 \times \frac{7}{8} = 1.86 =$ pitch.

Take the nearest eighth above and we have pitch $= 1\frac{7}{8}$ inch.

Then taking strength of plate at 60,000 lbs. per square inch we have:

Strength of Plate in Joint $= (1\frac{7}{8} - \frac{7}{8}) \times \frac{1}{2} \times 6{,}000 = 30{,}000$.
Area of $\frac{7}{8}$ inch rivet $= .60$ sq. in.
Strength of rivet $= .60 \times 60{,}000 \times .821 = 29{,}550$.
Original strength of plate $= 1\frac{7}{8} \times \frac{1}{2} \times 60{,}000 = 56{,}250$.
Efficiency $= 29{,}550 \div 56{,}250 = .525$.

Similarly, if we should take 15-16 inch rivets, we should find for equal strength in plate and rivet a pitch of 2.07 inches. If we take a pitch of $2\frac{1}{8}$ inches we shall find an efficiency of .533. If we take the 2.07 exact the efficiency will be .548.

Joint B. Lap Joint. Double Riveted.

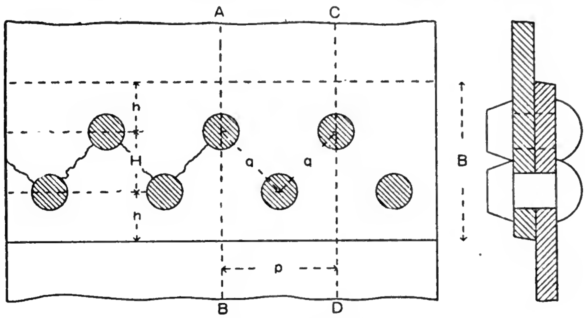


Fig. 32. Joint B.

$$h = 1\frac{1}{2}\,d$$

$$q \text{ not less than } (.6\,p + .4\,d)$$

$$\text{Hence } H \text{ not less than } \sqrt{(1.1\,p + .4\,d)\,(.1\,p + .4\,d)}$$

$$B = 3\,d + H$$

Where there are two or more rows of rivets they must be placed at a sufficient distance apart, so that there may be no danger of rupture along a zig-zag line, as indicated in the diagram. To this end the British Board of Trade rules give certain values for the distance q as given above for this case. This distance is known as the *diagonal pitch.* The rules are derived from experiment. The distances resulting may be considered as the smallest allowable. In practice the values of q are often made somewhat greater than would result from the rules. Having selected the distance q, the location of the second row of rivets is easily found from the first by constructing a triangle with base equal to p, and the two other sides each equal to q.

In this case the element $A\ B\ D\ C$ contains one whole rivet and two halves, or two rivets in all. We have then:

$$\textit{Strength of Plate} = t\,(p - d)\,T$$

$$\textit{Strength of Rivets} = \tfrac{1}{2}\pi d^2 S$$

For equal strength of plate and rivets:

$$\frac{p}{d} \text{ or } n = 1 + 1.29\,\frac{d}{t}$$

$$\text{Efficiency} = \frac{n - 1}{n} = \frac{1.29}{a + 1.29}$$

The values of $d \div t$ may vary through about the same range as in joint A, above, and for the same reasons as there explained. These operations may be expressed by a rule similar to that for joint A, the numbered sections referring to that rule as given above:

Rule:

(1) Same as for joint A.

(2) Use 1.29 instead of .645.

(3), (4), (5), (6) Same as for joint A.

(7) Take twice the strength of one rivet, found as for joint A.

(8), (9) Same as for joint A.

Example. To lay out a double riveted lap-joint for ½ inch plates, using ¾ inch rivets.

$$\text{Then } \tfrac{3}{4} \times 1.29 \div \tfrac{1}{2} = \frac{3 \times 1.29 \times 2}{4} = 1.935$$

And $1.935 + 1.00 = 2.935$.

And $2.935 \times \frac{3}{4} = 2.201 =$ pitch.

Taking the nearest eighth above we have $p = 2\frac{1}{4}$ inches.

Then taking strength of plate at 60,000, we have:

Strength of plate in joint $= (2\frac{1}{4} - \frac{3}{4}) \times \frac{1}{2} \times 60{,}000 =$ 45,000.

Area of ¾ inch rivet = .4418 sq. in.

Strength of rivets $= .4418 \times 2 \times 60{,}000 \times .821 = 43{,}520$.

Original strength of plate $= 2\frac{1}{4} \times \frac{1}{2} \times 60{,}000 = 67{,}500$.

Efficiency $= 43{,}520 \div 67{,}500 = .645$.

Similarly with ⅞ inch rivets, pitched 2⅞ inches, the strength of plate and rivets will be nearly equal, and the efficiency will rise to .687.

Joint C. *Lap Joint.* *Triple Riveted.*

$$h = 1\tfrac{1}{2}\, d$$

$$q \text{ not less than } (.6\, p + .4\, d)$$

$$\text{Hence } H \text{ not less than } \sqrt{(1.1\, p + .4\, d)\,(.1\, p + .4\, d)}$$

$$B = 3\, d + 2\, H$$

In this case the element $A\ B\ D\ C$ contains two whole rivets and two halves, or three rivets in all. We have then:

$$\textit{Strength of Plate} = t\,(p - d)\,T$$

$$\textit{Strength of Rivets} = \tfrac{3}{4}\pi d^2 S$$

For equal strength of plate and rivets:

$$\frac{p}{d} \text{ or } n = 1 + 1.935\frac{d}{t}$$

$$\text{Efficiency} = \frac{n-1}{n} = \frac{1.935}{a + 1.935}$$

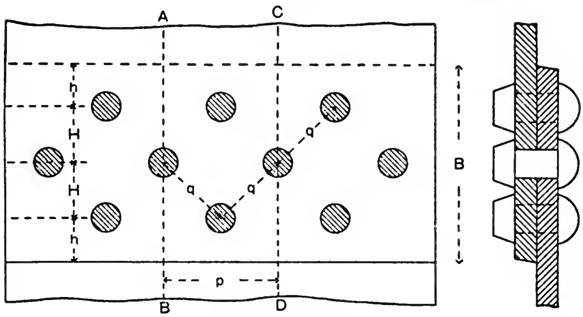


Fig. 33. Joint C.

The values of $d \div t$ may vary through about the same range as in joint A, above, and for the same reasons as there explained. These operations may be expressed by a rule similar to that for joint A, the numbered sections referring to that rule as given above.

Rule:

(1) Same as for joint A.

(2) Use 1.935 instead of .645.

(3), (4), (5), (6) Same as for joint A.

(7) Take three times the strength of one rivet found as for joint A.

(8), (9) Same as for joint A.

Example. To lay out a triple-riveted lap-joint for ½ inch plates, using ¾ inch rivets.

$$\text{Then } \tfrac{3}{4} \times 1.935 \div \tfrac{1}{2} = \frac{3 \times 1.935 \times 2}{4} = 2.903$$

And $2.903 + 1.00 = 3.903$.

And $3.903 \times \frac{3}{4} = 2.93$ inches = pitch.

Take $p = 3$ inches.

Then taking strength of plate at 60,000, we have:

Strength of plate in joint $= (3 - \frac{3}{4}) \times \frac{1}{2} \times 60{,}000 = 67{,}500$.

Area of ¾ inch rivet $= .4418$.

Strength of rivets $= .4418 \times 3 \times 60{,}000 \times .821 = 65{,}280$.

Original strength of plate $= 3 \times \frac{1}{2} \times 60{,}000 = 90{,}000$.

Efficiency $= 65{,}280 \div 90{,}000 = .725$.

Similarly with ⅞ inch rivets, pitched 3⅞ inches, the efficiency becomes .764.

Joint D. Lap Joint. Triple Riveted, with rivets in inner row spaced one-half p.

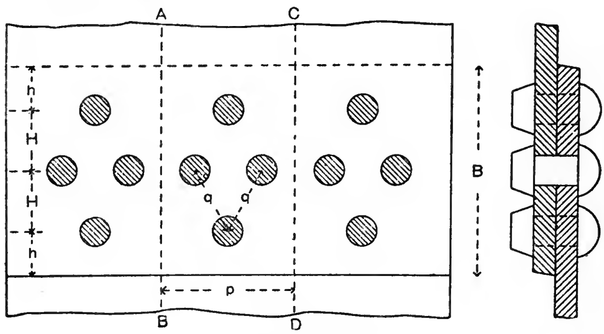


Fig. 34. Joint D.

$h = 1\frac{1}{2}\, d$

q not less than $(.3\, p + d)$

Hence H not less than $\sqrt{(.55\, p + d)\,(.05\, p + d)}$

$B = 3\, d + 2\, H$

As seen below, the efficiency of this joint is superior to that of joint *c*, but it is perhaps slightly inferior as regards tightness against leakage. We have in this case:

Strength of Plate $= t\,(p - d)\, T$

Strength of Rivets $= \pi d^2\, S$

For equal strength of plate and rivets:

$$\frac{p}{d} \text{ or } n = 1 + 2.58\,\frac{d}{t}$$

$$\text{Efficiency} = \frac{n - 1}{n} = \frac{2.58}{a + 2.58}$$

The values of $d \div t$ may vary through about the same range as in joint *A* above, and for the same reasons as there explained. These operations may be expressed by a rule similar to that for joint *A*, the numbered sections referring to that rule as given above.

Rule:

(1) Same as for joint *A*.

(2) Use 2.58 instead of .645.

(3), (4), (5), (6) Same as for joint *A*.

(7) Take four times the strength of one rivet found as for joint *A*.

(8), (9) Same as for joint *A*.

Example. To lay out a triple-riveted joint as in *D* for ½ inch plates, using ¾ inch rivets.

$$\text{Then } \tfrac{3}{4} \times 2.58 \div \tfrac{1}{2} = \frac{3 \times 2.58 \times 2}{4} = 3.87.$$

And $3.87 + 1.00 = 4.87$.

And $4.87 \times \frac{3}{4} = 3.65$ inches = pitch.

Take $p = 3$ 11-16 inches.

Then $\frac{1}{2} p = 1$ 27-32 inches.

Then taking strength of plate at 60,000 we have:

Strength of plate in joint $= (3\frac{11}{16} - \frac{3}{4}) \times \frac{1}{2} \times 60{,}000 = 88{,}125$.

Area of $\frac{3}{4}$ inch rivet $= .4418$.

Strength of rivets $= .4418 \times 4 \times 60{,}000 \times .821 = 87{,}050$.

Original strength of plate $= 3\frac{11}{16} \times \frac{1}{2} \times 60{,}000 = 110{,}625$.

Efficiency $= 87{,}050 \div 110{,}625 = .787$.

With ⅞ inch rivets spaced 2 7-16 inches in the middle row and 4⅞ in the outer rows, the strength of plate and rivets would be nearly equal, and the efficiency would rise to .81.

Joint E. Double Butt-Straps. Double Riveted.

$h = 1\frac{1}{2} d$

q not less than $(.6\, p + .4\, d)$

Hence H not less than $\sqrt{(1.1\, p + .4\, d)(.1\, p + .4\, d)}$

$B = 6\, d + 2\, H.$

Thickness of each butt-strap not less than ⅝ the thickness of plate.

The arrangement of rivets is duplicated on either side of the joint line *P Q*. We need only to investigate the part of the joint on one side of *P Q*. The element is then *A B D C*, as in joint *B*, except that the rivets are in *double* shear instead of *single* shear. For the total shearing strength of a rivet in double shear, as previously explained, it is customary to take 1¾ times the strength for single shear instead of 2 times, or to take the two strengths in the ratio 7 : 4.

We then have:

$$\textit{Strength of Plate} = t\,(p - d)\,T$$

$$\textit{Strength of Rivets} = \tfrac{7}{8}\pi d^2 S$$

For equal strength of plate and rivets:

$$\frac{p}{d} \text{ or } n = 1 + 2.26\frac{d}{t}$$

$$\text{Efficiency} = \frac{n-1}{n} = \frac{2.26}{a + 2.26}$$

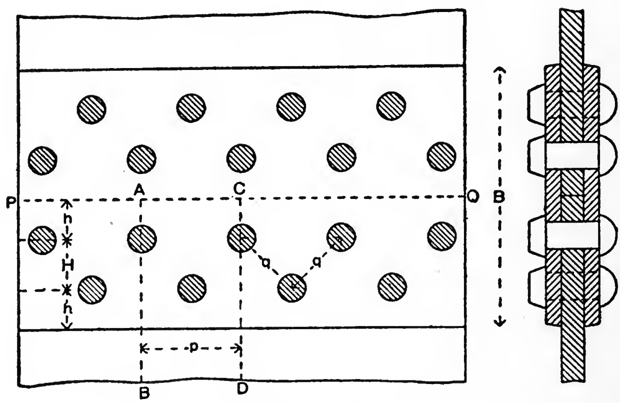


Fig. 35. Joint E.

In all double butt-strap joints $d \div t$ usually varies from 1 to $1\frac{1}{4}$. The lower range of values, as compared with joints in which the rivets are in single shear, is required in order to insure the joint against danger of failure by crushing.

These operations may be expressed by a rule similar to that for joint *A*, the numbered sections referring to that rule as given above.

Rule:

(1) Same as for joint *A*.

(2) Use 2.26 instead of .645.

(3), (4), (5), (6) Same as for joint *A*.

(7) Take $3\frac{1}{2}$ times the strength of one rivet, as found for joint *A*.

(8), (9) Same as for joint *A*.

Example. To lay out a joint as in *E* for 1 inch plates, using $1\frac{1}{8}$ inch rivets.

$$\text{Then } 1\tfrac{1}{8} \times 2.26 \div 1 = \frac{9 \times 2.26}{8} = 2.54.$$

And $2.54 + 1 = 3.54$.
And $3.54 \times \frac{9}{8} = 3.98$ inches = pitch.
Take pitch = 4 inches.

Then taking strength of plate at 60,000 lbs., as before, we have:

Strength of plate in joint $= (4 - 1\frac{1}{8}) \times 1 \times 60{,}000 =$ 172.500.

Area of $1\frac{1}{8}$ inch rivet $= .994$.

Strength of rivets $= .994 \times 3\frac{1}{2} \times 60{,}000 \times .821 = 171{,}350$.

Original strength of plate $= 4 \times 1 \times 60{,}000 = 240{,}000$.

Efficiency $= 171{,}350 \div 240{,}000 = .714$.

Similarly with 1 inch plates and $1\frac{1}{4}$ inch rivets, pitched $4\frac{3}{4}$ inches, the strength of plate and rivets will be about the same, and the efficiency is .737.

Joint F. Double Butt-Straps. Triple Riveted.

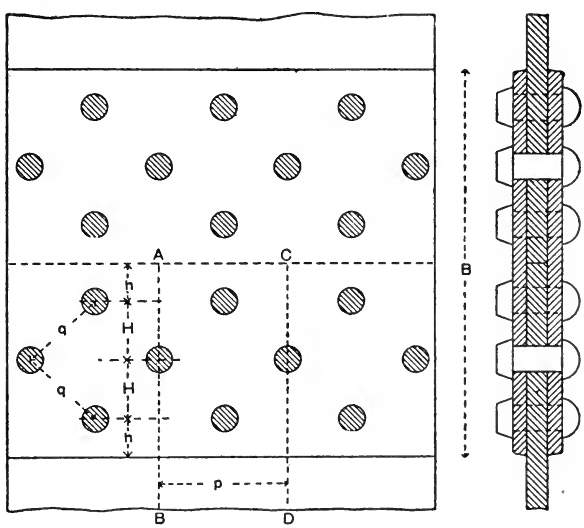


Fig. 36. Joint F.

$h = 1\frac{1}{2}\,d$

q not less than $(.6\,p + .4\,d)$

Hence H " " " $\sqrt{(1.1\,p + .4\,d)\,(.1\,p + .4\,d)}$

$B = 6\,d + 4\,H.$

Thickness of each butt-strap not less than $\frac{5}{8}$ the thickness of plate.

The element of the joint is $A\ B\ D\ C$, as in joint C, except that the rivets are in double shear. Taking, as before, the strength in double shear to that in single in the ratio 7 : 4, we have:

$$\textit{Strength of Plate} = t\,(p - d)\,T$$
$$\textit{Strength of Rivets} = \tfrac{21}{16}\,\pi d^2\,S$$

For equal strength of plate and rivets:

$$\frac{p}{d} \text{ or } n = 1 + 3.39\,\frac{d}{t}$$

$$\text{Efficiency} = \frac{n-1}{n} = \frac{3.39}{a + 3.39}$$

In this joint $d \div t$ usually varies from 1 to 1¼, as explained for joint *E*. These operations may be expressed by a rule similar to that for joint *A*, the numbered sections referring to that rule as given above.

Rule:

(1) Same as for joint *A*.

(2) Use 3.39 instead of .645.

(3), (4), (5), (6) Same as for joint *A*.

(7) Take 5¼ times the strength of one rivet, as found for joint *A*.

(8), (9) Same as for joint *A*.

Example. To lay out a joint as in *F* with 1 inch plates, using 1 3-16 inch rivets.

$$\text{Then } 1\tfrac{3}{16} \times 3.39 \div 1 = \frac{19 \times 3.39}{16} = 4.03$$

And $4.03 + 1 = 5.03$.

And $5.03 \times 1\frac{3}{16} = 5.97$ inches = pitch.

Take pitch = 6 inches.

Then with strength of plate at 60,000, as before, we have:

Strength of plate in joint $= (6 - 1\frac{3}{16}) \times 1 \times 60{,}000 = 288{,}750$.

Area of $1\frac{3}{16}$-inch rivet = 1.108.

Strength of rivets $= 1.108 \times 5\frac{1}{4} \times 60{,}000 \times .821 = 286{,}500$.

Original strength of plate $= 6 \times 1 \times 60{,}000 = 360{,}000$.

Efficiency $= 286{,}500 \div 360{,}000 = .796$.

Joint G. Double Butt-Straps. Rivets as in Joint D.

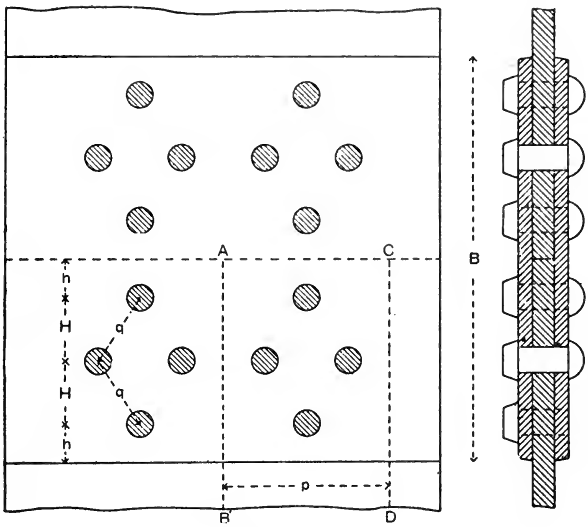


Fig. 37. Joint G.

$h = 1\frac{1}{2}\, d$

q not less than $(.3\, p + d)$

Hence H " " " $\sqrt{(.55\, p + d)\,(.05\, p + d)}$

$B = 6\, d + 4\, H.$

Butt-straps to be of thickness not less than as given by the formula:

$$\text{Thickness of strap} = \frac{5\,(p-d)}{8\,(p-2d)} \times (\text{thickness of plate})$$

The element of the joint is $A\,B\,D\,C$, as in joint D, except that the rivets are in double shear. Taking, as before, the strength in double shear to that in single shear in the ratio $7 : 4$, we have:

Strength of Plate $= t\,(p - d)\,T$

Strength of Rivets $= \frac{7}{4}\pi d^2 S$

For equal strength of plate and rivets:

$$\frac{p}{d} \text{ or } n = 1 + 4.52\frac{d}{t}$$

$$\text{Efficiency} = \frac{n - 1}{n} = \frac{4.52}{a + 4.52}$$

In this joint $d \div t$ usually varies from about 1 to $1\frac{1}{4}$, as explained for joint E. These operations may be expressed by a rule similar to that for joint A, the numbered sections referring to that rule as given above.

Rule:

(1) Same as for joint A.

(2) Use 4.52 instead of .645.

(3), (4), (5), (6) Same as for joint A.

(7) Take 7 times the strength of one rivet, as found for joint A.

(8), (9) Same as for joint A.

Example. To lay out a joint as in G with $1\frac{1}{2}$ inch plates, using $1\frac{5}{8}$ inch rivets.

$$\text{Then } 1\tfrac{5}{8} \times 4.52 \div 1\tfrac{1}{2} = \frac{13 \times 4.52 \times 2}{8 \times 3} = 4.9$$

And $4.9 + 1 = 5.9$.

And $5.9 \times 1\frac{5}{8} = 9.6$ inches = pitch.

Take pitch for outer rows $9\frac{5}{8}$ and for inner rows $4\frac{13}{16}$

Then, with strength of plate at 60,000, we have:

Strength of plate in joint $= (9\frac{5}{8} - 1\frac{1}{2}) \times 1\frac{1}{2} \times 60{,}000 = 731{,}250$.

Area of $1\frac{1}{2}$ inch rivet = 2.074.

Strength of rivets $= 2.074 \times 7 \times 60{,}000 \times .821 = 715{,}200$.

Original strength of plate $= 9\frac{5}{8} \times 1\frac{1}{2} \times 60{,}000 = 866{,}250$.

Efficiency $= 715{,}200 \div 866{,}250 = .826$.

Joint H. Double Butt-Straps. Triple Riveted, with double spacing in outer row on each side.

$h = 1\frac{1}{2}\, d$

q not less than $.3\, p + .4\, d$

q_1 " " " $.3\, p + d$

Hence H " " " $\sqrt{(.55\, p + .4 d)\,(.05\, p + .4\, d)}$

" H_1 " " " $\sqrt{(.55\, p + d)\,(.05\, p + d)}$

$B = 6\, d + 2\, H + 2\, H_1$

Thickness of butt-straps found by same formula as for joint G.

The element of the joint is $A\ B\ D\ C$, containing four whole rivets and two halves, or five in all. These are all in double shear. Taking, as before, the strength in double shear to that in single in the ratio 7 : 4, we have:

$$\textit{Strength of Plate} = t\,(p - d)\,T$$
$$\textit{Strength of Rivets} = \tfrac{35}{16}\pi\, d^2\, S$$

For equal strength of plate and rivets:

$$\frac{p}{d} \text{ or } n = 1 + 5.64\,\frac{d}{t}$$

$$\text{Efficiency} = \frac{n-1}{n} = \frac{5.64}{a + 5.64}$$

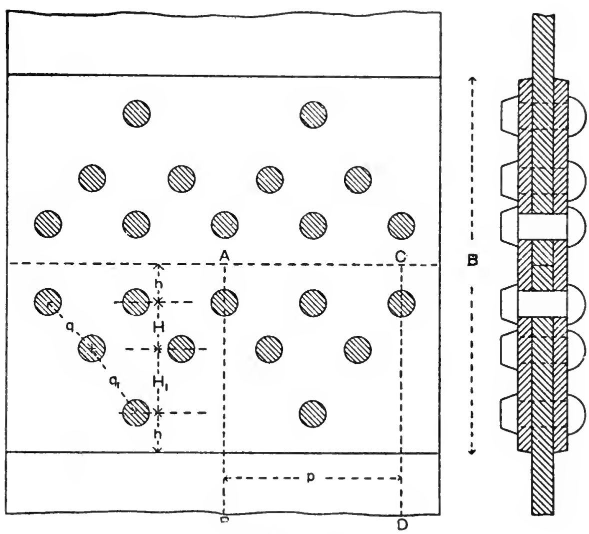


Fig. 38. Joint H.

In this joint $d \div t$ usually varies from 1 to $1\frac{1}{4}$, as explained for joint *E*. These operations may be expressed by a rule similar to that for joint *A*, the numbered sections referring to that rule as given above.

Rule:

(1) Same as for joint *A*.

(2) Use 5.64 instead of .645.

(3), (4), (5), (6) Same as for joint *A*.

(7) Take $8\frac{3}{4}$ times the strength of one rivet, as found for joint *A*.

(8), (9) Same as for joint *A*.

Example. To lay out a joint as in *H* with $1\frac{3}{8}$ inch plates, using 1 7-16 inch rivets.

Then $1\frac{7}{16} \times 5.64 \div 1\frac{3}{8} = \dfrac{23 \times 5.64 \times 8}{16 \times 11} = 5.9$

And $5.9 + 1 = 6.9$.

And $6.9 + 1\frac{7}{16} = 9.92$ inches = pitch.

The limiting pitch by the Board of Trade rule for this case would be 9.87. This means that a pitch of 9.92 or larger would not be passed without special permission. If necessary to reduce below the limit, the joint should be re-designed with a smaller rivet. This case illustrates the point that these limiting values of the pitch, if rigidly adhered to, would prevent the attainment of the best joint efficiencies with thick plates. We shall here assume the right to proceed with the pitch derived from the formula which we will take as 10 inches for the outer and 5 inches for the inner rows.

Then taking strength of plate at 60,000 we have:

Strength of plate in joint $= (10 - 1\frac{7}{16}) \times 1\frac{3}{8} \times 60{,}000 = 706{,}400$.

Area of $1\frac{7}{16}$ inch rivet $= 1.623$.

Strength of rivets $= 1.623 \times 8\frac{3}{4} \times 60{,}000 \times .821 = 699{,}600$.

Original strength of plate $= 10 \times 1\frac{3}{8} \times 60{,}000 = 825{,}000$.

Efficiency $= 699{,}600 \div 825{,}000 = .848$.

Joint I. Double Butt-Straps. Triple Riveted, outer row on each side being double spaced, and passing through inside butt-strap only.

$$h = 1\tfrac{1}{2}\, d$$

q not less than $.3\,p + .4\,d$

q_1 " " " $.3\,p + d$

Hence H " " " $\sqrt{(.55\,p + .4\,d)(.05\,p + .4\,d)}$

" H_1 " " " $\sqrt{(.55\,p + d)(.05\,p + d)}$

$$B = 6\,d + 2\,H$$

$$B_1 = 6\,d + 2\,H + 2\,H_1$$

Thickness of butt-straps found by same formula as for joint *G*. The element of this joint is *A B D C*, with four rivets in double shear and one in single shear. Taking, as before, the strength in double shear to that in single in the ratio 7 : 4, we have:

$$\textit{Strength of Plate} = t\,(p - d)\,T$$

$$\textit{Strength of Rivets} = 2\,\pi d^2\,S$$

For equal strength of plate and rivets, we have:

$$\frac{p}{d} \text{ or } n = 1 + 5.16\,\frac{d}{t}$$

$$\text{Efficiency} = \frac{n-1}{n} = \frac{5.16}{a + 5.16}$$

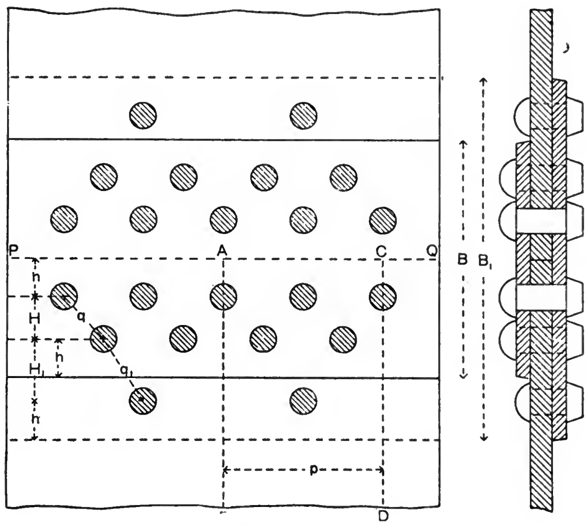


Fig. 39. Joint I.

In this joint $d \div t$ usually varies from 1 to $1\frac{1}{4}$, as explained for joint *E*. These operations may be expressed by a rule similar to that for joint *A*, the numbered sections referring to that rule as given above.

Rule:

(1) Same as for joint *A*.

(2) Use 5.16 instead of .645.

(3), (4), (5), (6) Same as for joint *A*.

(7) Take 8 times the strength of one rivet, as found for joint *A*.

(8), (9) Same as for joint *A*.

Example. To lay out a joint as in *I* with $1\frac{3}{8}$ inch plates, using 1 7-16 inch rivets.

$$\text{Then } 1\tfrac{7}{16} \times 5.16 \div 1\tfrac{3}{8} = \frac{23 \times 5.16 \times 8}{16 \times 11} = 5.40$$

And $5.40 + 1 = 6.40$.

And $6.40 \times 1\frac{7}{16} = 9.2$ inches = pitch.

We will take $9\frac{1}{4}$ for pitch of outer row, and hence $4\frac{5}{8}$ for pitch of inner rows. Then, taking strength of plate at 60,000, we have:

Strength of plate in joint $= (9\frac{1}{4} - 1\frac{7}{16}) \times 1\frac{3}{8} \times 60{,}000 =$ 644,530.

Area of $1\frac{7}{16}$ inch rivet $= 1.623$.

Strength of rivets $= 1.623 \times 8 \times 60{,}000 \times .821 = 639{,}600$.

Original strength of plate $= 9\frac{1}{4} \times 1\frac{3}{8} \times 60{,}000 = 763{,}125$.

Efficiency $= 639{,}600 \div 763{,}125 = .838$.

An examination of the values of the efficiency will show that these various joints for the same value of $d \div t$ stand, in this respect, in the order:

$$H,\ I,\ G,\ F,\ D,\ E,\ C,\ B,\ A.$$

Sec. 16. MATERIALS AND CONSTRUCTION.

[1] Materials.

Open hearth mild steel is used almost universally as the material for boiler construction, and in standard practice is used exclusively for shells, drums, heads, furnaces, combustion chambers and braces. Both steel and wrought iron are used for tubes, though solid drawn steel tubes may be considered as the better representing advanced engineering practice. Wrought iron is also used to some extent for rivets, though in the best modern practice steel rivets are preferred.

[2] Joints.

The various plates of a boiler are fastened together by riveted joints. These are of several varieties, as discussed in Sec. 15, and to which reference may be made.

The holes in the plates are either drilled or punched. The former method is much the better. In the operation of punching, a thin skin of metal about the hole is so severely strained that its strength, and especially its ductility and toughness, are reduced far below what they are in the remainder of the plate. This is not the case with the operation of drilling, or, at least, not to anything like the same extent. Drilled holes may also be located more accurately than punched holes, and thus with the former the parts of a riveted joint may be more perfectly fitted than with the latter. The operation of drilling leaves, however, a sharp edge, which should be removed by a reamer in order to avoid any tendency to cut the rivet. In spite of the greater cost of drilled holes they are now generally accepted as the best for all high-class work, and in many specifications no holes are allowed to be punched.

Riveting is either by hand or by machine; usually hydraulic. The latter gives much the better result, and is preferred where the machine can be made available. In many cases the construction is such that the jaws of the machine cannot be brought to bear on the joint, and in consequence hand riveting must be employed.

After being riveted the joints are calked to insure tightness against leakage. This operation consists in beating down the edges of the metal against the face of the opposite plate by means of special pneumatic driven or hand tools, as shown in Fig. 40. These are known as *calking* tools, and are of two

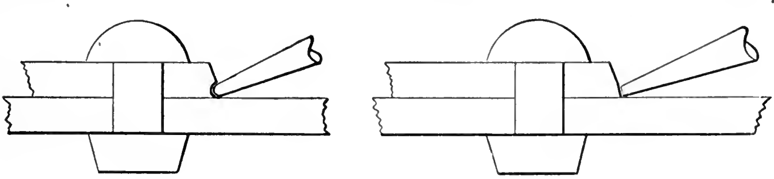


Fig. 40. Calking Tools.

types, square and round nosed, as shown in the figure. The latter form is usually employed in modern practice. For operations on board ship the common hand tool is, of course, most commonly used; but for extensive calking, as in the construction of boilers in the shop and where compressed air is available, the pneumatic driven tool is very largely displacing the hand tool.

[3] Construction of Fire-Tube Boilers.

We will now consider the chief features of the construction of a Scotch boiler. This will, at the same time, sufficiently illustrate the operations involved in the construction of other types of fire-tube boilers.

In the best practice the longitudinal joints are double butt-strapped and triple-riveted in order to give to the boiler in this direction the highest possible proportion of the strength of the plate itself. The circumferential joints, those which run around the shell, are lapped and double or triple riveted. So far as internal pressure is concerned the boiler is twice as strong to resist rupture around the girth as lengthwise so that a lapped circumferential or girth joint is quite enough for strength alone, and it only remains to make it steam and water tight and to insure the necessary stiffness of the boiler as a whole. (See Sec. 63.) Single-ended boilers are usually made with two courses of plates, as in Fig. 11. Double-ended boilers are usually made with three courses. Each course consists of two or three sheets, varying with the diameter of the boiler. The heads are flanged, as shown in Fig. 11, and thus secured by riveting to the shell. In some cases the shell has been flanged instead of the head, but such form of construction is rare. The head flanges are sometimes turned out, and sometimes in, as shown by the figure. Where machine riveting is to be used they must be turned out in order to allow the riveter to do its work. The back head is made usually in two pieces, with double or triple-

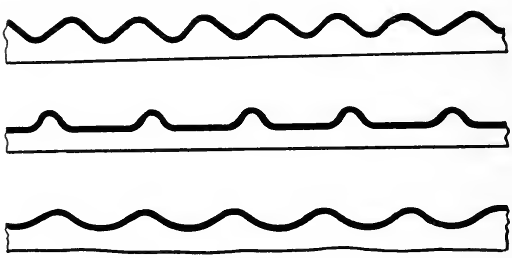


Fig. 41. Styles of Corrugation.

riveted lap joints. The front head is made in two or three pieces, according to size of boiler, usually with double-riveted lap joints.

The furnaces, as shown, are corrugated in order to give greater strength and elasticity. There are three styles of corrugation in common use, as shown in Fig. 41. The furnaces are riveted to flanges formed on the front furnace sheet, and are connected by flanging to the sheets of the combustion chamber. Several different modes of connection are in use for this purpose. In one the furnace end is left plain and the flange is all on the combustion chamber sheet, as in Fig. 9. In another the combustion chamber sheets are left plain and the flange is on the furnace, as shown in Figs. 11, 42, 44. In some forms provision is made for removal and renewal without disturbing the furnace head sheets. Thus, in Fig. 9 the diameter at the front is the same as, or slightly larger, than that on the outside of the corrugations, and so the furnace may be withdrawn through the opening in the front sheet. In other forms of connection, where the furnace is flanged, especial provision must be made for removal, as shown in Fig. 43.

Here the back end of the furnace is necked in on the bottom and sides, and a flange is thus obtained which only extends outside the outer diameter of the corrugation at the top. This

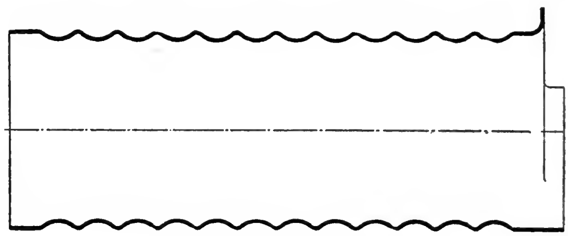


Fig. 42. Flanged Furnace.

flange serves to attach the furnace to the combustion chamber, and on cutting the joint loose the furnace may be taken straight to the front until the upper flange strikes the front sheet, and then swung upward and out of the front opening, as may be readily seen.

In some cases where it is difficult to obtain the necessary room on the front head for the greater diameter of the outside

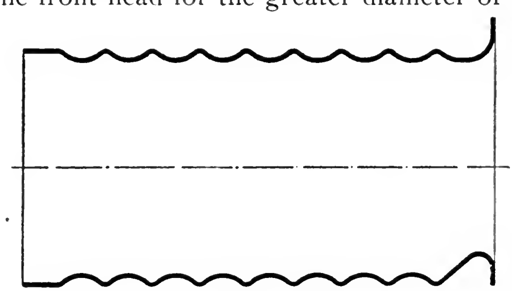


Fig. 43. Removable Furnace.

of the corrugation, or where, for other reasons, it is not considered preferable to have the furnaces removable without disturbing the front sheet, the furnace end at the front runs out on the smaller diameter, as shown in Fig. 44. Some one of the forms favoring easy removal may be recommended as preferable in all ordinary cases.

The combustion chamber, as shown, is built up of steel plates flanged and riveted together. The details of the construction vary somewhat with the form of furnace attachment adopted, with the size of the boiler, and with the choice of the designer. The front plate is known as the *back tube sheet.* The top of the combustion chamber is sometimes flat, as in Figs. 9, 65, and sometimes rounded up, as in Figs. 11 and 45.

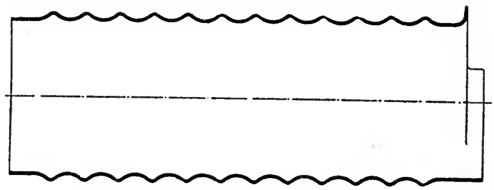


Fig. 44. Non-Removable Furnace.

The tubes are secured into the tube sheets by "expanding," and "beading" or turning over at the back or at both the back and front ends. See Figs. 46, 47, 48. Tubes are expanded by means of a tool as shown in Fig. 49, representing the Dudgeon expander. The tool is introduced into the mouth of the tube and the small steel rolls are forced out by means of the tapering steel mandrel on which they rest. The mandrel is then turned

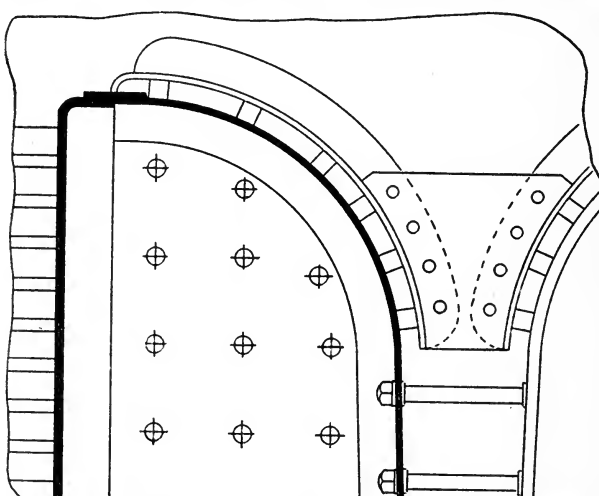


Fig. 45. Rounded Top Combustion Chamber.

around, and this by means of the frictional contact with the rolls causes them to turn also, and thus to roll around on the inner surface of the tube, carrying the whole tool slowly round and round. The mandrel is continually forced in and thus the rolls are forced outward against the tube. The action is thus a rolling of the tube out against the tube sheet, and in this way the joint is made thoroughly tight.

The Prosser expander, which was generally employed in former years, is now but rarely used. It consists, as shown in

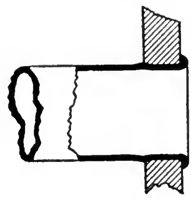


Fig. 46. Tube End.

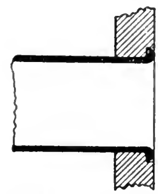


Fig. 47. Tube End.

Fig 50, of a hollow tapering plug divided up into separate elements or sections which are held together by a steel band. These are forced outward against the inner surface of the tube by driving a taper mandrel into the hollow between the ele-

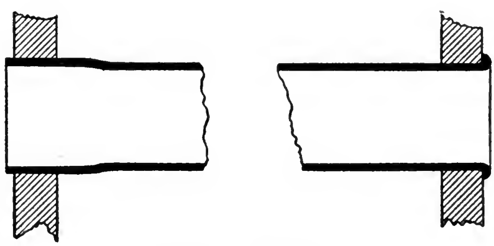


Fig. 48. Tube Ends.

ments. The action of the expander is thus to force the metal of the tube out against the edges of the sheet in a form of circular ridge as shown in Fig. 46.

Beading over the tube ends is usually done with a tool, as

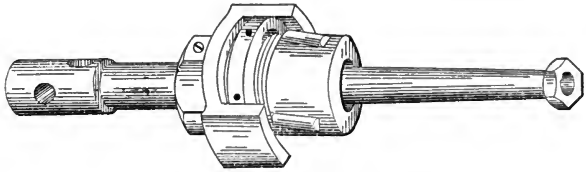


Fig. 49. Roller Tube Expander.

shown in Fig. 51, and the result is as shown in Figs. 46-48. In some cases the tube sheet is recessed out for the beaded end of the tube, as shown in Fig. 47. The front ends of the tubes, as shown in Fig. 48, are usually swelled slightly larger than the rear ends to facilitate removal. The thickness of the metal of plain boiler tubes is usually from 8 to 12 wire gauge, or from about .17 to .10 in.

In addition to the plain tubes fitted as before described,

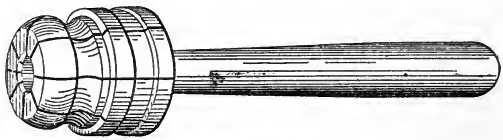


Fig. 50. Prosser Tube Expander.

stay-tubes are also frequently fitted. These are of extra heavy metal, usually about ¼ in. thickness, and specially fitted to the

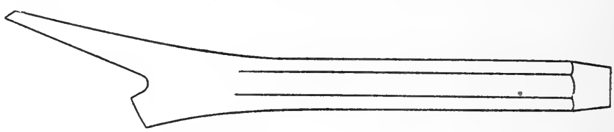


Fig. 51. Beading Tool.

tube sheets by screw joints, as shown in Fig. 52. These tubes act as stays between the tube sheets. Further reference to this

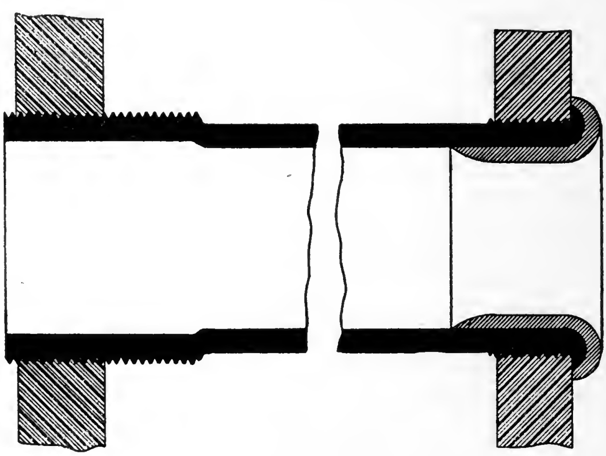


Fig. 52. Stay Tube with Ferrule.

point will be found under the head of *bracing*. When stay tubes are fitted, it is customary to bead over only the back ends of the plain tubes, as in Fig. 48. Not infrequently, however, no stay tubes are fitted, and in such case the plain tubes must be beaded over on both ends in order that they may securely support the tube sheets. Instead of the ordinary form of boiler tube, the *Serve* tube of cross-section, as shown in Fig. 53, is frequently fitted. The ribs of metal reach down into the column of hot gas moving through the tube and furnish additional surface to absorb the heat and help it through into the water. The surface on the fire side is thus much greater than the surface on the water side, while with the plain tube it is somewhat less.

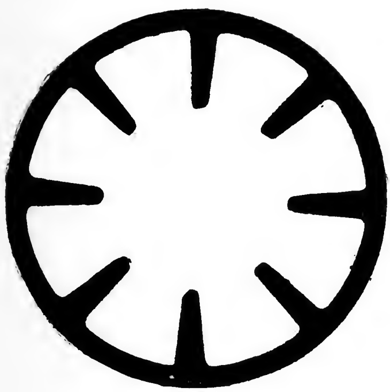


Fig. 53. Serve Tube.

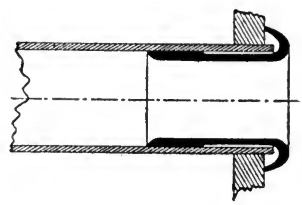


Fig. 55. New Admiralty Ferrule.

Such tubes usually show an increased evaporation per square foot of surface measured on the water side, of from 15 to 20 per cent. Their increased weight, however, offsets in a measure this increase of evaporative efficiency per square foot of surface.

Reference may also be made at this point to the use of *retarders* in boiler tubes. These are long twisted strips of thin sheet steel, as shown in Fig. 54. They are simply laid in the tubes and serve to give the gases more or less rotary motion

Fig. 54. Retarder.

and to assist in throwing them outward against the surface of the tube. With forced draft and high rates of combustion the use of retarders has been accompanied with a marked increase of economy. In some cases both Serve tubes and retarders have been fitted, but the special advantages of the combination may be called in question.

As a measure of protection for the back ends of tubes under forced draft, cast iron ferrules are sometimes fitted. Fig.

52 shows the so-called *Admiralty* ferrule in place in a stay tube. In Fig. 55 is shown an improved type of ferrule which by reason of the air space is believed to act still more efficiently to protect the tube sheet than the form shown in Fig. 52.

Bracing.—We must now consider the bracing needed to make the boiler perfectly secure and safe under the pressures

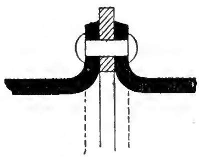


Fig. 56. Adamson Ring.

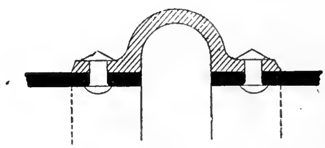


Fig. 57. Bowling Ring.

to which the various parts will be subjected. The general principles to be kept in mind are as follows: (*a*) Cylindrical surfaces subjected to pressure on the concave side are not helped by bracing. They must be made sufficiently strong by giving to the material a suitable thickness. (*b*) Cylindrical surfaces subjected to pressure on the convex side may be stayed like a flat

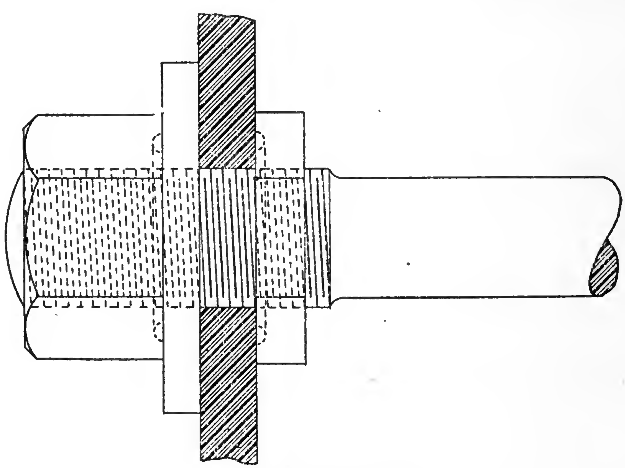


Fig. 58. Main Head Brace.

surface, or they may be stiffened by ribs running around them in planes at right angles to the axis. (*c*) Flat surfaces will support themselves if their area is sufficiently small in relation to their thickness and to the load per square inch, and it follows that large, flat surfaces must be sub-divided into parts of such size that they may thus become self-supporting.

As an illustration of (*b*), furnaces were formerly strengthened in this way, and the long favorite Adamson ring, as shown in Fig. 56, or the Bowling ring, as shown in Fig. 57, may be taken as good illustrations of this mode of adding support to cylindrical surfaces loaded on the convex side. The present corrugated furnace, especially the Purves type, as shown in Fig. 41, may be considered as a further illustration of the same principle. In modern marine boilers, aside from the furnaces, this mode of support is chiefly used to stiffen the bottom of single combustion chambers where screw stay bolts could not be readily fitted, and also in some cases the curved tops of combustion chambers. See Fig. 45.

Coming next to flat surfaces as referred to under (*c*), the necessary sub-division is provided by the fitting of braces connecting the part to be supported to some point where the support can be provided, or by connecting together two surfaces urged by the steam pressure in opposite directions, as for example the two opposite heads of a boiler, as shown in Figs. 9, 11. Occasionally also flat surfaces are aided by attaching to them stiffening ribs of angle or tee bar, as on the front tube sheet, between the nests of tubes, or between the tubes and the shell.

Plates which are subjected to the direct action of the fire, as in the furnace and combustion chambers, are made relatively thin. This is done because a thin plate transmits heat better than a thick one, and is subjected to less severe internal stresses due to the difference in temperature of its two faces. The thinner the plate, however, the less the area which will be self supporting. Hence the braces for thin flat plates are relatively small and closely spaced, while those for thick plates are larger and spaced at greater intervals.

The main head braces are secured as shown in Fig. 58. A washer is fitted on the outside to increase the supported area, and a nut is fitted both inside and outside so that the joint may readily be made tight, and that the brace may, if needed, act as a strut against pressure from without as well as a tie against pressure from within. In some cases a relatively thin plate is supported by a brace connecting it to a thicker or perhaps to a double plate, or to a place not requiring support itself, but which furnishes a convenient point for carrying the load. In such case the attachment to the thin plate is often made, as shown in

Fig. 59, in order the better to sub-divide and distribute the support. In double-ended boilers, certain parts of the head, as for example those between the furnaces, are supported by braces running obliquely back to the shell and attached as shown in Fig. 60. It often thus happens that braces must run at a slight obliquity in order to connect the parts to be supported with convenient points of support. Other instances are

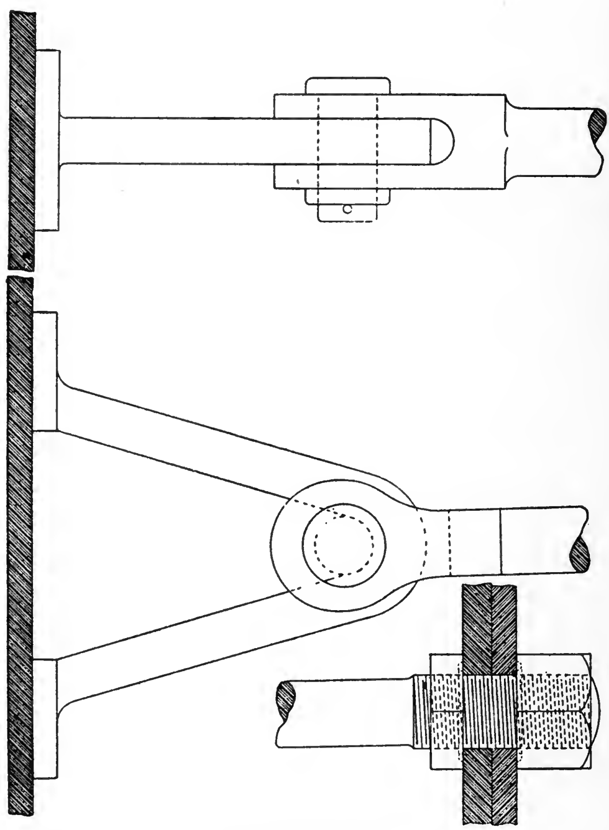


Fig. 59. Forked End Brace.

often found in the braces connecting parts of the back tube sheet below or between the tubes to the boiler head. In all such cases, wedge-shaped washers, as shown in Fig. 61, must be fitted under the nuts in order to get a good bearing between the nut and the shell.

The braces connecting the relatively thin plates of the combustion chamber to the back head and shell of the boiler and to each other, are fitted by screwing them through into both plates, as shown in Fig. 62. The ends are sometimes riveted over and sometimes fitted with nuts. In some cases they are left thread-

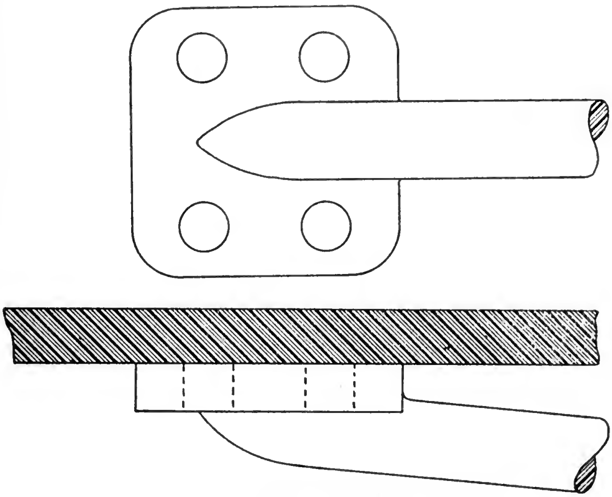


Fig. 60. Flange Foot Brace.

ed the entire length, in others the threads are raised on the ends, as in the main head braces. The latter practice is much to be preferred. These braces are commonly known as "screw

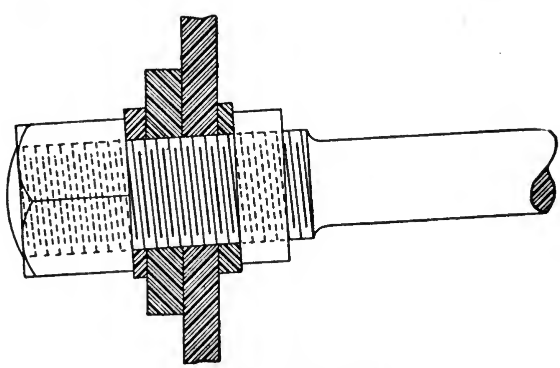


Fig. 61. Oblique Brace.

stays," or "screw staybolts." This mode of fitting enables the screw stay bolt to act both as strut and as tie, or to resist pressure in both directions. In some cases the older form of "socket bolt," as illustrated in Fig. 63 is still fitted. In such case the head is riveted and the part of the bolt between the plates is provided with a hollow "socket." This acts as a strut and as a protection to the bolt proper. In modern approved practice screw stay bolts are either hollow or are drilled in at

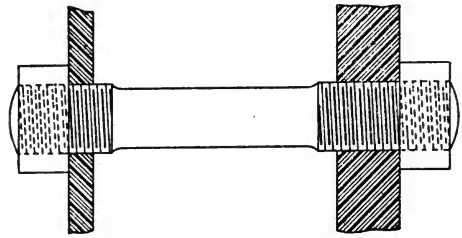


Fig. 62. Screw Stay Bolt.

each end (see Fig. 64), to a point well beyond the inner face of the supported plate. The expansion and contraction of such parts of the boiler often have the effect of bending these bolts

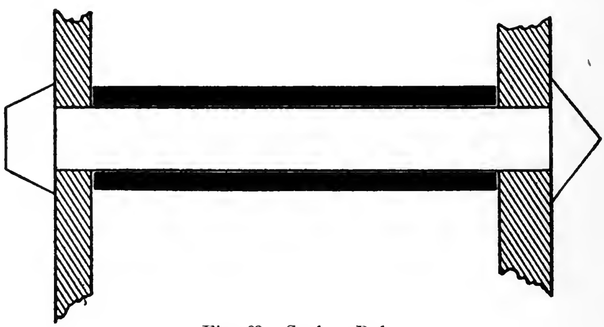


Fig. 63. Socket Bolt.

back and forth, and they may thus in time become broken off, the break naturally occurring near the thicker of the two sheets where the bolt is held more rigidly. If this should occur, or

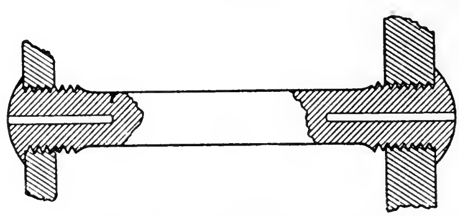


Fig. 64. Improved Screw Stay.

if the bolt should become badly corroded or pitted, especially near the plate, warning will be given of the fact by the escape of water or steam, and proper means must be taken for replacing the bolt. In this way timely warning may be given of a condition of affairs, which if allowed to go unnoted, might result in a collapse of the plate, or in a disastrous explosion of the boiler as a whole.

The usual spacing of stays, such as that shown in Fig. 58,

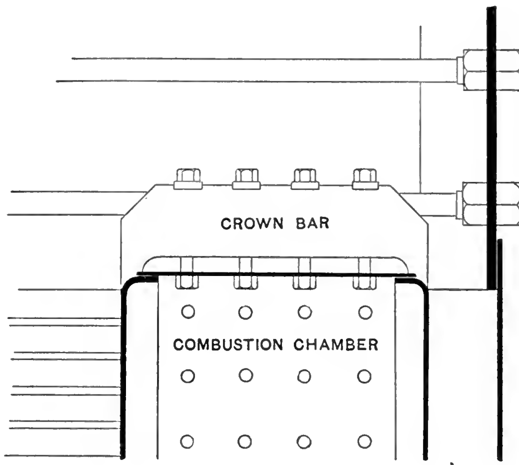


Fig. 65. Girder Brace or Crown Bar.

and supporting plates not directly exposed to the hot flames or gases is from 14 to 16 inches between centers, while for screw

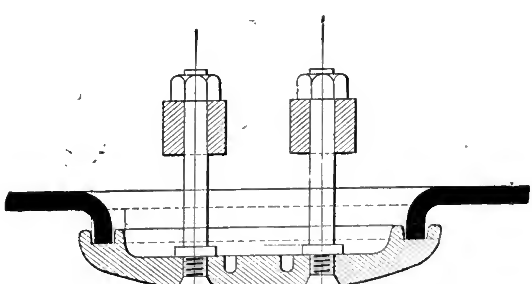


Fig. 66. Flanged Manhole and Fitting.

stays supporting plates more or less directly exposed to the fire, the spacing is usually between 6 to 8 inches. See also on this point Sec. 62.

For the support of the top of the combustion chamber, girders or crown-bars are used, see Fig. 65. The load is transferred by means of the bolts from the combustion chamber plate to the

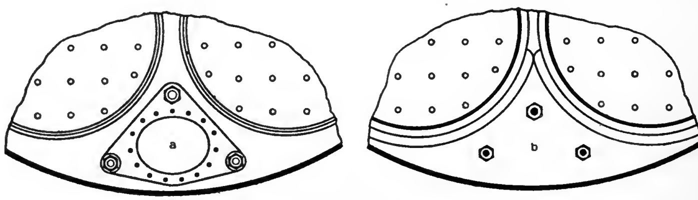


Fig. 67. Reinforce Plate.

girder, while the latter is supported by the edges of the vertical plates forming the front and back of the chamber.

These girders are made of two pieces of steel plate, usually from ½ to ¾ in. thick, bolted or riveted together with distance

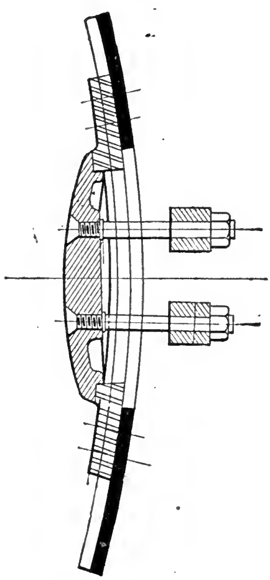


Fig. 68. Manhole and Fitting in Shell of Boiler.

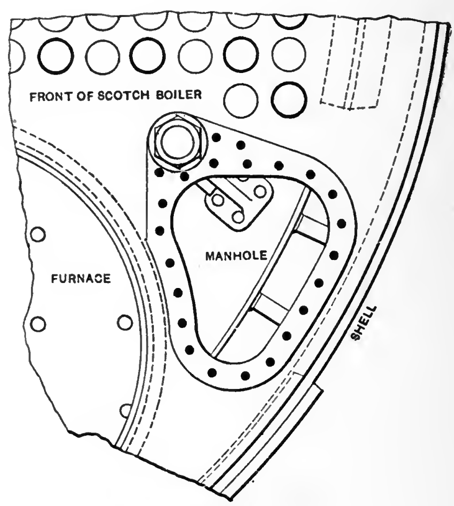


Fig. 69. Reinforce Plate.

pieces between so that the bolts which take the load from the flat plate may pass up between them as shown in the figure.

The combustion chamber is sometimes secured to the back head of the boiler, or in double-ended boilers the two combustion chambers are secured together by plate braces fitted as shown in Fig. 45. Such are usually called *gusset* braces.

In the general internal arrangement of the tubes, furnaces and combustion chambers, care must be taken to allow, as far as possible, a ready examination of the various parts. In good modern practice a space of from 10 to 12 inches is left between the nests of tubes and between the tubes and shell, so as to allow the passage of a man from the steam space down through these spaces to the furnaces.

Manholes and Covers.—For the purpose of entering, examining and cleaning the interior of a boiler, man or hand holes are cut in the head or shell. These are then covered by *manhole covers, plates* or *doors,* as they are variously called. These

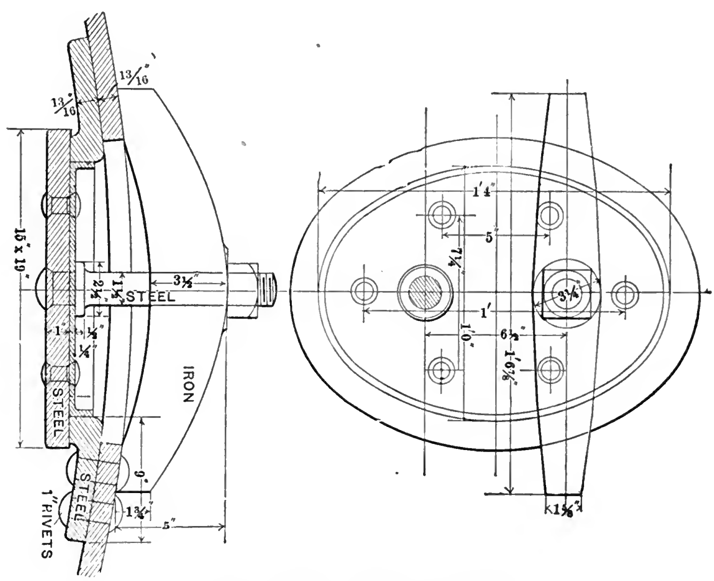


Fig. 70. Manhole Plate and Fittings.

are secured by bolts and dogs, as shown in Figs. 66-70. The usual size of a manhole is 11 by 15 inches, which are the dimensions required by the U. S. rules. It is of oval or elliptical shape, so that the cover with its lip extending over the edge may be gotten in and out without difficulty. The joint is made on the inside in order that the pressure may tend to keep the joint tight. A hand-hole is entirely similar in shape and fitting, and is simply smaller in size. In order to provide local strength and stiffness and to help support the load which comes on the feet of the dog, and also when the hole is cut in the shell to restore in some measure the metal taken out, a reinforce ring of metal is fitted about the hole. Such a ring of cast steel for a hole in the shell is shown in Fig. 68. The inner face is planed, so that the joint with the cover is readily made. In order that the removal of the metal may affect as little as possible the strength of the shell, the longer axis of the hole should run around the boiler rather than lengthwise. For holes in the head of a boiler the metal is often flanged inward, as shown in Fig. 66, the joint being made against the dressed edge of the ring. Where a manhole or handhole comes close to through braces, as, for example, near

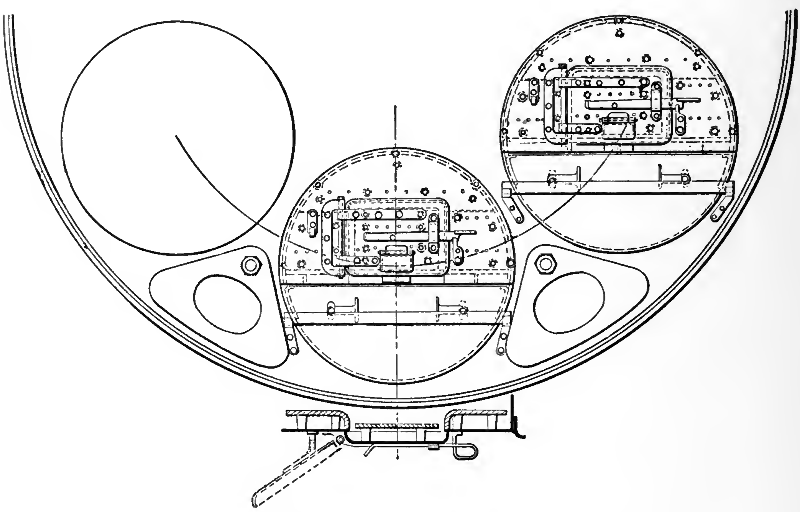


Fig. 71. Furnace Front.

the furnaces, the reinforce plate may be formed, as shown in Figs. 67 and 69. At the angles or corners the plate is of sufficient width to let the threaded end of the brace come through, and the outside nut is then jammed down on the ring as shown. For heavy pressure the fitting illustrated in Fig. 70 may be recommended. The reinforce ring is of flanged steel, and the cover of steel plate also, somewhat thicker than the metal of the shell. An angle iron, as shown, is riveted to the cover, making a neat fit within the reinforce ring, and keeping the plate accurately to its seat.

Furnace Fronts and Doors.—The furnace front is a fitting attached to the mouth of the furnace, and carrying the furnace door. In Figs. 71, 72 a common form of arrangement is shown. The front consists of a steel plate forming the outer part, and made with lugs or a flange for attachment to the furnace. The opening for the door is formed, as shown, within this front or door frame as it is sometimes called. Attached to this frame and with a space between is a second plate of cast iron forming the inner wall. This is pierced with a large number of small holes, while the frame is provided with a smaller number of larger holes. These are provided for the purpose of admitting air to the furnace above the grate. The inner plate is subject to the direct action of the fire, and although cooled somewhat by the air passing through, it is liable to burn out from

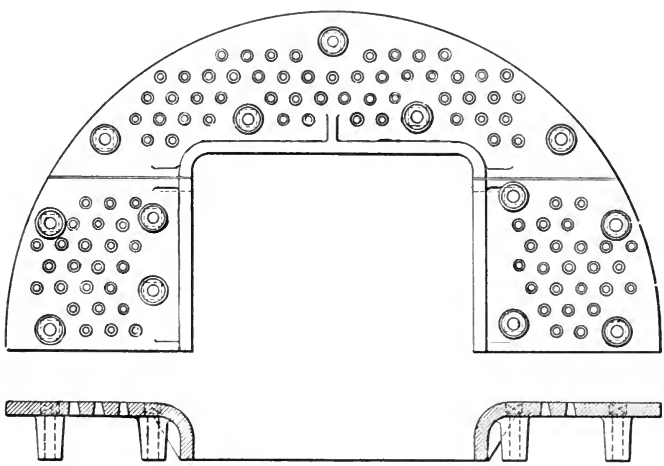


Fig. 72. Detail of Furnace Front.

time to time. It is for this reason that it is made as a separate piece, and so is readily replaced as occasion requires. The door is formed in much the same way as the frame, and is provided with holes in a similar fashion and for the same purpose. Often a small covered peep-hole is provided for examining the fire without opening the door. In some cases also a small opening is made through which a slice bar may be introduced for stirring or breaking up the fire without opening the door. A form of slide or gridiron is also sometimes fitted so as to control the amount of air entering above the grates. In some cases the doors and frames are made entirely of flanged steel plates instead of cast iron, while much variety exists in the arrangement of the holes for the introduction of the air. Certain special fittings necessary to adapt the furnace fronts and doors to the application of forced draft will be referred to at a later point.

The ash-pit door usually consists simply of a plate of thin sheet steel provided with the necessary lugs and handles, and covering the front opening in the furnace below the grate bars. It is used chiefly as a damper in connection with closed stoke-hold forced draft.

Grate and Bridge Walls.—The general arrangement of the inside of the furnace is illustrated in Fig. 73. The grate extends from the front of the furnace to the bridge wall as shown. The bottom of the door frame extends back a little

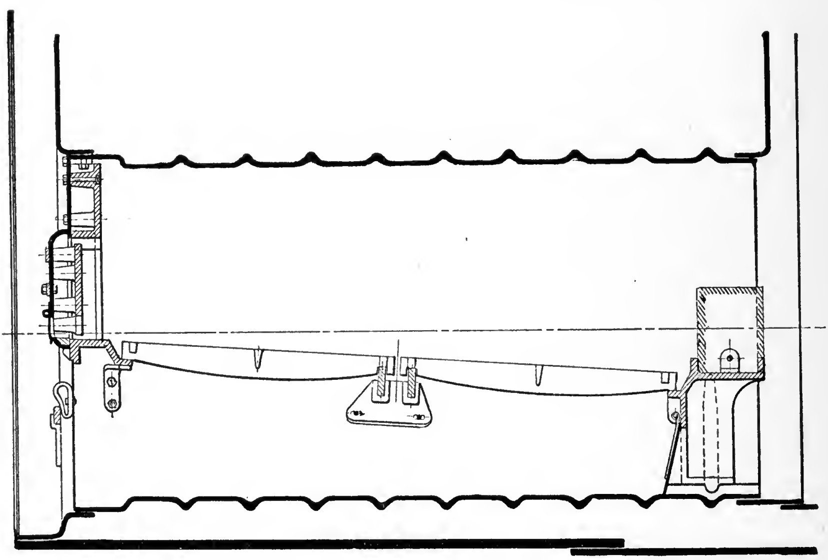


Fig. 73. Section of Furnace and Grate.

way and drops down, forming a kind of shelf for the support of the front ends of the grate bars. In some cases this extension of the door frame extends back some distance, forming the so-called *dead plate*, upon which bituminous coal may be piled when first fired, so as to provide for the gradual distillation and combustion of its gases.

The great bars may be made in a large variety of forms. In Fig. 74 is shown the standard type of cast iron bar. There are usually two lengths of bar in the length of the furnace, supported by the door frame in front and bridge wall at the rear, and by *bearing bars* in the middle. These latter in turn are supported at their ends by attachment to the furnace. The bars are usually cast double, as shown, while for convenience in fitting grates of varying widths, a small number of single bars are usually provided. The width of air space between the bars is usually made about equal to the width of the bar, or about one-half of the entire grate area, although this proportion should vary somewhat according to the fuel in use. The surface of the grate usually slopes slightly from front to rear, from 1 in 24 to 1 in 12, covering the usual range of angle. Cast iron grate bars often have a shallow groove running along the top. This fills with ashes and tends to prevent the clinkers adhering to the grate.

In addition to the type of bar shown in Fig. 74, square wrought iron bars running the whole length of the furnace are sometimes used, and there is a large variety of patent and special kinds of shaking grate. The purpose in grates of this character is to provide means for breaking up and working the fire without the need of opening the door. Many of them accomplish this end to a considerable extent, but the greater simplicity and cheapness of the plain cast iron grate, as in Fig. 74, insures for the latter a wide use, and it is still the favorite in ordinary practice.

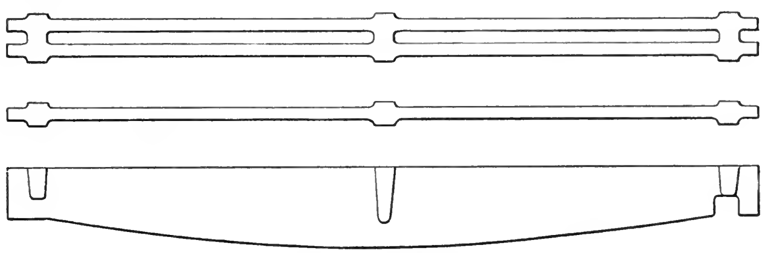


Fig. 74. Grate Bar.

Turning now to the bridge wall, a common arrangement is shown in Fig. 73. A casting extends across the back of the furnace and is supported by attachment at the sides. This supports the back ends of the grate bars, as already referred to, and also a wall of fire brick which forms the back limit of the grate, and over which the products of combustion pass on their way to the combustion chamber.

Instead of fire-brick, the use of cast iron for bridges is becoming frequent in modern advanced practice. Such bridges are of ribbed or channeled form, and in use they become sufficiently covered with ashes to form a protection against the heat of the fire.

Front Connections and Funnel.—After leaving the tubes at the front end the gases and smoke must be guided to the base of the funnel. This is done by the front connections or smoke boxes and uptakes, as shown in the diagrams. Fig. 75 shows the connection made between two single-ended boilers and one funnel, used in common by both. The boilers are placed front to front in an athwartship fire room. Fig. 76 shows the connections between one double-ended boiler and the smoke pipe.

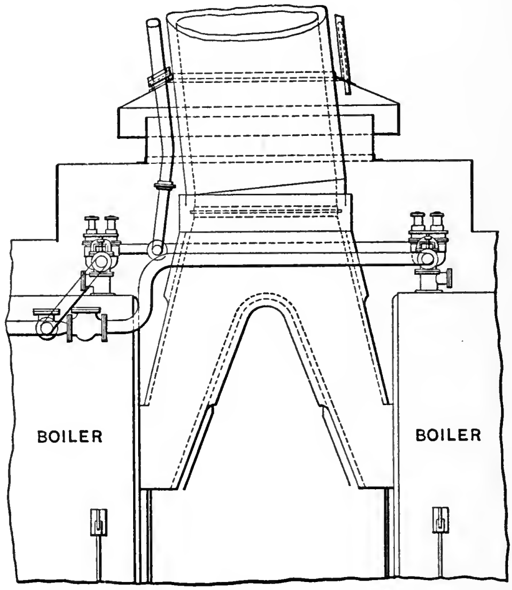


Fig. 75. Front Connections, Uptakes and Funnel Base.

These connections are formed of sheet metal riveted up in two or more thicknesses with an air space or non-conducting material between. The term *front connection* refers more especially to that part of the passage directly in front of the tubes. This is provided with doors swinging upward to allow examination, cleaning and repair of the tubes. A swinging damper is often placed in the uptakes for controlling the draft as may be desired, especially where two or more boilers are connected to one stack. The funnel or stack is also made of sheet metal riveted up, and in good practice in two thicknesses with a considerable air space between. This tends to prevent loss of heat by radiation, and thus the temperature of the gases is kept as high as possible while in the funnel, as is necessary for good draft. It may be remembered that for boiler economy the temperature of the waste gases at the front connection should be as low as possible, while for the sake of the draft all further loss of heat while in the funnel should be prevented. Around the base of the funnel is fitted an additional air screen or passage, known as the *air casing*. See Fig. 77. This serves to ventilate

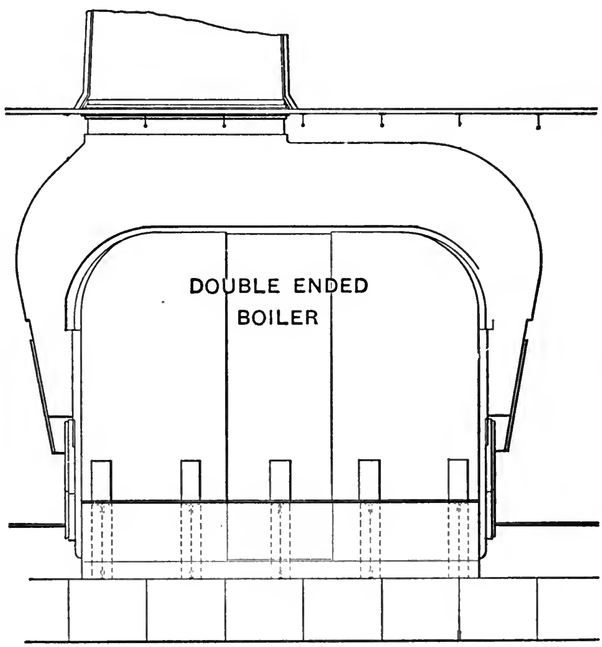


Fig. 76. Front Connections and Uptakes.

the fire-rooms and to protect the neighboring parts of the ship from the heat radiated by the funnel. The air casing is protected from the weather by a sloping ring of metal attached to the funnel, as shown in the figure, and known as the *umbrella*.

The weight of the funnel is usually carried by straps or lugs attached to the structure of the ship, and it is furthermore stayed by guys on deck in order to provide the necessary steadiness and support in a sea-way. In small craft, however, the weight of the funnel is often taken simply by the uptakes and boilers.

The funnel is often provided with a cover, which may be placed over the top when the ship is laid up, or when for other reason the funnel is not in use. The cover is usually kept a little distance above the top so as to allow the escape of smoke from small fires used for warming and airing the boilers. A ladderway should also be provided on the funnel to assist in examination, adjustment of guys, fitting of cover, etc. In small

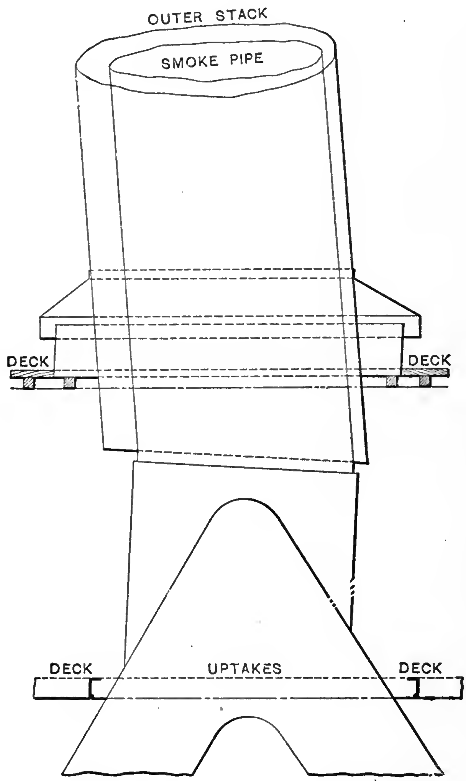


Fig. 77. Funnel.

craft a damper is often fitted in the funnel near the base, to assist in controlling the draft.

[4] Construction of Water-Tube Boilers.

Only a few points will require special notice under this heading. We have already seen that many types of water-tube boiler consist of one or more cylindrical drums above and one or more below, joined by a series of tubes. See Figs. 15-26. These drums, which are rarely more than 18 to 24 in. diameter, are made from steel plates usually by flanging and riveting in the usual manner. The heads alone of such drums require consideration as regards bracing. If of sufficient size to require it they may be braced by through bolts as with boiler heads. In most cases, however, the heads are bumped or made of dished form, either concave or convex on the outside. The latter is preferable, as the pressure is then carried on the concave side, and according to the United States law such heads are allowed without bracing a pressure the same as that for a cylindrical shell of a diameter equal to the radius of the sphere of which the head forms a part.

It is much preferable to form the heads in this way, avoiding the need of bracing, and thus leaving the interior of the drum free for examination, at least so far as bracing is concerned. To allow access to the interior, manholes or handholes with appropriate cover plates are fitted to the heads. Instead of forming these drums with riveted joints, drums with welded joints have recently come somewhat into use, especially in naval practice.

With boilers having headers formed by the space between two parallel sheets, the necessary arrangements are quite different. These sheets require special support, and this is usually provided by screw stay-bolts or other equivalent stays worked between the two sheets attached to the tube sheet between the tubes as convenient, and securely tying the two sheets together. In some cases the outer sheets are supported by rod stays passing from head to head through the tubes. In such case the tube sheets are left to be supported by the tubes which are thus thrown into compression, and the tubes must, therefore, be carefully expanded, especially on the inner side of the sheet, in order to give sufficient hold to support the sheets in this direction.

In Sec. 14 [6] in speaking of the operation of water-tube boilers, reference was made to a separation in the upper drum of the water and the steam as they are delivered from the upper ends of the tubes. This is usually effected by some form of baffle plate, as illustrated in Fig. 17. A plate pierced with small holes is placed just in front of the tube openings, and against this the escaping jets of water and steam are directed. The water is supposed to collect on the plate and to run down to the lower edge, or to the water in the lower part of the drum, while the steam passes through the holes and enters the steam pipe beyond. In the Bellville boiler, as in Fig. 26, there is a series of baffle plates forming a more or less tortuous passage through which the steam must pass on its way to the outlet, while at the last there is a plate with holes which exercise a straining action in the manner described just above. Special separators, as described in Sec. 35, are sometimes fitted in addition to these internal separating devices.

The tubes of water-tube boilers are of wrought iron or steel, and welded or solid drawn. For the bent-tube boilers solid drawn steel tubes are to be preferred. For straight tube boilers welded iron tubes are still in common use. The tubes are secured to the tube sheets either by expanding or by special fittings with screwed joints. In general there is a force tending to draw the tubes out of the tube sheet or junction box or other form of header, equal for each tube to its cross-sectional area multiplied by the steam pressure. This force must be resisted by the tube fastening, and while it is not usually serious in amount, its existence should not be forgotten, and the need of care in the fastening is shown.

The furnaces of water-tube boilers are formed of grate-bars with a space below for ash-pit, all inclosed in the same general casing which surrounds the boiler as a whole, and as shown in the various figures referred to in the foregoing.

Often a considerable amount of fire-brick is used as a lining to the furnace, and for protection to the lower ends of the tubes.

Due to the great variety of forms of water-tube boilers, the details of construction often present the widest variation, and they cannot be so readily reduced to standard forms as in boilers of the fire-tube type.

[5] Common Sizes and Dimensions of Scotch Boilers.

The furnace diameter for Scotch boilers is usually found between 42 and 48 inches. The upper limit comes about in the following manner. Taking into account the extreme ranges of temperature to which this part of the boiler is subjected, and based on general experience, it is usually considered that from $\frac{1}{2}$ to $\frac{5}{8}$ inch is about as far as it is desirable to carry at present the thickness of the metal for the furnace.

Again the strength of the furnace increases with the thickness and decreases with the diameter. Hence for a given pressure and limit on the thickness, the diameter will be limited as well. With modern pressures and a general limit on the thickness as above, the limit on the diameter, therefore, results. Rarely furnaces are met with up to 54 in., but only with the more moderate steam pressures. The lower limit for furnace diameter is given by a consideration of the necessary space between the fire and the furnace crown. If this space or height is not sufficient the fires cannot be properly worked and the combustion will be incomplete, due to insufficient space for the admixture of air with the gases given off from the coal.

In fact for efficiency of combustion we should probably prefer the diameter of the furnace larger than we are actually able to fit. As a lower limit, however, it may be considered inadvisable to fit furnaces much smaller than 42 inches, though they are sometimes found down to 36 inches.

The length of fire grate is usually found between 5 and 6 feet, though occasionally it extends to 6 ft. 6 in., or may be found as short as 4 ft. 6 in. The chief limitation here comes from the limit in the capacity of the average fireman to efficiently work his fire beyond a certain length. For average practice 5 ft. 6 in. may be considered a good length, while it is doubtful if grate area added beyond this length will be of any great value for steam production. It is more than likely to become partially choked with ashes and clinker, while a shorter grate of 5 ft. or 5 ft. 6 in. length may be kept bright and efficient over its entire surface.

The length of the furnace itself being equal to the tubes will be somewhat longer than the grate. The difference is usually from 12 to 24 or even 30 inches.

This gives for the usual length of furnace and of tubes from 7 to 9 feet.

The usual depth of the combustion chamber is from 24 to 30 inches. This will usually give a suitable volume, and will also provide a sufficient space within which a man may swing a hammer or make use of such other tools as may be necessary in caring for the back ends of the tubes.

The usual thickness of the water leg or space between the back of the combustion chamber and the back head of the boiler is from 6 to 9 inches.

It will thus be seen that the usual length of a Scotch single-end boiler will be found between 10 and 12 feet; 10 ft. 6 in. and 11 ft. are quite common values.

Comparing the construction of a single and double end boiler it is clear that the length of the latter will be slightly less than twice the length of a single end. This gives for the usual length of a double-end boiler from 18 to 21 feet.

The diameter of the boiler will depend largely on the number of furnaces to be fitted, and, of course, on the steam pressure and on any limitations in the thickness of the plates employed. Modern four-furnace boilers are usually found between 15 and 17 feet in diameter. For three-furnace boilers the diameters will similarly range from 13 to 14 feet, while for two-furnace boilers the diameter may vary from 10 feet or less to 11 or 12 feet.

The usual diameter for tubes is from 2¼ to 3 inches. The smaller sizes are used with forced draft and the higher rates of combustion. For natural draft 2¾ and 3 inches are common sizes.

The thickness of boiler tubes is usually specified by sheet metal gauge number. Plain tubes are usually No. 8, 10 or 12, corresponding to .17 to .10 inch. Stay tubes are usually about No. 3 or ¼ inch in thickness.

[6] Common Proportions for Scotch Boilers.

Grate Surface (G. S.) 10 to 15 I. H. P. per sq. ft. G. S.

Heating Surface (H. S.) 2 to 5 square feet per I. H. P. or 25 to 40 square feet per sq. ft. G. S.

Coal Burned. 15 to 30 lbs. per sq. ft. G. S. per hour, or ½ to 1 lb. per sq. ft. H. S. per hour.

Water Evaporated. 6 to 10 lbs. per lb. of coal, or 4 to 10 lbs. per sq. ft. H. S. per hour.

Section of Passage Over Bridge Wall. 1-6 to 1-8 G. S.

Sectional Area of Tubes. 1-5 to 1-7 G. S.

Sectional Area of Funnel. 1-6 to 1-8 G. S.

Volume of Combustion Chamber. 3 to 4 cu. ft. per sq. ft. of G. S.

Steam Volume. .3 to .4 cu. ft. per I. H. P.

[7] Weights of Boilers.

A modern four-furnace single-end Scotch boiler will weigh in the neighborhood of 40 tons, or upward, while the water will weigh not far from 20 tons, making 60 tons or more for the boiler as a whole. A four-furnace double-end boiler will similarly weigh not far from 70 tons, while the water will weigh not far from 40 tons, making 110 tons more or less for the boiler as a whole. For a modern three-furnace single-end boiler the weights would be similarly about 25, 15 and 40 tons, respectively, for boiler, water and total, while for a three-furnace double-end boiler they would be about 45, 25, and 70 tons, respectively, for boiler, water and total. Wide variations, of course, are found in the weights of boilers, and the above figures are only given to show the general nature of the weights involved. The weight of boilers with and without water, per square foot of heating surface, has already been noted in Sec. 14. From these various figures and proportions it results that Scotch boilers may be expected to develop from 20 to 30 I. H. P. per ton according to conditions, while for water-tube boilers the figures will run from 30 or 40 for the heavier types to 60 or 70 and even more for the lighter types, and with extreme rates of forced draft.

[8] Western River Boat or Flue Boilers.

As noted in Sec. 14 these boilers are in common use on the western rivers of the United States. A few additional points may here be given regarding their construction and installation.

The length of such boilers varies from 20 to 30 feet, with a diameter of about 4 feet and with from 4 to 6 flues 10 to 14 inches in diameter. The shells are made up of several courses as shown in Fig. 14, the circumferential seams being single riveted and the longitudinal seams double riveted. The flues are usually made also in lengths, lap welded and telescoped together.

Such a boiler, for example, of 26 feet length and 47 inches diameter with six 10 inch flues has about 580 sq. ft. heating surface and will provide steam for about 275 I. H. P. in engines of the type commonly used and described in Sec. 24.

When such boilers are arranged in battery they are placed side by side and are usually provided with a single setting, thus giving a common furnace for the entire battery. The length of grate bar is short, being usually about 4 ft. 6 in. The boilers are furthermore usually connected by a steam drum on top and by one or more mud drums at the bottom. The steam drum for the size of boiler referred to above may be from 18 to 24 inches in diameter, connected by legs 12 to 16 inches in diameter and spacing the boilers so as to give flame room of 9 to 12 inches between the shells.

In order to make the setting of such boilers as light as possible the brick work may be kept down to a single thickness of fire brick supported by a sheet iron casing. The ash pan is preferably of steel plates lined with fire brick laid in cement. An interesting feature of the ash pan is the ash well which is frequently fitted. This consists of a 10 or 12 inch cylindrical passage leading from the surface of the ash pan down through the bottom of the boat, and through which the ashes are discharged overboard without further handling. The boiler fronts are of cast iron with suitable fire-door and ash-pit openings. The uptakes, and funnels, or chimneys as they are more commonly called, are made of heavy sheet iron and are supported by bracing carried down to the main deck beams which carry the boilers themselves.

As noted below the engines work non-condensing, and the exhaust, as a rule, is led first through the feed heaters to the "Doctor" as described in Sec. 24, and then to the base of the chimneys for forming a blast and forcing the combustion. Connections may also be provided for carrying the exhaust to an exhaust pipe leading to the air, and also in part to the stern wheel if desired, in order to prevent the formation of ice in cold weather.

Lever safety valves, as illustrated in Fig. 78, are usually employed on these boilers, while gauge cocks, fusible plugs, steam gauges, and blow-off cocks are provided in accordance with usual practice.

The steam piping is usually of lap-welded wrought iron with flanged joints. One of the chief features of western river practice is the flexibility of the boat under different conditions of lading, and the necessity for allowing for such flexibility in the connections between the boilers and engines, and between the engines and wheel. Between the boilers and engines this is usually provided by the introduction of long bends or special connections intended to allow for changes due to expansion, contraction, twisting, etc.

In addition to the flue boiler, as illustrated above, the direct fire tubular or locomotive type of boiler, as illustrated in Fig. 12, is sometimes used on the western rivers, each boiler being thus self-contained, and the brick-work setting of the flue boilers being dispensed with. The flue boiler, however, is the more used and must be considered as the typical boiler in this field of practice.

Sec. 17. BOILER MOUNTINGS AND FIRE ROOM FITTINGS.

[1] Safety Valves.

The purpose of the safety valve is to provide for the escape of the steam in case the pressure should tend to rise above the safe working limit for which the valve is set. There are two kinds of safety valves, known as *lever* and *spring* valves, according as the valve is kept down on its seat by a weight on a lever or by a powerful spring under compression. Fig. 78 shows the construction of the standard U. S. lever valve.

The valve itself has a plain conical face, and fits to a corresponding seat as shown. In its motion up and down it is guided by the double stem, so that it can by no means become jammed

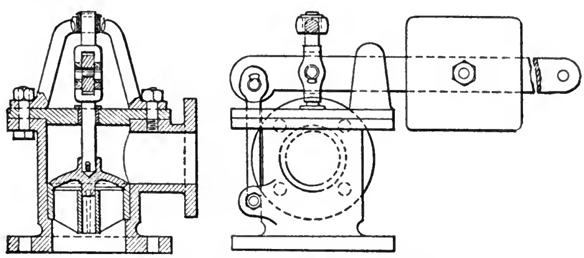


Fig. 78. Standard Lever Safety Valve.

in the chamber. The pressure of the steam comes on the bottom of the valve, and as it reaches or passes the limit for which the adjustment is made the valve lifts and the steam escapes about the edge, and thence is led by the escape pipe to the deck. The actual lift of a safety valve is very small, $\frac{1}{8}$ in. being usually a large lift. The opening for the escape of the steam depends, therefore, on the circumference of the valve and on the lift, rather than on the area and lift. Safety valves are, however, usually designated and determined according to their area. The weight acts by means of the lever, as shown, and may be adjusted so as to allow the valve to open at the pressure desired. On this point consult further Sec. 61.

In modern practice the lever valve is infrequently used, except in vessels engaged in smooth water (river) service, the spring valve being fitted almost universally. In this form of valve, which is shown in Fig. 79, the chief point of difference is in the substitution of the spring for the weight and lever. The tension of the spring is adjusted by a screw at the top, so that the valve will not open until the limiting pressure is reached or exceeded. It is readily seen that as soon as the valve rises the spring is compressed and the tension is increased. It is also found that with the plain form of valve, as shown in Fig. 78, the

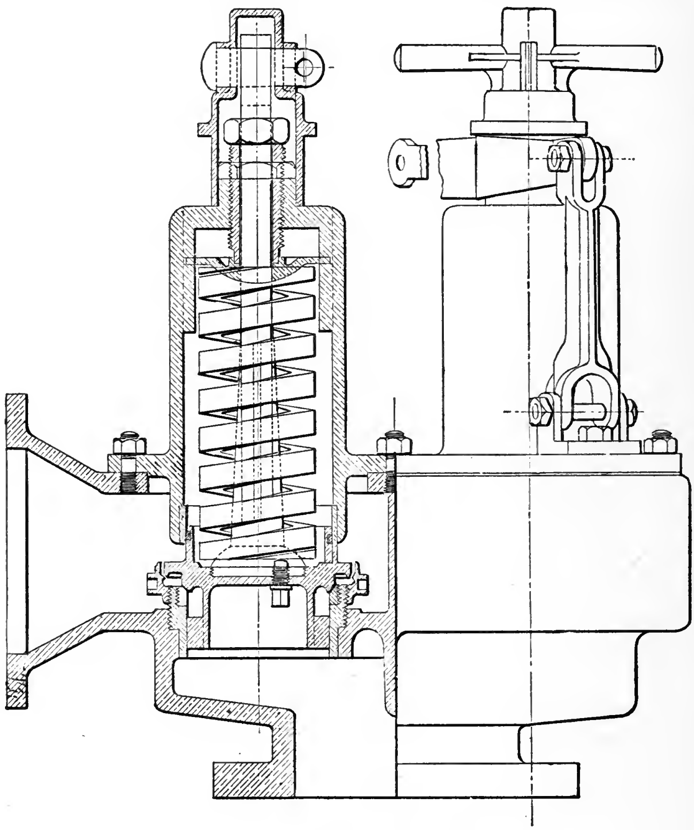


Fig. 79. Double Spring Safety Valve.

pressure on the face decreases the instant the valve lifts. Due to these facts it follows that such a valve, especially when controlled by a spring, is apt to seat itself the instant after rising, lifting again the next instant in answer to the restored value of the pressure. This irregular action will lead to a rapid opening and closing of the valve, producing a chattering noise very undesirable in passenger boats, and interfering with the continuous and regular escape of the steam. To avoid this a lip of one form or another, as shown in Figs. 79, 80 and 81, is fitted to the valve at or beyond the edge, so that it may catch the escaping jet of steam, and thus increase the effective area of the valve after it has lifted from the seat. In such case the valve is forced farther from the seat, and while it still vibrates, it remains definitely open until the pressure has fallen some 4 or 5 lb. below that for which it opens. The valve then touches the seat in one of its vibrations downward, and remains closed until the pres-

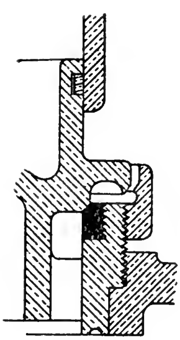


Fig. 80. Enlarged Section of Lip.

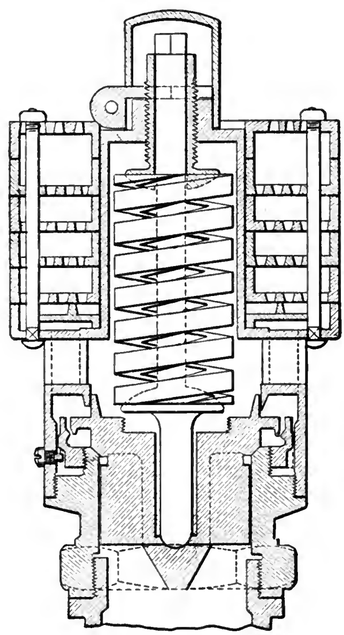


Fig. 81. Safety Valve and Muffler.

sure again rises to the point for which it is set. The safety valve should always be fitted with a hand lifting gear, so that it may be opened by hand when desired, and the spring adjustment should be protected by lock and key, so that it cannot be changed by unauthorized persons.

For large boilers, instead of one large valve, safety valves are often fitted in groups of two or three. This reduces to the smallest possible limit the danger from sticking or other derangement of the valve. The safety valve or valves should always be attached to a fitting leading direct to the boiler, and with no possibility of closing it off by a stop valve. If both stop and safety valves are attached to the same fitting the latter must always be placed inside or nearer the boiler than the former.

[2] Muffler.

This fitting, though not in the fire-room, may be properly referred to at this point. It consists of a metal chamber filled with bits of metal or stone, marbles, wire-gauze, small spiral springs, or with thin plates in layers pierced full of holes and arranged in staggered fashion so as to provide a series of zig-zag passages for the steam. The steam from the safety valves and escape pipe makes its way to the air through this chamber, the purpose of the filling being to muffle or deaden the noise, which might otherwise seriously interfere with the giving of orders on deck. Fig. 81 shows a combined safety valve and muffler, the latter with plates as above described. Such an arrangement would be applicable for small or open craft having but one boiler, such as launches, small yachts, etc.

[3] Stop Valve.

Each boiler is connected through a separate boiler steam pipe to the main pipe. The entrance of steam to this pipe is controlled by the boiler stop valve, which thus provides for the regulation of the supply of steam to the engine, and for closing the boiler off entirely from the main steam pipe if necessary. The usual type of valve employed is shown in Fig. 82, and consists of a valve disc guided to its seat by wings, and raised or lowered by its connection with the screw spindle and handle as shown. As also shown in the figure the valve seat with wings and guide for the valve is very commonly a separate piece of gun metal or bronze, specially fitted for its strength and wearing qualities.

Commonly in warship practice, and to some extent in mercantile practice, such valves are made self-closing in case of rupture of the boiler. In fundamental principle such a valve is a form of *non-return* valve, as illustrated by the check valve of Fig. 83. The screw stem does not open the valve, but limits simply the extent to which the valve can open. A second plain stem passing through the first then allows of the valve being pulled open by hand, even if there is no definite difference of pressure to force it open. In case of accident which reduces the pressure back of the valve so that the rush of steam is in the reverse direction to its usual flow, the valve will be closed by this rush and held securely on its seat by the excess of pressure on its outer face, thus shutting off the injured boiler, and retaining the others intact for use. If such an arrangement is not fitted and the valve cannot be closed by hand, or until it can be thus closed, an entire battery of boilers may be thrown out of

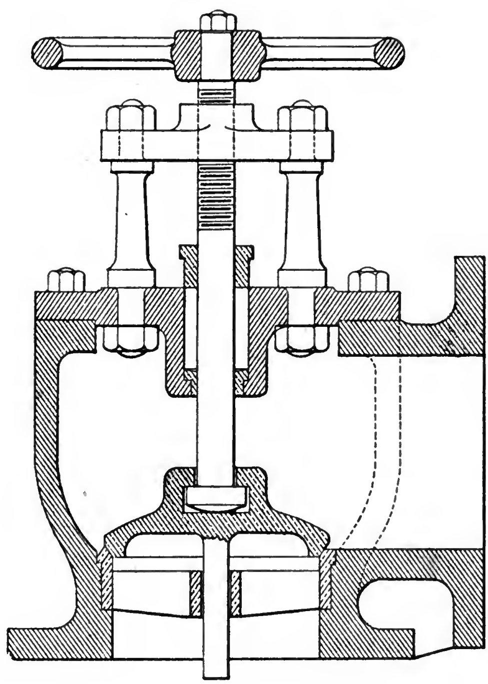


Fig. 82. Boiler Main Stop Valve.

use by the rupture of any one of them, all of the steam formed escaping through the one opening. Such form of valve should be placed with the spindle horizontal, so that its own weight may not enter as a direct factor in the movement of the valve toward and from its seat.

In warships, where the pipes or boilers may be pierced by the fragments of exploding shell, such a safety provision may be of the utmost importance and value.

[4] Dry-Pipe, or Internal Steam Pipe.

This is a pipe of relatively thin metal placed within the boiler, extending lengthwise, and close to the top of the shell. At the inner end it is closed, and at the outer end connects with the pipe leading to the safety valve chamber, stop-valve and boiler steam pipe. Along the top of the pipe are cut a large number of narrow slits, through which the steam enters the

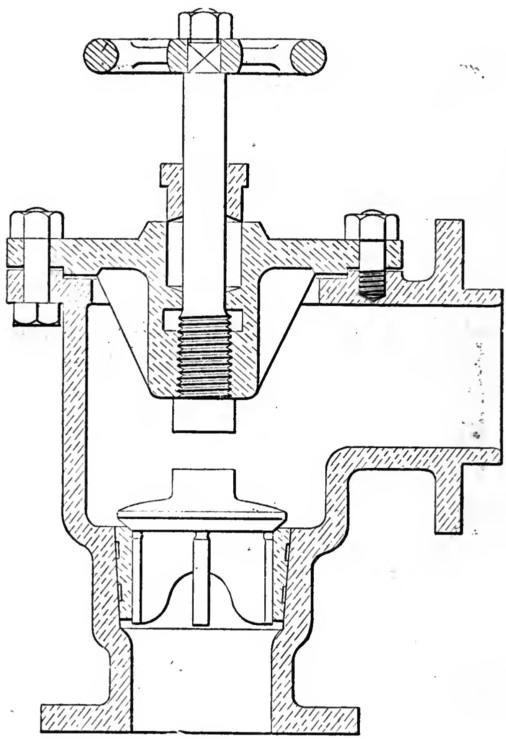


Fig. 83. Boiler Check Valve.

pipe. This arrangement has the effect of drawing the steam from the highest part of the steam space, and of straining out some part of the entrained water. A small hole in the bottom of the pipe provides for draining off the water which may gradually collect.

The uniform draft of steam from the whole length of the boiler tends also to prevent the priming which might be caused by drawing it all from one point.

[5] Feed Check Valve and Internal Feed Pipe.

The water from the feed pump comes to the boiler through the feed pipe, and then at the boiler passes through the feed-check. This is a screw-down, non-return valve, as shown in Fig. 83. The valve itself is entirely disconnected from the spindle, and the latter simply limits the height to which the valve can rise, while by screwing down sufficiently, the valve may be forced shut and held there. This construction is adopted so that should the feed pump stop working or between the strokes of the pump there may be no escape of water backward from the boiler into the pipe. Two such check valves are usually

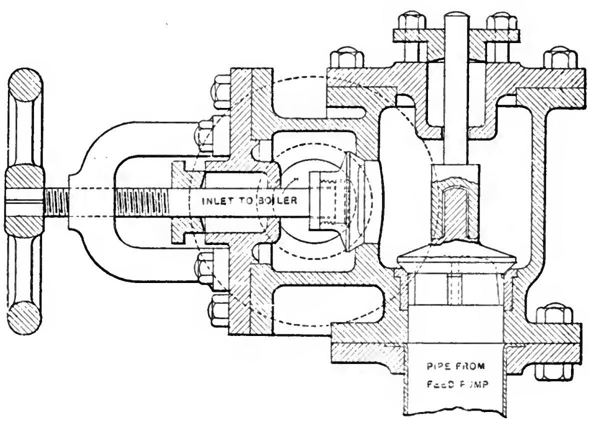


Fig. 84. Combined Check and Stop Valve.

fitted to each boiler, one connecting with the main and the other with the auxiliary feed pumps.

In modern practice a stop valve is usually fitted between the check valve and boiler, in order that, if necessary for examination or repair, the check may be shut off from communication with the boiler. Such combined stop and check valves are frequently fitted in a single casing, the stop valve, of course, being placed next the boiler (see Fig. 84.)

After passing through the check-valve the water enters the internal feed pipe, by which it is led to the point or points of delivery. The end of the pipe is usually closed, and the water is delivered through a large number of small holes distributed along the pipe. The delivery is usually below the water level, and often between the nests of tubes where it meets with the rising currents of water heated by them. In some cases it is led to the bottom of the boiler, where it mixes with the relatively cool water there found; but this plan cannot be recommended, as it retards rather than assists circulation. In some cases also the water has been introduced as a spray into the steam space, but, while this plan has some advantages, it has not met with general favor.

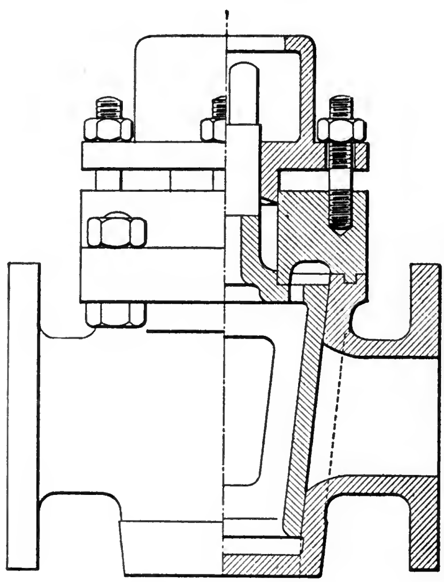


Fig. 85. Blow-off Cock.

In water-tube boilers the feed water is usually fed into the upper drum, whence it joins the circulation in the boiler as noted in the description of boilers of this type.

[6] Surface and Bottom Blows.

Cocks or valves and connecting pipes are fitted for blowing grease and scum, or mud sediment and water, out of the boiler into the sea—Fig. 85. The surface blow consists of a valve or cock attached to an internal pipe lying just below the normal water level, and either perforated with holes or leading to a shallow open pan. Outside the boiler there is a discharge pipe leading to an outboard valve, through which the discharge is effected. The scum and grease which collect on the surface of the water may by means of this arrangement be blown out of the boiler, and thus disposed of. In early engineering practice the bottom blow was of great importance, as it was used not only to discharge mud and sediment, but also the relatively dense water in the boiler when blowing down to reduce concentration, or when emptying the boiler of water for purposes of examination or cleaning. In modern practice with the surface condenser and the evaporator, blowing off to reduce concentration is no longer necessary, and blowing the water out of a boiler with its own steam is no longer considered good practice. The preferable plan is to allow the steam to condense and the water to cool down, and to then run it into the bilge or remove it by pump connections suitably arranged. Due to these facts, bottom blow valves have been sometimes omitted. There may still, however, be occasion to use such valves for the discharge of mud and sediment, and, therefore, they are still quite generally fitted. In any event, there should be some valve and pipe connected with the lowest part of the boiler, and through which it can be emptied in one way or another.

Both surface and bottom blows are usually fitted to water-tube boilers, especially to those types consisting of upper and lower drums with sets of connecting tubes. The surface blow is for scum and grease, while the bottom blow is essentially for mud and sediment, and is often attached to a special *mud-drum* provided to collect such substances.

In many cases the inner end of the surface blow pipe terminates in a shallow pan located near the low water level in the boiler. The water within this pan will tend to remain more quiet than that outside, and the scum and impurities will thus collect therein, ready for removal by the use of the blow. The arrangement thus serves as a collecting pan for the surface blow, and by most engineers is believed to be quite efficient for the purpose in view.

In other cases the pipe terminates in a closed end and is provided with a number of longitudinal slits through which the scum is drawn from the surface of the water.

The cross-sectional area of the bottom blow may be so proportioned as to give about one square inch for every 5 tons of water contained by the boiler. with perhaps somewhat larger area in the case of small boilers. The area for the surface blow may be usually made from ½ to 1-3 that of the bottom blow.

[7] Steam Gauges.

The steam pressure within the boiler, or rather the excess of the pressure within over the atmospheric pressure without, is shown by some form of steam gauge, of which the best-known and most used are those employing a *Bourdon tube*. In Fig. 86 is shown such a gauge and tube, the cross section of the latter

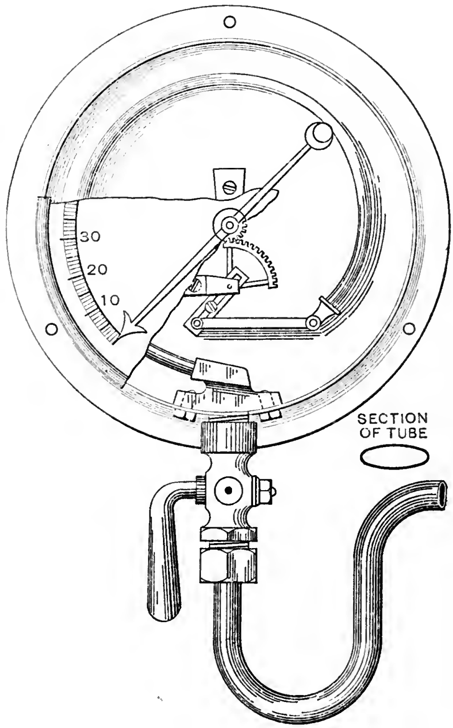


Fig. 86. Bourdon Steam Gauge.

being an ellipse as shown. When the inside of the tube is subjected to the pressure of the steam it tends to become round in section, and as a result of this the tube as a whole tends to straighten out. This carries the free end outward, and this movement, by means of suitable connections, is made to give motion to the needle. These gauges are graduated by comparison with a mercury column or other form of gauge tester, or with a standard gauge which has been thus graduated. Steam should not be allowed to enter these gauges, as the change in temperature may affect the accuracy of the reading. To prevent this the pipe leading to the gauge is always provided with a loop or U bend, called a "goose neck," which serves as a trap for the water condensed beyond this point. In this way the Bourdon tube and part of the connecting pipe are kept filled with water, which in turn is acted on by the steam, and thus the pressure is indicated without the actual presence of steam within the gauge. Steam gauges require comparison with a standard gauge from time to time, in order to make sure that their indications are correct. They are often provided in duplicate, and

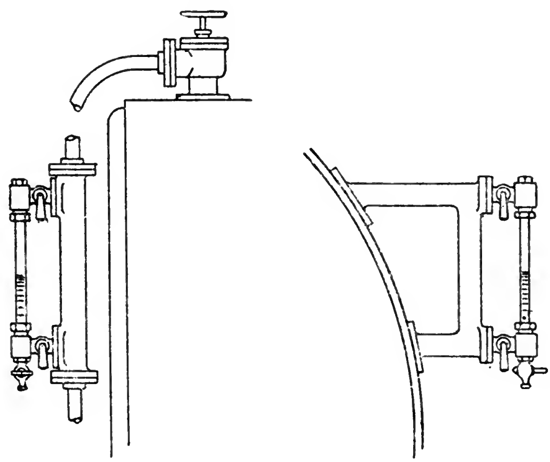


Fig. 87. Water Gauges.

frequently one gauge, at least, is provided of sufficient range to allow of use in the hydrostatic boiler test.

[8] Water Gauge and Cocks.

The level of the water within the boiler is shown by a vertical glass tube connected to fittings at each end, which in turn connect the one with the steam space and the other with the water space. As shown in Fig. 87*a*, the entire arrangement of glass and fittings is attached to a hollow mounting called the *stand-pipe, water-column*, or *water-gauge mounting*. To the top and bottom of this are attached pipes, one leading to the steam space at or near the top, and the other to the water space at or near the bottom. In connecting the pipe with the steam space care must be taken that the opening is not near a steam outlet, as the rush of steam past such an opening might disturb the pressure and render the indications inaccurate by showing a higher level of water in the glass than actually exists in the boiler. At the bottom of the mounting a drain cock and pipe are provided, so that the glass may be blown through and cleaned as occasion requires. Screw plugs are also fitted above and below in a line with the base of the tube, so that if necessary a wire and swab may be run through the glass. Instead of the connections, as shown in Fig. 87*a*, and which are to be considered as preferable, the ends of the water column are sometimes connected by horizontal passages directly to the boiler, as in Fig. 87b, which shows the fitting attached to the curved shell of boiler. With such a mounting the level of water in the glass is more liable to fluctuation and disturbance due to rolling of the ship or to priming than with the arrangement of *a*.

Gauge glasses are usually from 12 to 15 in. in length, and 5/8 to 3/4 in. diameter. Due to the fluctuations in temperature and the accompanying expansion and contraction, they are liable to occasional breakage. To avoid danger or trouble from the escaping jet of water and steam, it is quite customary in modern practice to fit the connection carrying the ends of the glass with ball non-return valves, working on a similar principle with the safety stop valve described above. So long as the glass is in place and the pressure equalized, the balls by their weight remain away from the seat and leave the passages open. Upon the breakage of the glass, however, they are carried by the rush of water and steam, each against its seat, thus closing the openings and stopping the escaping jets of water and steam.

In addition to the gauge glass, small cocks, three or four in number, are usually provided. In some cases such cocks are attached to the mounting, and in other cases to the boiler itself. These cocks serve as a check on the gauge glass, or for use in case the glass is not to be depended upon. The glass is usually so adjusted that when the water is at the bottom it is still some 3 or 4 in. above the level of the highest heating surface. The water cocks cover about the same vertical distance, though in some cases the lowest cock is placed nearly on a level with the top of the heating surface. On single-end boilers two such water gauges are often fitted, one on either side at the front, and with water cocks at the back. Similarly on double-end boilers three would be fitted, two on one end and one on the other.

[9] Hydrokineter.

This is an appliance used to force the circulation of the water in the boiler, more especially when raising steam. It consists, as shown in Fig. 89, of a steam jet and series of nozzles

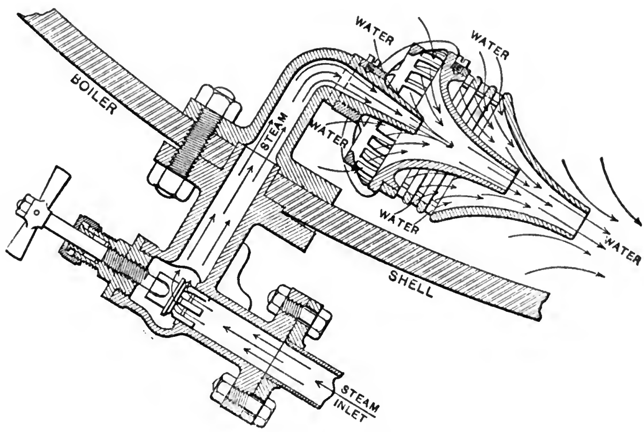


Fig. 88. Hydrokineter.

with frame perforated at the back for the entrance of water. The steam is furnished from another boiler, and by its inducing action a current of water is set up and driven along, as shown by the arrows. This arrangement is placed near the bottom of the boiler, and thus serves to drive out the cold water which tends to collect there, and which is only slowly heated by the operation of natural circulation.

[10] Hydrometer.

The density of the water in the boiler is determined by an instrument known as the *hydrometer*, and shown in Fig. 90. It

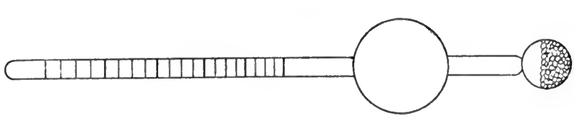


Fig. 89. Hydrometer.

may be of either glass or metal, and consists essentially of two bulbs with stem as shown. The upper and larger bulb is filled with air, and serves to give buoyancy to the instrument; while the lower and smaller bulb is weighted and keeps it in the upright position. When a body floats freely, wholly or partly immersed in a liquid, the weight of the body equals the weight of the liquid displaced. Hence, in this case the denser the water the less the volume displaced, and the higher the stem out of water. Average sea water contains about 1 part in 32 of solid matter, and hydrometers are usually graduated relative to this as a unit. That is, 2 means twice as much solid matter relatively as sea water; 3, three times as much, etc., while 0, of course, means fresh water. The density of water depends, furthermore, on its temperature, so that the scale on the hydrometer can only be used with the temperature for which it was graduated. This is usually 200 deg. F., though frequently three scales are provided; for 190 deg., 200 deg. and 210 deg., respectively. The water is drawn from the boiler through an appropriate pipe and connections into a deep, slender vessel called a *salimometer pot.* Soon after drawing, the water cools down through the temperature corresponding to the hydrometer scales, and thus its density is observed.

[11] Boiler Saddles.

The weight of the boiler is supported on *saddles,* or *bearers,* which in turn are attached to the structure of the ship. A modern form of boiler saddle is shown in Fig. 90, and consists

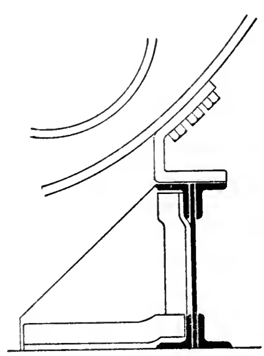


Fig. 90. Boiler Saddle.

of two or more supports on each side of the boiler of the form shown, and extending each one for some little distance longitudinally. An older form shown in Fig. 91 consists of a plate on edge connected with the structure of the ship, extending transversely under the boiler, and cut out to fit the round of the shell. The upper edge of this plate is fitted with angle irons on one or both sides to give a broader surface of support for the boiler. The form of saddle shown in Fig. 90 gives a better longitudinal support, and, moreover, makes access and examination of the bottom of the boiler more easy than with the other

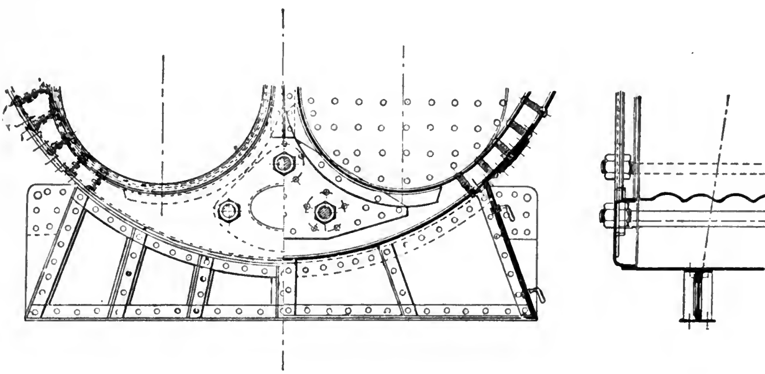


Fig. 91. Boiler Saddle.

form. Single-end boilers usually have two such saddles on each side, while double-end boilers are given three or four.

In addition, the boiler is held in place in its saddles by stays adjustable by screw turnbuckles, or by other like means. A knee-piece, or chock, is also often riveted to the structure of the ship, projecting just above the end of the boiler at the bottom, and thus preventing endwise motion.

[12] Boiler Lagging.

To prevent loss of heat by radiation the boiler is covered with non-conducting, non-combustible felting, which in turn is held on by iron straps, or in some cases by a complete covering of sheet metal. This covering is known as *boiler lagging.*

Sec. 18. DRAFT.

Draft is due to a difference in pressure between the uptakes or base of the stack, and the ashpits. Due to this difference the air is driven up through the grate, thus supplying the amount required for combustion, see Sec. 11 [2]. We must first inquire what it is that causes this difference of pressure. To make the case simple let AB Fig. 92, denote a grate with burning fuel, and ACDB the funnel. Then the pressure downward on the top of the fuel will be equal to the weight of a column of air and gas of cross section equal to AB, and extending up to the limits of the atmosphere. The pressure upward at the bottom of the grate will be the regular pressure of the external air, and this will equal the weight of a column of air of the same cross-sec-

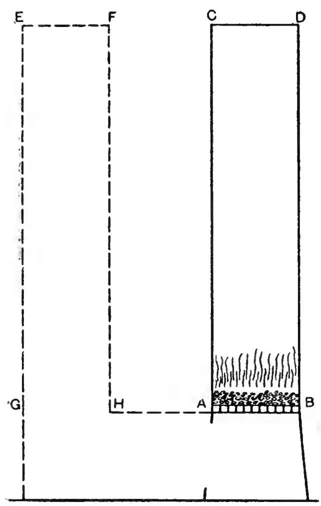


Fig. 92. Showing Principle of Natural Draft.

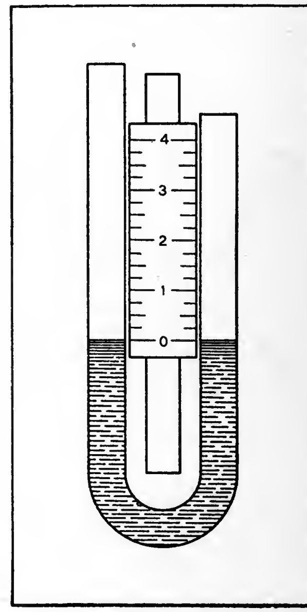


Fig. 93. Draft Gauge.

tion GH, and extending also up to the limit of the atmosphere. The difference in the weight of these two columns is seen to lie in their lower ends, the bottom of one being composed of hot gas and the other of common air. The difference in pressure will, therefore, equal the difference in weight between the column of hot gas CDBA, and that of air EFHG. Actually the column of hot gas will extend some distance above the top of the funnel before losing its heat or mingling with the air, so that the real height of the column is greater than the funnel. This difference is, however, usually neglected, and the difference in pressure is usually taken as the difference in weight between the column of hot gas extending from the grates to the top of the funnel, and a like column of external air. The pressure per square inch will, of course, be likewise equal to the difference between two similar columns of one square inch section. This shows the conditions for producing the so-called natural or funnel draft pressure.

In order that the combustion may proceed, however, it is not enough to produce simply a column of hot gas. Care must also be taken to provide for a free and full inflow of the outside air to the grates in order that the amount necessary for combustion may be on hand as required.

Draft pressures are usually measured by an instrument known as a draft gauge. As illustrated in Fig. 93, it consists of a bent U. tube partly filled with water as shown, and with a scale between the legs. In use the two legs are connected by appropriate means to the two places between which the difference of pressure is desired, as for example, the ash-pit and funnel-base, or external air and funnel-base. In the latter case one leg is open to the air and the other is connected by a flexible pipe or other similar means to the uptake or funnel-base. With equal pressure in both places the water will stand at the same height in both legs, but with a difference of pressure it will rise in one leg and fall in the other, the movement being toward the lesser pressure. The difference in pressure is then measured by the weight of a column of water equal to the difference in height between the two legs. This is usually read in inches, and hence draft pressures are usually expressed in inches of water. With ordinary funnel draft the pressure is usually from 1/4 to 1/2 or 3/4 inch, with assisted or light forced draft from 1/2 inch to 1 inch, with forced draft on large ships from 1 to 3 inches, while on fast yachts and torpedo boats the pressure may rise to 5 or 6 inches or more. In this connection it may be well to remember that an inch of water pressure is equal to a pressure of about 2-3 oz. per square inch.

Since as above explained, natural draft is dependent on the difference in weight between the hot gas in the funnel and the outside air, it follows that the lighter, and, therefore, the hotter the gas the stronger the draft; also the higher the funnel the greater the difference and the stronger the draft. A strong natural draft with moderate height of funnel requires, therefore, a high temperature of escaping gases, and since these carry away heat to the outside air, this means a loss of heat and hence of economy. Strong natural draft requires, therefore, either a very high funnel, or a very high temperature of escaping gases with the resulting loss in economy. The usual temperature of the gases in the funnel base is from 600 deg. to 800 deg. Fah. At a lower temperature the draft will be very poor, while with a higher temperature, the increase of draft will be obtained at the expense of economy. With natural draft the rate of combustion will usually range from 12 or 13 to 20 lb. of coal per square foot of grate surface, dependent on the quality of the coal and other circumstances.

From the preceding it is clear that with natural or funnel draft the power which can be obtained from a square foot of grate surface soon reaches a limit, and under present conditions this is usually found at from 10 to 12 I. H. P. In order to obtain more power per square foot of grate area, or in general more per pound of boiler, some form of assisted or forced draft is necessary. In all cases where very high speed is required as in torpedo boats, fast launches, yachts, etc., the application of forced draft is a necessity, as the boilers required to develop the power under natural draft would occupy far more weight and space than could be assigned them. With assisted or moderately forced draft the power per square foot of grate surface may be raised to from 15 to 18 I. H. P., and if properly installed, without sacrifice of economy. With harder forcing the power may be raised to 25 or 30 I. H. P. per square foot of grate surface, or even more in extreme cases, but necessarily at the expense of a loss in economy.

The immediate object of all forced draft appliances is to increase the difference in pressure between the ash-pit and uptake over what it would be with the funnel alone, at the same time taking care to provide for the full supply of air to the grates as required by the rate of combustion desired. To this end there are four fairly distinct means as follows:

(1) Closed fire room.

(2) Closed ash-pit.

(3) Exhaust fans in the uptakes, or between them and the funnel.

(4) Steam jets in base of funnel.

In the closed fire-room system the air is forced by means of blowers into the fire-room, which is closed air tight except for the outlet into the furnaces. The fire-room is hence under a pressure greater than the other parts of the ship, and to enter or leave it, an air lock is necessary, as illustrated in Fig. 94. Small air valves are provided by means of which the pressure in the lock may be equalized with that on either side, as may be desired, and this being done the door may be opened. To leave the fire-room, for example, both doors of the lock being closed, the pressure inside is equalized with the fire-room and the door being opened the person enters and closes it behind. The pressure in the lock is then equalized with that outside, the door is opened, and thus exit is effected. The chief advantages of this system lie in the fact that the boilers are left unchanged as for

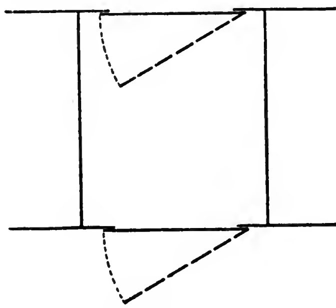


Fig. 94. Showing Principle of Air Lock.

natural draft, and the shift from one system to the other is readily made. The necessary structural arrangements are also sometimes more readily effected than for the other systems, especially in warship practice, and this reason may in some cases largely determine the choice. Its chief disadvantages lie in the difficulty of making the fire-rooms air tight, in the necessity of fitting air-locks as above described, and in the more severe strain placed on the fire-room force than with other systems.

In the closed ash-pit system the air is forced by means of blowers into conduits leading directly to the ash-pits and furnaces, which are closed air tight from the fire-room. The latter are, therefore, under a pressure greater than in the fire-room, and if the furnace doors were opened with the draft on, the flames and gas would be driven out into the fire-room. To prevent this the draft must be shut off when the furnace or ash-pit doors are opened, and to avoid accidents a locking arrangement is often provided which prevents the door from being opened while the draft is on, or the draft from being turned on till the doors are closed. The *Howden forced draft*, which is representative of this system, provides also for heating the air by means of the waste furnace gases before it enters the ash-pits and furnaces. The general arrangement for the Howden draft is shown in Fig. 95. In the uptake is fitted a nest of vertical tubes through which the furnace gases pass on their way to the funnel. The air from the blower is delivered into a conduit which leads

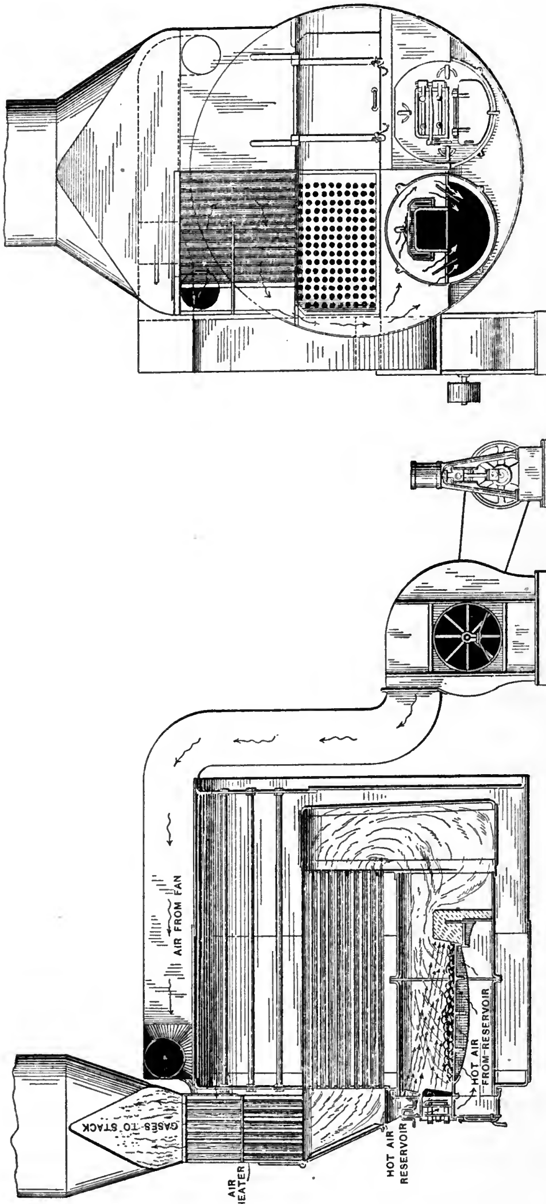


Fig. 95. Arrangement for Howden's Forced Draft.

across the front of the boiler and within which these tubes are placed, as shown in the figure. The furnace gases pass, therefore, through the tubes while the entering air passes about them on the outside, and thus absorbs a part of their heat which would otherwise escape through the funnel. The heated air then passes downward to a kind of special front over the ash-pits and furnaces. From these passages openings controlled by either sliding or hinged valves lead into the ash-pits and into the furnaces above the grates. The supply of air to the fire may thus be regulated, and the relative amounts delivered above and below the grate may be adjusted as required for the best combustion. The ash-pit and furnace doors are, of course, air tight to the fire-room, and arrangements may be provided for insuring the closure of the valves before opening the doors as above explained.

Induced draft is represented by the *Ellis and Eaves* system, as illustrated in Fig. 96. A large exhaust fan is so placed as to draw the gases along the uptakes and discharge them to the funnel, thus producing the draft by means of a defect of pressure in the uptakes, rather than by an increase in the ash-pit or fire-room. Considering the uptakes and funnel base as representing the main passage, the fan is so placed as to draw from the former and deliver into the latter. Between the inlet and delivery is a slide or damper in the main passage, while the fan inlet may similarly be closed off. When the blower is in operation the former of these is closed while the latter is open and the products of combustion are thus drawn from the uptake and delivered to the funnel-base on the other side of the main slide. On the other hand, if it is desired to run without the fan the inlet may be closed off and the main slide or damper opened, thus giving the usual arrangement for natural draft.

The entering air which in this case is supplied either by natural ventilation or by special blowers is heated before reaching the furnaces usually by being drawn around nests of tubes through which the gases pass on their way to the fan. These tubes are sometimes arranged vertically in front of the boiler as in the Howden system, and as shown in the figure, and sometimes horizontally in the spandrels over or between the boilers. The air thus heated is then led through passages at the side of the front connections to the ash-pits, and to the spaces around the furnace frames. The furnace and ash-pit fronts are closed

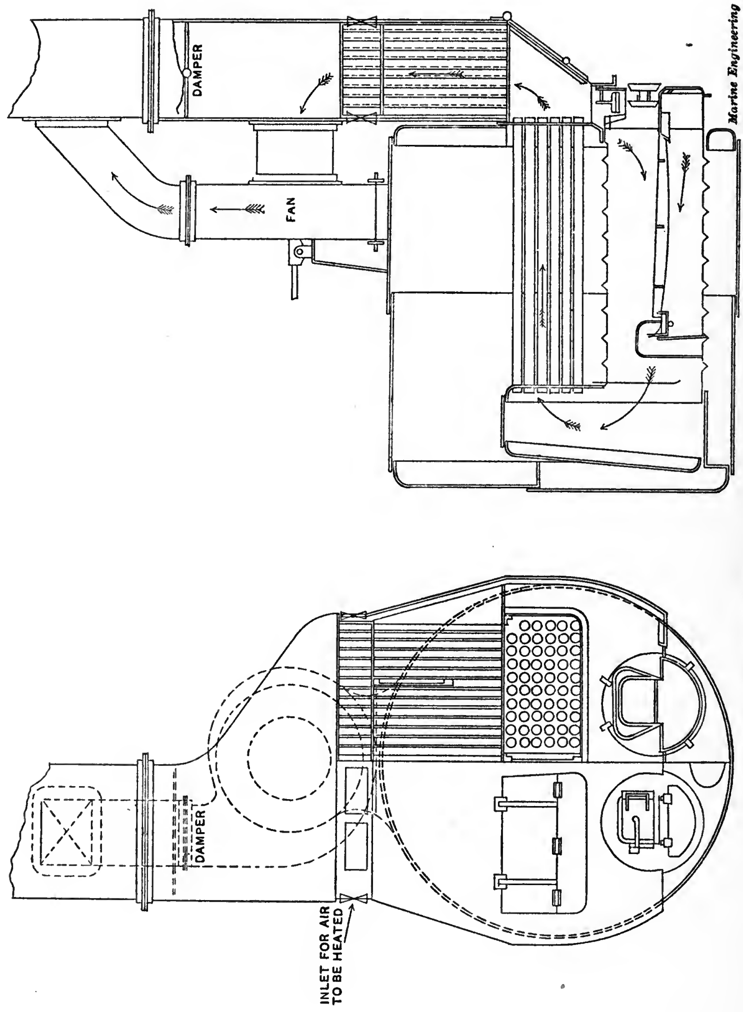


Fig. 96. Arrangement for Ellis and Eaves Forced Draft.

from the fire-room, and a certain proportion of the air is admitted above and below the grates according to the needs of the combustion. This system has the advantage of leaving the fire-room open and of working the ash-pit under practically atmospheric pressure. Furthermore all leaks between the fire-room and the fire side of the boiler are inward, and thus the fire-room is kept free from escaping gas, or from flame and gas when the furnace doors are opened. Its chief disadvantage lies in the large size and weight of fan necessary to handle the gases, as compared with the smaller size needed for the air alone in the closed fire-room and closed ash-pit systems.

The action of steam-jets in the base of the funnel is to produce a defect of pressure in the uptake, thus giving a form of induced draft. Turning the exhaust of a non-condensing engine into the funnel produces the same result, and may also be considered as a form of induced draft. The latter arrangement is sometimes met with in tugs and other small craft, while the steam jet is much used throughout the whole range of tugs, yachts, launches, tenders and all forms of small craft. One of the special advantages of the steam jet is its readiness for use as soon as a small head of steam is formed, and independent of any special auxiliary machinery. It is, however, a wasteful and expensive mode of obtaining increased combustion, and is only to be recommended when simplicity and the saving of weight and space are of more importance than economy of steam.

In this connection it must not, of course, be forgotten that the operation of the blowers used in the closed fire-room, closed ash-pit, or induced systems of draft requires also the consumption of steam and hence of coal, so that in no case can forced draft be obtained without paying for it in one form or other. General experience shows, however, that for a given increase in the rate of combustion, the use of a steam jet requires more steam than blowers, so that the latter are to be preferred except in such special cases as are referred to above.

Reference may also be made to the practice of introducing jets of air under considerable pressure into the combustion chambers or furnaces of steam boilers. The result of such an arrangement is two fold:

(1) It places the air at the immediate point where it may be of the most value in aiding to complete the combustion.

(2) Its introduction under pressure insures its thorough

mingling with the gases and the latter with each other, and thus still further aids in bringing about the conditions necessary for complete combustion. The introduction of air in this manner has met with considerable favor in many cases where it has been tried, especially in certain forms of water-tube boilers where the volume available for the combustion of the gases before they pass among the tubes is often limited or insufficient in amount.

Sec. 19. BOILER DESIGN IN ACCORDANCE WITH THE RULES OF THE U. S. BOARD OF SUPERVISING INSPECTORS OF STEAM VESSELS.

In the present section extracts are given from the United States Rules relating to the design of Marine boilers. To give here the Rules entire would require more space than is available, but those of chief importance are given, though in some cases the wording is slightly changed to aid in condensation.

Rivet Holes to be Drilled.

All boilers built for marine purposes after July 1, 1898, shall be required to have all the rivet holes "fairly drilled" instead of punched.

Pressure Allowed on Cylindrical Shell Boilers.

Rule.—Multiply one-sixth (1-6) of the lowest tensile strength found stamped on any plate in the cylindrical shell by the thickness—expressed in inches or parts of an inch—of the thinnest plate in the same cylindrical shell, and divide by the radius or half diameter—also expressed in inches—and the quotient will be the pressure allowable per square inch of surface for single riveting, to which add 20 per cent for double riveting, when all the rivet holes in the shell of such boiler have been "fairly drilled" and no part of such hole has been punched.

Test Pressure.

The hydrostatic pressure applied must be in the proportion of 150 pounds to the square inch to 100 pounds to the square inch of the steam pressure allowed.

Butt Straps.

Where butt straps are used in the construction of marine boilers, the straps for single butt-strapping shall in no case be less than the thickness of the shell plates; and where double butt straps are used, the thickness of each shall in no case be less than five-eighths (5/8) the thickness of the shell plates.

Stays and Flat Surfaces.

In allowing the strain on a screw stay bolt, the diameter of the same shall be determined by the diameter at the bottom of the thread.

No braces or stays hereafter employed in the construction of boilers shall be allowed a greater strain than six thousand (6,000) pounds* per square inch of section, and no solid or hollow screw stay bolt shall be allowed to be used in the construction of marine boilers in which salt water is used to generate steam unless said screw stay bolt is protected by a socket. But such screw stay bolts without socket may be used in staying the fire-boxes and furnaces of such boilers, and elsewhere when such screw stay bolts are drilled at each end with a hole not less than 1/8 inch diameter to a depth of at least 1/2 inch beyond inside surface of sheet, when fresh water is used for generating steam in said boilers (to take effect on and after July 1, 1898, on all boilers contracted for or construction commenced on or after that date). Water used from a surface condenser shall be deemed fresh water. The flat surface at back connection or back end of boilers may be stayed by the use of a tube, the ends of which being expanded in holes in each sheet beaded and further secured by a bolt passing through the tube and secured by a nut. An allowance of steam shall be given from the outside diameter of pipe. For instance, if the pipe used be 1 1/2 inches diameter outside, with a 1 1/4 inch bolt through it, the allowance will be the same as if a 1 1/2 inch bolt were used in lieu of the pipe and bolt. And no brace or stay bolt used in a marine boiler will be allowed to be placed more than 10 1/2 inches from center to center to brace flat surfaces on fire-boxes, furnaces, and back connections; nor on these at a greater distance than will be determined by the following formulas.

Flat surfaces on heads of boilers may be stiffened with doubling plate, tees, or angles.

The working pressure allowed on flat surfaces fitted with screw stay bolts riveted over, screw stay bolts and nuts, or plain bolt with single nut and socket, or riveted head and socket, will be determined by the following rule:

When plates 7-16 inch thick and under are used in the construction of marine boilers, using 112 as a constant, multiply

* This limit is understood to refer only to the use of *iron* as a material.

this by the square of the thickness of plate in sixteenths of an inch. Divide this product by the square of the pitch or distance from center to center of stay bolt.

Plates above 7-16 inch thick, the pressure will be determined by the same rule, excepting the constant will be 120.

On other flat surfaces there may be used stay bolts with ends threaded, having nuts on same, both on the outside and inside of plates. The working pressure allowed would be as follows:

A constant 140, multiplied by the square of the thickness of plate in sixteenths of an inch, this product divided by the pitch or distance of bolts from center to center, squared, gives working pressure.

Flat part of boiler-head plates when braced with bolts having double nuts and a washer at least one-half the thickness of head, where washers are riveted to the outside of the head, and of a size equal to ⅞ of the pitch of stay bolts, or where heads have a stiffening plate either on inside or outside covering the area braced, will equal the thickness of head and washers, the head and stiffening plate being riveted together * * * * shall be allowed a constant of 200, rivets to be spaced by thickness of washer on the stiffening plate. Boiler heads so reinforced will be allowed a thickness to compute pressure allowed of 80 per cent of the combined thickness of head and washer, or head and stiffening plate.

Plates fitted with double angle iron and riveted to plate with leaf at least two-thirds thickness of plate and depth at least one-fourth of the pitch would be allowed the same pressure as determined by formula for plate with washer riveted on.

But no flat surface shall be unsupported at a greater distance in any case than 16 inches, and such flat surfaces shall not be of less strength than the shell of the boiler, and able to resist the same strain and pressure to the square inch.

Steel stay bolts of a diameter of 1¼ inches and not exceeding a diameter of 2½ inches at the bottom of the thread may be allowed a strain not exceeding 8,000 pounds per square inch of cross-section. Steel stay bolts exceeding a diameter of 2½ inches at bottom of thread may be allowed a strain not exceeding 9,000 pounds per square inch of cross-section, but no forged or welded steel stays will be allowed.

Any steel stay brace of the Huston type, or similar thereto, prepared at one heat from a solid piece of plate without welds, intended for use in marine boilers, to be allowed a strain exceeding 6,000 pounds per square inch of cross-section, shall be tested as hereinafter provided for steel bars intended to be used as stay bolts; and any brace formed in this way, with an area of cross-section of 1.227 and not exceeding an area of 5 inches, may be allowed a strain not exceeding 7,000 pounds per square inch of cross-section; exceeding this area, may be allowed a strain not exceeding 8,000 pounds to the square inch.

All steel bars intended for use as stay bolts to be allowed a strain exceeding 6,000 pounds per square inch of cross-section shall be tested by the inspectors, in lots not to exceed fifty bars, in the following manner: Inspectors shall promiscuously select one bar from each lot and bend one end of such bar cold to a curve, the inner radius of which is equal to one and one-half times the diameter of the test bar; and should any such test bar break in the bending process the lot from which the bar was taken shall not be allowed to be worked into stay bolts for marine boilers.

Corrugated Furnace Flues.

Corrugated furnace flues constructed with corrugations 8 inches from center to center, the radius of outer corrugation being not more than one-half of the reverse or suspension curve, the plain parts of the ends not exceeding 9 inches in length, made of plates not less than five-sixteenths of an inch thick, when new, corrugated with practically true circles, shall be allowed a steam pressure in accordance with the following formula:

$$\text{Pressure in pounds} = \frac{15000}{D} \times T$$

Where T = thickness in inches.

D = mean diameter, in inches.

The strength of all corrugated flues, other than described in the preceding paragraph, when used for furnaces or steam chimneys (corrugation not less than 1½ inches deep, and not exceeding 8 inches from centers of corrugation) and provided that the plain parts at the ends do not exceed 9 inches in length and the plates are not less than five-sixteenths inch thick when new, corrugated, and practically true circles, to be calculated from the following formula:

$$\frac{14000}{D} \times T = \text{pressure.}$$

T = thickness, in inches.

D = mean diameter, in inches.

Ribbed Furnace Flues.

The strength of ribbed flues, when used for furnaces or steam chimneys (rib projections not less than 1⅜ inches deep), and not more than 9 inches from center to center of ribs, and provided that the plain parts at ends do not exceed 9 inches, and constructed of plates not less than seven-sixteenths inch thick, with practically true circles; and

The strength of corrugated flues, when used for furnaces or steam chimneys, corrugated by sections with flanged ends overlapping each other and riveted with ¾ inch rivets, 2 inch pitch, corrugated projection not less than 2½ inches from inside of flue to outside of lap, and not more than 18 inches between centers of corrugation, provided plain parts at ends do not exceed 12 inches in length, constructed of plates not less than 7-16 inch thick, with practically true circles; and

The strength of ribbed flues, when used for furnaces or steam chimneys, when made in sections of not less than 12 inches in length, measuring from center to center of said projections, and flanged to a depth not exceeding 2½ inches, and substantially riveted together with wrought-iron rings between such flanges, and such rings have a thickness of not less than double the thickness of the material in the flue, and a depth not less than 2½ inches, when straight ends do not exceed 12 inches in length, shall, in each of the above cases, be calculated from the following formula:

C = 14000, a constant.

T = thickness of flue in decimals of an inch.

D = diameter of flue in inches.

P = pressure of steam allowable.

Formula : $P = \frac{C \times T}{D}$

When plain horizontal flues are made in sections of less than 8 feet in length and flanged to a depth of not less than 2½ inches, and substantially riveted together with wrought-iron rings between such flanges, and such rings have a thickness of not less than half an inch and a width of not less than 2½ inches, or, in lieu thereof, angle-iron rings are employed, and such rings have a thickness of material of not less than double the thickness of the material in the flue and a depth of not less than 2½ inches, and substantially riveted in position with wrought-iron thimbles between the inner surface of the ring and the outer surface of the flue, at a distance from the flue not to exceed 2 inches, with rivets having a diameter of not less than one and one-half times the thickness of the material in the flue, and placed apart at a distance not to exceed 6 inches from center to center at the outer surface of the flue, the distance between the flanges, or the distance between such angle-iron rings, shall be taken as the length of the flue in determining the pressure allowable, which pressure shall be determined in accordance with the following formula:

$$P = \frac{89600 \times T^2}{L \times D}$$

Where P = pressure of steam allowable in pounds.
T = thickness of flue in decimals of an inch.
L = length of section in feet.
D = diameter of flue in inches.

All vertical boiler furnaces constructed of wrought iron or steel plates, and having a diameter of over 42 inches or a height of over 40 inches, and crown sheets of flat-sided furnaces, if made with a radius of over 21 inches, and all cylindrical shells of back connections having a radius of over 21 inches, shall be stayed as provided * * * * * for flat surfaces. But the cylindrical shell or bottom of back connections may be stiffened by angles or tees secured with rivets spaced no more than 6 inches from center to center, the distance from center of rivets at edge to center of tee, or from center to center of tees, not to exceed 24 inches, tees and rivets to be of suitable section for the pressure and radius of surface braced. And the thickness of material required for the shells of such furnaces shall be determined by the distance between the centers of the stay bolts in the furnace and not in the shell of the boiler; and the steam pressure allowable shall be determined by the distance from center of stay bolts in the furnace, and the diameter of such stay bolts at the bottom of the thread. Where steam chambers are formed in such vertical boilers at the upper end thereof by a sheet in form of a cone between the upper tube sheet and upper head of such boiler, the pressure allowed shall be determined by the diameter of such cone at the central point between the tube sheet and upper head of such boiler.

Steam chimneys or superheaters formed of a flue, with an inclosing shell, shall be built as follows:

The outer shell subject to internal pressure shall be constructed under rules governing the shells of boilers, without allowance for any bracing to lining or flue.

For Linings.

The lining of flue subject to external pressure shall be constructed as follows:

Plates under 30 inches in diameter shall be at least 5-16 inch thick.

Thirty inches and under 45 inches diameter, plates shall be at least ⅜ inch thick.

Forty-five inches and under 55 inches diameter, plates shall be at least 7-16 inch thick.

Fifty-five inches and under 65 inches diameter, plates shall be at least ½ inch thick.

Sixty-five inches and under 75 inches diameter, plates shall be at least 9-16 inch thick.

Seventy-five inches and under 85 inches diameter, plates shall be at least ⅝ inch thick.

Eighty-five inches diameter a corresponding increase in thickness of plate of 1-16 inch for every 10 inches increase in diameter.

The linings of flues shall be braced as follows:

On or for all boilers using salt water, carrying a steam pressure of 60 pounds and under per square inch, the lining shall be braced with socket bolts, with heads, and with ends of bolts threaded for nuts, with plate washers not over 12 inches between centers (or equivalent) on the inside of the lining; bolts to be at least 1 inch diameter.

On or for all boilers using salt water, carrying a steam pressure over 60 pounds per square inch, the lining shall be braced with socket bolts, with heads, and with ends of said bolts threaded for nuts, with plate washers not over 10 inches between centers (or equivalent) on the inside of lining; bolts to be at least 1⅛ inch diameter, the diameter of the bolts to be determined by the diameter at the bottom of the thread of said bolts.

On or for all boilers using fresh water, the lining may be braced as described for boilers using salt water, or as hereafter described (or equivalent thereto), viz., with iron or steel angle rings, properly riveted to lining, and properly connected to outer shell by plate braces. These plate braces shall be of sufficient number and width to make space between plates not over 20 inches on the lining; the angle rings shall be at least 2½ inches by 2½ inches on lining, 5-16 inch and ⅜ inch thick; 3 inches by 3 inches on linings, 7-16 inch and ½ inch thick; 3½ inches by 3½ inches on linings, 9-16 inch and ⅝ inch thick, and 4 inches by 4 inches on linings, 11-16 inch or more in thickness. *Provided, however*, That lining of steam chimney, between 24 inches and 32 inches diameter and ⅝ inch thick, and lining between 32 and 46 inches diameter, 11-16 inch thick, may be used in lengths not exceeding 8 feet, without bracing.

The pressure of steam to be allowed on linings shall be determined by the following formula, viz:

Constant, 89600.
D = diameter in inches.
T = thickness in decimals of an inch.
L = length in feet.
P = pressure of steam allowable in pounds.

Formula : $$\frac{89600 \times T^2}{L \times D} = P.$$

And the length of the lining or flue shall be the distance between center and center of angle rings, or center of angle rings to center of nearest row of rivets holding head, but in no case shall this distance be greater than 2½ feet, except as otherwise provided.

Corrugated or ribbed flues may be used as lining to steam chimney or superheaters under the same rules and conditions as apply to their use in the furnaces of steam boilers.

Crown Bars.

$$\text{Working pressure} = \frac{C \times d^2 \times T}{(W-P) \times D \times L}$$

Where W = Width of combustion box in inches.
P = Pitch of supporting bolts in inches.
D = Distance between girders from center to center in inches.
L = Length of girder in feet.

d = Depth of girder in inches.
T = Thickness of girder in inches.
C = 550 when the girder is fitted with one supporting bolt.
C = 825 when the girder is fitted with two or three supporting bolts.
C = 935 when the girder is fitted with four supporting bolts.

Boiler Heads (Special Class).

All heads employed in the construction of cylindrical boilers for steamers navigating the Red River of the North, and rivers whose waters flow into the Gulf of Mexico, shall have a thickness of material as follows: For boilers having a diameter exceeding 32 inches and not exceeding 36 inches, not less than half an inch; for boilers exceeding 36 inches in diameter and not exceeding 40 inches in diameter, not less than nine-sixteenths of an inch; for boilers exceeding 40 inches in diameter, not less than one-sixteenth of an inch additional thickness for every 8 inches additional diameter, required for boilers 40 inches in diameter.

And the heads of steam and mud drums of such boilers shall have a thickness of material not less than half an inch.

Bumped heads may have a manhole opening flanged inwardly, when such flange has sufficient depth and thickness to furnish as many cubic inches of material as was removed from the head to form such opening.

Bumped Heads of Boilers.

Multiply the thickness of the plate by one-sixth of the tensile strength, and divide by one-half of the radius to which head is bumped, which will give the pressure per square inch of steam allowed.

To find the radius of sphere of which the bumped head forms a part, square the radius of the head. Divide this by the height of bump required. To this result add height of bump. This will give diameter of sphere, one-half of which will be the radius required.

Unstayed Flat Heads.

The pressure on unstayed flat-heads, when made of stamped material, on steam drums or shells of boilers, when flanged and made of wrought iron or steel or of cast steel, shall be determined by the following rule:

The thickness of plate in inches multiplied by one-sixth of its tensile strength in pounds, which product divided by the area of the head in square inches multiplied by .09 will give pressure per square inch allowed. The material used in the construction of flat-heads when tensile strength has not been officially determined shall be deemed to have a tensile strength of 45,000 pounds.

When such heads are stayed or braced, the pressure allowed shall be determined as above for flat surfaces.

Pressure Allowable for Concaved Heads of Boilers.

Multiply the pressure per square inch allowable for bumped heads attached to boilers or drums convexly, by the constant .6, and the product will give the pressure per square inch allowable in concaved heads.

Manholes.

All manholes for the shell of boilers over 40 inches in diameter shall have an opening not less than 11 by 15 inches in the clear, except that boilers 40 inches diameter of shell and under shall have an opening of not less than 9 by 15 inches in the clear in manholes.

When holes exceeding 6 inches in diameter are cut in the boilers for pipe connections, man and hand hole plates, such holes should be reinforced, either on the inside or outside of boiler, with reinforcing plates, which shall be securely riveted to the boiler, * * * * * such reinforcing material to be of wrought iron or steel rings of sufficient width and thickness of material to equal the amount of material cut from such boilers, in flat surfaces; and where such opening is made in the circumferential plates of such boilers, the reinforcing ring shall have a sectional area of at least one-half the area of material there would be in a line drawn across such opening parallel with the longitudinal seams of such portion of the boiler. On boilers carrying 75 pounds or less steam pressure a cast iron stop valve, properly flanged, may be used as a reinforce to such opening. When holes are cut in any flat surface of such boilers, and such holes are flanged inwardly to the depth of not less than 1½ inches, measuring from the outer surface, the reinforcement rings may be dispensed with.

Also plates constructed of plate steel of corrugated form, without opening in plate for bolt, corrugation forming support for bolt, will be allowed for use for manhole and hand-hole openings.

No connection between shell of boiler and mud drum exceeding 6 inches in diameter will be allowed.

Safety Plugs.

All steamers shall have inserted in their boilers plugs of Banca tin, at least one-half inch in diameter at the smallest end of the internal opening, in the following manner, to wit: Cylinder boilers with flues shall have one plug inserted in one flue of each boiler; and also one plug inserted in the shell of each boiler from the inside, immediately below the fire line, and not less than 4 feet from the forward end of the boiler. All fire-box boilers shall have one plug inserted in the crown of the back connection or in the highest fire service of the boiler. All upright tubular boilers used for marine purposes shall have a fusible plug inserted in one of the tubes at a point at least 2 inches below the lower gauge cock, and said plug may be placed in the upper head sheet when deemed advisable by the local inspectors. All fusible plugs, unless otherwise provided, shall have an external diameter not less than that of a 1 inch gas or steam pipe screw tap, except when such plugs shall be used in the tubes of upright boilers, plugs may be used with an external diameter of not less than that of a three-eighths of an inch gas or steam pipe screw tap, said plugs to conform in construction with plugs now authorized to be used by this Board; and it shall be the duty of the inspectors to see that these plugs are filled with Banca tin at each annual inspection.

Gauge Cocks.

All steamers having one or two boilers shall have three suitable gauge cocks in each boiler. Those having three or more boilers in battery shall have three in each outside boiler and two in each remaining boiler in the battery; and the middle gauge cocks in all boilers shall not be less than 4 inches above the top of the flues, tubes, or crown of the fire box.

Safety Valves.

Lever safety valves to be attached to marine boilers shall have an area of not less than 1 square inch to 2 square feet of the grate surface in the boiler, and the seats of all such safety valves shall have an angle of inclination of 45 degrees to the center line of their axis.

The valves shall be so arranged that each boiler shall have one separate safety valve, unless the arrangement is such as to preclude the possibility of shutting off the communication of any boiler with the safety valve or valves employed. This arrangement shall also apply to lock-up safety valves when they are employed.

Any spring-loaded safety valves constructed so as to give an increased lift by the operation of steam, after being raised from their seats, or any spring-loaded safety valve constructed in any other manner so as to give an effective area equal to that of the aforementioned spring-loaded safety valve, may be used in lieu of the common lever-weighted valve on all boilers on steam vessels, and all such spring-loaded safety valves shall be required to have an area of not less than 1 square inch to 3 square feet of grate surface of the boiler, except as hereinafter otherwise provided for water-tube or coil and sectional boilers, and each spring-loaded valve shall be supplied with a lever that will raise the valve from its seat a distance of not less than that equal to one-eighth the diameter of the valve opening, and the seats of all such safety valves shall have an angle of inclination to the center line of their axis of 45 degrees. All spring-loaded safety valves for water-tube or coil and sectional boilers required to carry a steam pressure exceeding 175 pounds per square inch shall be required to have an area of not less than 1 square inch to 6 square feet of the grate surface of the boiler. Nothing herein shall be construed to prohibit the use of two safety valves on any water-tube or coil and sectional boiler, provided the combined area of such valves is equal to that required by rule for one such valve. But in no case shall any spring-loaded safety valve be used in lieu of the lever-weighted safety valve without first having been approved by the Board of Supervising Inspectors.

The first paragraph of this section applies to valves constructed in material, workmanship, and principle according to the drawings for a safety valve printed with these rules, and all common lever safety valves to be hereafter applied to the boilers of steam vessels must be so constructed.

Copper Steam Pipe.

All copper steam pipes shall be flanged to a depth of not less than four times the thickness of the material in the pipes, and all such flanging shall be made to a radius not to exceed the thickness of the material in such pipes. And all such pipes shall have a thickness of material according to the working steam pressure allowed, and such thickness of material shall be determined by the following rule:

Rule.—Multiply the working steam pressure in pounds per square inch allowed the boiler by the diameter of the pipe in inches, then divide the product by the constant whole number 8000, and add .0625 to the quotient; the sum will give the thickness of material required.

The flanges of all copper steam pipes over 3 inches in diameter shall be made of bronze or brass composition, shall be securely brazed to pipe, and shall have a thickness of material of not less than four times the thickness of material in the pipes plus .25 of an inch; and all such flanges shall have a boss of sufficient thickness of material projecting from the back of the flange a distance sufficient to be properly riveted to the pipe, and of a thickness of not less than one-half inch; and all such flanges shall be counterbored in the face to fit the flange of the pipe; and the joints of all copper steam pipes shall be made with a sufficient number of good and substantial bolts to make such joints at least equal in strength to all other parts of the pipe.

Steel and Iron Pipe.

The terminal and intermediate joints of all wrought iron and homogeneous steel feed and steam pipes over 3 inches in diameter, other than on pipe or coil boilers or steam generators, shall be made of wrought iron, homogeneous steel, or flanges of equivalent material; and all such flanges shall have a depth through the bore of not less than that equal to one-half of the diameter of the pipe to which any such flange may be attached; and such bores shall taper slightly outwardly toward the face of the flanges; and the ends of such pipes shall be enlarged to fit the bore of the flanges, and they shall be substantially beaded into a recess in the face of each flange.

But where such pipes are made of extra heavy lap-welded steam pipe up to and including 5 inches the flanges may be attached with screw threads; and all joints in bends may be made with good and substantial malleable iron elbows or equivalent material.

All feed and steam pipes not over 2 inches in diameter may be attached at their terminal and intermediate joints with screw threads by flanges, sleeves, elbows, or union couplings; but where the ends of such pipes at their terminal joints are screwed into material in the boiler, drum, or other connection having a thickness of not less than ½ inch, the flanges at such terminal joints may be dispensed with. Where any such pipes are not over 1 inch in diameter and any of the terminal ends are to be attached to material in the boiler or connection having a thickness of less than ½ inch, a nipple shall be firmly screwed into the boiler or connection against a shoulder, and such pipe shall be screwed firmly into such nipple. And should inspectors deem it necessary for safety they may require a jam nut to be screwed onto the inner end of any such nipple.

The word "terminal" shall be interpreted to mean the points where steam or feed pipes are attached to such appliances on boilers, generators, or engine, as are placed on such to receive them.

All lap-welded iron or steel steam pipes over 5 inches in diameter or riveted wrought iron or steel steam pipes over 5 inches in diameter, in addition to being expanded into tapered holes and substantially beaded into recess in face of flanges, shall be substantially and firmly riveted with good and substantial rivets through the hubs of such flanges, and no such hubs shall project from such flanges less than 2 inches in any case.

No cast iron nozzles, branch pipes, or elbows shall be used in connecting steam drums, superheaters, branch pipes, or steam pipes to boilers, and in no other part of steam pipes. Flanges welded to wrought iron, Bessemer, or other steel pipe may be used. No cast iron flanges will be allowed to be used on boilers for marine purposes unless such cast iron has been officially tested and test on record in the office of the local inspectors where boiler with such appliances was constructed, and no cast iron with a tensile strength of less than 30,000 pounds will be permitted to be used for such purposes. Semi-steel of not less than 24,000 pounds tensile strength may be used for nozzles, stop-valves, branch pipes, elbows, slip-joints, flanges to boilers, tee pipes, and water and gauge cock pipes or columns, when said semi-steel has been officially tested, and test on record in the office of the local inspectors, same as is required of cast iron.

Coil and Tubulous Boilers.

All coil and pipe boilers hereafter made, when such boiler is completed and ready for inspection, must be subjected at inspection to a hydrostatic pressure double that of the steam pressure allowed in the certificate of inspection—to take effect on and after July 1, 1897.

The use of cast-steel manifolds, tees, return bends, or elbows in the construction of pipe generators shall be allowed, and the pressure of steam shall not be restricted to less than one-half the hydrostatic pressure applied to pipe generators, unless a weakness should develop under such test as would render it unsafe in the judgment of the inspector making such inspection.

All drums attached to coil, pipe, sectional, or water-tube boilers not already in use or actually contracted for, to be built for use on a steam vessel, and its building commenced at or before the date of the approval of this rule, shall be required to have the heads of wrought iron or steel or cast steel, flanged and substantially riveted to the drums, or secured by bolts and nuts of equal strength with rivets, in all cases where the diameters of such drums exceed 6 inches.

Except steam drums not exceeding 15 inches diameter attached to coils or pipe generators may be used when heads are made of malleable iron or cast steel, said drums being threaded on outside of such shell with a good full U. S. standard thread, eight to the inch, for a distance of at least 1 inch on such shell, the thread on head to correspond with the same and well fitted; the end of shell projecting beyond the threaded part and screwed against a packing that will prevent water or steam to come in contact with the threaded part:

Provided, Such steam drums are placed outside of and not brought in contact with the heat or gases used in generating steam, and have been subjected to a hydrostatic pressure of double the steam pressure allowed.

Drums and water cylinders constructed with bumped head at each or either end, any opening in the shell or heads to be reinforced as required by the rules of the Board, the circumferential and horizontal seams to be welded and properly annealed after such welding is completed, and when tested with a hydrostatic pressure of at least double the amount of steam pressure allowed, may be used for marine purposes.

CHAPTER IV.

Marine Engines.

Sec. 20. TYPES OF ENGINES AND ARRANGEMENT OF PARTS.

The various types of marine steam engine may be classified in different ways, according to the particular feature under special consideration. A typical modern marine engine as in Fig. 98 may be defined as a *vertical, inverted, direct-acting, multiple-expansion, condensing,* engine. Let us first examine the significance of these various terms.

In the early days of marine engineering the engines were often horizontal, as shown in Figs. 101, 102, and such are still met with occasionally in special types of warship practice and elsewhere. An intermediate type, as shown in Fig. 103 and known as the inclined or diagonal engine, has been used to a considerable extent with paddle wheels. In modern practice, with rare exceptions, the marine screw engine is vertical, as in Figs. 97-100.

In the earlier vertical marine engines the cylinder was at the bottom and the motion of the parts proceeded upward either directly to the crank shaft, as in the oscillating engine, Fig. 104, or to a beam or intermediate mechanism, Fig. 105, whence it came back to the shaft. In the modern engine the cylinders are on top and the motion of the parts proceeds downward to the shaft. Hence in comparison with the earlier types the modern engine is called *inverted.*

Where the connecting-rod and crank lie beyond the cross head or farther end of the piston-rod, as in Figs. 97-101, the engine is said to be *direct-acting.* In certain early types of horizontal engines in single screw ships, as represented in Fig. 102, the cylinder was sometimes placed close to the shaft and two piston-rods were fitted passing beyond the shaft, one above and the other below. Then from a crosshead at this point the motion came back to the crank pin by a connecting-rod in the usual way. Such engines were called *return connecting-rod* or *back-acting*. In still earlier times the same type of engine placed on end, with the cylinder at the bottom, and known as the stee-

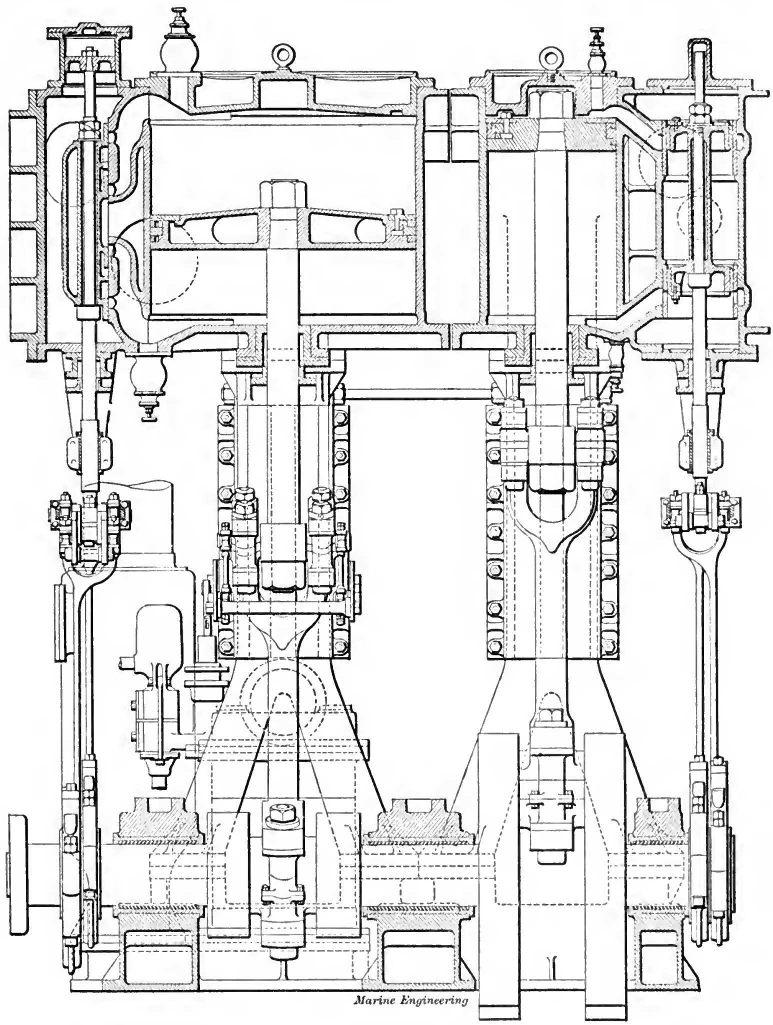


Fig. 97. Longitudinal Section, Compound Engine, Mercantile Type.

ple engine, was frequently fitted in side-wheel paddle steamers, and a modification of this is occasionally met with abroad at the present time.

In early marine engines the expansion of steam always

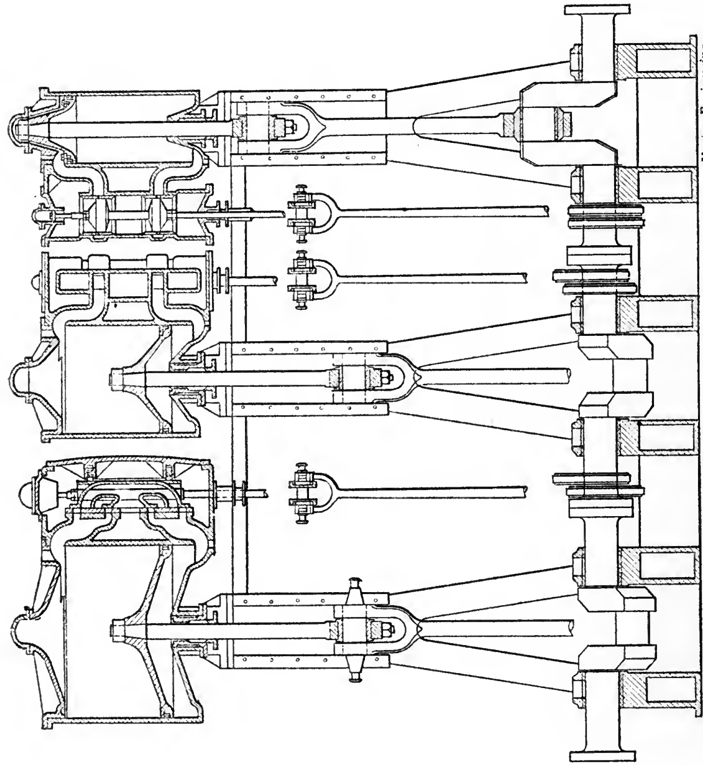


Fig. 98a. Triple Expansion Engine, Longitudinal Section.

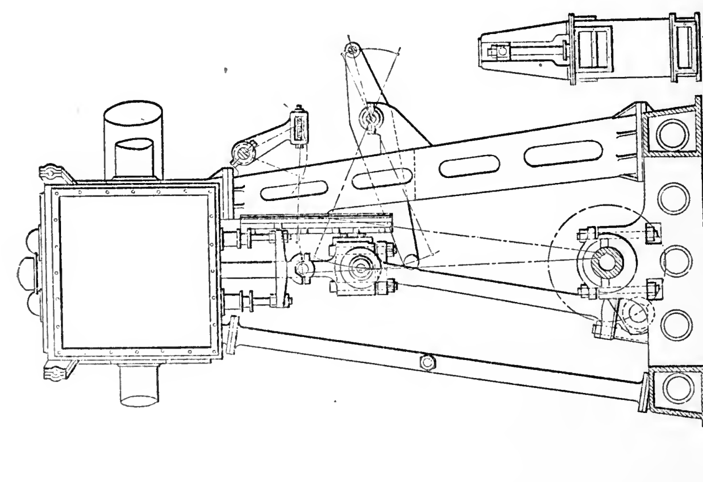
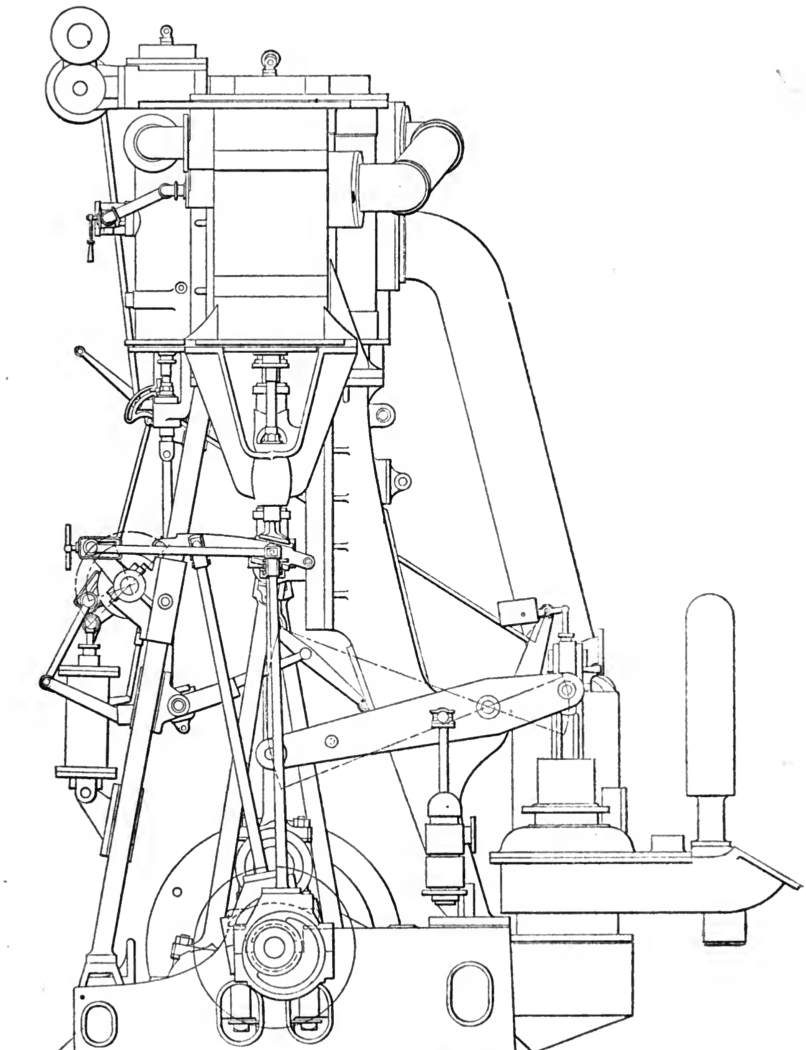


Fig. 98b. End View Looking Forward.

took place in one cylinder only. In the typical modern engine the steam is passed through a series of cylinders from one to another of increasing size. Such engines in general are termed



Marine Engineering

Fig. 99. Triple Expansion Engine, End View Looking Aft.

multiple expansion. If the steam is thus used successively in two cylinders or the expansion occurs in two stages, the engine is said to be a *compound;* if in three cylinders or three stages, it is a *triple* or *triple expansion;* if in four cylinders or four stages, it is a *quadruple* or *quadruple expansion,* etc.

Where the steam after being used in the cylinder is ex-

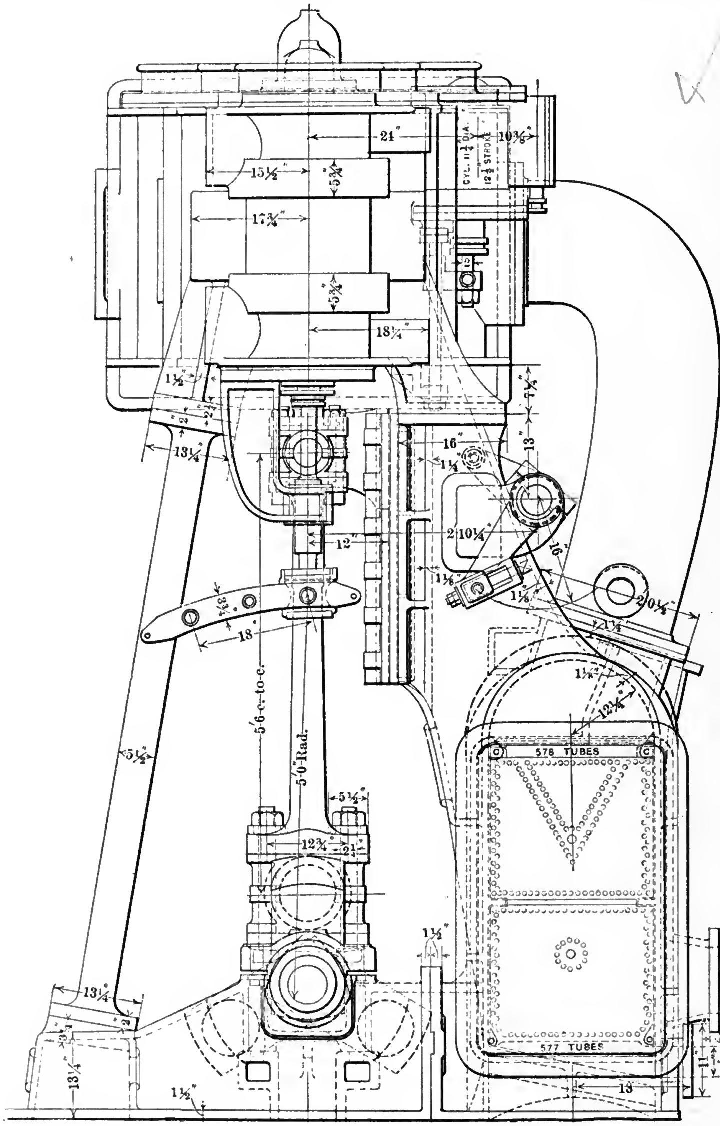


Fig. 100. Triple Expansion Engine; End View Showing Condenser in Back Framing.

hausted into the air, the engine is said to be *high-pressure* or *non-condensing.* In the typical modern engine the steam is exhausted to a condenser, thus giving the advantage of an increased ratio of expansion and a decreased back pressure. Such engines are called *condensing*.

Engines are often given special names according to the nature of the mechanical movements employed. In the usual type, as we have already seen, the motion is direct-acting and proceeds through piston, piston-rod, crosshead, connecting-rod,

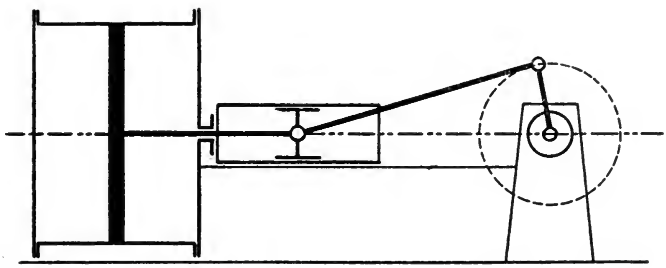


Fig. 101. Horizontal Direct Acting Engine, Outline.

crank-pin and crank-shaft. In the beam engine, as shown in Fig. 105, the motion passes from the piston-rod to a crosshead and then by link or parallel motion to the beam. Thence from the other end of the beam it passes by the connecting-rod to the crank-pin and crank-shaft. Such engines are especially suited to side-wheel paddle steamers, and for many years were

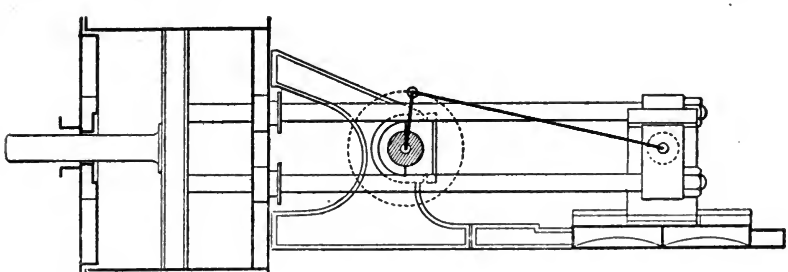


Fig. 102. Horizontal Back Acting Engine, Outline.

considered the standard engine for use on river, bay and lake steamers. In more recent practice, however, the vertical direct-acting engine with screw propeller is to a considerable extent displacing the beam-engine with paddle-wheel, even in its own territory.

In the oscillating engine, a favorite in British practice for side-wheel paddle steamers, the cylinders are located below the shaft and are swung on trunnions, as shown in Fig. 104. The piston-rod is connected directly to the crank-pin, the piston-rod and connecting-rod forming thus but one member. This motion is made possible by swinging the cylinder on trunnions, as may be readily seen by the diagram. The trunk type of horizontal engine, as shown in outline in Fig. 106, was often fitted in former years where economy of transverse or athwartship dimension was necessary. In this engine the use of the piston-rod was avoided by the large trunk, to which the connecting-rod was directly attached, as shown.

The stern-wheel western river boat engine, as shown in Fig. 155, is a direct-acting horizontal engine connected to the stern wheel, and provided with a peculiar type of valve gear.

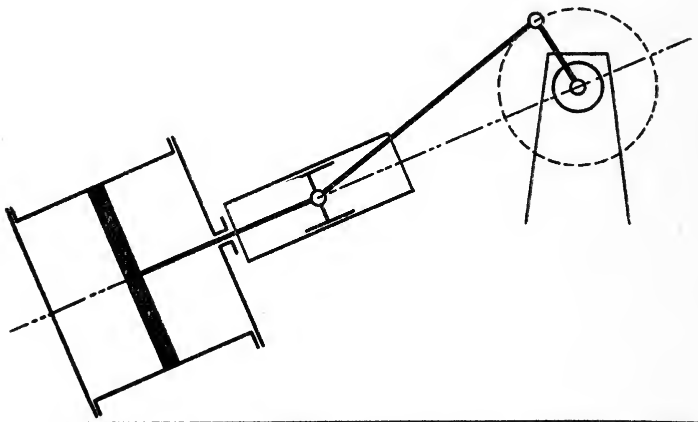


Fig. 103. Inclined Engine, Outline.

Further reference to some peculiar features of this engine will be made in Section 22.

The various members of a multiple-expansion marine engine may be arranged in a great variety of ways as regards the location of the cylinders, the crank angles, and the way in which the cranks follow each other around in the revolution. These are illustrated in Figs. 107-110. Of the many combinations which might be made, only the more important are mentioned. Throughout these diagrams the high-pressure cylinder is denoted by *H*, the low-pressure cylinder by *L*, the intermediate cylinder of a triple-expansion engine by *I*, and the first and second intermediates of a quadruple-expansion by I_1, and I_2, respectively. Where the total cylinder volume is divided between two, each of half size, both of the latter are given the same letter. The course of the steam through the engine is also indicated by the arrows. For compound engines the usual arrangements are illustrated in Fig. 108. We may have two or three cylinders and one, two or three cranks. In the latter case the entire volume of low-pressure cylinder is divided between two cylinders, each of half the total volume. The first arrangement with high-pressure cylinder on top of low-pressure is known as a single-crank *tandem* compound, but is rarely met

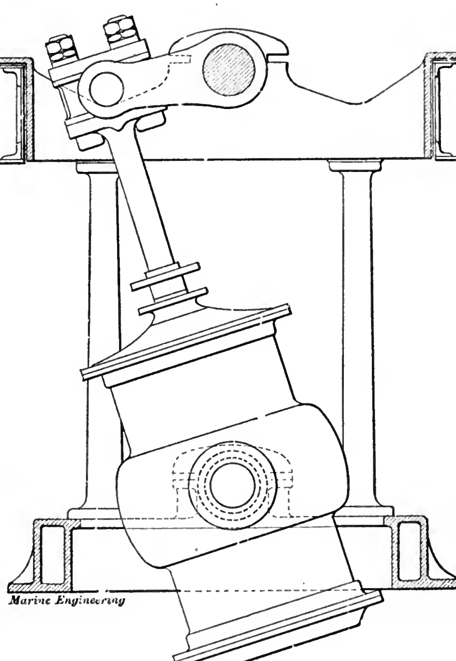


Fig. 104. Oscillating Engine, Outline.

with in marine practice. The other arrangements may be placed, of course, with either end forward. The various crank angles are shown in Fig. 107 at 1, 2, 3, 4 and 5, the crank marked *I* in No. 5 being in this case for one of the L. P. cylinders. With two cranks the angle between may be either 90 deg. or 180 deg., or slightly greater or less than 180 deg., as 175 deg. or 185 deg. The 90 deg. angle is undoubtedly the best for all-around service. The 180 deg. angle gives a better balance to the moving parts and admits of a simplification of valve gear, and is sometimes preferred for these reasons. There is, however, a liability of the engine's sticking on the center and the general readiness of handling is less than with cranks at 90 deg. To overcome this, angles of 175 deg. or 185 deg., as shown at 2, are sometimes used, the balance of moving parts in such case being substantially as good as with an angle of 180 deg.

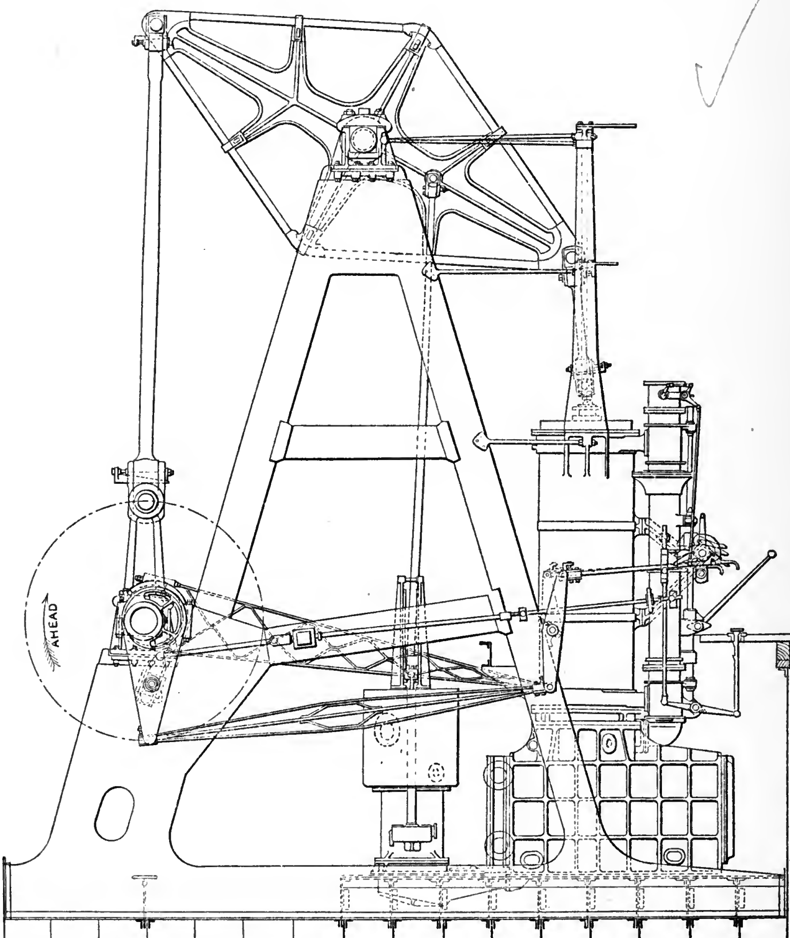


Fig. 105. Beam Engine, Side Elevation.

With three cranks the angles are usually equal, and hence 120 deg. each. Occasionally they are slightly varied from these values in order to give a more uniform rotative effort, or to give a better balance to the forces causing vibration.

For the triple-expansion engine the more important arrangements of cylinders are shown in Fig. 109. We may have three cylinders or more, and two, three or more cranks. The

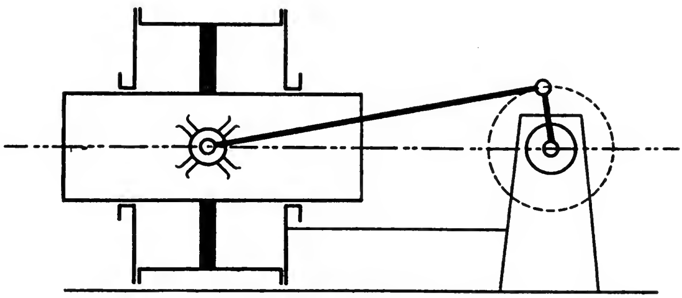


Fig. 106. Trunk Engine, Outline.

most common types have either three or four cranks, in the latter case the total L. P. volume being divided between two cylin-

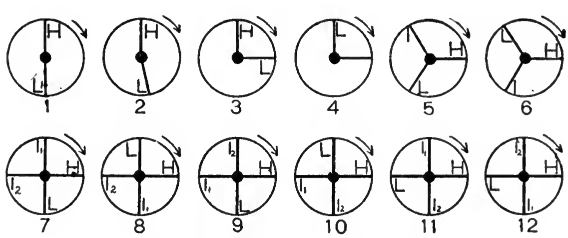


Fig. 107. Various Crank Angles.

ders, each of half the total volume. The crank angles are usually 120 deg. with three cranks, and 90 deg. with four,

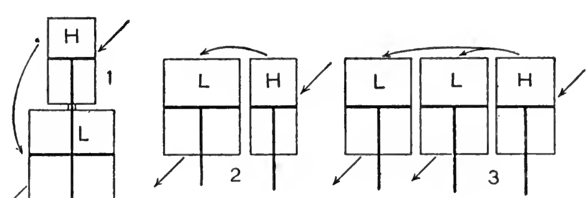


Fig. 108. Cylinder Arrangements for Compound Engine.

though occasionally slight variations from these values are adopted in order to obtain a better balance of the forces causing vibration. Of the various arrangements of cylinders shown in

Fig. 109, each may, of course, be placed either end forward in the ship. We may also have the various sequences and arrangements of cranks as indicated in Fig. 107, the changes of lettering where necessary being readily seen.

For the quadruple-expansion engine the more important cylinder arrangements are shown in Fig. 110. The number of

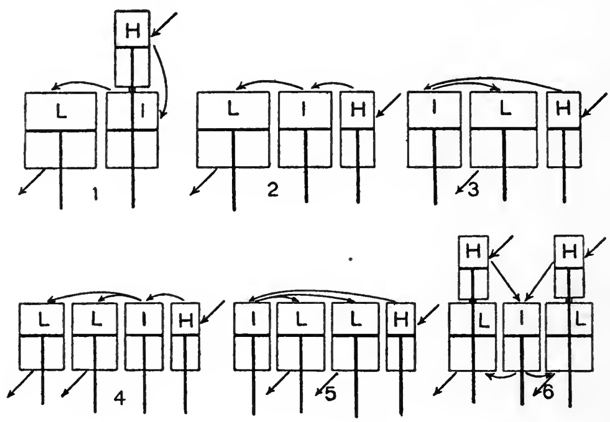


Fig. 109. Cylinder Arrangements for Compound Engine.

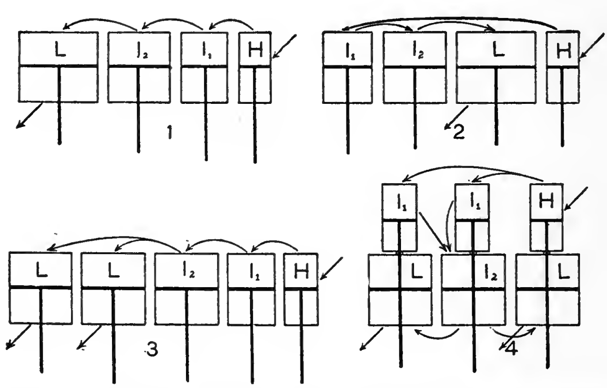


Fig. 110. Cylinder Arrangements for Quadruple Expansion Engine.

cylinders may be four, five or six, with four or five cranks. With five cranks the angles are usually equal, and hence of 72 deg., though as with three and four cranks slight departures might be made to obtain a better balance of the forces producing vibration. The arrangements of cylinders shown in Fig. 110 may be placed in the ship either end forward, and various crank sequences in addition to those shown in Fig. 107 may be easily arranged. One of the chief tendencies of modern practice is to pay especial attention to the balancing of the forces producing vibration. The use of irregular crank angles in this connection has been already referred to. In addition, and of not less importance, the larger cylinders with the larger and heavier pistons are now frequently placed inside, with the lighter moving parts on the outside, as in Fig. 109, Nos. 3 and 5, or Fig. 110, No. 2.

Sec. 21. DESCRIPTION OF PRINCIPAL PARTS OF A MARINE ENGINE.

In describing the principal parts of the typical modern marine engine, we may take first the stationary, and then the moving parts.

[1] Cylinders.

As shown in Figs. 97-100, the cylinders are at the top of the engine and consist each of a cylindrical chamber containing the moving piston. The steam is received from the steam chest alternately in either end and thus forces the piston up and down. The motion is then transmitted through the piston-rod and connecting-rod and thus the revolution of the crank and the crank-shaft is produced.

Cylinders are made of cast iron of the highest grade. Each one, as shown in the figures, consists essentially of a cylindrical body or barrel, with which is usually cast the *lower* or *bottom head*. With the barrel are usually cast also the valve casings and chests and all ports and passages, as well as the necessary feet for attachment to the columns, lugs for attaching braces, etc. The *top head* or *cover* is cast separately and is secured to an appropriate flange on the barrel by means of stud-bolts. In some cases the head is made in a single thickness, conical in form to correspond to the piston, and ribbed on top for strength. In other cases it is made by a double shell or in two thicknesses with connecting ribs between. The lower head is formed in the same general way, but, as noted above, is usually cast in one piece with the barrel.

In many cylinders, as shown in Fig. 111, liners are fitted within the barrel or cylinder proper. These are of extra hard and fine grained iron, and are fitted for one or both of the following purposes: (1) To provide a working surface admitting of replacement in case of excessive wear. (2) To provide a jacket space between the barrel and liner in case the cylinders are to have steam jackets. The space thus formed is filled with steam from the boiler, thus providing a jacket or layer of steam entirely surrounding the steam cylinder. Such an arrangement is known as a *steam jacket*, and is used to increase the economy of the engine as noted in Section 59. The liners are usually secured at the lower end by a flange, as shown in Figs. 111, 113,

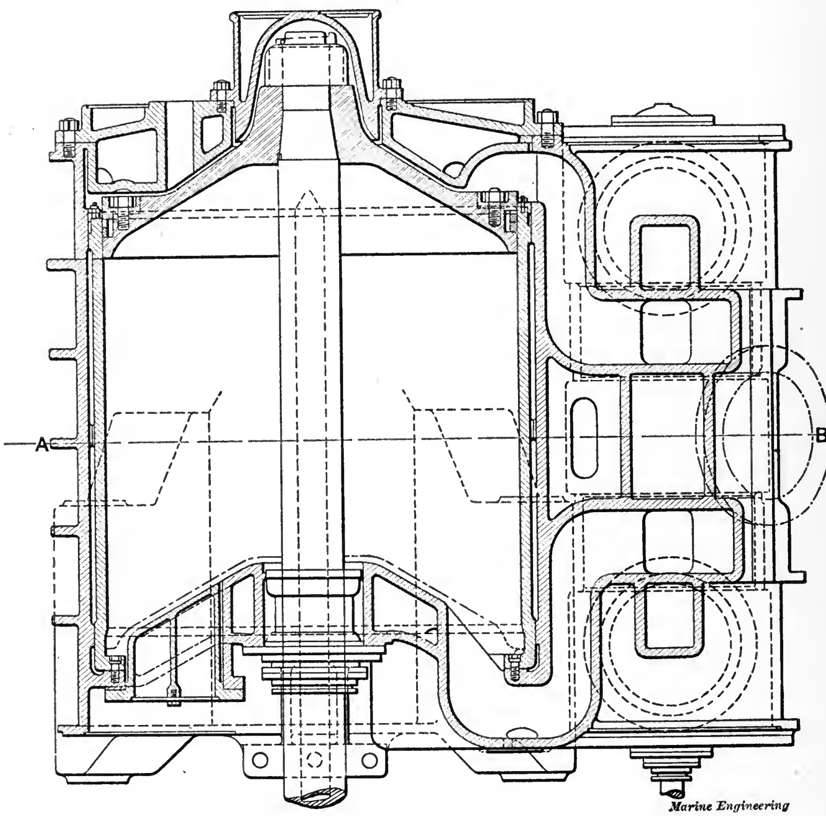


Fig. 111. Cylinder with Liner and Double Valve Chests.

the joint between the end faces of the liner and barrel being carefully made in order to prevent leakage, especially if the space between the barrel and liner is to be used as a steam jacket. At the upper end the joint between liner and barrel may be made in a variety of ways.

As shown in Fig. 112, a packing space is formed between the liner and barrel. This is filled with some form of elastic packing held in place by a ring attached to the liner as shown. In this way the upper end of the liner is free to come and go as expansion and contraction may require, while the packing maintains the joint steam tight. In another mode of fitting, a groove of dovetailed cross section is turned out partly in the liner and partly in the barrel, and a ring of soft metal or packing is expanded into the space thus formed.

The bore of the cylinder or liner is made uniform except near the top and bottom, where it is counterbored out slightly larger, so that at the extreme ends of the stroke the piston rings may overrun the counterbore, and thus avoid wearing a shoulder in the metal.

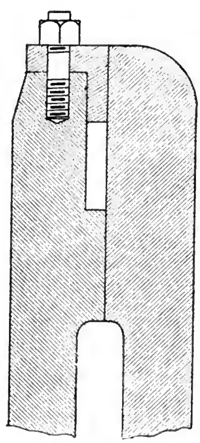


Fig. 112. Joint Between Liner and Barrel, Top.

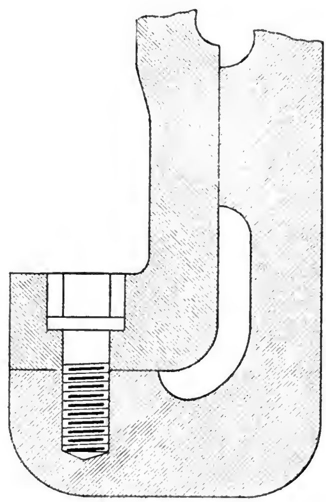


Fig. 113. Joint Between Liner and Barrel, Bottom.

Cylinders as well as steam jackets are usually provided with *drain cocks* and *valves* with suitable piping, so that water collecting within them may be drained away In addition, *automatic relief cocks* or *valves* (see Sec. 24) should be fitted, set to open under an appropriate pressure, and thus furnishing relief in case a large quantity of water may find its way into the cylinder.

The cylinders are supported directly upon the columns which are attached to *facings* on the lower head, or to lugs cast on the lower part of the barrel in case its diameter is not sufficient to reach out over the tops of the columns. See Fig. 114. For mutual support the cylinders are quite commonly tied together by braces, or flanged and bolted to each other. In some cases, however, the cylinders are allowed to stand alone and independently, while in the other cases of recent practice a form of connection has been adopted, consisting of a vertical tongue and grooved joint. This allows differences of expansion vertically and fore and aft, but provides mutual support transversely.

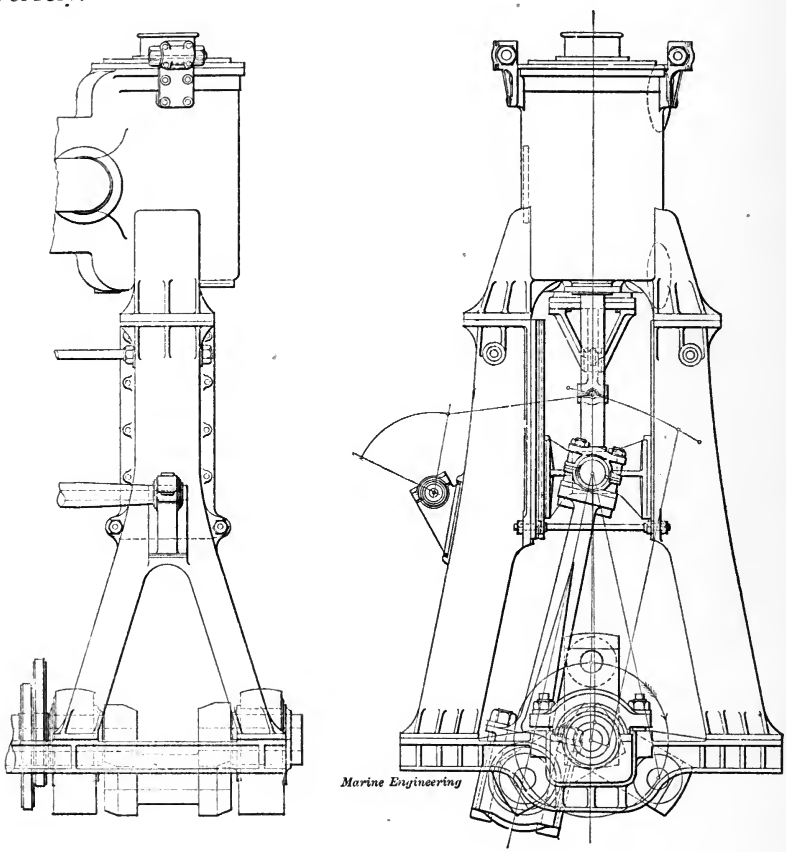


Fig. 114. Double Inverted Y Columns.

The valve chests with the various ports, passages, etc., are also cast with the cylinders, as shown in the figures. These parts will receive further notice in connection with valves. See Sec. 46.

[2] Columns.

The columns serve to support the cylinders and to connect them with the bed-plate. They also serve to support the guide surfaces for the crossheads, and thus receive the transverse thrust of the connecting-rods. Columns are made either of cast iron, cast steel or forged steel. When of cast metal they are usually in the form of an inverted Y, as shown in Figs. 114, 115, and of a box or I-formed section. When forged, the columns are usually cylindrical or slightly tapering, and sometimes hollow. Cast inverted Y columns both front and back of the engine, as shown in Fig. 114, for many years constituted standard practice. More recently, however, cast inverted Y columns at the back of the engine and cylindrical forged columns in front,

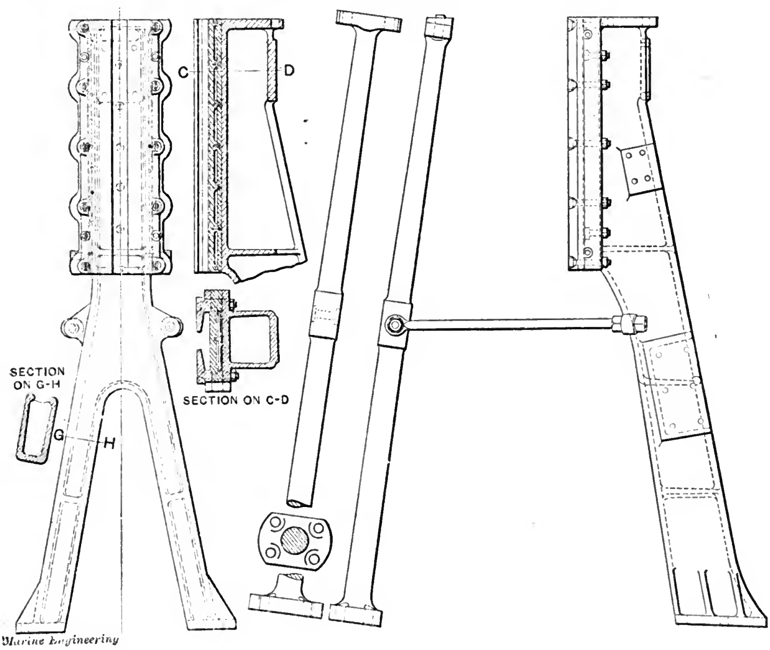


Fig. 115. Inverted Y and Cylindrical Columns.

as in Figs. 115, 116, are commonly employed in representative marine practice. In such case either one or two columns may be fitted in front and one in the rear. When the columns are all cylindrical, it is customary to provide four for each cylinder. Such columns are usually placed vertical, as in Fig. 117, though occasionally they are spread somewhat at the base, as in Fig. 115.

In some cases of modern practice four vertical columns of I section have been provided for use with a crosshead as shown in Fig. 134. The columns stand in pairs, one forward and one aft, and the wings of the crosshead carrying the slide surfaces work between them on the guides carried on their inner faces.

In some cases the condenser is placed back of the engine

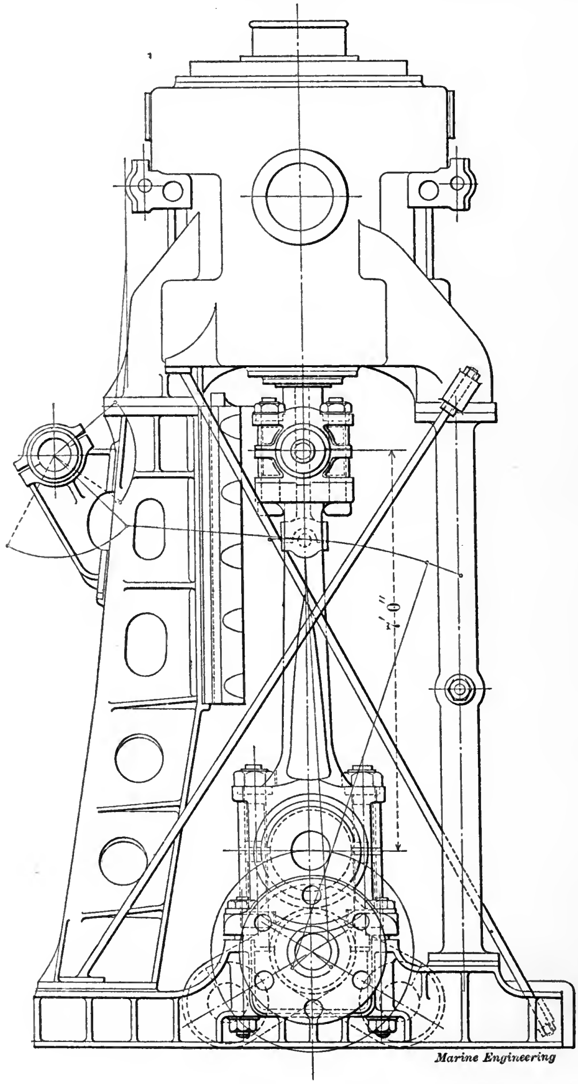


Fig. 116. Inverted Y and Cylindrical Columns, Warship Type.

and on the bed-plate, as in Fig. 100. In this case the back columns are either cast with the condenser shell or consist of short vertical columns standing on top of the condenser; which thus constitutes a part of the support of the cylinders as shown.

To resist the racking and cross-breaking stresses to which the columns may be subject, it is necessary, especially with plain cylindrical columns, to provide transverse and even longitudinal ties and braces. The usual arrangement of such bracing is shown in Figs. 115, 116, 117. It will be noted in particular that the transverse bracing between a pair of columns as in Fig. 117 unites them into a single girder, thus providing vastly more strength to resist lateral stresses due to rolling of the ship, etc., than could be furnished by the columns themselves and without the assistance which the bracing is able to provide.

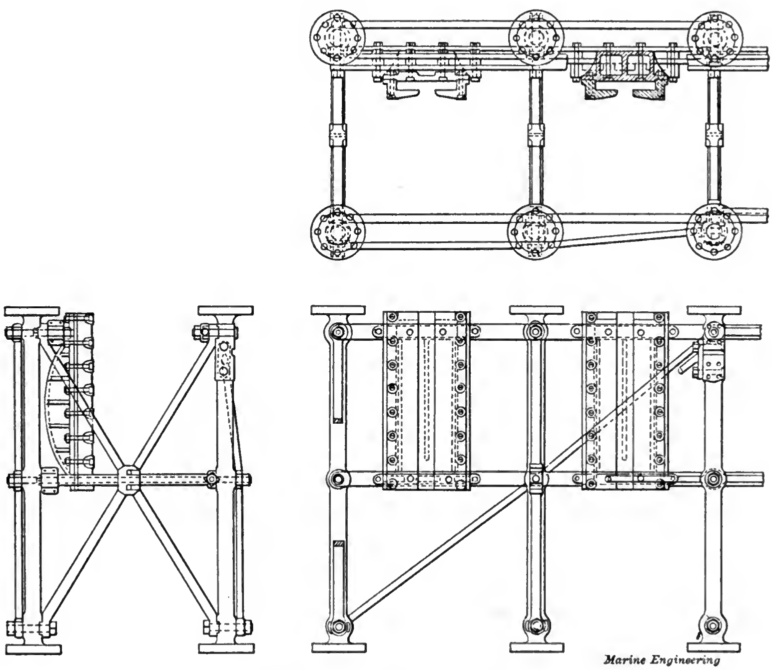


Fig. 117. Cylindrical Columns.

The guide surface for a crosshead is fitted in various ways according to the style of crosshead, the style of column and type of practice. The simplest arrangement is as shown in the cross-section of Fig. 118, in which the guide surface is fitted directly on the inner face of the Y column. In the arrangement of Fig. 115 the guide surface is fitted on a separate slab of rather harder and finer grained cast iron, and hence better adapted for bearing purposes. Between this slab and the face of the column a space is left as shown, and through this may be circulated a stream of water to absorb the heat generated by the friction, and thus to keep the bearing surface cool. With cylindrical columns the guide surface must be fitted as a separate slab for each crosshead, and usually in the manner shown in Fig. 117. These slabs may be of cast iron, steel or bronze, and are carried on longitudinal bars attached to the columns. The form of cross-section may be either hollow for water circulation, or plain or ribbed on the back for strength, as the case may require. A common form is that shown in Fig. 117, thinner towards the

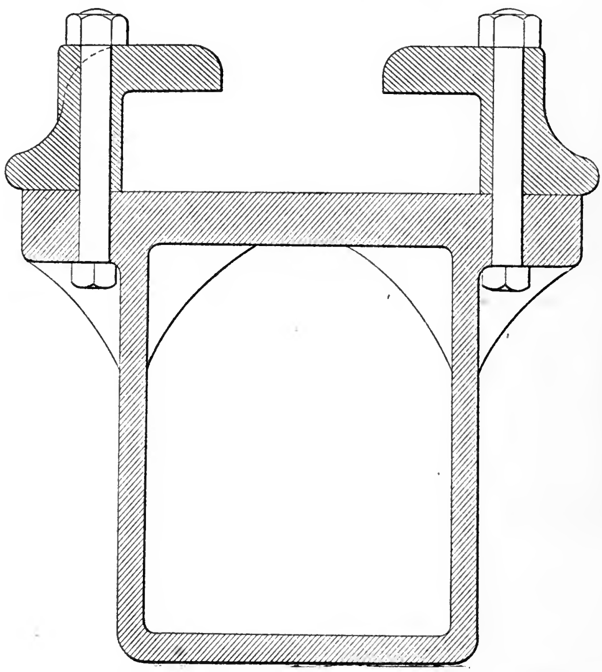


Fig. 118. Section of Cast Column Showing Guide Surface.

ends and thicker in the middle as a girder, to provide the necessary strength at this point.

For further details of the guide surfaces which depend on the form of crosshead used, reference may be made to [7].

[3] Bed-Plates.

The purpose of the bed-plate is to support the feet of the columns, and thus to carry the weight of the cylinders and attachments, to provide seatings and support for the crank-shaft bearings, and generally to serve as the foundation piece upon which the engine rests, and through which its weight and the various stresses developed are transferred to the structure of the ship.

As usually formed it consists of a series of transverse box

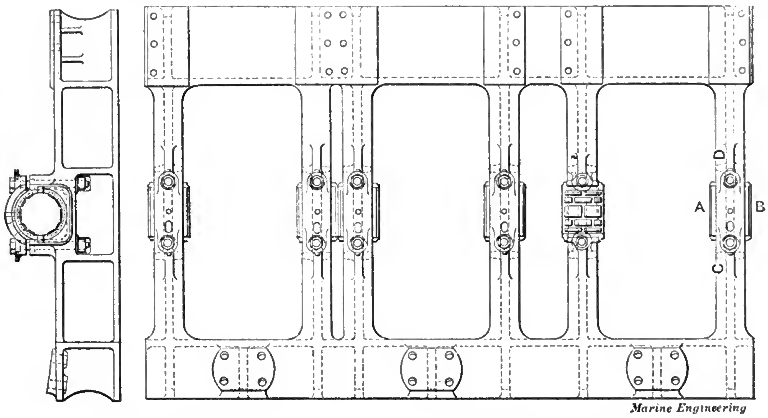


Fig. 119. Bedplate for Triple Expansion Engine in one Casting.

or I girders, one for each crank-shaft bearing, these being connected together by fore and aft members, as shown in Figs. 119, 121.

Bed-plates are usually made of cast iron or cast steel.

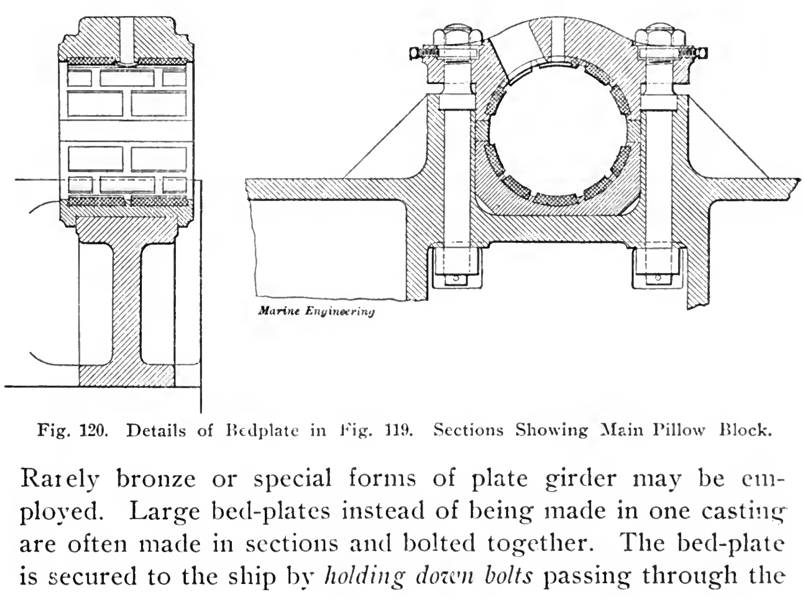


Fig. 120. Details of Bedplate in Fig. 119. Sections Showing Main Pillow Block.

Rarely bronze or special forms of plate girder may be employed. Large bed-plates instead of being made in one casting are often made in sections and bolted together. The bed-plate is secured to the ship by *holding down bolts* passing through the flanges of the plate and of the specially strengthened structure of the ship underneath, known as the engine seating or foundation. Further examples of bed-plates may also be noted in Figs. 97-100, 114, 116.

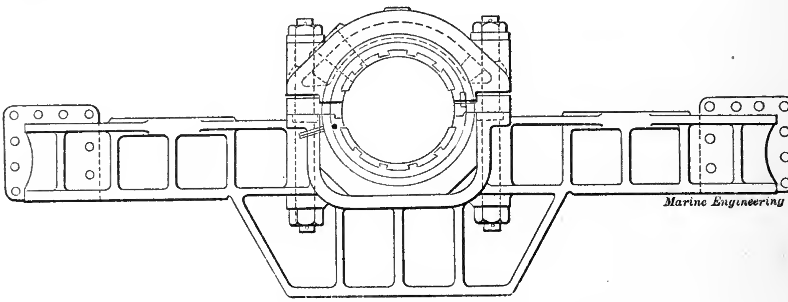


Fig. 121. Bedplate in Sections. End View of one Section.

[4] Engine Seating.

This structure is a part of the ship, and serves to give the final support to the weight of the engine, and to lead the stresses due either to its weight or to its operation, into the structure of the ship as a whole. The usual character of the seating is shown in Fig. 122. It consists of a cellular construction formed

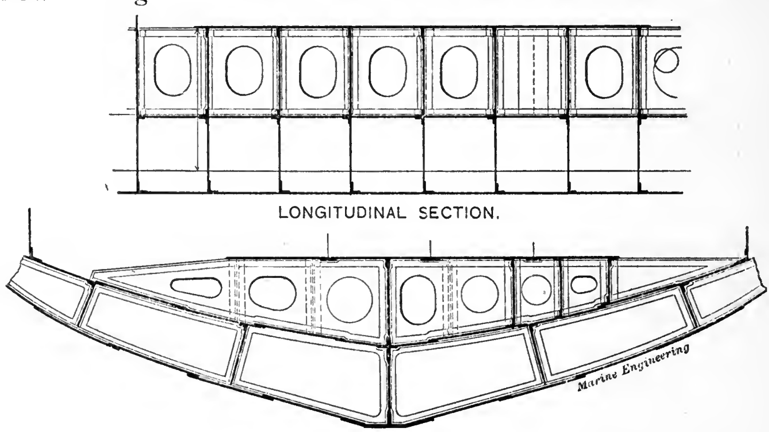


Fig. 122. Engine Seating.

by longitudinal and transverse vertical plates, stiffened and connected at the corners by angle irons, and usually forming a continuous structure with a part, at least, of the regular internal members of the ship itself.

We will now turn to the chief moving parts of the engine.

[5] Pistons.

The piston is the moving part of the engine upon which the steam directly acts, and which by the steam pressure is driven back and forth in the cylinder, and from which, through the piston-rod, crosshead and connecting-rod, the motion is transmitted to the crank and crank-shaft. The requirements for the piston are therefore: (1) It must be able to support the load which the steam pressure brings upon it. (2) It must be of such form as to admit of movement up and down in the cylinder, at the same time making a steam tight joint between its outer edge and the cylinder walls. (3) Provision must be made for its secure attachment to the piston-rod, through which the forces are transmitted to the remaining moving parts of the engine.

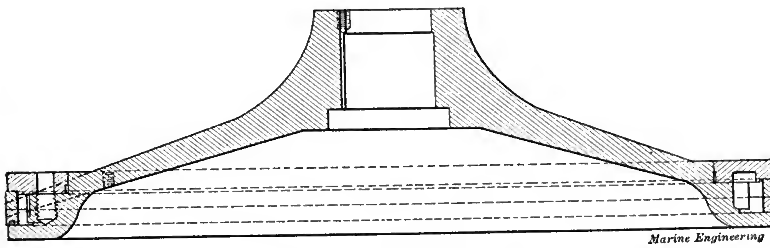


Fig. 123. Conical Marine Piston.

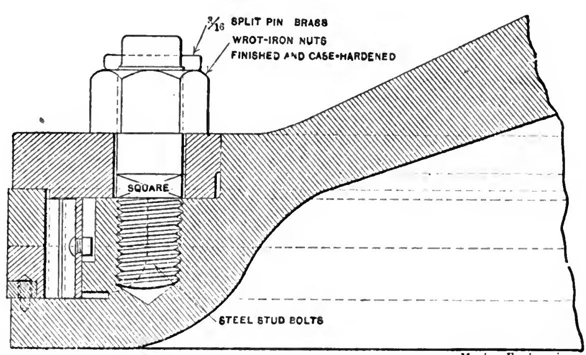


Fig. 124. Marine Piston, Enlarged View, Showing Packing Rings and Follower Plate.

The usual form of marine piston is shown in Fig. 123, and consists of a shell of conical form with a central boss or body for carrying the piston-rod as shown. Around the outer edge of the piston the metal is thickened up to provide for the packing rings, which are fitted to make a steam tight joint between the piston and cylinder walls. The fitting of these rings is shown in Fig. 124. The rings are usually two in number, and are formed

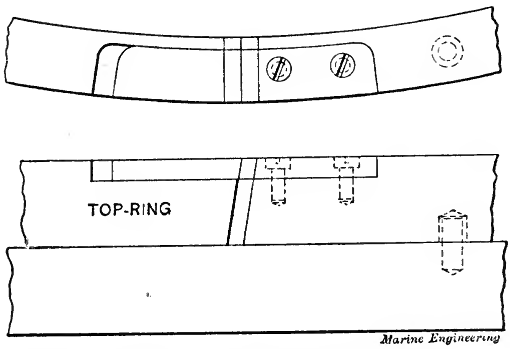


Fig. 125. Marine Piston, Joint in Packing Rings.

of cast iron turned first to an outside diameter slightly larger than the bore of the cylinder. They are then cut as shown in Fig. 125, and enough is taken out so that they may be sprung together sufficiently to allow their entrance into the cylinder bore. Care is taken to so locate the two rings that the cuts

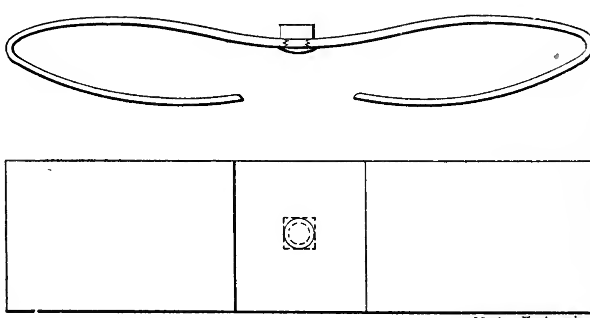


Fig. 126. Marine Piston, Steel Springs for Packing Rings.

shall not come opposite, and thus the opportunity for a direct leak through from one side to the other is avoided. In order to still further prevent such leakage, a tongue as shown in the figure is usually fitted across the opening. The tongue piece, which is usually of brass, is attached to the ring and overlaps the slit, as shown. The joints between the ring and tongue piece are carefully fitted so that in this way the ring may open and shut as circumstances may require, while the opening into the slit remains closed to the entrance of steam. When the piston is of any considerable size it is customary to aid the natural elasticity of the rings by steel springs, as shown in Fig. 126. These bear on the bottom of the recess formed in the piston, and on the inner surface of the rings, and thus the latter are forced outward against the surface of the cylinder.

The body of the piston itself, as shown in Fig. 124, is turned slightly smaller than the diameter of the cylinder, so that it clears the latter at all times, while the rings extend beyond and make the joint with the cylinder wall. The rings and springs

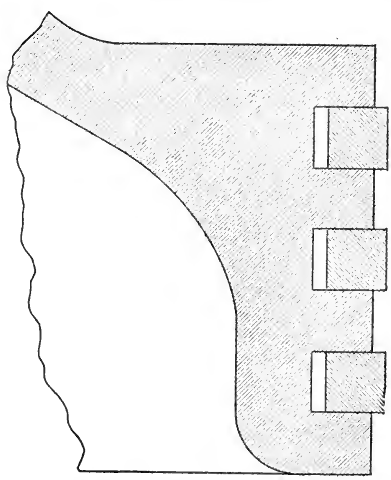


Fig. 127. Ramsbottom Rings.

are fitted as shown between the lower flange of the piston body and a plate known as the *follower plate* or *ring*. By removing the latter the rings and springs may be removed when necessary for overhauling and refitting. The follower plate is secured to the piston by stud bolts and nuts, as shown in the figure. In the best class of work all joints between the piston and rings, between the follower plate and rings and between the two latter are carefully made by hand scraping and fitting, in order to reduce the chances of leakage to the smallest possible limits.

Many variations are met with in the details of the form and fittings of pistons. In some cases they are flat and either solid or hollow, as shown in Figs. 97, 128.

In some cases ramsbottom rings are fitted instead of the rings of Fig. 123. These consist of two or three narrow rings turned slightly larger than the cylinder with a piece cut out so that they may be sprung on over the body of the piston, and into grooves, as shown in Fig. 127. No special springs are fitted, and the natural elasticity of the rings is depended upon to give the necessary pressure between the ring surface and the cylinder.

It is easily seen that no follower plates being fitted, the rings cannot be examined or removed without removing the piston. To avoid this difficulty the arrangement of Fig. 128 is sometimes used. Here the rings are carried on a larger solid ring, as shown, and sometimes known as a bull ring. This is carried between the faces of the piston flange and follower plate, and thus by the removal of the latter the whole arrangement may be withdrawn and examined. There is usually some clearance between the inner surface of the bull ring and the body of the pis-

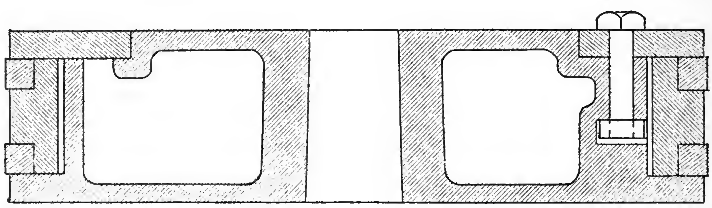


Fig. 128. Piston with Ramsbottom Rings on Bull Ring.

ton, as shown in the figure. This allows the whole arrangement of rings to move transversely independent of the piston body, thus making allowance for lack of alignment between the axis of the piston-rod and the axis of the cylinder, or for wear in the latter.

While light packing rings of cast iron fitted as above described without the assistance of steel springs may prove satisfactory for small pistons, the more standard method of Fig. 123 is to be recommended for all cases where the pistons are of any considerable size.

In present practice pistons of the form shown in Fig. 123 are made of cast steel. Pistons of the form shown in Figs. 97, 128, are more commonly made of cast iron.

The chief advantage of the conical form of piston lies in the saving of weight for the necessary strength and stiffness, as compared with other forms. This superiority has gained for it almost universal adoption in modern practice, and it may be considered as the present day representative form of marine piston, and the one which will naturally be adopted unless there may exist special reasons for the adoption of the older type.

[6] Piston-Rods.

The piston-rod is that member of the moving parts which serves to support the piston, to carry the forces due to the steam pressure through the stuffing box outside the cylinder, and through the crosshead to communicate them to the connecting-rod and other moving parts. The requirements are therefore as follows: (1) It must have sufficient strength and stiffness to safely carry the load coming from the piston. (2) It must be provided at the upper end for attachment to the piston, and at the lower end to the crosshead. (3) It must be of such form as to admit of readily making a steam tight joint where it passes out of the cylinder. To fulfil these conditions the piston-rod, as

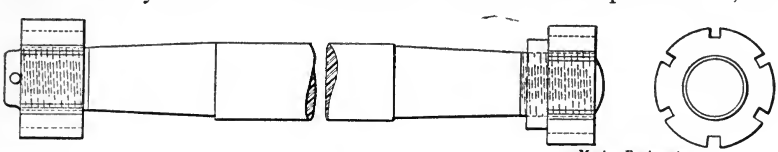


Fig. 129. Piston Rod.

shown in Fig. 129, has the form of a uniform cylindrical rod except at the ends where it joins the piston and crosshead. The common form of attachment to the piston is shown in the figure. The rod is sometimes tapered where it lies in the piston, and sometimes parallel. It is often relieved from direct bearing except near the top and bottom, so as to give definite points of bearing where it is most needed. A shoulder or ring is also fitted, as shown, so as to give a definite stop against which the body of the piston rests. The end of the rod is threaded and a nut on top completes the fastening. This nut is sometimes hexagonal and sometimes cylindrical with longitudinal grooves, a spanner wrench being used in the latter case to set the nut down.

The fitting at the lower end of the piston-rod depends on the style of crosshead used, and may be more appropriately described under that heading.

In modern practice with conditions requiring the highest grade of material and most careful design, the piston-rod is often made hollow. This practice also extends to most of the other cylindrical elements of the engine such as cylindrical columns, crosshead pins, connecting-rods, crank-pins, crank, line,

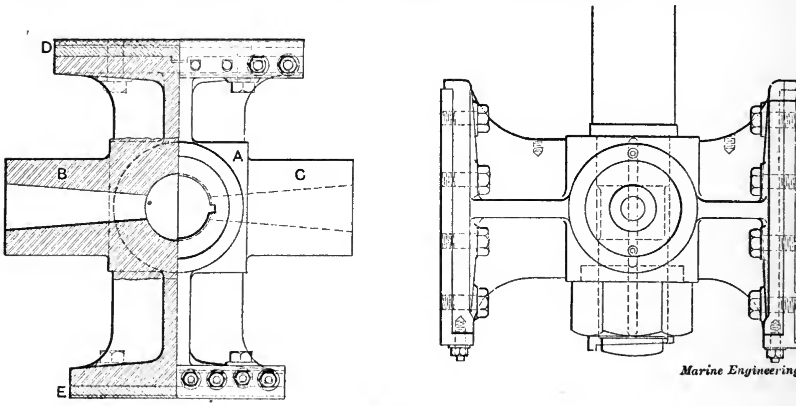


Fig. 130. Marine Crosshead.

thrust, and propeller shafts. Inasmuch as this style of construction was first commonly introduced in connection with shafting, the reasons for such practice and its advantages may properly be discussed under that heading.

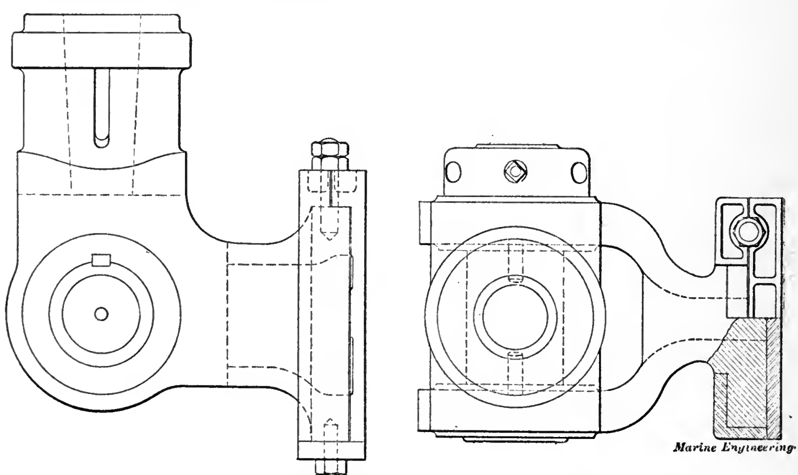


Fig. 131. Marine Crosshead, Slipper Type with Cotter Fastening for Piston Rod.

[7] Crossheads.

There are several types of crosshead to be met with in marine practice. In Fig. 130 is shown one of the more common forms. It consists essentially of a cubical body A, through which in a vertical direction is the hole for the piston-rod. Extending out on either side longitudinally are the two crosshead

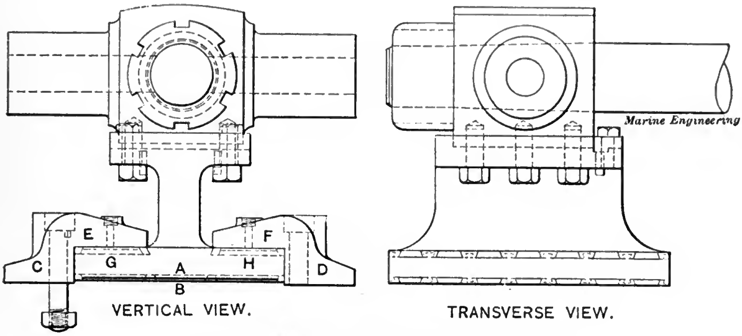


Fig. 132. Marine Crosshead, Slipper Type.

pins B and C. Then attached to the two remaining sides transversely are the slides as shown. The connection between the crosshead and the piston-rod is commonly by means of thread and nut, as shown in the figure, in the same way as for the con-

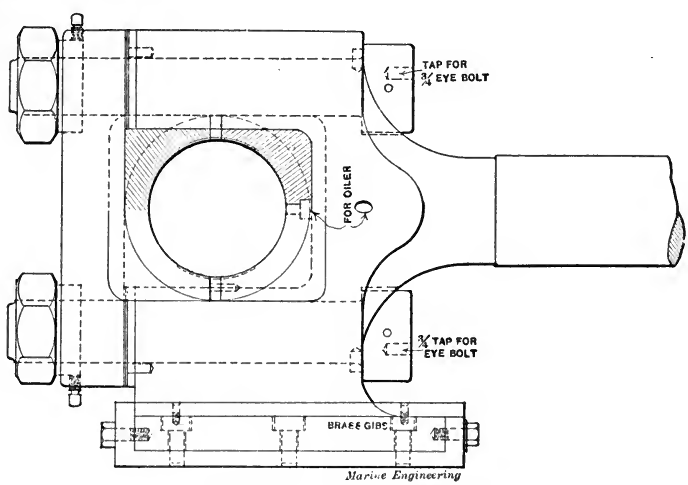


Fig. 133. Crosshead formed on Lower End of Piston Rod.

nection to the piston. In some cases a pin or cotter joint, as shown in Fig. 131, is used instead of the thread and nut on the end. The slide surfaces D and E rest on the guide surfaces of the columns, as above described, one side taking the load when going ahead and the other when backing. A crosshead of this type is therefore suitable for double inverted Y columns where there is a guide surface on both back and front sides of the engine. Where cylindrical columns are used, as in Figs. 115-117, the slipper form of crosshead is commonly fitted. This is shown in Fig. 132, and so far as the part connected with the piston-rod and carrying the crosshead pins is concerned may be the same as in Fig. 130. Instead of two wings carrying slides, however, there is but one with the form shown in the vertical view. The corresponding form of guide is shown in Figs. 115, 117.

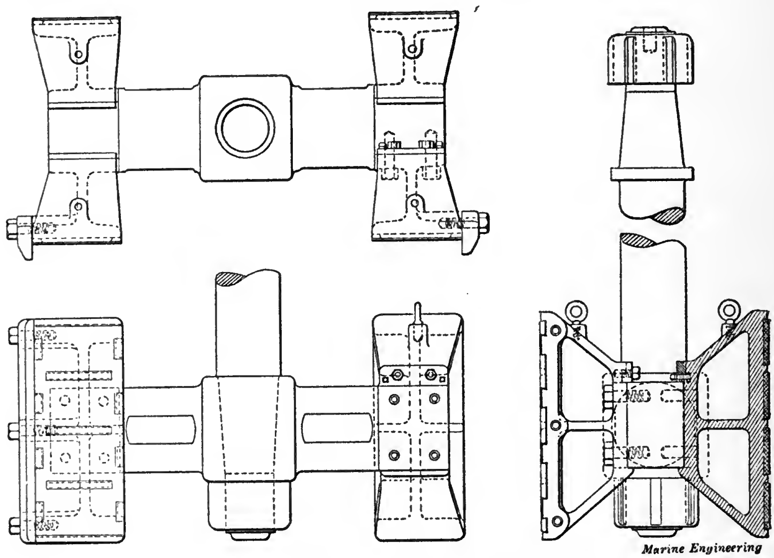


Fig. 134. Marine Crosshead, Special Type.

When going ahead the face A of the slipper bears against the face B of the guide. Cheek pieces or gibs C and D are secured to the column, thus forming guide surfaces on their inner faces E and F. Against these the faces G and H of the slipper bear when in backing motion. The go-ahead surface is therefore formed on the faces A and B of the guide and slipper, as with Fig. 130, while the backing surface, instead of being provided on the opposite column and on another slide piece on the other side of the crosshead, is formed on the reverse side, G and H, of the slipper, and on the cheek pieces, as shown.

These types of crosshead are suited to the so-called forked type of connecting-rod, as described below, for which the crosshead pins are a part of or fast in the crosshead, and are naturally two in number, one on either side fore and aft, as shown. In the other type of connecting-rod which is frequently met with, the rod is not forked, and the pin is fast in the upper end, and is single rather than double, while the crosshead member furnishes the bearing. This arrangement is shown in Fig. 133. The crosshead body is usually forged up on the lower end of the piston-rod, and together with a suitable cap and bearing brasses forms the bearing for the pin which is fast in the upper end of the connecting-rod, as shown. The wings for carrying the slides may be attached and the slides may be fitted in the same

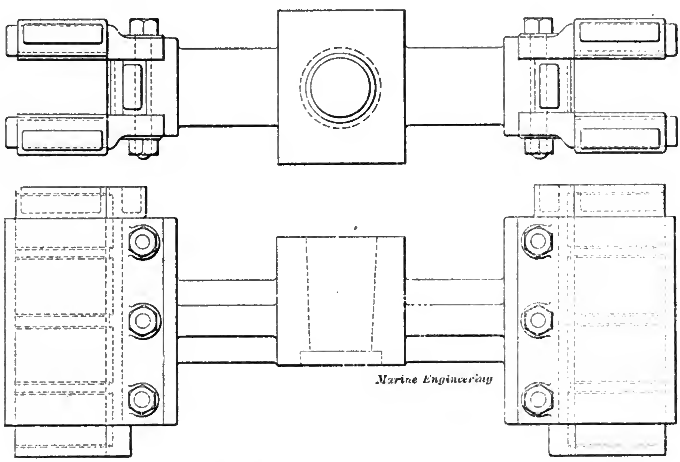


Fig. 135. Marine Crosshead, Special Type.

general manner as in Figs. 130, 132, either double or of the slipper type.

A third form of crosshead occasionally found in modern practice was referred to in [2], and is here shown in Fig. 134. This type of crosshead is a marine adaptation of a type very common in stationary engine practice. The slide surfaces are formed on the opposite faces of webs or wings extending out from the body of the crosshead, and bearing on the guide surfaces formed on the columns. In Fig. 135 is shown a somewhat different form of the same type of crosshead, the latter being suited to two columns and the former to four.

As noted in [6], the crosshead-pins, in the most advanced practice, are often made hollow. In some cases the hole is parallel; in others its diameter decreases from the outer end inward, thus giving the most metal at the inner end, where the greatest stresses are likely to be found. See Fig. 130.

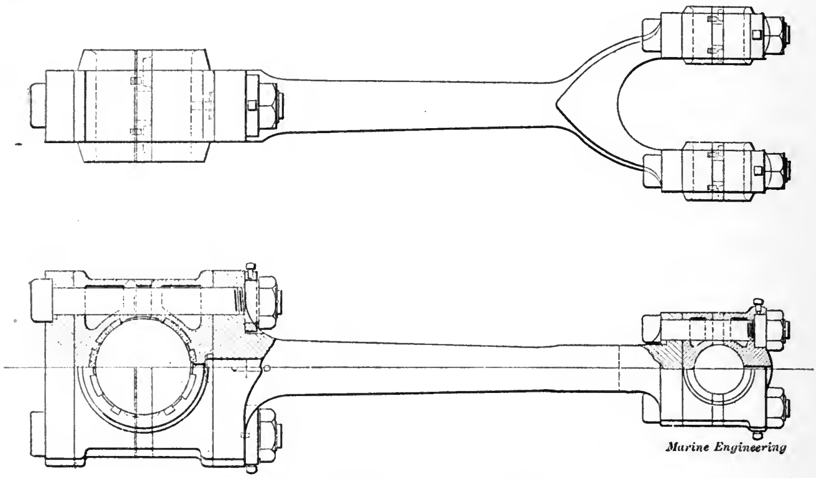


Fig. 136. Marine Connecting Rod.

[8] Connecting-Rods.

Fig. 136 illustrates perhaps the more common type of connecting-rod. At the upper end it is forked or formed into a U

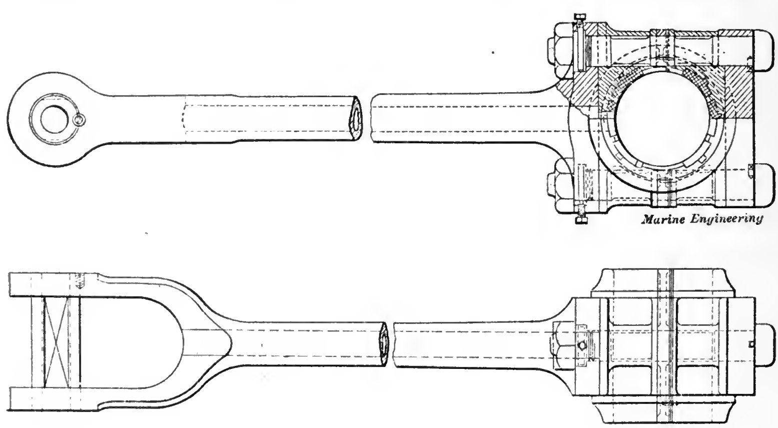


Fig. 137. Marine Connecting Rod.

shape, each branch being provided with a bearing and connections for one of the crosshead-pins. This type of end corresponds therefore to the type of crosshead shown in Figs. 130, 132.

For connection to the crank-pin, the lower end of the rod is fitted with brasses and cap, all secured to the forged out foot of the rod by through bolts as shown.

For the type of crosshead shown in Fig. 133 the rod is formed, as shown in Fig. 137, with a U-shaped upper end fitted to receive the two ends of the crosshead-pin, which is thus made fast to the rod. This pin is then seated in a bearing in the crosshead, as described under that heading. The lower end of the rod is usually of the same form as shown in Fig. 136.

Rarely the gib and key form of connecting-rod end as illustrated in Fig. 138, is found in marine practice.

In external form marine conecting-rods usually increase in transverse dimension from top to bottom. In some cases they are given a uniform taper from one end to the other, as in

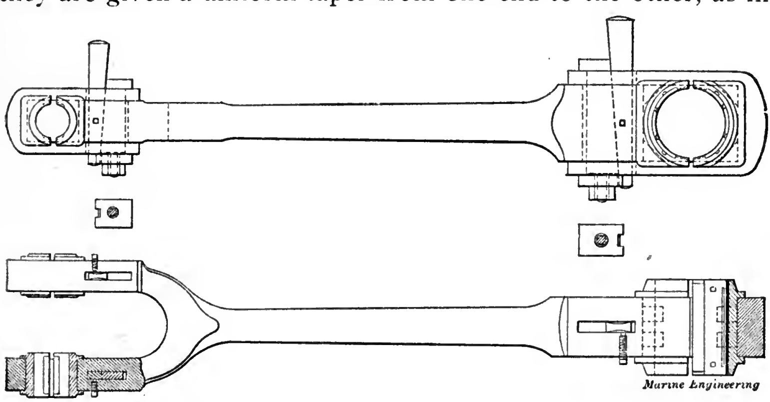


Fig. 138. Marine Connecting Rod with Gib and Key Connections.

Figs. 136-138, while in others the extra metal is slabbed off on the forward and after sides until the thickness in the fore and aft direction is uniform from top to bottom.

As noted in [6], the connecting rod, in the most advanced practice, is often hollow, a hole of uniform bore being drilled from one end to the other, as shown in Fig. 137.

[9] Crank Shafts.

Modern marine crank shafts are of two principal types, forged and built up. Fig. 139 shows a portion of a built-up crank shaft. It consists, as shown, of two crank *webs* or *throws*, A and B, one crank-pin C and two portions of shaft D, E. Built-up crank-shafts are usually made contiuous for the whole engine, and in such case the piece of shafting D connects the crank shown with the one next to it, and thus serves as a common member for the two.

In this type of crank-shaft the various sections of shaft, the crank-pin and the webs, are all made separately, and then fitted and secured together. This is usually done by shrinking and keying the various cylindrical members into the sections of web as shown in the figure.

Fig. 140 shows a section of a forged crank-shaft. In this case a forging of suitable form is made, and the various parts are then formed by cutting out and machining this forging. In many cases, moreover, the series of such sections for the entire engine are forged and machined in one piece, the result being a continuous forged crank-shaft. In other cases the section for each crank, as shown in the figure, is forged and made sep-

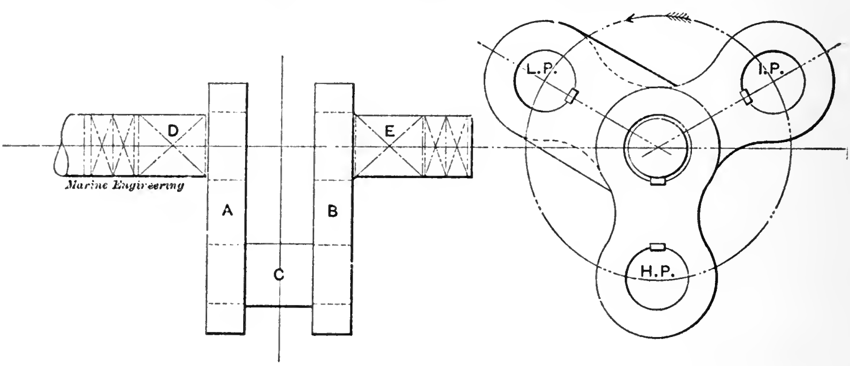


Fig. 139. Section of Built Up Crank Shaft.

arately, the various sections being then secured together by flange couplings, as shown in Figs. 140, 141. The advantage of making the shaft in sections lies in the fact that in many cases the sections may be made interchangeable, and thus a single spare section is sufficient for the replacement of any section which may become disabled through accident, and in any event a break will usually require the refitting of a single new section instead of an entire shaft.

Forged crank-shafts are commonly used in naval practice, and in general where the type of construction is of specially high grade and the saving of weight an important feature. Their use in all departments of marine practice seems, moreover, to be on the increase. Built-up crank-shafts, however, are still much used in the mercantile marine, especially where the conditions are easily fulfilled, and their somewhat greater weight is not a serious objection.

As noted in [6], the cylindrical members of marine engines are often made hollow, especially in the more advanced types of design. Fig. 140 shows a crank-shaft section with hollow pin and shaft. As this feature was first commonly introduced in connection with shafting, and is more often met with here than elsewhere, the advantages of such construction may be now considered.

The advantages of a hollow cylindrical member such as a piston-rod, connecting rod, crank-pin, or length of shafting, are two in number. (1) It is stronger for a given weight, or for a given strength less weight is required. (2) The central core of metal is removed, and this is the most liable to contain cracks or flaws, which might in time extend out into the remaining

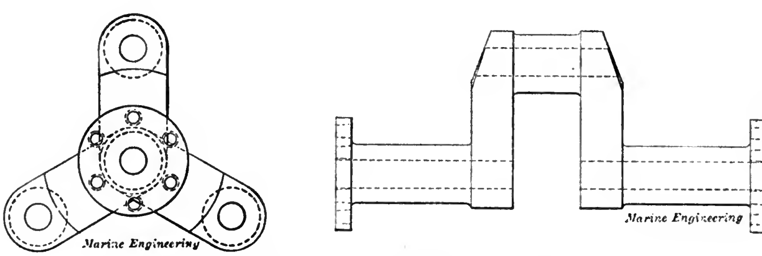


Fig. 140. Section of Forged Crank Shaft.

metal, and thus seriously weaken the member. Furthermore, the hole gives opportunity for the inspection of the metal on the inside, and thus increases the opportunity for the detection of a flaw which might not extend to the outer surface, or which might there be so small as to be overlooked.

For cross-breaking or for torsion the metal in the interior of a cylindrical member is of comparatively small value. Thus in a 10-inch shaft, the inner core 5 inches in diameter is worth no more than a shell of metal about .16 inch thickness lying next the outer surface. Or, as a further illustration, a 16-inch shaft with a 10-inch hole is equal to a 15-inch solid shaft. In other words, a shell of metal 1-2 inch in thickness all around added on the outside of the 15-inch shaft will make up for the removal of the inner core of 10 inches diameter. In the latter case the hollow shaft would weigh about 65 per cent of the equivalent solid shaft. The saving in weight for a desired strength may thus be very considerable, but it is probable that the advantages noted above under (2) are of still greater importance, and in some cases might justify the added cost of making the member hollow where such addition could not be justified by the saving of weight only.

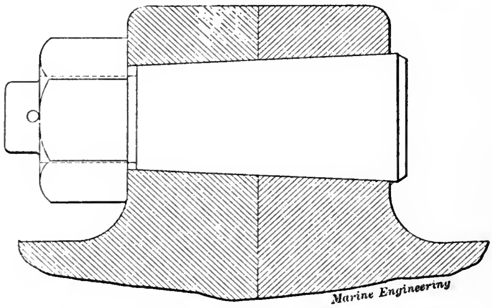


Fig. 141. Detail of Flange Coupling and Bolt.

[10] Line Thrust and Propeller Shafts.

From the crank-shaft the motion is carried on to the propeller by means of a number of lengths or sections of shafting according to the distance from the engine to the stern of the ship. Of these sections the one to which the propeller is attached is known as the propeller shaft. One length must also

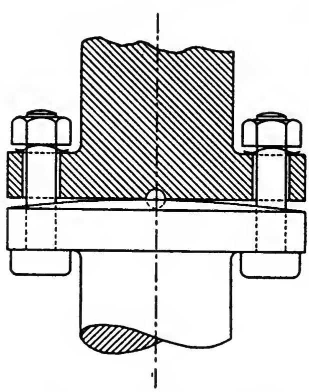


Fig. 142. Flexible Coupling.

be specially fitted to transmit the forward thrust to the thrust bearing and thence to the ship. This section is known as the *thrust shaft*. Other intermediate lengths form the *line shafting*. These various lengths, with the exception noted below, are usually connected by flange couplings, as in Figs. 140, 141. The coupling from the engine to the next section of shafting aft is often made of such form as to allow a certain degree of flexibility between the line shafting and the crank shaft. A form of such coupling is shown in Fig. 142. One of the coupling flanges is faced off, as shown like the segment of a sphere, with a ball and socket joint at the center to keep the two parts in line. The coupling bolts are then set up with nuts bearing on some form of spring washer which will take up the slack as the shaft revolves, even when not exactly in line. The action of the coupling will be readily seen from a study of the figure. Various other styles of coupling are in use, but the one shown will sufficiently illustrate the principles involved.

The thrust shaft will more naturally find its description with the thrust bearing. See [11].

The propeller shaft is formed with the after end tapered

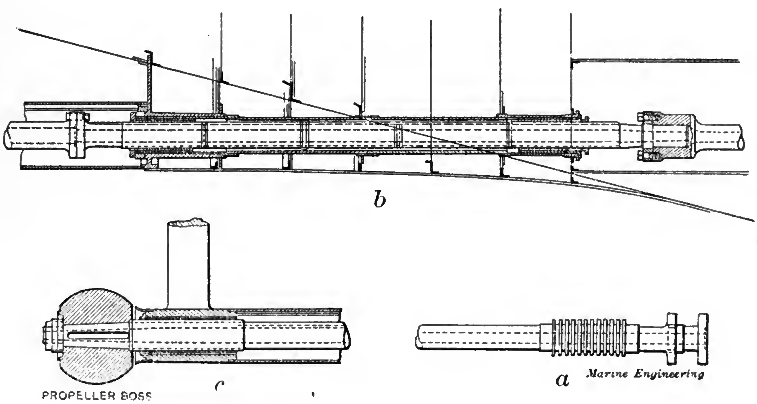


Fig. 143. Outboard Shaft for Twin Screw Ship.

and fitted with screw thread for a nut, as shown in Fig. 143c. The propeller is fitted with a corresponding taper, and is held in place by a nut and prevented from turning on the shaft by one or more keys, as indicated in the figure.

In the case of twin screws, where the propeller shafts pass outside the skin of the ship some distance forward of the stern, it often becomes necessary to form the outboard shaft in more than one length, coupled together by flange couplings as above described. In all cases it is necessary to form one end of the section of shaft which passes through the skin of the ship with a plain end, so that it can be passed through the outboard bearing as described in [11]. In cases therefore where the propeller cannot be attached directly to the plain end, as in single screw ships, it becomes necessary to provide a special form of socket coupling connecting the after end of the last inboard length of shafting with the forward end of the length which passes through the ship. The general plan of this arrangement is shown in Fig. 143b, and the details of the coupling in Fig. 144. Such couplings vary somewhat in detail, but the form shown in

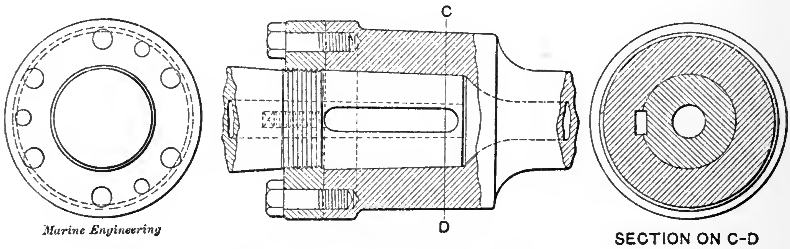


Fig. 144. Detail of Socket Coupling.

the figure will serve to illustrate the type. It consists, as shown, of the enlarged end of the inboard shaft, in which is bored out a tapering socket of appropriate size to take the tapered forward end of the first outboard length of shaft. The two are then secured together by keys and locking ring, as shown in the figure.

[11] Bearings.

The various types and forms of bearing and bearing surface to be found in a marine engine may be conveniently examined under one heading.

(1) *Crosshead and Guides.* The stationary part of this bearing has been already referred to in [2], and as there noted is usually of a hard and fine grained cast iron. The moving surface on the crosshead is usually of brass, bearing-bronze or white metal. When of brass or bronze it is in the form of a bearing piece secured to the crosshead, as shown in Fig. 130. When of white metal a suitable slab of brass, cast iron or cast steel, with shallow pockets formed in its surface, forms the bearing piece. These pockets have slightly overhanging edges, and into them molten white metal is run, the general layout and arrangement being similar to that for the main pillow block bearings shown in Fig. 146. The white metal is then machined down to a bearing surface, in some cases being hammered with a round pene hammer in order to compress or harden the metal.

The spaces between the pockets thus become spaces between the sections of white metal, and serve for the circulation and supply of oil to all parts of the bearing surface. In some cases shallow channels or oil grooves are cut in the guide surface or stationary part as well, but this is the least necessary with the arrangement of white metal as described.

Liners or packing pieces are often placed between the bearing piece and the crosshead, so as to allow for adjustment and take up in case of wear.

(2) *Crosshead Pins.* The general arrangement for the bearings are indicated in Figs. 133, 136, 137. The crosshead

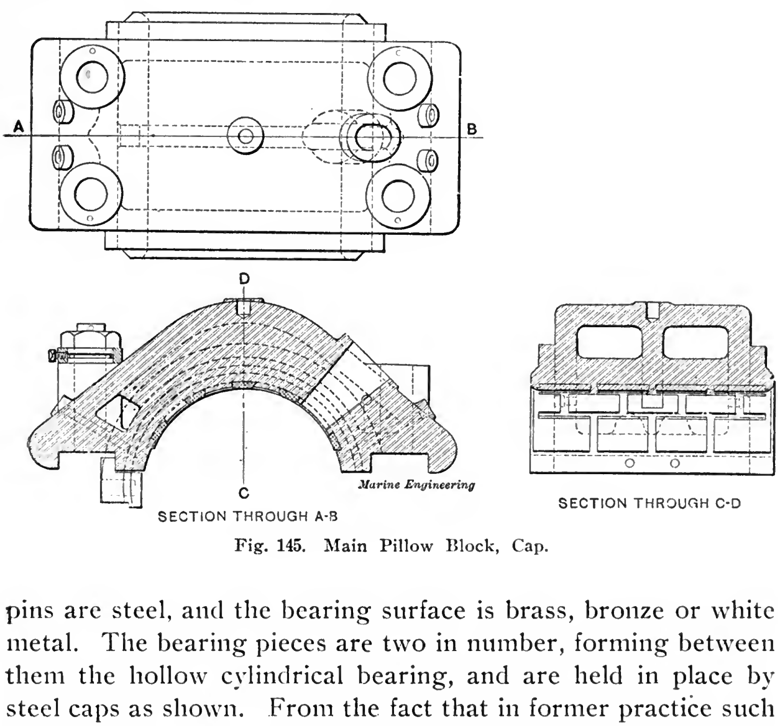


Fig. 145. Main Pillow Block, Cap.

pins are steel, and the bearing surface is brass, bronze or white metal. The bearing pieces are two in number, forming between them the hollow cylindrical bearing, and are held in place by steel caps as shown. From the fact that in former practice such bearing pieces were almost universally made of some grade of brass, they are still usually known as *brasses.*

When white metal is used it may be fitted in the same general manner as above described for the crosshead slides, or in some cases, especially for small surfaces, the white metal sections are turned down until a continuous bearing surface is obtained on both white metal and brass. For the distribution of oil, grooves or channels are then cut to serve in place of the channels between the sections of white metal, as in Fig. 146.

(3) *Crank Pin.* The usual arrangement of this bearing is sufficiently shown in Figs. 136, 137. In modern practice the material used is commonly white metal in a brass backing or bearing piece, as already described. In older practice brass or bronze was commonly employed.

(4) *Pillow Block or Crank Shaft Bearings.* The usual arrangements are shown in Figs. 120, 145, 146. Here likewise in modern practice the usual surface is white metal in a brass backing piece. In all bearings for cylindrical elements, as the

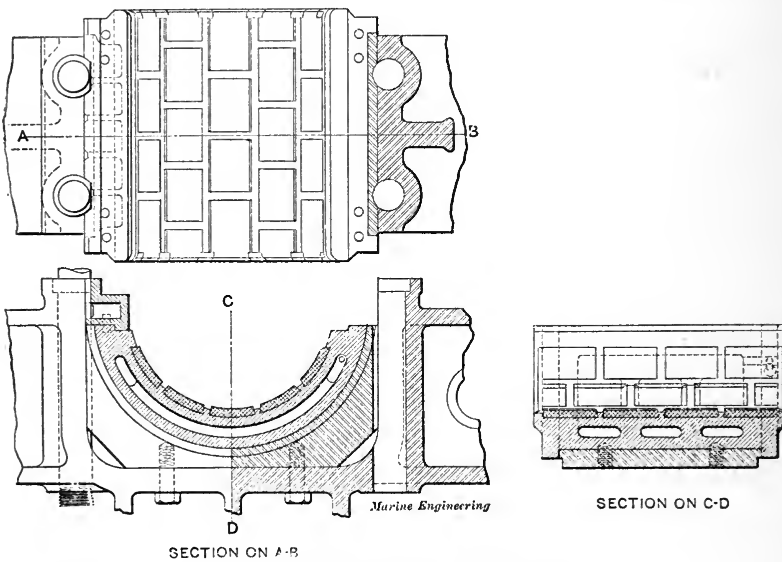


Fig. 146. Main Pillow Block Bearing.

crosshead pin, crank-pin, crank-shaft, etc., the two brasses are held from separating by a cap and bolts as shown, and from pinching the pin or shaft too tightly by filling pieces or liners. These by adjustment allow of take up for wear. In modern practice the lower pillow block brass is often made in the form of a half cylinder, as shown in Fig. 146, so that by the removal of the cap and upper brass the lower one may be slid around the shaft and so removed for adjustment or repair, or a new brass replaced without disconnecting the crank shaft and lifting it from its bearings, as is necessary when the lower brasses are of the shape shown in Fig. 120.

(5) *Line-Shaft or Spring Bearings.* In these bearings the chief load is the weight of the shaft, at least so long as the bearings and shaft are in line and adjustment. It is therefore quite common to provide for such bearings simply a lower brass or bearing piece, usually in modern practice of white metal backed by brass, or in some cases by cast iron or steel. A bearing cap or cover is then fitted, not in contact with the shaft, and serving

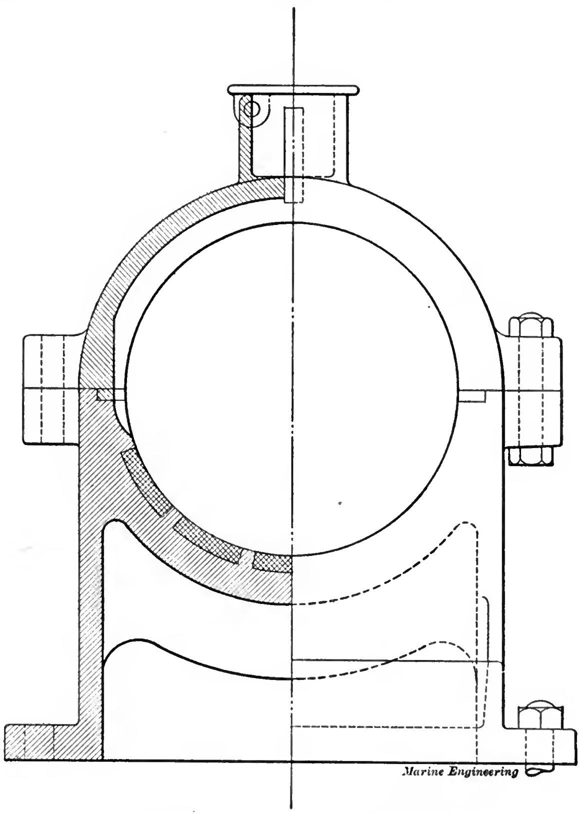


Fig. 147. Plain or Spring Bearing.

simply to protect the bearing surface and to support grease cups or other lubricating arrangement. A bearing of this type is shown in Fig. 147.

(6) *Thrust-Bearing.* At this bearing the thrust coming from the propeller is taken off the shaft and transferred to the ship. The length of shafting which is specially fitted for this purpose is known as the thrust-shaft. The special provision on the thrust-shaft consists of a series of rings or collars, as shown

in Figs. 148-150, while the bearing of the type shown in Fig. 149 has a corresponding series of channels into which the shaft rings enter when the thrust-shaft is in place. The bearing thus comes on the forward faces of the shaft rings and after faces of the intermediate bearing rings when the propeller is turning ahead, and *vice versa* when backing. The faces of the bearing rings are usually of white metal, thus giving a steel on white metal pair of surfaces. In order to take the weight of the thrust-shaft, a support of brass or white metal of the usual spring bearing form is usually provided at the forward and after ends of the bearing casing. The casing is furthermore commonly made in the form of a rectangular box, so that it can be filled with oil and thus flood the bearing with lubricant. Where the shaft passes through the ends of the casing, a stuffing-box or form of packing ring is provided to prevent the oil from leaking through. At the bottom of the casing a hollow space is often provided connecting freely with the general oil receptacle above, and through

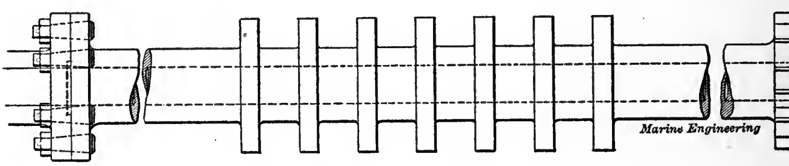


Fig. 148. Thrust Shaft.

which a pipe of copper or thin brass is led back and forth. Through this pipe cold water is circulated for cooling down the oil, and thus absorbing the heat of the bearing. The base of the bearing is secured to the ship through a seating specially strengthened and stiffened to take the thrust from the shaft and thus transfer it to the structure of the ship.

In Fig. 150 is shown a bearing somewhat more modern in type and very commonly met with in present day practice. It is known as the *horseshoe collar* bearing. The shaft is fitted the same as in Fig. 149, but the bearing, instead of being fitted with series of fixed rings and intermediate channels, is provided with a series of separate collars of the form shown in Fig. 151. These collars are provided with ears or lugs *A* and *B*, by means of which they are carried on side rods attached to the bearing casing, as shown in Fig. 150. These lugs in turn bear against adjusting nuts on the side rods as shown. In operation the thrust is transferred from the shaft rings to the faces of the col-

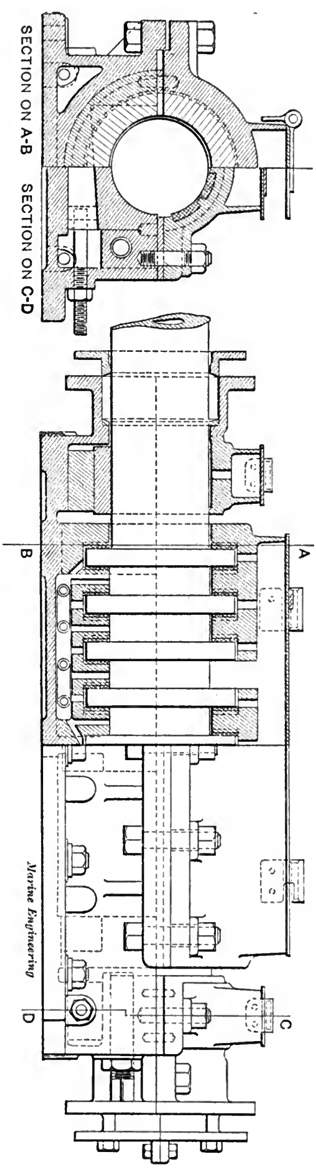


Fig. 149. Plain Thrust Bearing.

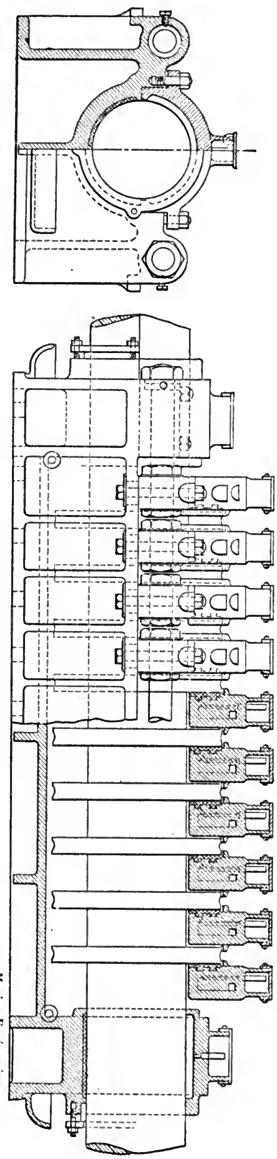


Fig. 150. Collar Thrust Bearing.

lars, thence through the lugs and nuts to the side rods, thence to the bearing casing, and thence to the ship.

It is readily seen that this arrangement allows of the individual adjustment of each collar as may be required by wear, or, if need be, of its removal and replacement by a spare collar even when under way, and without interfering with the action of the other parts of the bearing.

The collars are usually of cast steel, or, in some cases, of brass or bronze, and the bearing surface of white metal, carried in pockets, as explained above.

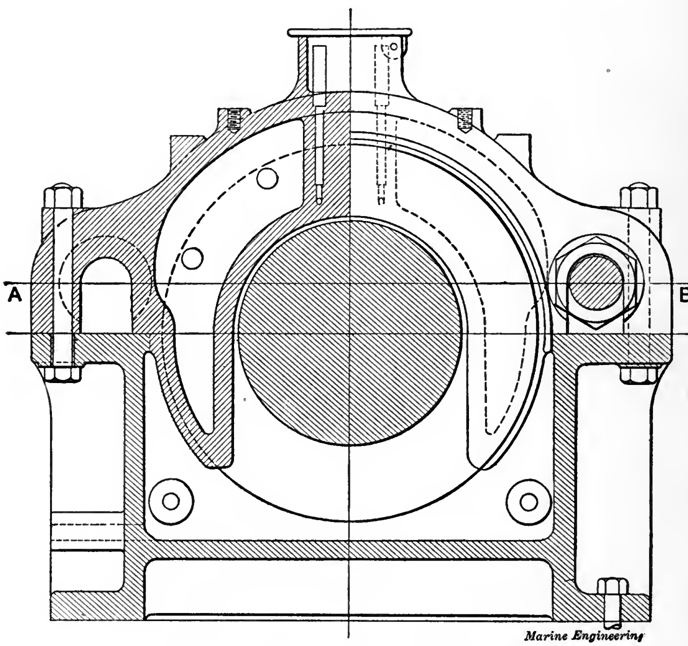


Fig. 151. Detail of Collar Thrust Bearing.

The arrangement of the casing as a receptacle for oil, the provision for spring bearings at the ends, and the provision of circulating pipes for cooling water, are similar to those above described in connection with Fig. 149.

The thrust bearing is variously located. In some cases it is placed immediately aft of the engine with its base connected directly to the engine bed-plate. In other cases it is placed at the after end of the inboard shaft, or just forward of the after shaft-alley bulkhead, and in others at some intermediate point.

(7) *Stern Bearing.* The general arrangement of the stern

bearing for a single screw ship is shown in Fig. 152. The entire distance through the stern of the ship from the stern-post to the after bulkhead of the shaft alley is lined with a tube *A B*, usually of cast iron. Within this are placed brass tubes *C D*, each perhaps about one-third the total length. These bearing tubes are provided with longitudinal channels slightly dovetailed in section, as shown in the figure, and into these are forced blocks of lignum vitae for a bearing. The arrangement is therefore somewhat similar to that for white metal, as described above, except that lignum vitae is substituted for white metal, and is placed in continuous channels running the entire length of the bearing tube with intermediate spaces between. The shaft itself is cased with a brass sleeve or casing, so that the bearing surfaces are brass on lignum vitae. It is found by experience that water is a lubricant for such a pair of surfaces, and it is chiefly for this reason that lignum vitae is so commonly used as the material for the stationary part of the bearing. The brass sleeve, which preferably extends the whole length of that part of the shaft within the tube, also protects the shaft from corrosion. This part of the shaft is known either as the *tail* shaft or *propeller* shaft. It is usually made a little larger than the line shaft to provide for corrosion, and also for the more violent shocks to which it is subject. At the forward end of the tube *A* is fitted a stuffing-box, through which the shaft passes to the after end of the shaft alley. At the after end of the tube the water enters freely through the

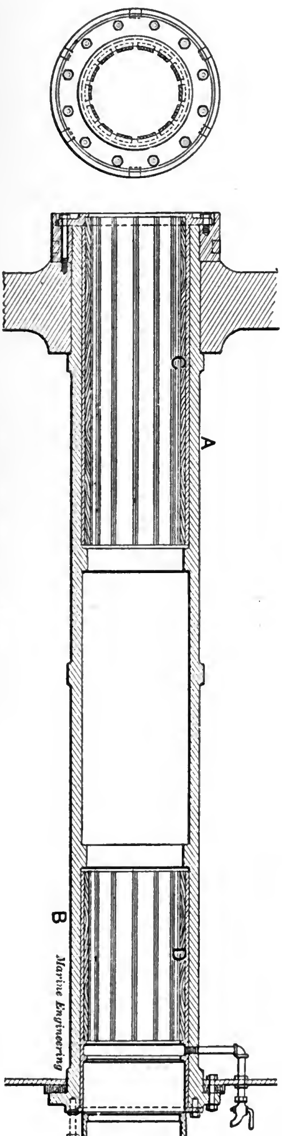


Fig. 152. Stern Tube and Bearing.

spaces between the lignum vitae and flows forward, thus serving to cool and lubricate the bearing. At the forward end the stuffing-box prevents leakage through into the ship. It is desirable, however, to fit a small pipe and cock so that water may be drawn from the tube as desired, in order to judge by its temperature as to the condition of the bearing. Instead of a pipe and cock the stuffing-box follower is sometimes loosened up so as to allow a sufficient leakage to insure circulation through the tube, and to serve as an index of the condition of the bearing.

In some cases, instead of water lubrication, the after end of the stern tube is closed against the water, and the tube is filled

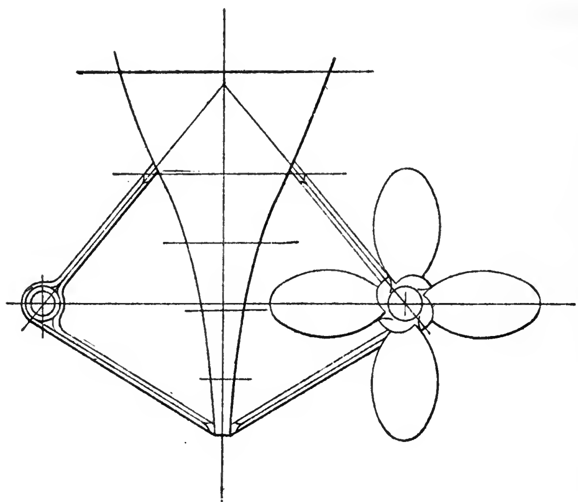


Fig. 153. Stern Brackets.

with heavy oil or tallow. Or if desired a stand pipe may be run up to a sufficient height, so that when filled with oil it will produce a pressure in the tube slightly greater than that of the water outside, and thus the leakage will be outward rather than inward. With oil lubrication the lignum vitae bearing surface is replaced by white metal.

In small craft the steel shaft without casing is often fitted directly in a brass bushing or bearing. In such case oil lubrication is to be preferred, but very commonly the bearing is left to run with such lubrication as the water can provide.

For twin screw ships, as shown in Fig. 143, the same general arrangement is used, except that the length of the tube where it passes through the skin of the ship is shorter, and frequently the lignum vitae bearing extends the entire length instead of over a part of the forward and after ends. A similar form of bearing is also provided in the shaft brackets or struts just forward of the stern post. The general form of such brack-

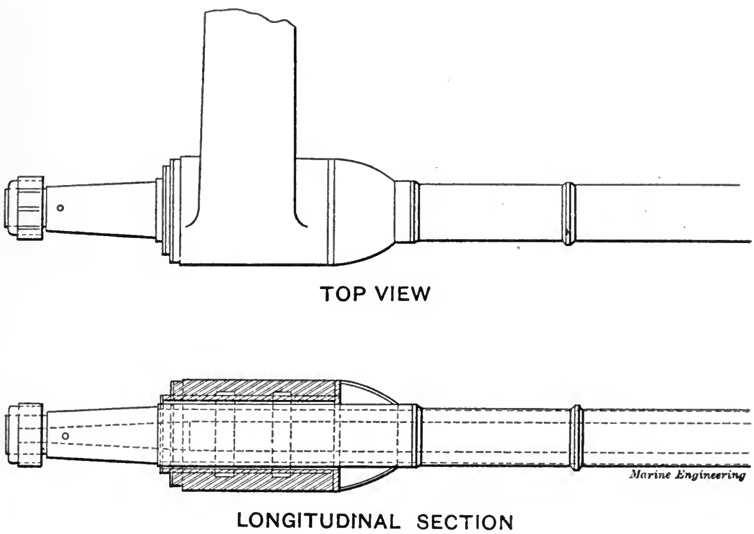


Fig. 154. Stern Bracket Bearing.

ets is shown in Fig. 153. On each side is a heavy steel casting secured firmly at top and bottom to the structure of the ship, and carrying at the apex a boss for the bearing; as shown in Fig. 154. This bearing is formed by a tube carrying lignum vitae strips as previously described, and in this way, with twin screws, the extreme after ends of the shafts are supported.

Sec. 22. WESTERN RIVER BOAT PRACTICE.

The peculiar conditions existing on the western rivers of the United States have resulted in the development of a special type of boat and propelling machinery. In the early days of river navigation the raft was first employed, and then came the flatboat, which has stood as the type of all later developments. On the rivers where at certain seasons of the year the water is shallow, the current swift and the channel narrow and tortuous, the usual style of keel boat would be of small service, while the light draft flat bottom craft seems admirably adapted for navigation under such difficulties.

Of the two varieties of boat, side wheel and stern wheel, the latter is preferred as on the whole the better suited to the all-around conditions of river navigation, and the flat-bottomed stern wheel craft may to-day be considered as the typical boat for western river navigation. Indeed this type of boat has met

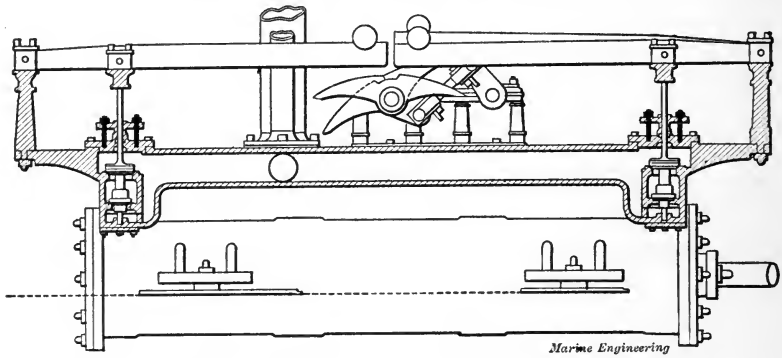


Fig. 155. Western River Engine, Elevation and Section through Valve Chests.

with much favor for river navigation in all parts of the world, and especially in South America, where they are largely employed.

The type of engine used on western river boats is shown in Figs. 155-157. It is horizontal and of the simple non-condensing type. Two such engines are usually employed, one on each side placed close to the guards, with the axis of the cylin-

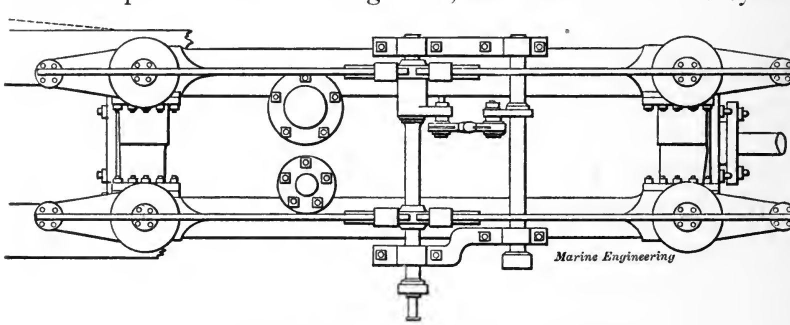


Fig. 156. Western River Engine. Top View.

ders fore and aft, and with the connecting-rods coupled to the cranks on the stern wheel paddle shaft.

The cylinders are of relatively small diameter and long stroke, the dimensions in a typical case being 24-inch diameter by 96-inch stroke. The most peculiar feature of the engine, however, is found in the valve gear. The valves themselves are usually of the double-beat poppet form, as shown in Fig. 155, and each cylinder is provided with four, two for steam and two for exhaust. These valves are actuated by a cam valve gear mechanism, as briefly described below.

The steam valves with their connecting pipes are located on one side of the cylinder, while the exhaust valves and connections are on the other side. Each set of valves is operated by separate rocking cams or levers, which receive their motion through rockers and connections from a special cam located on the main paddle shaft.

The cam type of valve gear possesses peculiar advantages, especially for long stroke, slow revolution engines such as are used in these cases. The motion of the valve may thus be made intermittent, giving a quick opening and closure, with interme-

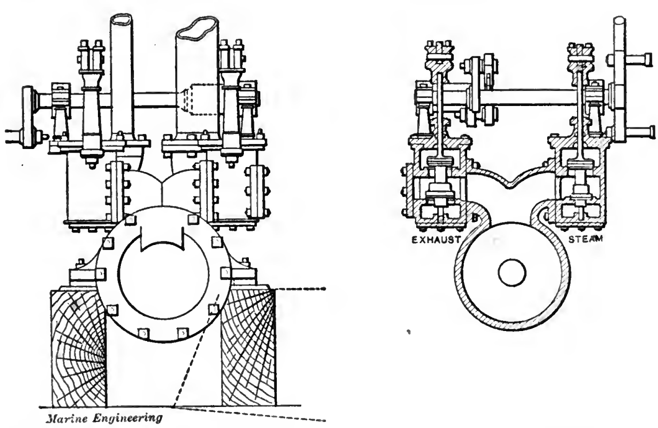


Fig. 157. Western River Engine, End View and Section Showing Valves.

diate periods of rest or very slow motion. It is also peculiarly adapted to the elastic movements of the boat during the process of loading, unloading, etc., movements which continually vary the distance between the main shaft and rock shaft, and which, with almost any other type of gear, would introduce serious disturbance into the movement of the valve and the distribution of the steam.

For the operation of these valves in the common type of gear two cams are used; one known as the *full stroke* cam and one as the *cut-off* cam. When the engine is in full gear the full stroke cam operates all four valves, raising one exhaust and one receiving valve at opposite ends of the cylinder at the same moment, and alternately at each end, thus distributing the steam as required to carry the piston back and forth continuously. The one cam does all the work in the full gear motion of the engine both ahead and astern, and is hence in its neutral position when the crank is at its dead point. The cut-off cam is so arranged as to be hooked on after the full stroke cam has given

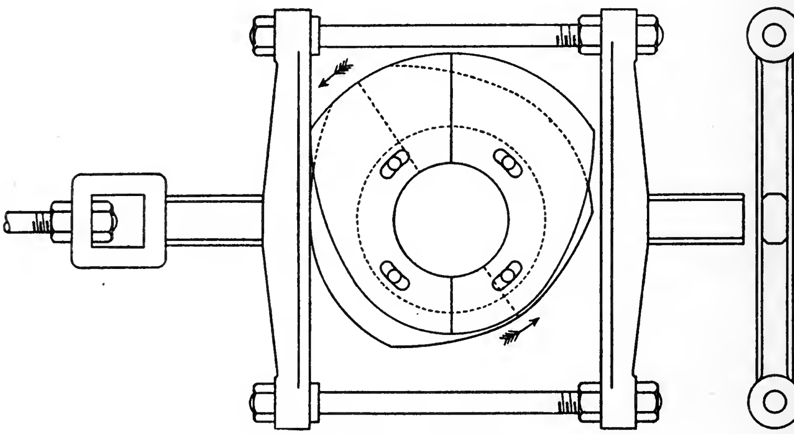


Fig. 158. Full Stroke Cam with Yoke.

headway to the boat, and is used in the go-ahead motion only. This cam is so designed that the steam is cut off at any designated point in the stroke, as at ½, ⅝, ¾, etc.

The form of a full stroke cam with its yoke is shown in Fig. 158, and of a ⅝ stroke cut-off cam in partly dotted lines.

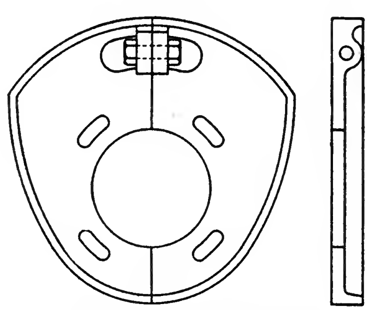


Fig. 159. Full Stroke Cam.

In Fig. 159 is shown the usual type of construction of the full stroke cam, and in Fig. 160 similarly the ¾ cut-off cam.

With this arrangement of gear the exhaust is opened and closed just at the end of the stroke, and hence neither early exhaust opening nor closure for cushion can be obtained. A means of obtaining the former has been found by blocking up the exhaust lifters somewhat, so that the valve will be slightly open when the engine is on the dead point.

This insures an earlier opening of the exhaust and so clears the cylinder for the return stroke, but it gives likewise a later exhaust closure, so that with the engine on the center both exhaust valves are slightly open, and in full gear operation a slight "blow through" will occur. This disappears, however, when the cut-off cam is engaged, because the opening movement of the latter is much slower than that of the full stroke cam.

Various modifications of this simple cam gear have been introduced with a view of improving the general operation, especially by the provision of means for obtaining both steam and exhaust lead and compression, as well as independent movements for the go-ahead and backing motions.

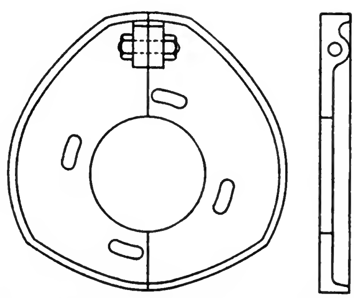


Fig. 160. Three Quarter Cut-off Cam.

In the Sweeney valve gear two full stroke cams are employed, one for go ahead and one for backing, each set so as to give suitable exhaust lead and compression, while a separate cut-off cam is fitted for the go-ahead motion.

The crossheads of these engines are usually of the locomotive type, with long brass gibs bearing on the top and bottom guides. The connecting-rods are commonly of wrought iron or wood, with iron or steel fittings, and form one of the most peculiar features of these engines. Wood is often thus preferred over metal because it seems to be better capable of standing the shocks and peculiar twisting strains which come upon the rod, and in spite of the strangeness of the combination, we find in some modern boats a fluid compressed nickel steel paddle shaft with a wooden connecting-rod. The rods are very long, frequently as much as eight times the crank, and the best rods are made of Oregon fir, reinforced with iron straps which are let into the body of the rod and through bolted. The ends of the rods are fitted with brass boxes with straps, gibs, keys, etc., in the usual manner of fitting up such form of rod, and as illustrated in Fig. 138.

In some cases of modern river boats on the Pacific Coast many changes have been introduced looking toward a closer approach to usual marine practice. In a typical example of such improved practice the engines are horizontal tandem compound, the high-pressure cylinder having piston valves and the low-pressure cylinder slide valves, both operated by excentrics and link work in the usual way. In this case there are two engines developing about 1,500 I. H. P. each. The cylinders are 22½ in. and 38¾ in. diameter, with a stroke of 8 feet, and are intended to make thirty revolutions per minute.

The crank shaft for such engines is built up in structure, the two cranks being separately forged and secured to the paddle shaft by shrinking and appropriate keys. The shaft is usually fitted with hexagonal bosses where the wheel flanges are to be secured. The latter are usually of cast iron, heavily ribbed and reinforced by wrought iron bands shrunk on their hubs and outer circumference. These flanges are fitted to the hexagonal bosses on the shaft, and are secured with suitable keys. They are provided on one face with sockets for the wheel arms, which are of wood. These latter are further strengthened by circular bands of iron bolted near the outer ends, and also by oblique bracing which is worked between them.

The buckets are also of wood, 2-inch oak plank of suitable width and length being a standard material. They are secured to the wheel arms by special clamp bolts, and are so located relative to the draft of the boat as to be immersed only some 4 to 6 inches when the steamer is running light. In some cases the buckets are divided at the center, forming really two sets, staggered with reference to each other, and thus reducing the shock of the wheel as it enters the water.

DOCTOR.

This peculiar feature of western river practice as illustrated in Fig. 161, is a combination of feed pump and feed water heater. As here shown, the doctor consists of a vertical beam engine with crank and flywheel operating four pumps. Two of these are simple lift pumps drawing water from the river and delivering it into the heating chambers overhead, while the other two are feed pumps proper, taking their supply from the heaters and forcing the water into the main boilers. Each lift and force pump is designed of sufficient capacity to supply the entire battery of boilers, so that one of either kind may be dis-

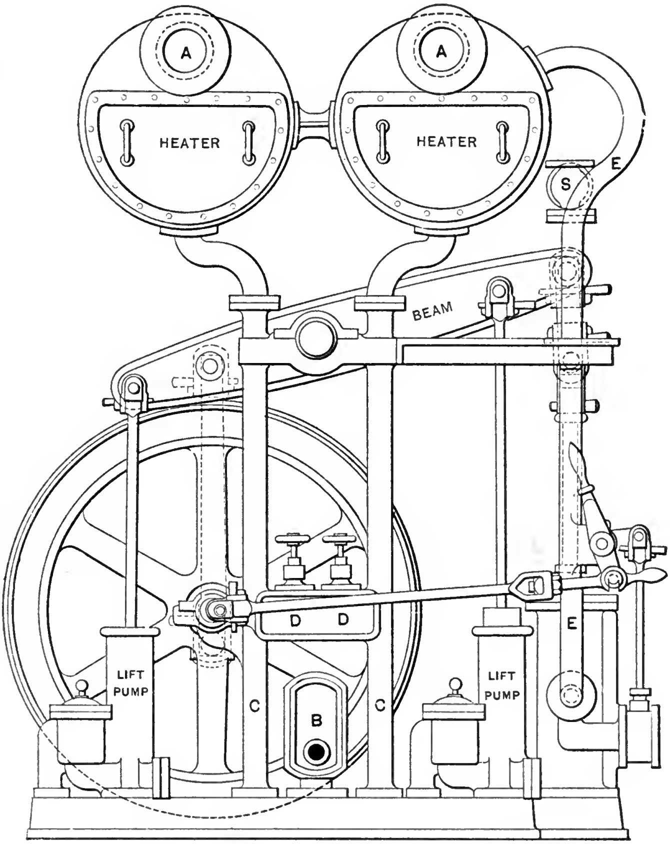


Fig. 161. Western River Boat "Doctor."

connected for examination or repair without disturbing the regularity of boiler feed supply. The various parts of the machine are erected on a deep cast iron base plate which contains various ports and passages, forming the water connections between the various pumps.

The suction pipe from the river is connected with a vacuum chamber, and communicates through a passage in the base casting with the suction side of the lift pump. The discharge from these pumps is then led by other passages to the columns, which serve as discharge pipes, supports for the engine beam and for the heaters. Valves are also located in these columns, by closing which the water in the heaters may be prevented from returning at such times as it is necessary to open up a pump for examination or repair.

The heaters themselves consist of wrought iron shells riveted to cast iron heads, through which the exhaust steam from the main engines is led on its way to the exhaust pipe. The exhaust steam thus comes in direct contact with coils of copper pipe that lie in the lower part of the heaters, and through which the feed water is forced and finally discharged below a diaphragm. Beyond this the exhaust steam and water are to some extent in direct contact, the latter being finally lead down through the pair of columns on the opposite side of the machine to the feed pump inlet valves in the base-plate. The head of water in the columns is thus sufficient to flood the valves and prevent the pump from missing stroke, even with the hottest feed water which the heaters can furnish.

The lift pumps are fitted with long pistons having either cup leather or square gum packing, while the feed pumps are of the common plunger type. The pump valves are flat disks of brass made quite thick so as to avoid the need of springs, and also to allow metal for re-facing. The engine part of the doctor is very simple and will call for no special comment, consisting simply of a steam cylinder for actuating the beam and thus giving motion to the four pumps as described.

In some cases of recent river practice injectors have taken the place of the "doctor," and if they can be depended upon they are of course much preferable, being easy to handle and occupying little or no space otherwise valuable. While it is probable that they can thus be used to advantage on certain of the upper portions of the western rivers, it is hardly possible that they could be used at all in many other localities on account of the sand and grit which is held in suspension by the water, and which would cut out the injector tubes so rapidly that their use would be out of the question. For this reason it seems likely that the "doctor" will hold its own in all such localities and that it will continue to be an important detail of western river practice.

Sec. 23. THE STEAM TURBINE.

Within the past few years the application of the steam turbine to marine propulsion has produced results which have attracted world-wide attention, and it seems at present not unlikely that the part taken in the future developments of marine propulsion by this form of motor will be one of increasing importance. The present development is represented by the Parsons' steam turbine as fitted on the *Turbinia*, and later on the British torpedo-boat destroyers *Cobra* and *Viper*, and Clyde passenger steamer *King Edward.*

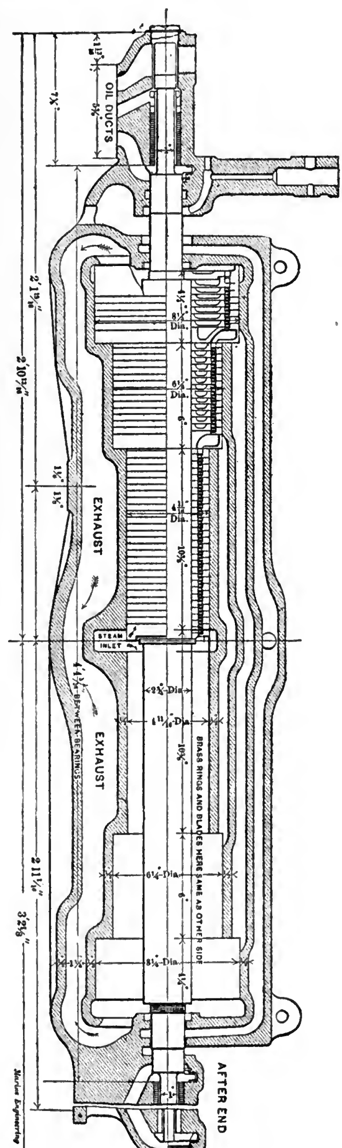


Fig. 162. Parson's Steam Turbine.

This type of motor, one form of which is represented in Fig. 162, consists of a cylindrical case carrying rings of inwardly projecting oblique guide blades, while within these revolves a shaft carrying rings of outwardly projecting oblique blades. There is a clearance of about 1-16 inch between the successive rings of blades and guides, and between the ends of the former and the case, and the ends of the latter and the body of the shaft. There is thus left between the shaft and the case an annular space filled with alternate rings of blades attached to the shaft and guides attached to the case. The steam when admitted passes first through a ring of fixed guides by means of which it is given a rotational motion, and then projected on to the first ring of blades. It is then thrown on to the following ring of guides, by means of which its direction is again changed, and it is then thrown in the same direction as at first upon the second ring of blades, and so on, passing from one ring to the next, and giving to each a part of its energy and thus maintaining the rotation of the shaft. It must be especially understood that the steam in a turbine acts by impact or reaction, and not by pressure. As the steam rushes against the blades and its direction of flow becomes thereby changed, it exerts a reaction on the blades, and this constitutes the force which produces the rotation. The energy which the steam possesses in virtue of its temperature and pressure cannot therefore be directly used in the turbine as in the common steam engine, but it must first be transformed into the energy of motion as in the rushing jet. This transformation is effected by the gradual expansion of the steam as it passes through the turbine. This gradual expansion keeps up a continual transformation of heat energy into the energy of motion, and this energy is as constantly transferred to the moving blades, and thus the original heat energy of the steam is transformed into mechanical work. In order that the expansion may be carried on continuously, the cross-sectional area occupied by the blades is increased from time to time, as shown in the figure, thus making a series of steps of increasing diameters for the rings of blades and guides. In a moderate sized motor there may be 50 to 80 successive rings thus arranged in from three to five groups.

The steam pressure acting on the annular rings separating these steps, and also on the blades themselves, would set up a severe end thrust. This may be balanced, as in Fig. 162, by arranging two sets of such steps on the same shaft varying in opposite directions, and thus balancing each other in end thrust.

In another method special disks are fitted on the shaft with grooves in their outer surface, within which project rings carried in the frame. These rings and grooves form a nearly steam tight joint, and the annular area is so adjusted in amount that the steam acting on it will balance the thrust acting on the blades and corresponding annular area of the main shaft. In addition a special thrust bearing is fitted for taking any residual end thrust and for making the necessary adjustments.

The arrangement thus briefly described constitutes a simple turbine. By combining two or three such in series and leading the steam from one to another through them, the so-called compound turbine is formed. In a simple turbine the steam may be expanded some six or eight times, so that by compounding a total expansion of, say, 30 to 60 could be obtained, and by a triple combination a total expansion of 200 and more may be effected.

The revolutions of the turbine are high. This is a necessity of its efficient performance. The *Turbinia* has three each of about 600 to 700 horse power, and the revolutions run up to about 2,000 per minute. In the *Viper* and *Cobra* there are four turbines, each of about 1,500 H. P., running at about 1,200 revolutions per minute. With the former a speed of about 34 knots has been reached, while with the latter speeds of over 35 knots were reached.

The efficiency of the compound type of turbine is not very different from that of good compound or triple expansion engines of the usual type. The weight for the same power is somewhat less, though the difference is not great when compared with light reciprocating engines forced in a manner corresponding to the conditions on these boats. In other words, the turbine shows no very great saving over the weights of the lightest types of reciprocating engines as designed for torpedo craft and fast launches.

The great advantages of the turbine are found in its compactness and absence of reciprocating motions, and thus in the entire absence of the forces which cause vibrations. With extreme speeds and in many conditions of modern practice the vibration forces become a serious question, and a motor entirely free from them becomes immediately of great importance in all such cases.

In regard to questions of durability, maintenance, liability to derangement or accident, etc., the experience available at present is too small, and we must wait for the future to answer these and many other questions which bear on the applicability of the turbine to the various conditions of marine practice.

Sec. 24. ENGINE FITTINGS.

[1] Throttle Valve.

The purpose of the throttle valve is to provide a means for quickly opening or closing the main steam-pipe near where it connects with the high pressure valve chest, and thus to provide for the quick control of steam to the engine when stopping and starting. A great variety of valves have been employed for this purpose. The necessity for quick operation, especially by hand gear, requires usually some form of balanced valve, though in very small sizes an ordinary globe or straight-way or gate valve, as shown in Figs. 166, 168 may be used. Of these the straight-way valve is much to be preferred, as when open it leaves practically an unobstructed passage for the flow of the steam.

(1) *Gridiron Valve.* The gridiron is another form of unbalanced valve sometimes employed as a throttle. This valve, as shown in Fig. 163, consists of a series of bars and ports corresponding to a like series in the valve chest, and giving a series of openings for the steam, wider or narrower according to the position of the valve. With such an arrangement a considerable area of opening may be obtained with a comparatively small

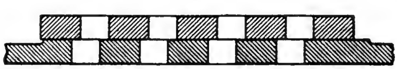


Fig. 163. Gridiron Valve.

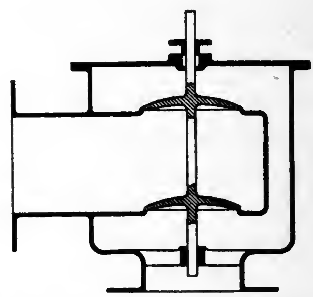


Fig. 164. Double Beat Poppet Valve.

movement of the valve, and a screw or some other form of slow motion gear may be employed without loss of quick opening and closure. This form of valve is, however, but rarely met with in modern practice.

(2) *Double Beat Poppet.* The double beat poppet valve, as shown in Fig. 164, has been much employed as a form of balanced throttle. The upper disc is slightly larger in area than the lower, so that if the live steam is on the outside the net load on the valve is that due to the difference of the two areas, and this may be made very small. The resistance to opening is thus no more than can be readily overcome with a direct hand gear, as for example, a simple lever or other like arrangement.

The chief difficulty with this valve is in keeping it tight, variations of temperature and the consequent expansions and contractions often tending to slightly unseat one disc or the other.

(3) *Butterfly Valve.* The butterfly valve has also been wide-

ly used as a balanced throttle. It consists of a disc of elliptical form carried on a spindle and swinging within a cylindrical casing. When closed it rests obliquely on the inner surface of the casing, thus closing the passage around its outer circumference. When full open it swings into a position with its plane lying along the pipe, thus leaving the passage nearly free for the flow of steam. This form of valve is quite perfectly balanced, but it is difficult to keep tight. If the angle of obliquity with the surface of the casing is too small, it may also be liable to stick fast, due to unequal expansion of the valve and casing. In another form of butterfly valve, as shown in Fig. 165, however, the

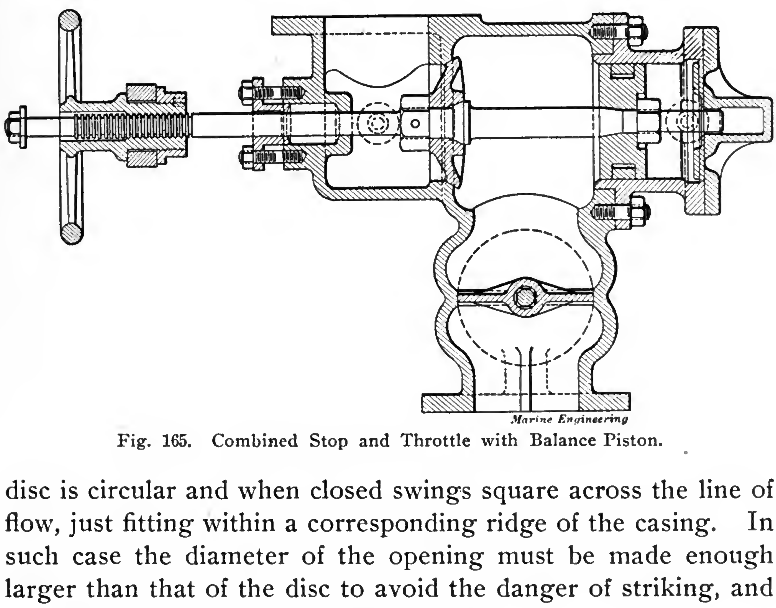


Fig. 165. Combined Stop and Throttle with Balance Piston.

disc is circular and when closed swings square across the line of flow, just fitting within a corresponding ridge of the casing. In such case the diameter of the opening must be made enough larger than that of the disc to avoid the danger of striking, and considerable leakage will usually result.

(4) *Disc Valve With Balance Piston.* A plain disc with balance piston attached to the stem is quite commonly employed in modern practice for the throttle or for the stop and throttle combined. Such an arrangement in combination with a butterfly valve is shown in Fig. 165. By this means the pressure on the piston nearly balances the load on the valve, and it may thus be operated by hand gear. Steam may also be admitted back of the piston by a pipe with stop-valve operated from the working platform. By this means the disc may be balanced when once off the seat, and closure effected as easily as opening.

(5) *Power Operated Throttle.* In some cases with large engines the throttle is operated by steam power instead of by hand, steam being admitted to an operating cylinder by means of a hand lever or other like arrangement. Here the steam acts upon an auxiliary piston and by suitable connections produces the movement of the throttle as desired. In such cases the connections are often of the "floating lever" type, as in the reversing

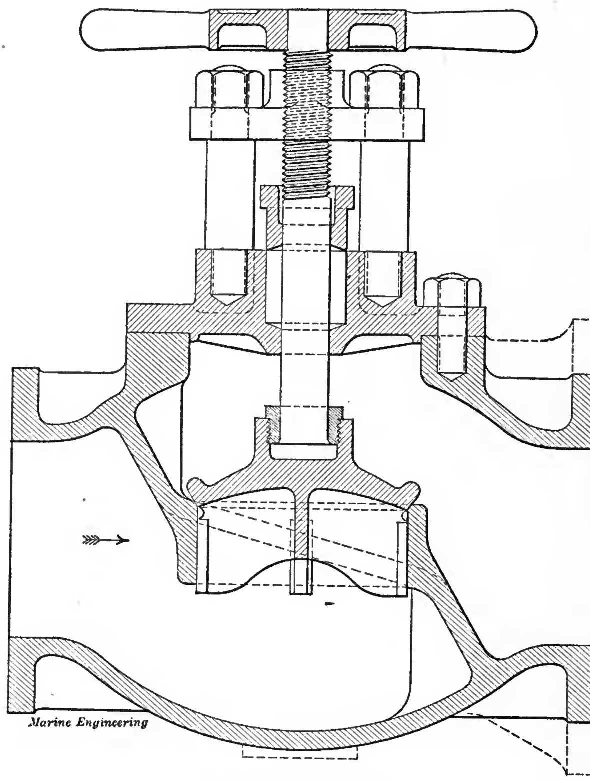


Fig. 166. Globe Valve.

gear described in [5,] so that the valve will follow the hand lever in its movement back and forth, and the combination becomes thus equivalent to a direct operation of the throttle by hand.

[2] Main Stop Valve.

The throttle valve from its construction can rarely be closed sufficiently tight to prevent leakage of steam, often considerable in amount. To provide a shut-off without sensible leakage a stop valve is often fitted in addition. Such valves may be of various types, as shown in Figs. 166, 167, 168.

(1) *Globe Valve.* This valve, as shown in Fig. 166, consists of a metal chamber of globular or spherical form with flanges for connecting to the line of piping. Within the body is a partition separating the portions connected with the two openings, and in this partition is a hole with conical seat upon which the valve with corresponding conical face bottoms when closed. The valve is attached to a threaded spindle which works in a nut either formed in the neck which contains the stem, or carried

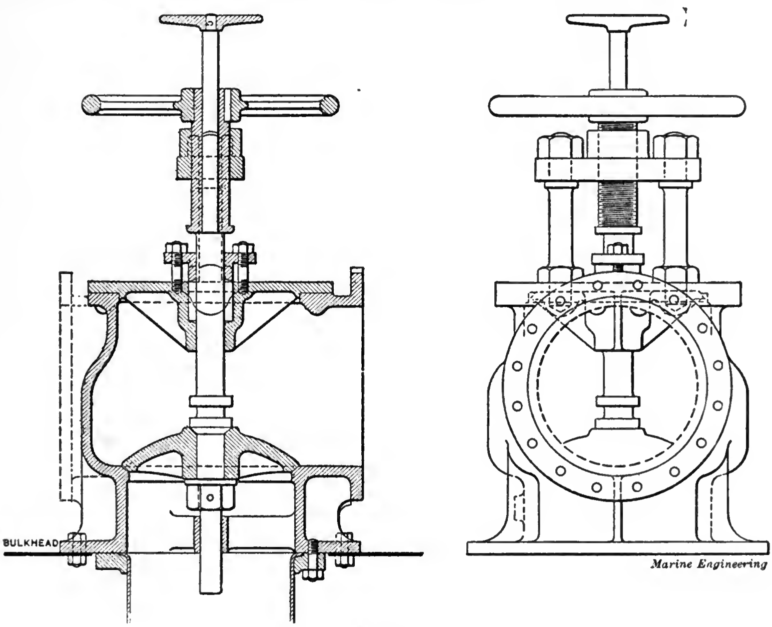


Fig. 167. Angle Stop Valve.

outside on a girder supported by stud bolts, as shown in the figure. To the end of the stem a handle is attached, and by this means the valve is opened or closed as desired. The stem is packed by means of a stuffing-box and soft packing compressed by a gland of the usual form as shown. In small sizes the gland is usually replaced by a form of nut threaded to the neck, which contains the stem, and compressing the packing between the nut and the bottom of the packing space.

(2) *Angle Valve.* In this type of valve, which is an angle or elbow and a valve combined, the seat and valve-face, as shown

in Fig. 167, are placed square across one of the openings, thus shutting off all flow through it when the valve is closed. When the valve is opened, however, the passage is left free, according to the degree of opening, for the flow of the liquid or vapor around the angle and on into the following section of pipe.

When the stop valve is of the disc form it is very commonly of the angle type and arranged to go in at a turn of the pipe, as shown in Fig. 167. In this case also the valve is attached to a bulkhead and the arrangement will serve to show the method of

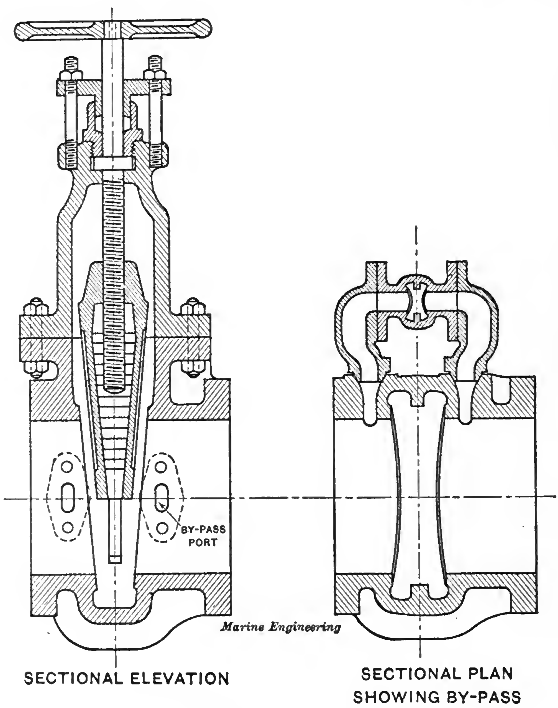


Fig. 168. Gate Valve.

carrying steam through a bulkhead and of making up the joints connecting together the steam pipe, the stop valve and the bulkhead plate.

(3) *Straight-way or Gate Valves.* In this form of valve, which is shown in Fig. 168, the moving part consists of a special form of slide which is moved by a screw back and forth across the opening of the pipe. There are various special forms and devices for securing tight contact between the valve and its seat when closed, and thus making the valve tight under steam pres-

sure. The general arrangement of Fig. 168 will, however, serve to show the main features of valves of this type. When closed and with pressure on one side of the slide only, there is sometimes some difficulty in opening the valve. To relieve this condition a small by-pass, as shown, is often fitted. This admits steam to the farther side of the valve, thus balancing the load and making the operation of opening much easier. In another form the valve slide is made of two parts, hinged together and with the end of the spindle working between them in such way that when screwed hard down it is forced as a wedge between the two parts thus forcing them against their seats. When the handle is turned in the reverse way the first action is to partly withdraw the stem from between the two parts of the slide, thus easing them from their seats and allowing them to be readily withdrawn as the stem is turned farther back.

When the throttle takes the form of a plain disc with balance piston, as in Fig. 165, no additional stop is thought necessary, and such an arrangement is often known as a combined stop and throttle valve. In such case, however, a screw stem may be provided with connections for bringing it into use when closing the valve down as a stop.

[3] Cylinder Drain Gear and Relief Valves.

A certain amount of water is likely to collect in the steam chests and cylinders, either carried in with, or condensed from the entering steam, especially when warming up the engine preparatory to getting under way. Provision must be made for getting rid of this water as occasion may require, and to this end the so-called *cylinder drains* and *relief valves* are fitted. The drains are usually plain cocks piped up and connected to the parts to be drained, and with the valve stems connected by levers and bell-cranks to operating handles at the starting platform. The drains in the bottom of the cylinder or valve-chest will naturally be placed at the lowest point at which water can collect, or as near to such point as is practicable. Those in the upper end of the cylinder will be placed at such a height that the opening will not be covered by the piston when at the top of the stroke.

For small engines, auxiliaries, pumps, etc., the drain valves are often plain globe valves piped into the cylinder at convenient points, and operated independently by hand.

The discharge of the drains is piped away either into the bilge, or into a fresh water collecting tank.

In addition to such gear, which is operated by hand, and when judgment may call for its use, it is necessary to provide automatic relief valves for the discharge of water in larger quantities should it find its way into the cylinder by priming or in other ways. Such a relief valve is in the form of a safety valve, and may be set to open at any pressure desired. Such valves are sometimes connected up with operating levers, also led to the starting platform, so that they may be operated by hand from that point. In such cases only the one set of valves is often fitted, automatic when necessary, and under hand control when desired. In some cases with large engines a double set of automatic relief valves is furnished, a pair of large valves not under hand control, and a smaller pair under hand control, as described above.

[4] Starting Valves.

In order to assist in starting the engine, especially if the high pressure piston happens to be on or near the center, a valve and pipe are usually provided for admitting steam direct from the steam pipe or high pressure valve chest, to the first receiver, or intermediate pressure valve chest. This will give sufficient load on the intermediate pressure piston to start the engine, and carry the high pressure piston off the center, and thus give the engine a chance to start in the regular way. In case the high pressure and intermediate pressure cranks should be opposite, and thus both pistons on or near the center at the same time, the auxiliary pipe will lead to the second intermediate piston, or to the first cylinder, whose piston is not on the center with the high pressure. In some cases the passage of the steam to the next cylinder beyond the high pressure is effected by the opening of a valve connecting the steam and exhaust sides of the high pressure valve chest. Such valve being opened the steam finds its way directly to the point where it is needed.

Valves for this general purpose are variously called *pass-over*, or *starting valves*, or *monkey tails*. They are either in the form of a cock or of a small slide valve, in either case admitting of full opening by a single short stroke of a convenient hand lever, to which they are connected by suitable rods and connections.

[5] Reversing Gear.

The various links of a Stephenson valve gear, as will be seen in Section 53, are connected by side or bridle rods to arms on the rock or "weigh" shaft. To reverse or link up with such a gear, therefore, it becomes necessary to provide some means for turning this shaft back and forth, and for holding it under complete control at any position desired. The form of reverse gear most commonly employed in American practice is of the so-called "floating lever" type, and is illustrated in Fig. 169.

It consists of a cylinder, AB, with piston and rod, D, connected by a link from E to an arm on the engine rock shaft, and thus connecting with the links.

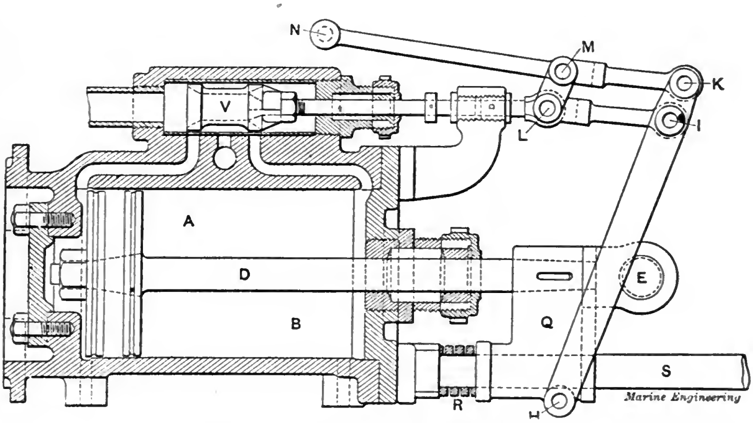


Fig. 169. Floating Lever Reverse Gear.

As the piston is moved back and forth by the steam this arm will evidently be carried with it, and the various Stephenson links, or like parts of other types of valve gear, will be moved as desired, each through its connection with the rock-shaft. The steam to the cylinder, AB, is controlled by a slide valve, V, either plain or of the piston type. This valve has very small lap so that from the position when covering both ports but slight movement is needed to uncover. To the stem of the valve is attached a link, LI, which at the latter point is joined to a bar, KH. The lower end, H, of this bar is attached to a lug, Q, on the piston rod, and, therefore, moves with the piston. The upper end, K, is connected through a link, KN, to a hand lever, which is provided with means for clamping in any position desired.

Suppose now the gear in the position shown and with the valve covering both ports. Let the hand lever be moved so as to throw KN to the left. For the moment H will be a fixed center, and with the connections shown the valve will be moved to the *left* also. With this arrangement of connections an inside valve (Section 46 [7]) must be used, and, therefore, steam will be admitted to the left hand end of the cylinder, AB, and the piston forced to the right. Let the hand lever, carrying with it the valve, be thus moved over a certain distance and then held or clamped there, thus fixing NK. The point, K, will thus become for the time a fixed center, and the movement of H to the right will carry the valve in the same direction, and thus finally close the ports, shutting off the supply of steam at one end and closing the exhaust at the other. The movement of the piston will thus be stopped and the gear will be held in the position reached. It is clear that for every position of the hand lever there will thus be some position of the piston, rock shaft arm and main valve gear, for which the valve will be brought to mid position and the piston and gear thus brought to rest, and that the steam will carry the gear to this position and then automatically shut off and stop. If the hand lever is moved but slightly so as to barely displace the valve, the piston will move but a small distance before again covering the ports and coming to rest. If the handle be moved to an extreme position the valve will be moved far over and the steam will rush the piston and gear over into the extreme corresponding position. In short, the position of the gear for equilibrium under steam will correspond exactly to that of the hand lever, and wherever the latter may be placed, the gear will run to the corresponding position and then stop. It is also clear that if the hand lever be moved slowly, the piston and main links will follow along at equal pace, stopping when the handle is stopped and moving when it moves. Also if the handle be slightly displaced and left to itself the friction of moving the valve will be usually more than that of moving the handle, and in consequence the point, I, will become for the time a fixed center, and the piston will move along, the valve remaining open and the connection, HKN, moving the handle over at equal pace with the link. This will continue till the handle comes against a stop at the end of its path. The point, K, will then become fixed, and the further movement of the piston will move the valve into mid-position, thus shutting off steam and bringing the gear to rest in the position corresponding to that of the hand lever.

To take up sudden shocks and provide a safeguard against putting the link over too rapidly and thus overrunning at the end through the inertia of the parts, spring stops or buffers, R, are provided on a rod, S, against which the lug, Q, comes at the end of its run.

It is thus seen that this gear furnishes a very perfect control over the main valve gear, the action being the same as for a man operating the gear directly, and thus giving him readiness of control with the least mental effort, and the least liability of error in a moment of hurry or excitement.

In addition to the spring buffers, as shown in Fig. 169, a form of plunger control is sometimes added. In this arrangement the piston rod of the reverse cylinder is continued backward and connected to a second piston or plunger working in a cylinder filled with oil. The operation of the plunger is to transfer the oil through a suitable pipe connection from one end of the cylinder to the other, and as this passage may be throttled at will by a stop valve, all possibility of slamming or of violent motion may be removed. A further advantage of this arrangement lies in the fact that with suitable pipe connections to a hand pump, the oil may be drawn from one side of the plunger and forced in on the other, thus giving a control over the valve gear by hand power in case of derangement of the power control.

Of other forms of reverse gear the so-called *all around* gear is quite commonly met with in English practice. The main links are connected up to a small engine which makes a large number of revolutions in running the link over from one extreme to the other. This engine is under the control of a small link which is directly operated by hand. A form of lever stop is usually provided which will either reverse or middle the small link and bring the engine to rest when the main links have reached either extreme of their travel.

In engines for small yachts, launches, etc., the links are placed directly under the control of a hand lever.

The various other types of valve gear may be operated by any of the forms of reverse described. With all valve gears, as described in Chapter VII, the reverse is effected by the movement of some piece of the gear from one position or location into another, and so back and forth for the various degrees of linking up, etc. By suitably connecting such piece to the power reverse gear the control may, therefore, be obtained in the same manner as for the Stephenson link as described above.

[6] Turning Gear.

It is always necessary to provide some means for turning the engine other than by steam on the main pistons. This is necessary for moving the engine when in port for adjustment of bearings, setting of valves, etc. The turning gear usually consists of a large worm wheel placed on the main shaft just aft of the bed-plate, geared down through worm and spur gearing to a small engine, usually a double simple engine with cranks at 90°. The gearing ratio is such that many hundred revolutions of the turning engine may be required to one of the turning wheel or main engine shaft. This gear must be so arranged as to be readily thrown in and out of connection with the main turning wheel. This is usually accomplished by carrying the main worm on a shaft which is pivoted, and which can thus be locked in either of two positions, in one of which the worm is in gear, and in the other out of gear, or else by driving the worm on a shaft with a feather, thus providing for endwise motion, and for fixing it in either of two locations on its shaft, in one of which it is in gear and in the other out of gear. The latter is the arrangement more commonly met with.

Where a turning engine is not provided the turning wheel is usually arranged for operation by hand through worm gearing operated by a lever with pawl and ratchet arrangement, or by some similar device.

In some cases the engine is turned by a hydraulic jack placed under a movable chock piece located in sockets cast in the turning wheel. This chock is shifted from one socket to another as the jack shoves it upward, and thus the engine is slowly turned.

In small engines the turning wheel is often simply a form of gear wheel with shallow teeth in which a pinch bar is worked, and by this means the engine may be slowly pried around. Such a wheel is known as a *pinch* wheel.

[7] Joints and Packing.

The joints to be considered under this head are of two kinds. (1) Fixed joints as those between a cylinder or valve chest cover and flange, and (2) Sliding or slip joints as those between a piston rod and the stuffing box, or the slip joint in a length of steam piping.

For making up stationary joints a great variety of packings are in use, the difference depending to some extent upon the temperature to which the joint is to be subjected. Thus for joints to stand high temperature, as with boiler manholes, cylinder heads, etc., sheet asbestos either plain or in combination with other materials is used. There are also various kinds of packing in which rubber in one form or another is used either in combination with some fibrous material as sheet canvas, or as a constituent of some form of compound. The tendency of rubber by itself is to grow dry, hard and brittle, especially under the action of heat, and the purpose of the modern forms of rubber compound is to avoid this tendency, at the same time retaining its elasticity and joint making qualities. For joints not subject to the action of high temperature, similar forms of packing are used, though with a greater proportion of rubber, if desired.

The strip or ring of packing which is cut out and fitted for the joint is called a *gasket*.

In making up such a joint it is well to smear the surfaces of the gasket with a mixture of black-lead and grease or oil. This will aid somewhat in making the joint, and very much in the removal of the cover and gasket at a later time without tearing the latter. With such precaution and when the temperature is not high the same gasket may be used several times over without loss of its joint making qualities.

In addition to gaskets made of such materials as described above, joints are also made with gaskets of corrugated sheet copper, or of plain copper wire. For high pressures such gaskets have proved quite successful. The soft copper is expanded between the harder metals of the flanges, and spreads, filling the surfaces where it touches, thus making a tight joint.

For sliding joints as between a piston-rod and stuffing-box, the greatest variety of packings is likewise in use. They may be broadly divided, however, into the two classes, fibrous and metallic.

The fibrous packings are made of the same material as the sheet packings above described, and are either round, square or triangular in section. For use they are cut to such lengths as may be necessary and placed in the stuffing-box in layers or turns, the joints between the ends being shifted so as not to come one above another. The stuffing-box, as shown in section in Fig. 170, consists of a cylindrical chamber or box, EF, with cavity B. This is bolted by means of the flange, F, to the lower cylinder head. The part, CC, is known as the gland or follower and is carried by two or more studs, as shown. At the bottom or upper end of the box is a ring, as shown, just filling in the space between the opening in the box and the piston rod. Fre-

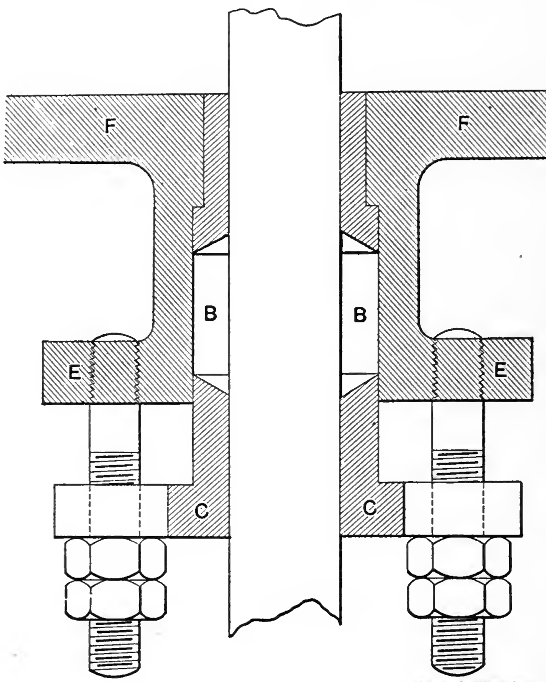


Fig. 170. Plain Stuffing Box.

quently this ring is omitted and the metal of the box fits about the rod. The packing is placed in the box as described above, thus filling the space, BB, between the bottom of the box and the gland. The packing may then be compressed as desired and as may be necessary by means of the nuts on the stud-bolts, thus forcing the gland down on the packing and making the joint tight. This is the general type of all such joints made with compressible packing, with, of course, variation in details.

Joints of this character are used for piston and plunger rods, slide valve stems, globe and disc valve-stems, joints about the shaft where it goes through a water-tight bulkhead, joint in the thrust bearing casing, as noted in section 21 [11], in slip and expansion joints, as noted in section 25 [2], etc., etc.

For metallic packing with joints of this character the form of the box is in general the same. In fact in some cases the box is so made that either soft or metallic packing may be used. Here again the greatest variety in detail is to be found, but a

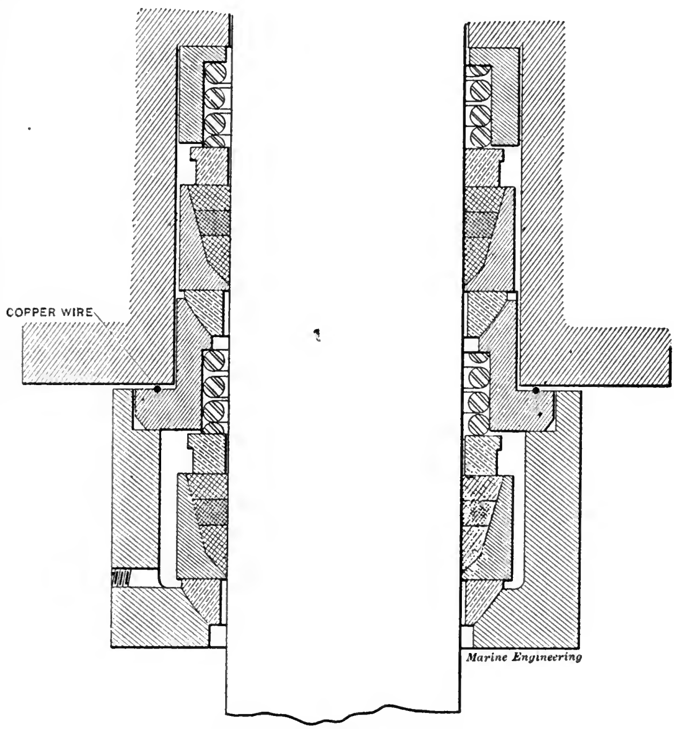


Fig. 171. Metallic Packing.

single instance will serve to illustrate the essential features of such packing.

In Fig. 171 is shown an example of metallic packing. The box or casing contains at its bottom, or upper end in the cut, a spiral spring, as shown. Next comes a brass ring, and next a series of babbitt or white metal rings carried in a casing or shell, as shown. These rings are conical on the outer circumference, and fit to a corresponding form of the containing shell. Next below is a second brass ring which supports the shell above, the joint between the two being ground to a tight fit. This ring rests on a casing below, the joint between the two being spherical and ground to a fit. The latter casing contains another spiral spring and then follows another series of rings, etc., similar to those above. The whole box contains, therefore, two similar sets of packing elements, each consisting of a spiral spring, white metal rings containing shell, etc. Each of the white metal rings consists of two separate halves, the whole arranged so as to break joints from one ring to the next. It is readily seen that the action of the spring is to crowd the white metal packing rings into the conical shell and hence against the rod, thus keeping the joint tight between the two. It is further seen that with this way of carrying the packing the latter is entirely unconstrained laterally, and may move in any way to accommodate itself to any slight irregularity in the rod without danger of disturbing the tightness of the joint. The two systems of packing, as a whole, are held up into place by an outer ring, secured to the cylinder head by stud bolts. The joint between the packing systems and the cylinder is made by a ring of copper wire, as shown, thus shutting off all leakage of steam from the packing space in this direction, while the various ground joints and packing ring surfaces close it off in other directions.

Among the various conditions which an ideal packing for piston and valve rods should fulfil those of chief importance may be stated as follows:

(1) The packing should make a steam tight joint between rod and stuffing box, at the same time opposing the minimum frictional resistance to the motion of the former.

(2) It must be durable even under the temperature of modern high pressure steam, and also easily removed or replaced with new when necessary.

(3) The packing should be free to move about transversely to a sufficient extent to follow the rod, even if it is slightly bent or out of line, at the same time maintaining the joints steam tight between the rod and the packing, and between the packing and the stuffing-box.

Requirement (3) has given the greatest trouble and has led to many varieties of design intended to cover the point, one of which is illustrated as above in Fig. 171. No packing can be considered satisfactory for modern requirements which does not possess in good degree the qualities detailed above.

[8] Reheaters.

A reheater is a collection of pipes placed in a receiver or exhaust passage from one cylinder to the next. Within these pipes high pressure steam is circulated, and around them the exhaust steam passes. The high pressure steam will, therefore, give up its heat to the cooler exhaust steam, and thus tend to dry or even to superheat it as it passes on into the next cylinder beyond. The office of the reheater is, therefore, to exercise a drying and heating action on the exhaust steam as it passes from one cylinder to another in a multiple expansion engine. Under most conditions this will exert a beneficial influence on the economy of the engine by decreasing the amount of cylinder condensation, and to such action may be referred the benefit which the reheater seems to give. See further Section 59.

[9] Governors.

In order to control the revolutions of the engine and to prevent violent increasing or *racing* when the propeller is partially lifted out of the water by the pitching of the ship, some form of governor is frequently fitted. The early types of marine engine were usually governed by hand at the throttle, which was commonly of the butterfly variety (see Section 24 [1]), though occasionally automatic means of moving the valve were employed. With modern multiple expansion engines, however, it is impossible to satisfactorily control the revolutions by the throttle. With such engines the control must come from the slide valve gear, the links of which may be linked up more or less, as required when the propeller is uncovered, and linked out again as it is submerged. Where no automatic governor is fitted, this must be done by hand control of the reversing gear. The modern automatic governor is intended to take the place of this hand control.

There are two modes of actuating marine governors.

(1) By utilizing the varying pressure under the stern.

(2) By utilizing a variation in the revolutions from the regular speed.

In the first type a pipe is run from the outside water at the stern to some form of pressure chamber near the engine, within which is a flexible diaphragm held in position by a spring or other equivalent means. The water being admitted to this pipe, the air within is compressed according to the head of water over the outer end. The apparatus is so adjusted that at normal draft the diaphragm is in equilibrium between the two forces, due to the water pressure on the one side and the spring on the other. A change in the depth of the water will cause a variation in the pressure which will be transmitted through the air and thus destroy the equilibrium, throwing the diaphragm in one direction or the other. This may be made to actuate a steam valve and thus through an auxiliary steam piston control the reversing lever and through this the links. In former practice this type of gear was sometimes made sufficiently large to actuate the steam throttle directly, or sometimes a like valve in the exhaust pipe.

The other type of governor is found in various forms. In many of them use is made of the centrifugal force of revolving balls or weights somewhat as in the ordinary stationary governor. Through the force thus available a small valve is operated, thus leading through a series of steps to the control of the reverse lever or other part which it is desired to operate. Again in other forms a revolving fan or propeller working in a box filled with liquid maintains the apparatus in a certain condition at a certain speed. With a sudden change of speed a corresponding change of resistance to the motion is met with, and this difference of force may be used to operate a small valve, and then, as before through the proper steps, the reversing lever is controlled. In another form a pump continually forces air into a chamber from which it escapes through a cock whose opening may be regulated at will. For a given size of outlet and speed of pump the pressure will rise until finally as much escapes as enters and the pressure remains constant. If the speed changes, however, the pressure will change correspondingly, and this difference of pressure will give a force which may be used as already explained.

A still different type of gear operated by a change of speed but not driven by the use of a belt employs the forces due to inertia. As usually installed it consists of a weighted vertical rod pivoted at the top so that if unrestrained it could swing to and fro between a pair of stops. This weight with its point of suspension is then given a movement of reciprocation horizontally by attachment to any suitable part of the engine. If not prevented, it would therefore swing to and fro between the stops, due to the change in momentum imparted by the reciprocating motion. It is, however, held by a spring against one of the stops, and the tension is so adjusted that movement will not result until the engine exceeds its normal speed, when the inertia forces overcome the spring, and the weight moves away from the stop. This motion, by means of an attached lever, may operate an auxiliary valve, piston, etc., and thus control the links. This type of governor is sometimes used only as a safety gear to quickly stop the engine in case of a breakage of the shaft or other accident permitting violent racing. In such case the gear is sometimes so arranged that the movement of the weight away from the stop will cause it to engage as a clutch with an arm or lever connected with the reverse lever, and so adjusted that the motion given will just bring the links to the mid position. The instant the weight leaves the stop, therefore, the levers will be suddenly thrown over, the links middled and the engine stopped.

All forms of marine governor are somewhat slow in controlling the variations of speed. The ideal governor would anticipate the motion and close down or open up just in advance of the rise in speed. Instead of this they act only after the stern has risen or fallen, or after the change of speed has become more or less pronounced. It is considered good practice, however, to fit some form of governor, at least as an emergency control, so as to prevent an excessive increase of revolutions from any cause whatever. For this purpose only those forms which depend on a change of speed are suitable, and such are commonly fitted in modern practice.

[10] Counter Gear.

The revolutions of the engine are automatically registered by a counter of the common type, and consisting of a series of discs with numbers from 0 to 9 on their circumferences. The motion for the counter is taken from any reciprocating piece which has a convenient location and a motion of small range. This is connected up to the counter by appropriate links, bell cranks, etc. The motion operates directly on the disc to the right and moves it along one notch or figure for each revolution. As each disc reaches 9 it engages with the one of next higher order on the left and throws it over, thus carrying the count continuously along the discs from the first to the last.

In this way the revolutions are registered one by one, the total number for any period of time being found by taking the difference of the two readings for the beginning and end of this period. By this means the revolutions per minute, per hour or per day are readily found as desired.

(11] Lagging.

The cylinders, cylinder-heads, valve chests and covers are usually provided with some form of covering intended to prevent the loss of heat, and thus conduce to economy as well as render the engine less disagreeable to work near and about. Such covering is known as *lagging* and consists usually of either wood in strips or polished sheet brass or Russia iron. When of wood the strips are narrow, 1 to 2 inches in width, and often of alternate light and dark color to give a pleasing effect. They are usually matched together and secured by bands of brass or polished iron, or by brass headed screws taking into foundation pieces held in place by countersunk tap bolts.

[12] Lubrication and Oiling Gear.

(1) *Lubricants.* For the lubrication of the various rubbing surfaces and turning joints, except within the cylinder, olive oil, castor oil, and the lubricating grades of mineral oil are used. Olive oil when pure is a most excellent lubricant, especially for machinery of moderate or light weight, but it is liable to adulteration by peanut oil or other oils of an inferior quality. For the lubrication of the internal surfaces nothing but the best mineral oil must be used. (See Sec. 40.) The grade commonly employed is known as cylinder oil, and is heavy and viscid at ordinary temperatures, becoming quite fluid, however, at the usual temperatures of the steam. It is usually fed in by means of a sight feed lubricator, as described in (9).

In addition to the liquid oils, various lubricating greases are used, often in combination with a certain proportion of graphite (black lead.) Graphite alone or in combination with oil is also used, and its lubricating qualities are of the highest order. It seems to possess the property, especially with cast iron, of filling the pores of the iron, and of thus forming a kind of graphite-metal skin on the surface, with a very small coefficient of friction.

The place where the lubricant should be supplied to the bearing is a subject which has attracted considerable attention in the past few years. It seems now to be very well established that the following principles should govern:

(a) The oil should always, where possible, be led into the bearing at a point which is under the smallest pressure.

(b) The continuity of the oil film, where it is under the greatest pressure, should not be interrupted by oil channels or grooves.

(c) The oil should be prevented, as far as possible, from escaping at those points which are under the greatest pressure.

For journals such as main pillow-blocks, etc., these principles are very commonly violated, and in fact it can hardly be said that practice has as yet come to act upon them, though their correctness seems to have been well demonstrated. According to these principles the oil for the main pillow block bearings should be introduced near the division between the upper and lower brasses, and the oil scores or grooves in the metal of the bearing at the top and bottom should be omitted. Similarly for the crank-pin and other cylindrical journals, the oil should be admitted at those points where there is the least pressure, and at the points where the pressure is greatest the bearing surface should be smooth and not interrupted by grooves, or scores, or oil channels of any kind whatever.

(2) *Amount of Lubricant Required.* As to the amount of oil to be used, practice differs widely, but from 5 to 8 lb. of oil per ton of coal may be taken as a fair allowance, or say from 5 to 8 lb. per 1000 I. H. P. per hour. In small sizes, for engines of the torpedo boat type, etc., the consumption will go up to considerably larger figures. Of this total amount some 5 to 10 per cent. may be required for internal lubrication, and the remainder for the various joints, bearings, etc.

In the opinion of many good engineers, after a ship has been at sea for a few days and the machinery has settled into a steady running condition, the amount of internal lubricant may be gradually decreased to perhaps one-half the amount first used.

With regard to the frequency of lubrication no definite rules can be given. The ideal system is, of course, as nearly continuous as possible. Where, however, the continuous system is not in use and intermittent oiling must be depended on, the various joints and stuffing boxes will require attention and a fresh supply of lubricant at intervals of from perhaps twenty minutes to one hour. For the various pin and turning joints, main guides, etc., the lubricant is usually supplied by oil cups or cans of character suited to the particular use for which intended, while the piston and valve rods are lubricated by means of a swab charged with cylinder oil or special grease. The main guides are also in some cases lubricated by the swab rather than the can.

There is also a growing tendency in engines of the high speed type as used on torpedo boats, etc., and where the steam is always more or less moist, to depend on water lubrication, and to avoid, so far as possible, the use of internal lubricant. This end may be furthered by careful workmanship in the fitting up of valves and pistons, and by driving the engine by belt or otherwise in the erecting shop with the surfaces charged with graphite. The minute pores of the metal are thus filled with the graphite, and rubbing surfaces are developed which run very well without further lubrication.

(3) *Adjustment of Bearings.* For a bearing in good adjustment the clearance or distance between the journal and bearing surface is proportioned to the size of the journal, and may be made about .002 or 1-5 of 1 per cent. of the diameter. With a lubricant of proper consistency and a load per square inch not too great, say not over 400 to 500 lb. per square inch of projected area, the film of oil will retain its place, and insure the proper lubrication of the bearing. If too thin a lubricant is used the bearing may heat and pound simply because the journal is not supported by the film of oil. The proper consistency of the oil as influenced by its natural viscosity and by the temperature of the bearing may, therefore, determine to a considerable extent the smooth running or pounding of the various joints and journals.

DEVICES EMPLOYED FOR SUPPLYING LUBRICANT TO BEARINGS.

We will now describe the more important devices for supplying oil and grease to the bearings, or to the points where required.

(4) *Wick Cup.* The plain wick cup consists of a receptacle, usually of cast brass, fitted with a cover, and placed at a convenient point for the delivery of the oil to the bearing. It is also frequently formed as a part of the bearing cap, as in Fig. 147.

A tube, as shown, enters through the bottom of the oil reservoir and rises within to a point above the level at which it is expected to carry the oil. This tube leads downward to the duct which carries the oil to the point of delivery to the bearing. The "wick" consists of a few threads of cotton wicking, one end of which is wrapped with a bit of wire, which then serves as a handle for pushing it down the tube or for pulling it out. In operation, one end of the wick is pushed down the tube and the other end dipped in the oil. Through the action of capillary attraction the oil rises in the wick on the outside, and then by a combination of capillary and siphon action descends and drips down the tube to the bearing. The end of the wick within the tube should be pushed down below the level of the bottom of the cup so that this shall form the longer leg of the siphon.

The size of the wick should be adjusted according to the amount of oil which it is desired to feed and to the quality of the oil as well. This adjustment of size is most easily effected by varying the number of strands of cotton in the wick. The

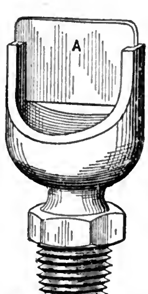


Fig. 172. Wiper.

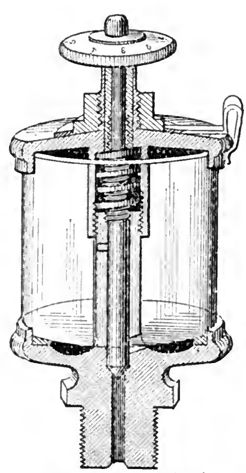


Fig .173. Oil Cup with Adjustable Feed.

amount fed may also be varied to some extent by regulating the distance to which the wick is pushed down the inner tube.

For a sudden flush of oil the cups may be filled until they overflow into the inner tube, by which means the bearing may be flooded if desired.

(5) *Wiper.* A wiper is shown in Fig. 172. It consists of

an oil cup with a central blade or plate, A, extending above the edge, and attached to one of the moving parts of the engine. At a convenient point is placed a strip of fibrous material on to which the oil is fed from the source of supply. The strip and wiper are so adjusted that the latter in its motion to and fro wipes or scrapes along the lower surface of the former, and thus as soon as the strip is saturated with oil the wiper takes off a drop or more which then runs down into the cup and thence to the surfaces to be lubricated. Naturally this mode of lubrication is more especially suited to parts having a horizontal motion.

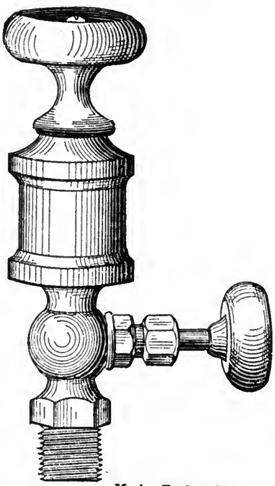


Fig. 174. Oil Cup with Screw Cover.

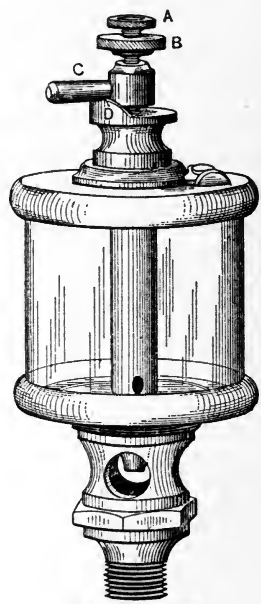


Fig. 175. Oil Cup with Sight Feed.

(6) *Plain Oil Cup With Adjustable Feed.* This consists of a simple cup, as shown in Fig. 173, mounted where convenient and connected by pipe or duct to the bearing to be supplied. The rate of feed is regulated by a needle or conical valve, which controls the size of opening through the discharge passage in the base. A cover is usually fitted to prevent spilling or the admission of impurities. Where such a form of cup is used to admit oil to a steam chest or other chamber under pressure, a strong screw cover is necessary, as shown in Fig. 174. To fill the cup in such case the valve is closed, the cover unscrewed, and

the cup filled. The cover is then replaced and the valve opened according to the rate of feed desired. Another variety of cup used for this purpose has a body of cylindrical or globular form terminating at top and bottom in a neck or tube, each of the latter being closed by a valve. The lower neck joins the tube leading to the bearing as in other cups, while to the upper is fixed a shallow open cup. The chamber between the two valves is filled by closing the lower valve, opening the upper and pouring into the shallow cup which serves thus simply as a funnel. When filled, the upper valve is closed and the lower one opened according to the rate of feed desired.

(7) *Sight Feed Oil Cup.* As shown in Fig. 175, this is essentially a plain cup with the addition of the "sight-feed" attachment or feature. When in adjustment, the flow of oil is regulated by the conical valve to a drop at a time at such interval as may be desired. The space below the outlet of the cup is cut away so as to show the drop as it falls into the mouth of the feeding tube. Very frequently a glass tube is fitted inside the brass framework, thus closing in the oil completely, but allowing the drop to be seen as it falls.

For a sudden flush of oil it is only necessary to open up the conical valve sufficient to let the oil descend in a stream and flood the bearing.

(8) *Compression Grease Cup.* In addition to oil, various forms of hard grease in cakes, or balls, or in bulk, are sometimes used for lubricating purposes. For feeding such lubricating material to bearings two means are made use of.

(1) If the bearing tends to become heated the heat developed will soften the grease and allow it to run to the spot where it is needed. (2) Compression cups are used containing a piston or plunger on top of the grease and acted on by a spring under control by a screw operated by hand. (See Fig. 176.) The grease is thus forced either automatically or by hand through the feeding tube and to the bearing. The spring arrangement may be made adjustable so as to force the grease more or less rapidly, according to its degree of hardness, and to the rate of feed desired.

(9) *Lubricator.* For the introduction of cylinder oil into the valve chests or steam pipes, an apparatus known as a sight-feed lubricator is very commonly employed. Such devices have been made in great variety of form, but the description of one will be

sufficient to show the principle upon which they operate. The lubricator, shown in Fig. 177, consists of a main chamber, A, with connections for attachment to the steam pipe or throttle valve casing at B, and for the attachment of a length of vertical pipe, P, leading in to the main steam pipe at C. D and E are two fittings for a short length of glass tube, as shown. From the top of the chamber a passage or pipe leads down to the lower fitting, E, while from D the passage leads into the steam pipe

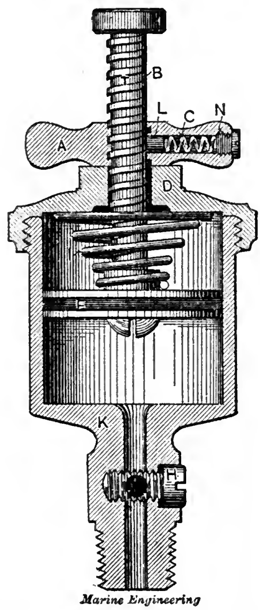


Fig. 176. Compression Grease Cup.

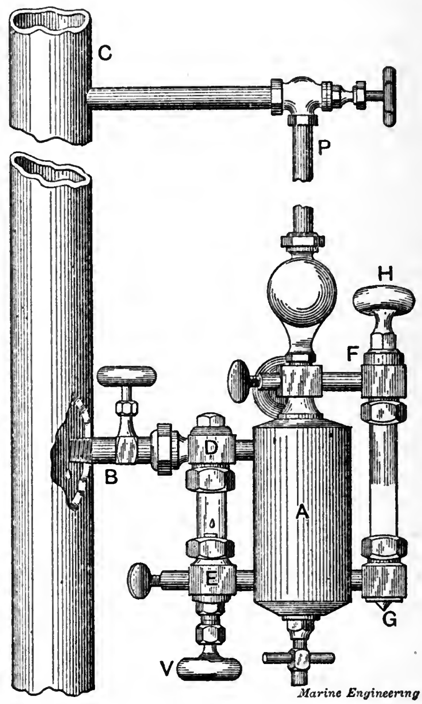


Fig. 177. Automatic Lubricator.

through the connection at B. The lower part of the chamber is connected to the vertical pipe leading up to C. There are also two connections, F, G, with glass gauge between to show the level of the oil and water within the chamber. The operation of the lubricator depends on the difference in density between the oil and water. The lubricator is first filled with oil through the plug H. The steam is then admitted to the pipe, P, where it will slowly condense and collect at the base of the chamber, in the pipe between D and E, and in the pipe P, thus furnishing a head of water acting on the oil and forcing it upward. As soon as the head is sufficient, oil will be forced a drop at a time as regulated by the valve V, out through the passage leading from the top of the chamber down to E and up through the water in the tube, DE, and so on to the steam pipe, where it is caught by the flow of steam and carried to the valve chest and cylinder. The passage of the oil drop upward is plainly seen, and thus the operation of the lubricator is under ready observation. Such lubricator may be placed on the steam pipe or throttle valve chamber, or at any convenient point where the oil will be carried by the inflowing steam to the points where needed.

(10) *Oil Pump.* A simple arrangement for forcing cylinder oil into a steam chest is sometimes used when it is not convenient to fit a lubricator. This consists, as shown in Fig. 178, of

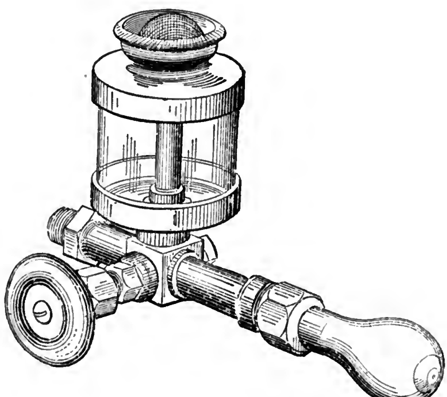


Fig. 179. Oil Pump.

an oil pump operated by hand. The chamber being filled with oil the delivery valve is opened and the oil forced in as may be desired, through a connection attached to the pump delivery, as shown.

(11) *Modern Systems of Oil Distribution.* In the preceding section we have described the principal devices used for supplying oil to a bearing, or to a steam pipe or chest. We will now describe briefly the general oiling system for the engine as a whole, involving such combination of these devices as may be found most desirable.

The leading features of the modern system consists in the provision of a small number of distributing centers from which oil is taken by piping as directly as possible to the various places where needed, each place having its own independent pipe and set of connections. Following is a brief description of such a system of oiling gear, and will serve to illustrate the methods now in use in good practice.

A light cast brass box is provided for each cylinder placed at a point higher than any joint or bearing to be reached by the oil, and having a capacity sufficient to last several hours without refilling. These oil boxes are provided with sight feed cups with protected glass tubes from which pipes lead to wipers on the moving parts, or to tubes in the bearings and guides. Union joints are fitted where necessary, so that the oil pipes may be quickly taken down and cleaned. With few exceptions the oil for the various moving parts of the cylinder is supplied from this box.

The main crank pin is oiled by means of a pipe and cup carried on the cross-head and taking oil from a drip supplied from the oil box as described. The pipe runs down the side of the rod, or frequently inside if the rod is hollow, and connects with the oil duct leading through to the pin. The cross-head guides are provided with oil through pipes connected with holes at about the middle of each forward and backing guide. The main pillow-blocks are oiled by one or more wick cups delivering the oil at the points desired.

A cup for tallow or grease is also usually provided, and likewise sometimes a hole through which the hand may be passed to feel of the shaft as may be desired. The presence of this hole, however, is not in accord with the principles given above in (1), and the practice cannot be recommended. If anything of the kind is to be fitted it is better to carry the hole simply through the cap, thus leaving the brass continuous. The latter may then be felt as desired, and a tendency to heat may be thus observed. The excentric straps are fed from long narrow oil cups, receiving their oil through the drip pipes from the reservoir. The length of the cup is made such that some part of it is always under the drip in any position of the excentric, and it will, therefore, always receive its supply. The various other parts of the gear are similarly supplied with oil either from a drip or a wiper as may be more convenient.

The chief advantages of such a system consist in the certainty and regularity of operation which may be assured with the minimum of time and attention on the part of the oiler.

Sec. 25. PIPING.

[1] Systems and Materials.

The principal systems of piping are as follows:

(1) The main steam piping from boilers to engine.

(2) The auxiliary steam piping from the auxiliary boiler or from one or more boilers specially selected, to the various auxiliaries which are to be operated by steam.

(3) The main exhaust piping from cylinder to cylinder and from the L. P. cylinder to the condenser.

(4) The auxiliary exhaust system, providing each auxiliary with its exhaust, either to the condenser or overboard, as desired.

(5) The feed system, main and auxiliary for returning the condensed steam to the boilers.

(6) The condensing system for bringing the condensing water to the condenser and for carrying it from the condenser to the sea again.

(7) The drainage and bilge pump delivery systems for leading water to the bilge pumps and from the pumps to the sea.

(8) The fire system for leading water to the fire pumps and from the pumps to the fire plugs.

(9) The sanitary system for delivering water to the W.C's. and heads.

(10) The steam heating system.

We may otherwise classify all pipes under three heads: *steam* pipes, *exhaust* pipes, and *water* pipes, while of the latter we have a further division into those carrying water to a pump or *induction* pipes, and those carrying water from a pump or *eduction* pipes.

For steam pipes the materials in present use are copper, wrought-iron, and steel. Copper pipes in small or moderate sizes may be made of seamless or solid drawn tubing; in large sizes they are made of sheet copper with brazed joints. Wrought iron pipes are lap welded, while steel pipes are also lap welded and are sometimes further fitted with a riveted longitudinal strap covering the line of the weld. Seamless or solid drawn steel pipes have also been made to some extent.

For the various junctions, elbows, bends, tees, etc., steel or malleable iron castings are used with steel pipe, while with copper pipe sheet copper is used for these parts, bent and formed up by hammering into shape, and secured at the joints by brazing.

The advantages of copper are its great ductility, freedom from corrosion, and the readiness with which it may be used to make pieces of an irregular form, such as the elbows, junctions, etc., referred to above. Its disadvantages are, greater cost, low tensile strength, the possibility of damage to the quality of the material in the process of pipe manufacture, and the possibility of a loss of ductility in service by repeated strains due to the expansions and contractions which result from changes in temperature.

The metal close about a flanged joint seems especially liable to lose its strength in this way. This is probably due to the concentration of the strains due to expansions and contractions in the vicinity of a rigid connection such as a flanged joint, and to the development in this way of a line of weakness running around the pipe.

To render copper pipe more secure under high pressure it has been wound with copper or steel wire, or reinforced by wrought-iron or steel bands. Such bands may be from ½ to 2 inches wide by ⅛ to ¼ inch thick, and spaced with intervals of from 6 to 10 inches. These methods, especially the latter, have proven quite successful in strengthening copper piping for modern advancing pressures.

The chief advantage of wrought iron and steel pipes are, less cost, greater tensile strength, less liability of the material to damage in quality in the processes of manufacture, and less liability to lose strength or ductility in service. Their disadvantages are, greater liability to corrosion, and greater weight of cast junctions and fittings than for copper.

Welded pipe is made from rolled strips the edges of which are machine beveled for a lap joint. The requirements of manufacture are such that for all except the largest sizes the thickness is in excess of that needed for strength alone, at least with the pressures at present in use, so that when such material is employed there is always an excess of strength.

With wrought-iron pipes the welded joint is trusted without reinforcement. With mild steel a covering strip or butt strap is sometimes riveted on, though with the later improvements in the welding quality of such material, experience shows that these joints are quite as reliable as those of iron. Flanges for wrought-iron or steel pipes may be welded on, but more commonly they are riveted or screwed to the pipes. In the latter cases the flanges are caulked on both sides to make a steam tight joint.

In small sizes and in a relatively cheaper grade of practice, ordinary commercial steam piping is used, fitted up with the usual fittings and screwed joints.

For exhaust piping the same general character of pipe is used as for steam, with such differences as the decreased strength necessary may indicate.

For water piping steel, iron and copper, and in cheaper practice the ordinary commercial pipe are all used. Steel and iron are usually considered less suitable for water than for steam piping on account of the greater danger from corrosion. This is especially true for feed piping, and in case such material is used for this purpose it is considered good practice to carefully galvanize the pipe both inside and out. It is likewise good practice to tin all copper piping which is under the floor plates or in the bilge, but this is rather to protect the ship than the pipe, the former being in danger of attack by electro chemical action (see Sec. 40) in case the copper and the metal of the ship obtain connection through a medium such as bilge water. In connection with piping see also that heading under Section 19.

[2] Expansion Joint.

The expansion and contraction of a length of piping under a change of temperature require some kind of joint or connec-

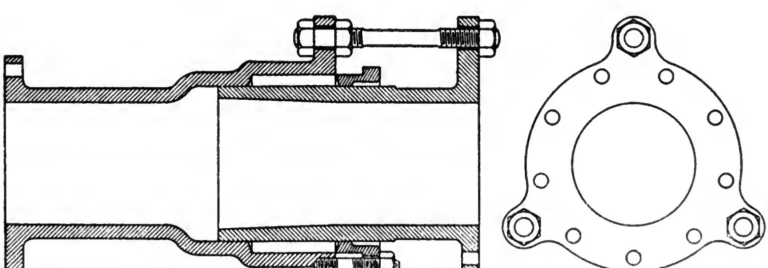


Fig. 179. Expansion Joint.

tion which will allow of the change of length without buckling or straining the pipe. This is usually provided by an *expansion joint*, as shown in Fig. 179. This consists of a recessed portion on one part of the joint into which the other fits, as shown. The space left between the two thus forms a stuffing-box into which packing is compressed by means of the gland. The two parts of the joint are thus free to slip a little way, one relative to the other, while the joint is kept tight by means of the stuffing box and gland in the usual manner. It is readily seen that the steam pressure within the pipe, especially if it contains a bend or elbow, will tend to force the two portions of the joint apart, and thus open the pipe at the joint. To guard against this, safety stays or ties should be fitted. In the figure one of three such stays is shown by the bolt at the top. Care must be taken in adjusting such stays that sufficient freedom is left for the expansion and contraction which the joint is intended to provide for.

In special and more complicated forms, known as balanced or equilibrium expansion joints, these forces are more or less completely balanced within the joint itself.

[3] Globe Angle and Straightway Valves.

For controlling the flow of a liquid, vapor or gas through a line of piping, various forms of valve are used. Chief among these are the Globe, Angle and Straightway or Gate types, as described in Section 24 [1], [2], and to which reference may be made.

CHAPTER V.

AUXILIARIES.

Sec. 26. CIRCULATING PUMPS.

The office of the circulating pump is to draw the condensing water from overboard, force it through the condenser tubes, as explained in Section 29, and thence overboard through the condenser discharge pipe. The principal resistance to be over-

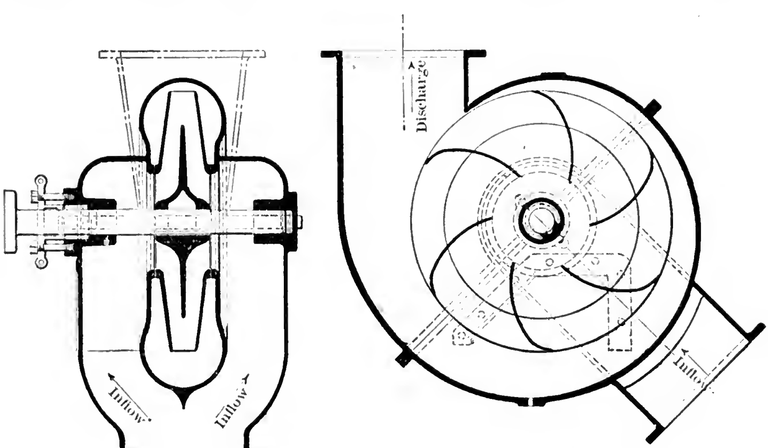


Fig. 180. Centrifugal Pump.

come by the pump is the resistance to the flow of the water through the tubes, and this is but slight measured in pounds per square inch or in feet of head. On the other hand, the quantity of water to be handled is large, and hence the requirement is for a type of pump which shall be able to handle large quantities of water against a small head or resistance. These requirements are very perfectly fulfilled by the centrifugal pump, as shown in Fig. 180. The moving part consists of a number of vanes or arms attached to a shaft and forming what is called the *runner*. This revolves within a casing furnished with inflow and outflow passages, as shown in the figure. The pump being primed or filled with water and started, the rotary motion gives rise to a centrifugal force, in obedience to which the water moves outward toward the tips of the blades, where it escapes through the outflow passage into the discharge pipe. There is a corresponding defect of pressure about the hub of the runner and a resulting inflow of water from the sea to take the place of that which leaves at the outflow. The operation thus becomes con-

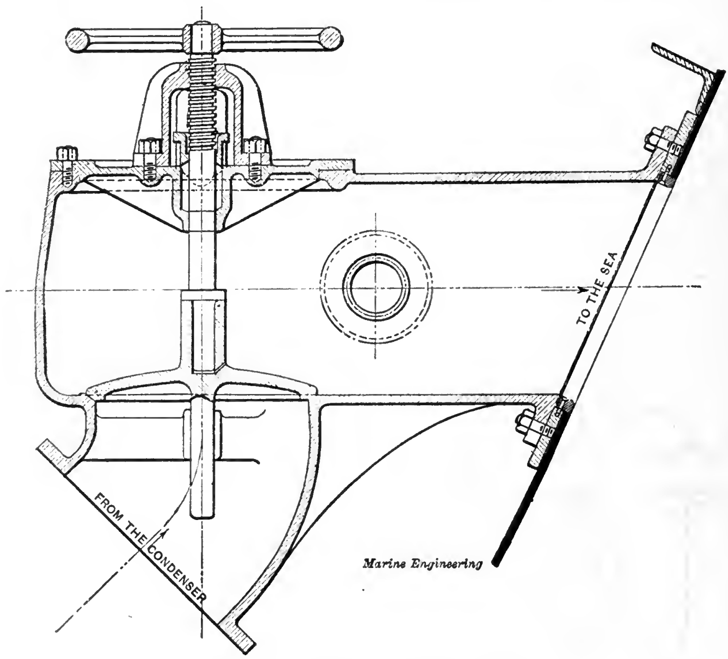


Fig. 181. Outboard Discharge Valve.

tinuous and results in a steady flow of water from the pump through the condenser tubes and back to the sea through the condenser discharge pipe and outboard delivery. In order to prevent the water from "short circuiting" or slipping back from the discharge space about the tips of the blades to the inflow space about the hub, a running fit is provided between corresponding faces on the runner and casing as indicated in the figure.

The discharge or outboard delivery valve is usually a plain type of angle stop valve, as illustrated in Fig. 181. Its office is simply to allow when open the discharge of the water from the condenser, and prevent when closed the inflow of the sea to the pump.

Sec. 27. CONDENSERS.

The purpose of the condenser is to provide for converting the exhaust steam back into water. Condensers are of two types—*jet* and *surface.*

The jet condenser consists simply of a chamber of rectangular or cylindrical form in which the steam and the condensing water are mingled together, the steam giving up its heat to the relatively cool water, and thus being reduced to the liquid state again. The water is usually led into the top of the chamber and allowed to fall upon a plate pierced with a large number of small holes, and known as the *scattering plate.* This divides the water into small streams or jets and enables it to mix intimately with the steam which enters just below the plate. The condensed steam and condensing water then fall together to the bottom of the condenser. From here the water is removed by the air-pump which delivers it overboard, the feed-pump in the meantime taking enough for the boiler feed and returning it to the boilers.

The usual type of surface condenser consists of a chamber commonly of cylindrical form if separate from the main engine (see Fig. 182), or rectangular if forming a part of the engine columns. (See Fig. 100.) This chamber contains, as shown, a large number of small brass tubes running between the inner walls of the heads, which are double, thus providing a connection between the ends of the various tubes. The condensing water is driven by the circulating pump through the tubes, the usual run being, as shown in the figure. The water enters at the lower left hand end and fills the lower half of the head, being prevented from filling the whole head by a partition half way up, as shown. It thus finds its way into the lower half of the tubes and flows through them to the right hand head. It then rises into the upper half of the tubes, flows back to the left and out at the opening in the upper part of the left hand head. The water thus traverses twice the length of the condenser forward and back, and from the bottom upward. The steam, on the other hand, enters at the top into the body of the chamber, and thus around the outside of the tubes. The steam and the condensing water are thus kept separate, and the steam is condensed simply by the surface action of the tubes. The steam thus condensed to water falls to the bottom of the condenser, whence with some air and vapor it is removed by the air-pump and delivered to the hot-well, whence it is taken by the feed-pump and sent back to the boilers. Baffle or diaphragm plates, as shown, are often fitted in the condenser to prevent the steam from rushing directly through from the inflow on top to the air pump passage on the bottom. The steam is thus forced to fill the condenser as completely as possible, and thus the condensing surface is more uniformly brought into action. In order to facilitate the rush of steam downward into the body of the tubes, thus bringing more

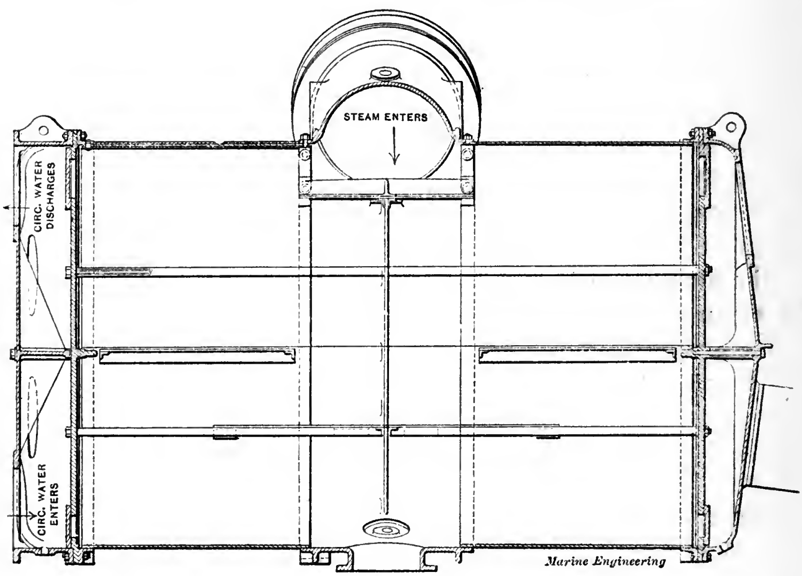


Fig. 182. Surface Condenser, Longitudinal Section.

quickly into action those in the lower part of the condenser, a few rows are often omitted in the upper part, thus forming branching passages leading from the top downward, as shown in Fig. 183. In order to support the heads against the pressure from without, a certain number of longitudinal braces or struts are necessary, as shown in the figures.

Condenser shells are made of cast-brass, cast-iron, or sheet brass or steel. When of rectangular form the sides are cast with the necessary webs to give them strength to stand the pressure from without. When of cylindrical form the necessary strength can be given by a suitable thickness of metal reinforced if necessary by ribs running around the shell. When relatively thin sheet metal is employed, as in torpedo-boat practice and the like, it is customary to fit one or more angle-iron or Tee-iron stiffeners running around the shell in order to provide the necessary strength.

Condenser tubes are of thin sheet brass, usually ⅝ to ¾ inch outside diameter. In order to make a water-tight joint between the condenser tubes and the inner heads or tube plates, and at the same time to avoid a rigid constraint of the tube, a great variety of condenser tube packings have been employed. The most common type of packing in present practice is shown in Fig. 184. The tube plate is counterbored, as shown, and threaded for a ferrule with a tapering outer end. The hole in the outer end of the ferrule is thus of about the same size as

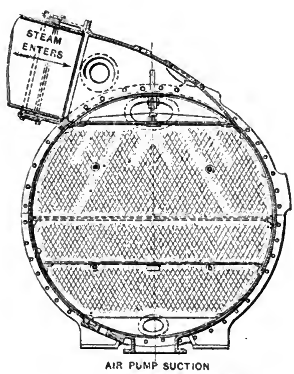


Fig. 183. Surface Condenser, Cross Section.

that inside the tube, and hence smaller than the outside of the tube. Between the other end of the ferrule and the bottom of the counterbore is usually a ring of rubber or a few turns of some other elastic or fibrous material as packing. Screwing down on the ferrule compresses the packing, and thus makes the joint, while at the same time the tube is free to expand and contract to a slight extent. The outer ends of the ferrules, however, prevent the tube from crawling to such an extent as to free either end, a result liable to occur without some method of prevention.

Sec. 28. AIR PUMPS.

It is the purpose of the air-pump to remove from the condenser the water and such small quantities of air as may enter by leakage or with the steam, and which would ultimately destroy the vacuum if not removed. As often stated, it is the office of the air pump to maintain the vacuum formed by the condensation of the steam.

The usual type of air-pump is shown in Fig. 185. A is the piston or bucket moving in the barrel B, and carrying *bucket valves,* D, opening upward, as shown. The *foot valves* at C also open upward and admit the contents of the condenser from below. Beginning with the piston in the position shown the operation is as follows:

As the piston rises, the air and vapor between its lower face and the foot valves become rarified with a resultant decrease

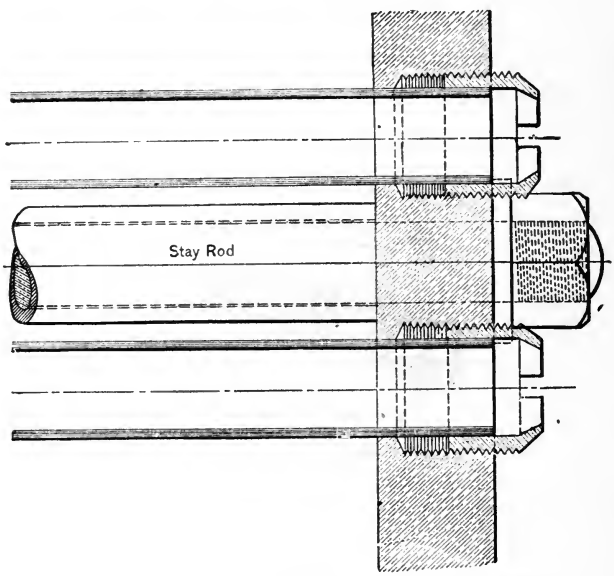


Fig. 184. Ferrule and Tube Packing, Surface Condenser.

of pressure. Soon a point is reached where the pressure in the condenser is decidedly greater than in the space above the foot-valves, and in answer to this difference of pressure the valves open and admit air, vapor and water from the condenser. This operation terminates with the piston at the top of the stroke. On the return stroke the foot valves close and the contents of the barrel are forced through the bucket valves at D to the space above. On the next stroke the contents are lifted and forced out through the *delivery* or *head valves* at E, where the air and vapor escape, and the water flows to the hot-well, whence it is sent by the feed-pump back to the boiler.

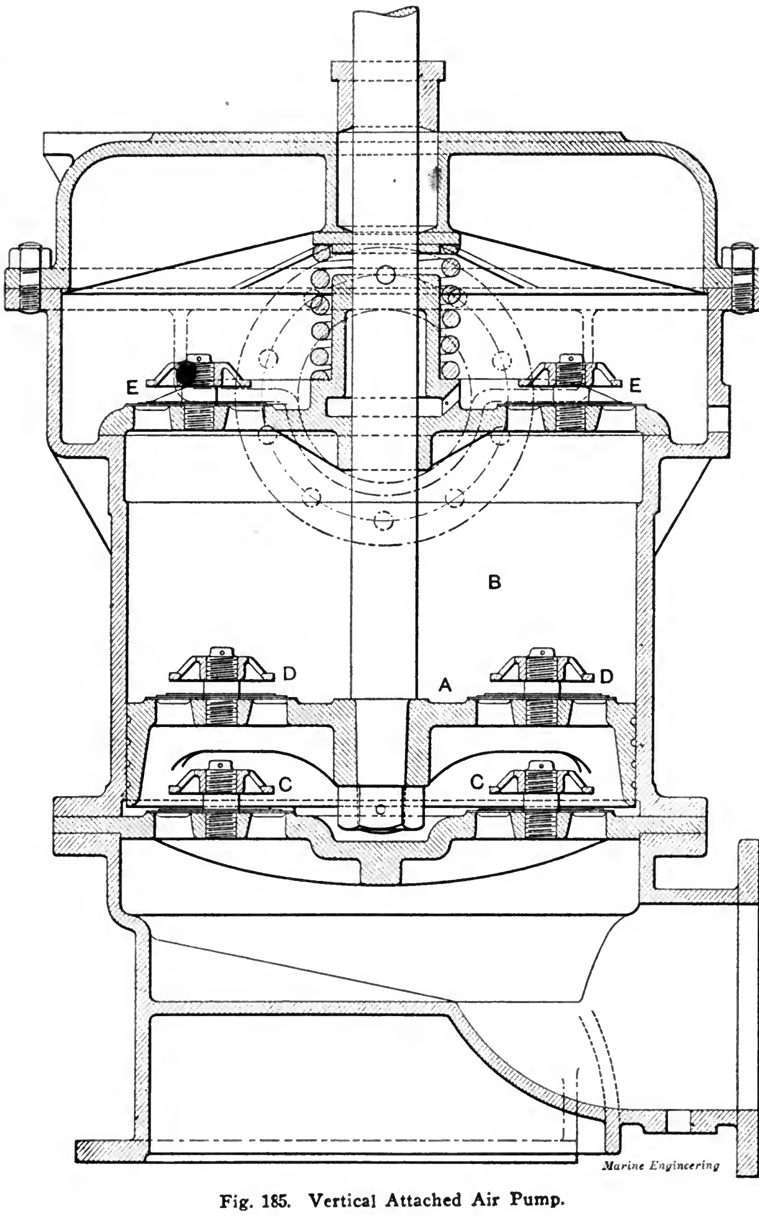


Fig. 185. Vertical Attached Air Pump.

It is evident that the pressure in the condenser cannot be reduced below that necessary to raise the foot-valves, so that for this reason, as well as for their more ready response to variation of pressure, they should be made as light as is consistent with a proper performance of their duty. As shown in Fig. 186, such valves in modern practice are usually made of light sheet metal discs controlled by spiral springs. In older practice vulcanized rubber was quite commonly employed. It is also evident that the foot valves will respond the more quickly, the more rapidly the pressure above them decreases as the piston begins to rise, and hence the less the clearance space between the valves and the piston when in its lowest position.

The air-pump has this peculiarity in its action, that the load per stroke and hence the resistance often decreases with an increase of speed, and increases with a decrease of speed. Hence when the pump is operated by an independent engine, it may

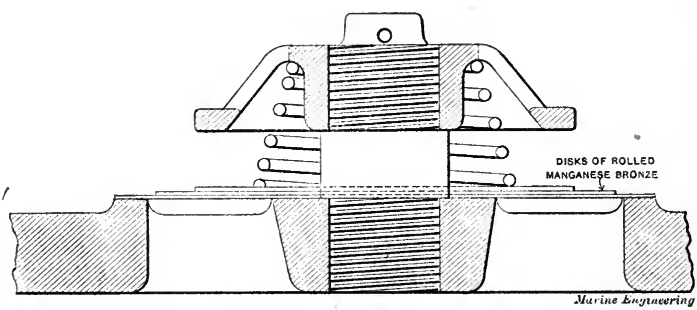


Fig. 186. Air Pump Valve, Guard and Spring.

be liable, unless carefully designed, to race or run away to excessively high speeds with an increase of pressure in the steam cylinder, or to slow down and stop with a corresponding decrease.

It is apparent also that a certain time will be needed for the foot valves to open, and for the air, water and vapor to flow through. Hence, with excessive speeds, the valves may not have time to open between strokes, the vacuum will become poorer, and the condenser may become flooded with water. It thus follows that with too high, as well as with too low a speed, the vacuum will be poor, and the operation of the pump unsatisfactory.

The air-pump may be driven either by an independent engine, or by attachment to the main engine. The attached air-pump is usually operated by *air pump levers*, which derive their motion in most cases from the L. P. cross-head, as shown in Fig. 99. The stroke of the pump is thus reduced to usually less than one-half that of the main engine. One of the advantages of the attached air-pump is that the number of strokes per minute is necessarily the same as for the main engine, and it can neither race nor slow down on account of variations in its own resistance. A further advantage is that the power required for its operation is obtained more economically than when operated by a separate engine. The chief disadvantage of the attached air-pump is that with the modern increase of revolutions, the speed may be too great for the best results from the pump as

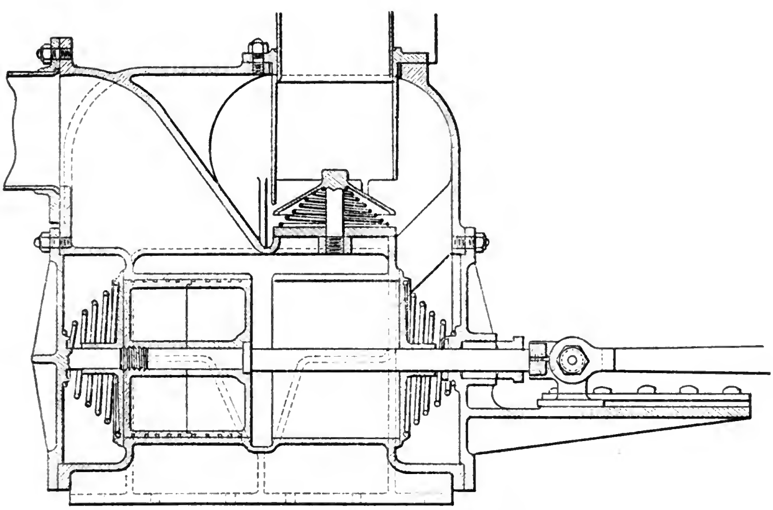


Fig. 187. Air Pump for High Speeds.

usually designed. This, together with the advantage of having the condition of the condenser under control independent of the main engine furnish the chief reasons for the use of the independent air-pump. When thus fitted as an independent auxiliary the number of double strokes per minute usually varies from 15 or 20 to 30 or more, while the revolutions of the main engine may be from 100 to 200 or more.

For special cases where the revolutions are very high, as in torpedo boat practice, but where for simplicity or for the saving of space it is desirable to use an attached air-pump, the Bailey type of pump is employed. In this pump, as shown in Fig. 187, the water flows by gravity into the barrel through ports alternately opened and closed by the piston itself, which thus serves as its own valve. The air and vapor naturally expand and enter with the water, and the whole contents are forced out of the end of the barrel through delivery valves similar to those in the pump of usual type. In some cases the delivery valves are car-

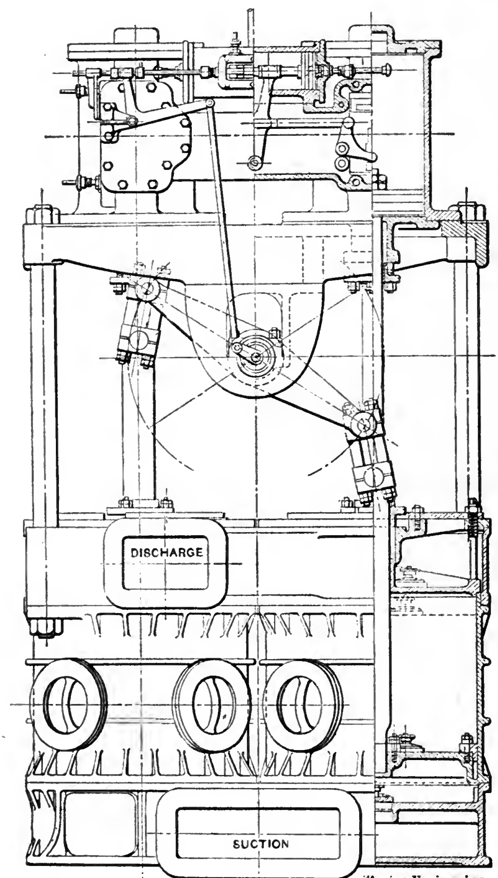


Fig. 188. Vertical Independent Air Pump.

ried on movable heads, which thus become valves in themselves, and available to relieve the barrel in case the smaller valves give insufficient opening. In some cases, as shown in Fig. 187, the smaller valves are omitted and the entire head serves as the delivery valve. With this design air-pumps may be successfully operated at speeds of 400 or 500 revolutions and higher.

When operated independently the air pump is very commonly made double, and operated by one form or another of special steam valve gear. A typical form of independent air pump is shown in Fig. 188. The general operation of the valve gear is as follows: The beam which positively connects the main piston rods of the pumps operates from a point near its center and by means of rod and bell cranks, the slide valve of the horizontal cylinder which lies between the main steam cylinders, as shown. The piston of this horizontal cylinder is really the driving engine of the main steam valves, a function which it performs by means of a system of internal levers. See further Section 33 regarding the operation of such types of pump valve gear. The adjustable collars on the valve stem of the "valve driving engine" afford a means for regulating for full stroke at any speed, while suitable cushion valves give a further control over the action during the stroke, in regulating the distribution of the work and preventing the slamming of the foot valves.

Sec. 29. FEED PUMPS AND INJECTORS.

Comparing the feed and circulating pumps we find that the former has to handle a very much smaller quantity of water—usually from 1-25 to 1-40 the amount—but against a very much higher pressure, viz., that in the boilers. In consequence an entirely different type of pump is required.

The feed-pump may be attached to the main engine, or run as an independent auxiliary. When attached to the engine it is usually of the type known as the *plunger pump* and shown in Fig. 189. The moving part consists simply of the plunger, AB, working in the stuffing box, KL, and operated usually from the air-pump levers. There are two valves or sets of valves, inflow and outflow, as shown at F and E. The level of the pump is usually below that of the hot-well so that the water stands ready to enter through the inflow valves as the plunger rises and makes room for it. This is aided by the partial vacuum formed within the barrel as the plunger rises. On the other hand, as the plunger descends on the next stroke the inflow valve closes and the water flows out through the outflow valve in order to make room for the descending plunger. It is thus seen that the pump is single acting; that is, that it delivers but once in two strokes, and that the amount delivered is measured by the volume of the plunger displacement.

This in turn equals the cross-sectional area of the plunger multiplied by the length of the stroke. The actual delivery per stroke will be somewhat less than this due to leakage, and to failure of the barrel to completely fill on the up stroke.

The stuffing box, K L, is of course accessible and adjustable from the outside, and with proper design the inflow and outflow valves may be examined by the simple removal of a bonnet. The strong points of this pump are its simplicity, and the ready accessibility for examination and adjustment, of all parts on which the operation of the pump may depend.

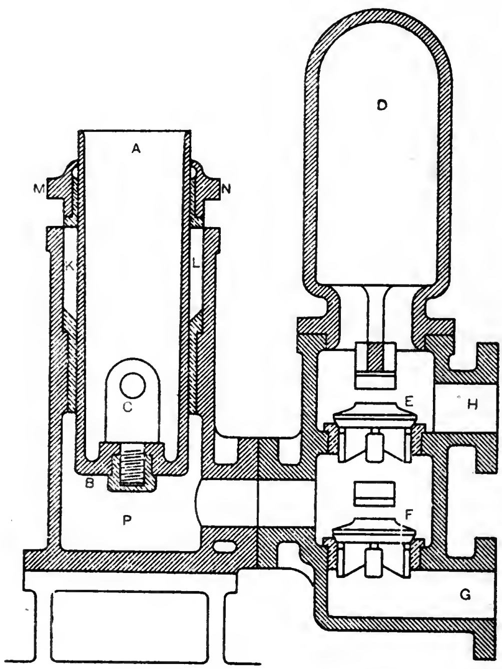


Fig. 189. Plunger Feed Pump.

In fitting up the attached feed pump it is necessary to provide it with some form of safety or relief valve, else should the discharge or check valve jam or fail to operate, the feed pipe or pump or some part of its operating gear would become broken. Such a relief valve is usually a simple form of spring loaded safety valve, and similar to the engine relief valves referred to in Section 24. In order that the valve may be effective in relieving the pump and entire line of pipe it should be placed on the pump chamber or in any event not beyond the pump discharge valve.

Where the feed pump is operated as an independent auxiliary it is usually of the direct acting or positive motion type, as described in Section 33, and to which reference for details may be made. For feed-pump purposes the area of the steam piston is made from 2 to 3 times the area of the water piston or plunger in order to give on the steam side a pronounced excess of total pressure over the resistance on the water side. This will enable the pump to overcome the resistance to the flow of the water through the feed-pipe and check valves, and thus to force water into the same boiler from which it draws its steam, or even into a boiler with a pressure somewhat higher than that from which its steam is drawn. For the general purposes of a feed pump the vertical or admiralty style, as illustrated in Section 33, has come to be very generally used. Its chief advantages over the horizontal type are two in number.

(1) It occupies less floor space and may be conveniently put up on a bulkhead or elsewhere, in such manner as to occupy but little space otherwise available.

(2) The valves in the water end, as shown in the figure, are more conveniently arranged for examination by the removal of a bonnet than with the horizontal type of pump.

These considerations and especially the latter, have made this general type of feed pump the standard in modern marine engineering practice.

In addition to feed-pumps of the plunger and piston types, an injector is often fitted as an auxiliary means of feeding the boilers. There are many different varieties of injector, but a description of one will suffice to illustrate the principles involved. Referring to the diagram, Fig. 190, S, is a nozzle connected with the upper pipe, B, leading steam from the boiler. When steam is turned on by means of the handle, K, and attached valve-stem and valve, it escapes in a jet which enters the slightly tapered passage VC. The air in the space around and between these two orifices is caught and drawn along with the jet, thus causing a reduction of pressure at this point. This space is connected through the lower pipe, B, to the water reservoir, and when the loss of pressure is sufficient, the water rises the same as in the case of a pump. The water and steam are thus brought into contact, and pass on together into the combining and delivery tube CD. The steam is here condensed and the resultant jet of water attains a very high velocity. A little further on, when this is reduced to the relatively low velocity of the water in the feed pipe, the pressure developed is sufficient to overcome the boiler pressure, to open the check valve, and to force the water into the boiler.

It may aid the understanding of this seemingly puzzling result if we remember that it is the *energy* of the steam which is the real motive power. This is transformed largely into motion in the combined jet, and this, when arrested, gives the pressure, as stated before. In the steam pump a steam piston is provided much larger than the water plunger in order to give a force sufficient to overcome the resistance of the feed pipe and at the check valve. So in the injector, in a somewhat similar manner,

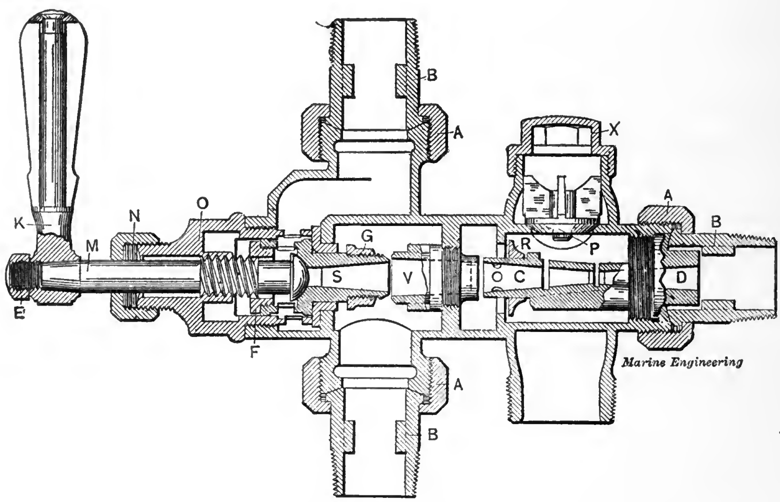


Fig. 190. Injector.

the energy of the steam in a relatively large pipe is concentrated on a small jet of water, giving it the high velocity and later the pressure, as described.

In passing it may be noted that as a boiler feeder a good injector has practically a perfect efficiency, all heat used being carried back again into the boiler, except the small amount lost by radiation from the instrument and connecting pipes.

An injector of the type shown in the figure is known as an *automatic* injector. This signifies that once the injector is adjusted and working, should the jet of water become broken by a jar or other accidental circumstances, it will restart itself without further adjustment. The capacity and working range of an injector are decreased as the lift is higher and as the water is warmer. With cold water and a moderate lift, say not exceeding 5 to 8 feet, a good automatic injector will start up with 25 or 30 lb. steam pressure, and will work with little or no further adjustment over a range of perhaps 100 pounds pressure. With feed water at about 100 deg. F, the same injector would start at 30 or 35 lb. steam pressure, and will work over a range of perhaps about 70 lb. or up to about 100 lb.

In addition to the automatic injector there is another type having two sets of tubes, one for lifting and one for forcing. Such instruments are often termined *inspirators* to distinguish them from the ordinary automatic injector. When properly adjusted the lifting set of tubes acts as a governor to the forcing set, supplying under a great range of steam pressure the proper amount of water to condense the steam in the final set of tubes. Such an injector handling cold water with a short lift, will work through a range of over 200 lb., while with water as hot as 100 deg. F and small lift it will work through a range of from 150 to 200 lb. The operation of each set of tubes is on the same general principles as above described for the automatic injector.

Sec. 30. FEED HEATERS.

The office of the feed heater is to raise the temperature of the feed-water from that of the hot-well as nearly to that within the boiler as may be practicable before the feed enters the boiler proper. Feed heaters are of two fundamentally different types, according as the heat used is drawn from waste furnace gases or from steam. If the former, as is very common with water-tube boilers, as described in Section 14, the feed heater is really a part of the boiler. The entire operation is thus performed in two stages, one in the heater, and one in the boiler proper. In the first stage it is sought to raise the water as nearly as possible to the boiling point by use of the furnace gases after they have passed the main steam generating tubes. In the second stage, carried out in the boiler proper, the previously heated water is transformed from liquid into vapor.

The resulting economy comes from being thus able to reduce the products of combustion to a temperature lower than they would otherwise have before finally getting rid of them. The addition of the heater will, of course, affect the draft, and the extent to which heating surface, either in the form of steam generating tubes or feed heating tubes, can be added without seriously interfering with the draft is a point which must receive consideration. Morever, since the feed-heating surface might be put into additional main boiler tubes, thus giving the same total surface without a feed-heater, the question may naturally be asked whether in such case the results would be as good. In other words, is it better to put the total heating surface all into main boiler tube surface, or to divide it up and put a part (usually quite small) into a feed-heater located beyond the main part of the boiler? Experience seems to indicate the latter as the better design of the two, and the fundamental reason is that given above—viz., that we are thus able to reduce the products of combustion to a lower final temperature than with main boiler tubes of the same aggregate surface. The reason for this is found in the fact that the temperature of the feed water as it enters the heater is much lower than that of the steam and water within the main tubes. Hence with such a heater the gases as they leave the boiler pass over relatively cool surfaces, and the flow of heat will be much more pronounced, and the gases will be more effectively cooled than by passing over an equal area of main boiler tube surface.

Turning now to the other type of heaters we have an entirely different mode of operation. Here the heat given the feed-water comes from steam which is drawn either directly from the main or auxiliary steam pipe, or from the receivers, or from the exhaust of some of the auxiliaries on its way to the condenser or to the escape pipe. There are two styles of heater working on this principle. In one the steam and feed-water are mixed together in the same chamber and the steam is condensed and thus joins the feed-water, raising its temperature as may be determined by the conditions of operation. In the other style the steam is on one side of a coil or nest of tubes and the feed-water on the other side, the heat passing through the metal of the tubes from the former to the latter while the steam condensed in consequence of the loss of heat is drawn or trapped out as may be required.

Where the steam and feed-water are mixed together the feed-heater consists essentially of a chamber or drum, as in Fig. 191, provided with means for introducting the feed as a spray or series of cascades, while the steam is introduced in jets, and the two thus become intimately mingled.

In the form here shown the feed enters through C and passing out through a valve, D, falls as a cascade through the annular space between the pipe and the steam delivery drum which is pierced, as shown, with small holes. The steam enters through B, and passing through the holes in small jets becomes mingled with the water and thus imparts to it its heat. The two then fall to the bottom of the chamber from whence the feed

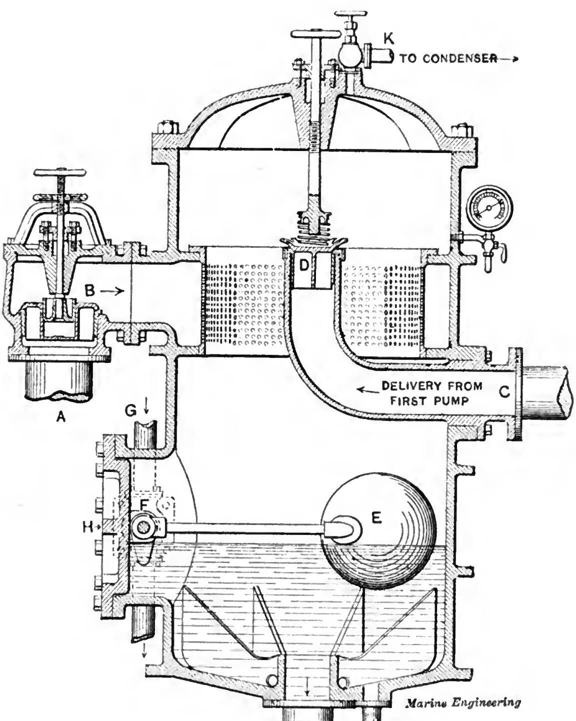


Fig. 191. Feed Water Heater, Direct Contact.

pump takes its supply, and by means of which the water is finally sent to the boiler.

The advantages of such a form of heater are simplicity in the apparatus itself and a quicker action than with tubes, as in the other form. The chief disadvantage lies in the fact that an additional feed pump must be provided; one for forcing the water from the hot-well to the feed-heater and the second for taking it from the heater and forcing it into the boiler.

Turning to the other style of heater, as illustrated in Fig. 192, we have a chamber or drum containing a nest of corrugated copper pipe. The feed-water passes on one side of the pipes and the steam on the other, as shown, the heat passing through from the one to the other as above described. Where such a heater is fitted to utilize the steam from an auxiliary exhaust, the heater forms simply a part of the exhaust passage and the steam passes through continuously, simply leaving a part of its heat behind.

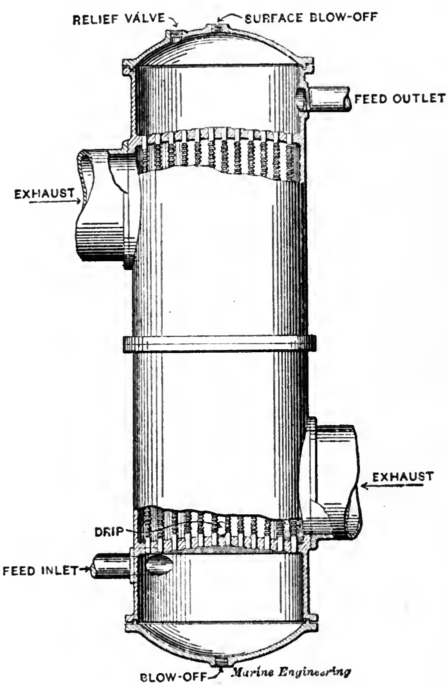


Fig. 192. Feed Water Heater, Surface.

In other forms of heater there is no continuous flow of steam through, but the steam is led to the steam side from the source selected and there gradually condensed by the loss of heat, while the water thus formed is drawn off as may be necessary. The heater is thus kept clear for efficient operation, and the water is returned to the hot-well or otherwise into the feed system.

In heaters of this type the flow of water is continuous from the feed-pump through the heater to the boiler, and no additional pump is required as with the other type referred to above. The difference is due to the fact that where the water and steam are separated, the water side of the heater can be operated under the full pressure in the feed-pipe and thus made part of the feed circuit. Where the water and steam are mixed the interior of the heater cannot be operated under any such pressure, and thus two separate pumps are required.

In heaters using steam from the steam pipe or from the receivers, it is clear that all such steam would ultimately go to the condenser and thence back to the boiler as feed-water, and hence that no heat is saved which would otherwise have been thrown away. In such cases it is not at first sight clear where the gain can come in, for the operation seems to be a simple shifting of heat from one part of the cycle to another without gain or loss in total amount. As a matter of fact the operation does consist of just such a shifting of heat, and here is where the gain in work comes in. This shifting of heat from one part of the cycle or routine of the steam to another introduces a change which brings the cycle a little nearer that for the highest efficiency as described in 59. While therefore there will be no saving of *heat* as such, the engine may be enabled to better use the heat which is provided and thus to show a larger return in useful work. These points cannot be here discussed in detail, but it seems at least worth while to note in general terms the chief source of the economy experienced with such heaters.

Especially will heaters of this type affect the routine of the steam favorably when they are arranged on the compound or step by step principle. In this arrangement the feed passes through a first chamber and receives heat from steam drawn from the low pressure receiver. It then passes on to a second chamber and there receives heat from steam drawn from the next higher receiver, and so on, in the last receiving a final addition of heat from steam of full boiler pressure. This type of heater, when of sufficient capacity to raise the temperature of the feed nearly up to that of the water in the boiler, will effect a marked economy in the engine, a saving presumably due to the change thus effected in the routine through which the steam is carried.

In addition to the saving thus effected by feed heaters, many engineers believe that they are of use in reducing the strain and wear and tear on the boilers by furnishing a hot rather than a cold feed, and hence that they are of distinct advantage to the boiler and well as to the engine.

Sec. 31. FILTERS.

Feed water filters are provided for removing oil from the feed water before it enters the boiler. See further on this point Section 41. Such filters are made in various forms, the chief features being the kind of filtering material employed, and the arrangement of the flow of water through it. Animal charcoal, sand, gravel, broken pumice stone, etc., form one class of substances, while fibrous materials such as sponges, bagging, toweling, etc., form another case. Of the first class, animal charcoal is the best, though somewhat expensive. It may, however, be

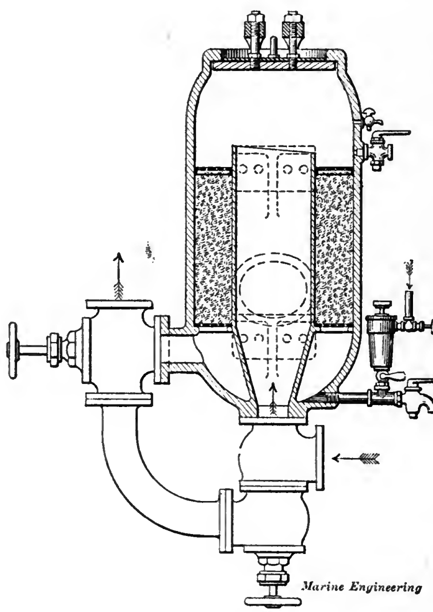


Fig. 193. Feed Water Filter.

removed from time to time, washed in lye water and replaced, and thus made to do duty for a long period of time. The various fibrous materials, such as sponges or bagging, soon become clogged also with oil and impurities, and require either replacement or washing and cleaning. After a few repetitions of cleaning in this manner, such material must be replaced with new.

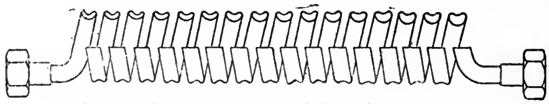
Filters also differ, according to whether the flow of water through the filtering material is forced by the pressure of the feed pump, or is due simply to the flow under the action of gravity. In gravity filters the flow may be up and down, two or three times through the bed of filtering material, the course being determined by suitable partitions or passages within the filter box. When the action is under pressure the filter forms a part of the feed pipe circuit and the water enters and passes through, urged by the action of the feed pump, though at a much lower velocity than through the feed-pipe itself. In such case, if the filter should become choked, excessive pressure might be developed between the feed-pump and filter with the possibility of a rupture of the latter. To avoid this danger a by-pass pipe and safety-valve may be arranged so that the valve will open under an excess of pressure and allow the water to flow around the filter to the continuation of the feed-pipe beyond. The safety-valve may also be maintained open by appropriate means and the filter shut off by stop-valves, thus sending the feed through the by-pass pipe and leaving the filter free for examination and repair.

In Fig. 193 a simple form of filter is shown with by-pass pipe and valves for controlling the flow of the feed. From the explanation above the operation of the filter will be readily understood.

Sec. 32. EVAPORATORS.

The office of the evaporator is to supply fresh water to make up the loss in boiler feed. Of the steam which the boilers supply not all can find its way back through the feed. Small steam leaks may occur at the various joints and stuffing boxes, some of the auxiliaries may not send their steam to the condenser, the whistle may be used (as in foggy weather), and so in various ways losses of fresh water will occur. The proportion of such loss varies widely with the circumstances, but will often amount to 5 per cent. and more. In order to avoid making up this loss with salt water the evaporator is provided.

A modern representative evaporator consists of a series of nests or coils of pipe contained within a chamber, as shown in Fig. 194. The chamber has a salt-water inlet, and steam from the boiler or from one of the receivers is passed inside the tubes. The heat in the steam passes through the tubes and forms steam or vapor of lower pressure on the salt-water side. The chamber is connected with the condenser or with the low pressure receiver, and the steam formed in the evaporator is thus passed into the main circuit and serves to make up the loss as specified before. At the same time the water formed in the coils by the loss of heat is drawn or trapped out as it accumulates, and is returned to the feed, so that all steam formed on the salt-water side is a net gain for the fresh-water account. The coils on the outer or salt-water side naturally become coated with scale, so that they must be cleaned from time to time. To this end they



CROSS SECTION OF EVAPORATOR COIL

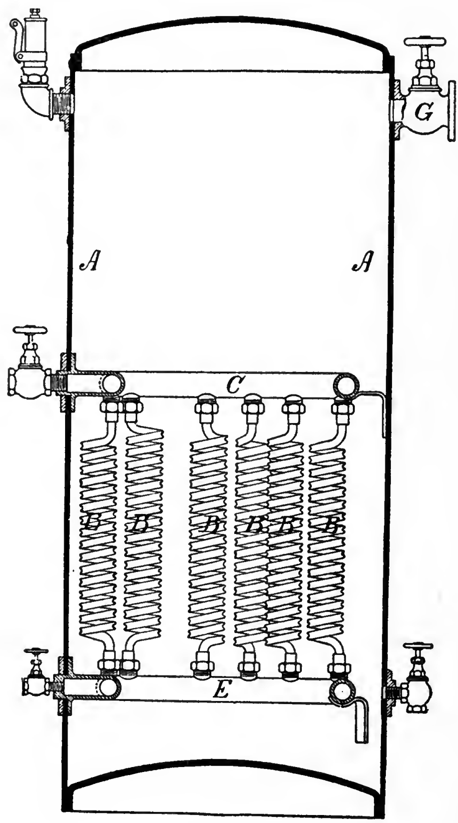


Fig. 194. Evaporater.

are usually made removable or arranged so as to be readily accessible through manhole openings in the shell. It is, of course, essential that the tubes be kept clean for the most efficient working of the evaporator.

In the operation of the evaporator the chief point requiring attention is the proper proportion between the amount and temperature of the inflowing steam and the pressure within the chamber on the salt-water side. If the pressure is low and steam is provided in excess, it may give rise to a violent ebullition or foaming, which will carry some of the salt water along with the vapor formed, and thus introduce salt into the circulating system. This condition must be guarded against by a proper control of the amount of steam admitted.

In the way of general maintenance, the tightness of the tube joints and the condition of the tubes as regards scale are the chief points requiring attention.

Sec. 33. DIRECT ACTING PUMPS.

In early marine practice the fly-wheel pump was a favorite type, and was used for all ordinary purposes where an independent pump was required, as for boiler feed, fire purposes, or for general purposes on shipboard. This pump consisted essentially of horizontal steam and water cylinders with the piston and plunger on a common rod and moving together. Attached to the rod was a cross-head with connecting rod leading to a crank and shaft carrying a fly wheel. The fly wheel served to carry the pump past the dead points, and the shaft served to carry an excentric which actuated a simple slide valve on the steam cylinder.

This type of pump, however, has almost entirely disappeared from modern practice, its place being taken by the direct acting pump with its greater compactness of form and better adaptation to the conditions of service.

We will now consider briefly the essential features of this type of pump, with a few examples drawn from modern practice.

As illustrated in Fig. 195, the pump is horizontal and consists of two cylinders, one for steam and one for water, carried on a common piston rod. The steam end is operated by means of a suitable valve gear as a simple reciprocating engine, and thus communicates the same movement to the pump plunger or piston. Each end of the water cylinder is provided with both inflow and outflow valves, as shown, and thus the pump becomes double acting,—that is delivering on each stroke alternately from one end and the other.

For operating the steam ends of pumps of this type, a great variety of ingenious valve gears have been devised.

The need for special device arises from the fact that there is no rotating part and no chance to use an excentric, and that the valve cannot be operated directly from the main piston rod. Where it is thus required that a single set of principal moving parts be self operating, the valve gear usually consists of the following chief features:

(1.) The main steam valve, often of special form, but usually operating as a simple slide valve.

(2.) An auxiliary plunger or piston moving in a cylinder formed in the valve chest, and coupled or connected to the main valve.

(3.) An auxiliary valve controlling steam and exhaust to and from the two ends of the auxiliary plunger cylinder.

(4.) Means for operating the auxiliary valve from the main piston-rod. Such means may consist of levers, links, rods, cams, etc., operated by tappets on the main rod, whose location or point of operation may be adjusted according to the length of the stroke desired.

The chain of operation is then in general as follows:

Just before the end of the stroke the tappet or other piece moved by the main piston rod gives motion to the auxiliary valve. This produces an adjustment of steam and exhaust for the auxiliary cylinder which results in a movement of the auxiliary piston and hence the movement of the main valve as desired. The motion of the main piston is thus reversed and the stroke takes place in the opposite direction, and so on continuously.

In Fig. 195 the lower section is horizontal and taken through the auxiliary piston and auxiliary slide valve operated by the levers and links as shown. The upper view shows in vertical section the auxiliary piston and main steam valve.

If two such pumps are placed side by side, it is found that the valve of each pump may be operated from the piston rod of the other. Hence by appropriate connections a pair of such pumps may be made self operative, the strokes being made alternately, and each piston rod running the valve gear of the other pump. Such an arrangement constitutes a *duplex* pump, a form which has enjoyed wide and continued favor among marine engineers for feed-pumps and for other purposes with generally similar conditions.

In the so-called "Admiralty" style of pump the motion of these parts is vertical, and the water valves are specially arranged with a view to ready examination and overhauling. As noted in Sec. 29, this general style is quite commonly used for feed-pump purposes. Such pumps may be either simple or duplex, but the duplex type is more commonly met with in this form. In Fig. 196 is shown one member of a duplex admiralty pump, the arrangement of the parts and operation of which will be apparent without further explanation. In Fig. 197 is shown similarly a single vertical type of feed-pump with independent valve gear,

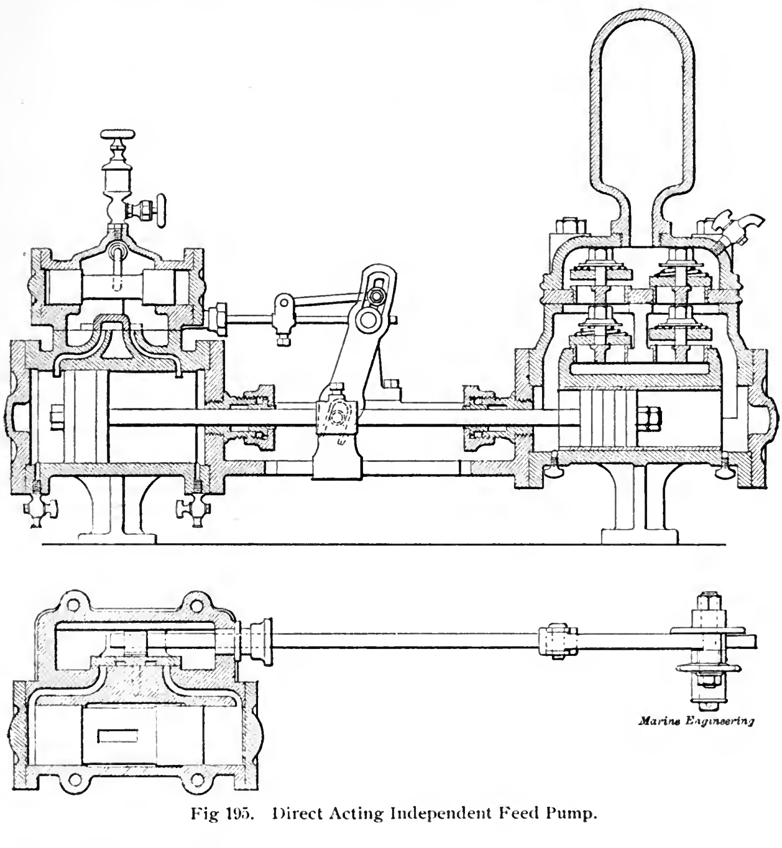


Fig 195. Direct Acting Independent Feed Pump.

the auxiliary piston being operated by the opening and closing of ports due to a rocking motion which is communicated to it by the levers and link work as shown.

Bilge pumps when independent, and all general service pumps, are usually of the direct acting form as above illustrated. The chief item of difference is found in the ratio of the areas of steam piston and water plunger. Where the water is to be delivered under considerable pressure, as for feed-pumps or for fire purposes, the area of steam piston will be from two to three times that of water plunger. Where the resistance to be overcome is less, as in a pump for freeing the bilge or for circulating

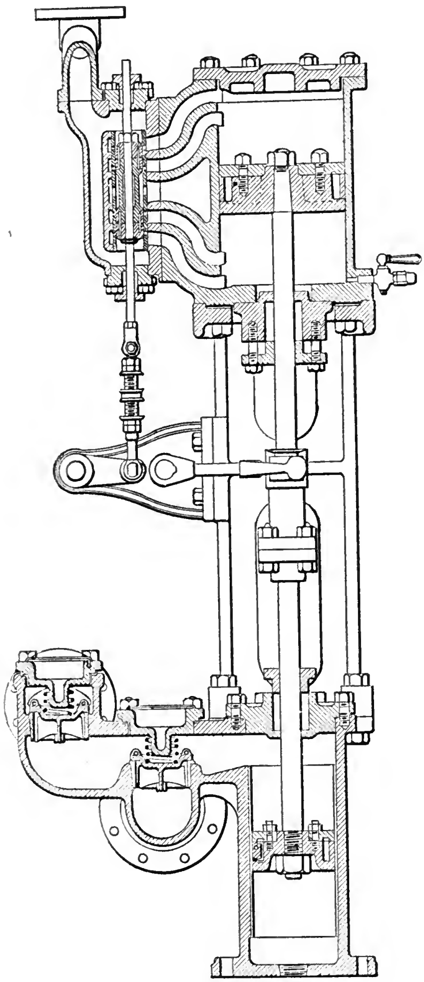


Fig. 196. Vertical Duplex Feed Pump. Admiralty Type.

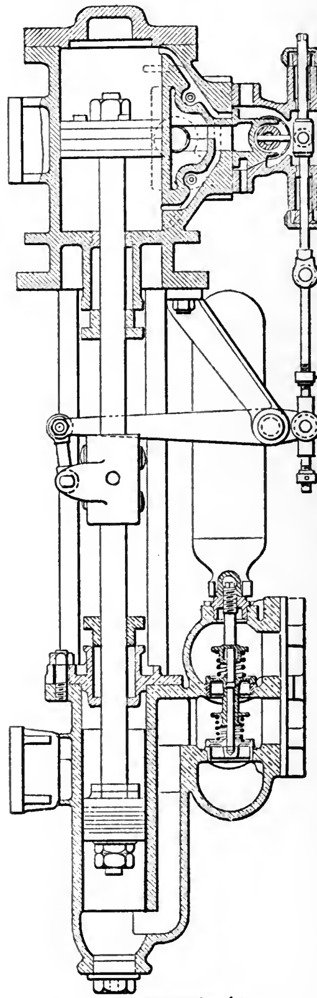


Fig. 197. Vertical Single Feed Pump.

water through distillers, evaporators, water-closets, etc., the water plunger may be relatively larger and we shall find such pumps with a water end only slightly smaller or equal in size or even larger than the steam end.

Such pumps are always so connected up, of course, as to enable them to be run from the auxiliary boiler.

Sec. 34. BLOWERS OR FANS.

The centrifugal blower is the type universally used on shipboard for all purposes requiring the handling of large quantities of air under light pressure, as for ventilation and forced draft.

As indicated in Figs. 95, 96, such a blower consists of a series of flat or nearly flat steel vanes carried on a shaft and surrounded by a casing. The principle of operation is the same as with the centrifugal pump, as described in Sec. 26. The rotation of the vanes sets up first a circular current or rotation of the air, and as a result of this motion, centrifugal force is developed which carries the air out toward the tips of the blades and develops an increase of pressure from the hub outward. If an outlet is then provided in the outer shell of the casing the air will be delivered at this point and the surrounding air will flow in to take its place at the intake about the hub. So long as the rotation is kept up these conditions will continue, and there will be a continuous flow of air in at the hub and out through the delivery passage under a pressure depending on the speed of rotation and other circumstances.

Blowers are driven by either steam engine or electric motor direct connected to the shaft, and are made of various forms so as to readily find a place in almost any position desired, thus requiring the smallest possible amount of otherwise valuable space.

In the operation of blowers the points of chief importance relate to the general care which must be given to the operating motor, whether electric or steam, and to the proper lubrication and care of the fan-shaft bearings.

Sec. 35. SEPARATORS.

In many types of water-tube boilers, special arrangements are provided for separating the steam from the water. These are usually located in the upper drum or chamber and consist commonly of one or more metal plates pierced with holes through which the steam passes to the stop-valve and steam pipe, and which exercise more or less of a straining or separating action on the water and steam. Reference has been made in Sec. 16 [4] to arrangements of this character.

In addition to such arrangements located in the upper drum of water-tube boilers, special devices known as *separators* are used wherever the steam is likely to have any considerable proportion of water. Such devices are found in great variety of form, and utilize various principles in their operation. The most successful are those which employ for separating the water from the steam the centrifugal force developed by a rotation of the steam as it enters or passes through the separating chamber.

The following description will serve to illustrate the operation of a typical separator of this character:

The separator, as shown in Fig. 198, consists of a vertical cylinder with an internal central pipe extending from the top downward, for about half the height of the apparatus, leaving an annular space between the two.

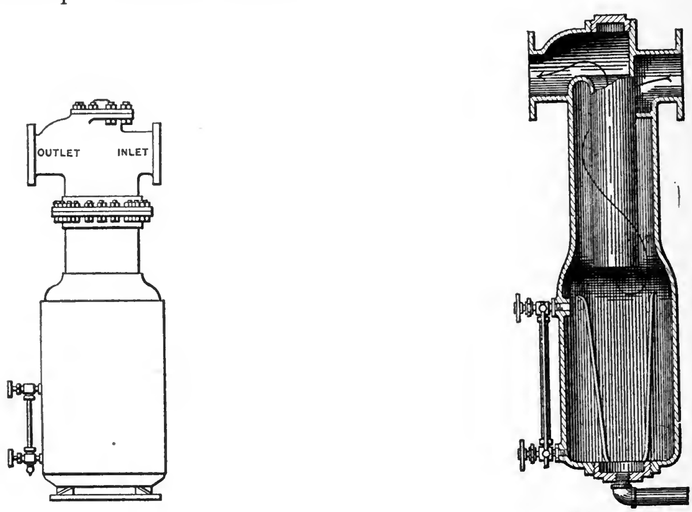


Fig. 198. Separator.

A nozzle for the admission of the steam is on one side, the outlet being on the opposite side or on top as may be most convenient in making the connections.

The lower part of the apparatus is enlarged to form a receiver of some considerable capacity, thus providing for a sudden influx of water from the boiler.

A suitable opening is tapped at the bottom of the apparatus for a drip connection, and a glass water gauge shows the level of the water in the separator at all times.

The current of steam on entering is deflected by a curved partition and thrown tangentially to the annular space at the side near the top of the apparatus. It is thus whirled around with the velocity of influx, and a centrifugal force is developed, which throws the particles of water against the outer cylinder. These adhere to the surface, so that the water runs down continuously in a thin sheet around the outer shell into the receptacle below, while the steam, following a spiral course to the bottom of the internal pipe, enters it abruptly, and in a dry condition passes upward and out of the separator, without having once crossed the stream of separated water, all danger of the steam taking up the water again after separation being thus avoided.

The water thus separated from the steam collects in the lower part of the chamber and may be drawn out from time to time or it may be led to a steam trap of approved form and

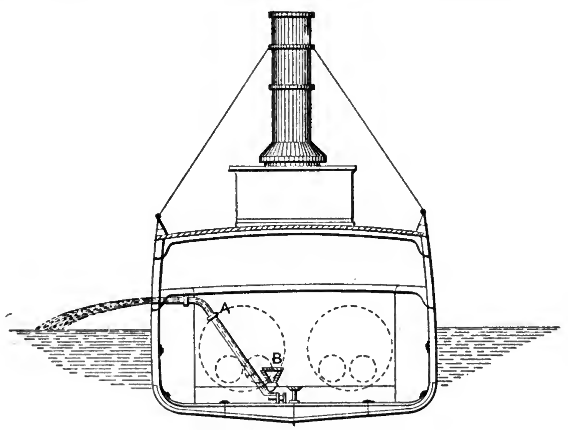


Fig. 199. Ash Ejector.

trapped out, thus making the operation entirely automatic. The water thus obtained will, of course, be of high temperature and should be led directly to the hot-well where it will aid in raising the temperature of the feed-water. The heat which it contains will thus be returned to the boiler, and saved, and all heat loss in connection with the operation will be avoided.

Sec. 36. ASH EJECTOR.

Ashes are either hoisted in a bucket by a special hoist to a point on the main deck level and there dumped into an ash-chute leading to the side of the ship and down into the water, or else disposed of directly from the fire-room by means of an ash ejector. Such a device is illustrated in Fig. 199. A represents a cast metal chute or pipe leading from the fire-room up and out through the side of the ship near or slightly above the water line. At the lower end this chute connects with a hopper, B, into which is led a pipe from the discharge of a pump. This pipe enters to a point near the lower end of the chute, into which its discharge is directed, and is contracted to a nozzle so that the water issues with a high velocity. The hopper may be closed by a cover, and if in this condition the discharge valve is opened and the pump started, a stream issues with high velocity from the upper and open end of the chute. If then the cover is removed

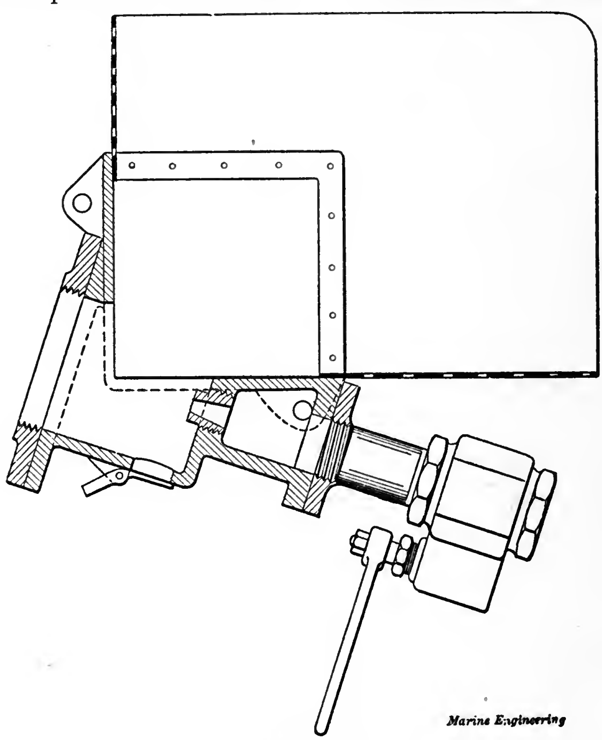
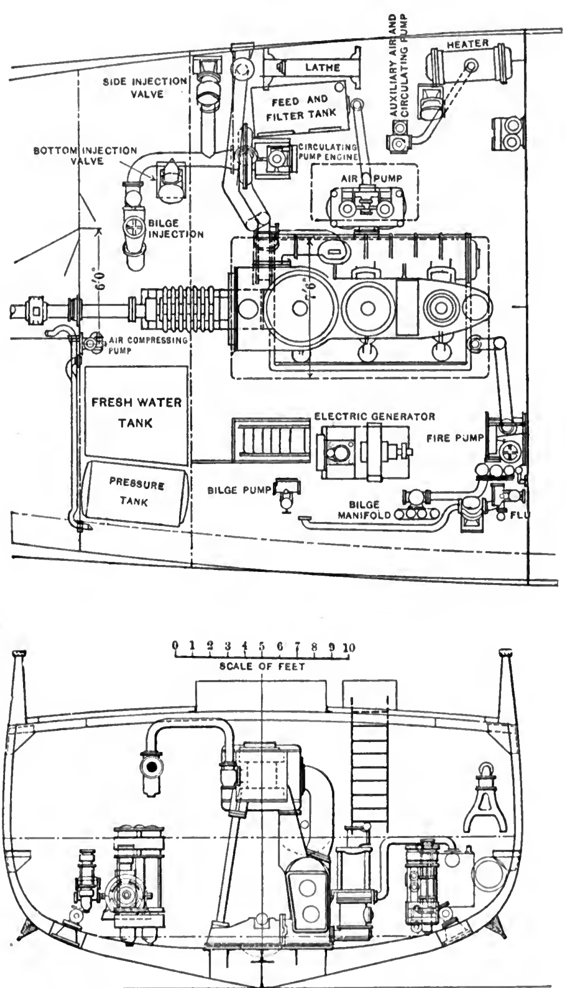


Fig. 200. Ash Gun.

and ashes shoveled into the hopper, they are caught by the stream, carried rapidly up, and ejected free of the ship's side in a mingled jet of ashes and water.

It is found by experience that at the upper bend the wear on the metal of the pipe, due to the scouring action of the ashes, is very rapid, and it is usually found necessary to make this bend in a separate piece of extra thick metal, and to provide by means of a proper arrangement of joints for its replacement as occasion may require.

A similar device known as the *ash gun* is shown in Fig. 200. Where possible the lead of pipe from the hopper to the ship's side is made straight so as to avoid all bends and elbows. The principles of operation are the same as above explained.



Figs. 201, 202. General Arrangement Plans.

Sec. 37. GENERAL ARRANGEMENT OF MACHINERY.

We shall not here discuss in detail the various questions which may arise in connection with the problem of the general arrangement of marine machinery. It will be sufficient for our present purpose to note the fundamental principles which must be held in view:

(1.) Each piece must be located so as to favor, as far as possible, examination and repair.

(2.) Each piece should be located with reference to handiness of care and control in routine operation, and in such way as to interfere to the smallest practicable extent with the routine care, examination and repair of other pieces.

(3.) Due regard must be had to economy of space and such combinations of the various pieces must be sought as will require the minimum total space, while giving the necessary freedom in accordance with the principles noted above.

(4.) The influence of the location of the various pieces on the arrangement of the piping must be carefully noted and due weight must be given to simplicity, shortness and directness of the various lines of piping.

In Figs. 201, 202 are shown illustrations of a general arrangement plan in which the condenser is located in the engine framing. The remaining features are independent, and include those most commonly met with in the auxiliary equipment of the engine room.

CHAPTER VI.

Operation, Management and Repair.

Sec. 38. BOILER ROOM ROUTINE.

In the present section it is the purpose to give brief hints and suggestions regarding the routine of operation and management in the fire-room in getting under way and on the voyage, first supposing that everything is working smoothly and without trouble, and then to notice the chief emergencies which may arise. We shall first suppose that fire-tube boilers are in use, and later give such supplementary suggestions as may be suitable for water-tube boilers.

[1] Starting Fires and Getting Under Way.

A general examination must first be made of the boiler and fire-room equipment in order to make sure that everything is in readiness for getting up steam. Among the more important points to be attended to the following may be mentioned:

See that the coal bunker doors are in proper working order and if the bunker is partly empty it may be well to air it by opening the door and taking off the deck plates.

See that the coal handling gear is on hand and in proper condition.

See that the necessary fire tools are provided, and distributed as needed.

Examine the grate-bars, bridge-walls and back-connections, and note whether the area of passage above the bridge-walls is properly proportioned. For usual conditions it should be from 1-5 to 1-7 the grate area.

Note the condition of the tubes both from the front and back connections.

Examine the dampers in uptakes and funnel to see if they are in working order, and open them preparatory to lighting the fires.

Examine carefully all valves, cocks, piping and connections and make sure that everything is connected up as it should be, and that no valves are open which should be closed nor closed which should be open.

See that no waste or other inflammable substances have been left about by workmen on the tops of the boilers.

If the water has not been previously run up in the boilers, this may be under way in the meantime. In modern practice the boilers are always filled with fresh water where possible, obtained from a hydrant on the dock or water-boat alongside, and put in usually by a hose through an upper manhole. If, however, the boat is lying in fresh water, or if by necessity the water is to be taken from overboard, it is then run in through the bottom blow and Kingston valve. In the meantime examine the connections leading to the water-gauge and cocks. Clean the glass if necessary, and make sure by means of a wire that the opening through the cocks is clear. The packing of the gauge glass should also be examined and renewed if necessary. When the water appears in the gauge glass and shows from one-half to two-thirds full in each boiler, it may be shut off.

All open manholes may then be closed, and the boilers are ready for the fires.

Notice of the time when steam is required should have been given not less than from six to eight hours in advance, and many engineers prefer a still longer time in which to bring along everything into working condition. With hard coal a certain amount of wood is necessary in starting the fires. With soft coal less wood is required, and if necessary oily waste may be made to answer the purpose. If fires are up in the donkey boiler a little live coal may be taken from them to assist in starting. As soon as the fires are going the hydrokineters are put on if such appliances are fitted. In some cases arrangements are made for drawing the water by the donkey or auxiliary feed-pump from the bottom of the boiler by the bottom blow and returning it through the feed-pipe, thus producing a forced or assisted circulation. Where there are no appliances for forcing the circulation during this period, it is considered good practice to light first the fires in the center furnaces, and later, by one or two hours, those in the wing furnaces. The natural circulation thus produced will more nearly even up the temperature within the boiler than if all fires are lighted at the same time. After the fires are fairly going the funnel or uptake dampers may be partly closed so as to hold the fires back, and bring them along at a moderate gait as desired.

While the boilers are thus warming up and before steam has formed, a last look may be given to the boiler mountings and their connections. The various cocks and valves should be worked, and especially the stop and safety valves, in order to make sure that none are jammed or in any way out of order. The oil lamps for the steam and water gauges may also be trimmed and lighted, or the electric bulbs cleaned, if such are provided.

During this period the steam-pipe drains and safety valves are usually open to allow of the escape of the air and of the condensed vapor as formed. In some cases, however, the safety valves are closed, and the stop valves being open, the air and vapor are expelled along the steam-pipe and through the engine, thus beginning the process of warming up. Many engineers, however, prefer to keep the boiler stop valves closed until steam is formed, and to discharge the air through the safety valve, or in some cases through a specially fitted air-cock. If steam is already up on some of the boilers or if there is no auxiliary steam-pipe and the pressure from the donkey boiler is on the main steam-pipe, then of course the stop valves must be closed on the boilers in which steam is being raised, and they must remain closed until the pressure on the boiler is equal to that in the steam-pipe. In opening a boiler stop valve connecting with a pipe in which there is no pressure the following precautions should be taken:

(1) The pipe should be thoroughly drained and especial care should be taken that there are no sags, bends or U's unprovided with proper drains, and in which a pocket of water may have collected.

(2) The valve should be very carefully eased from its seat and opened only from a quarter to a half turn until the pipe is under full boiler pressure and has taken the temperature of the steam, and the drains are discharging steam instead of water. In opening a boiler stop valve connecting with a pipe in which there is approximately the same pressure as in the boiler, it is

simply necessary to ease the valve from the seat and note by the sound whether there is a sufficient difference of pressure to cause any violent flow in one direction or the other. As soon as the absence of such evidence indicates an equality of pressure on both sides of the valve, it may be opened out as desired.

The two fundamental principles underlying much of this routine and detail are simply as follows:

(1) To prevent as far as possible *sudden* changes in the temperature condition of the boilers, piping and machinery, and

(2) To prevent throughout the steam-pipe system the accumulation of water at any point.

If these two points are kept clearly in view and good engineering judgment used in carrying them out, the life of the boilers and machinery will be prolonged, and danger of ruptured pipes through the effects of water hammer will be avoided.

After steam is formed and the pressure has risen to some 40 or 50 pounds the hydrokineters may be shut off, especially if the ship is to get under way as soon as ready. If, however, the boilers are to stand some time with steam up, it may be advisable to turn on the hydrokineters from time to time, at least as long as the pressure in the donkey boiler is sufficient for the purpose.

The fires in the meantime have been kept simply in good condition without forcing, and even if they work under a forced draft system, only enough air should be provided during this stage to bring them along at the gradual pace which will allow the boiler to properly accommodate itself to the change in temperature and other conditions.

The fire-room auxiliary machinery should also be examined during this period, and tested under steam from the donkey boiler if possible. The feed pumps should first receive attention, in order that there may be no question as to the proper supply of feed-water when required.

The fan engines should be examined, oiled and turned over under steam.

The ash-hoist gear and engine, or ash ejector and pump, should be examined and put in working order.

If steam for these purposes is not to be had from the donkey boiler, then as soon as a sufficient head is formed on the main boilers these auxiliaries must be examined, taking in all cases the feed pump first.

Soon after lighting fires it may be desirable to slacken up somewhat on the funnel guys on deck, in order that the expansion of the funnel may not bring an undue stress upon the guys and their connections, or upon the funnel and its supports. After the ship is away and the funnel has taken its temperature for running conditions, the guys may be tightened up so as to properly support the funnel in a sea way.

[2] Fire Room Routine.

At length we may suppose that full head of steam has been formed on the boilers, that the fires have been brought up to proper condition, and that the ship has gotten under way for the voyage. As soon as possible the operations in the fire-room should be brought to a regular routine. This will involve the following chief features, which we shall consider separately: (1) Firing. (2) Water tending. (3) Disposal of ashes. (4) Cleaning fires. (5) Sweeping tubes.

Firing.—The routine of firing should be so arranged that no two furnaces in boilers connected to the same stack shall be open at the same time. If this is not practicable, then care must be taken to avoid at least firing at the same time furnaces in opposite ends of double-end boilers, especially if there is a common combustion chamber. Two furnaces in a single-end boiler, or in the same end of a double-end boiler will, of course, never be fired at the same time. It is now well understood that firing light and often is better than heavy and at great intervals. There is, however, a limit to this, for the oftener the firing the more are the furnace doors open and the more is the draft subject to disturbance, while the arrangement of a suitable routine becomes more and more difficult.

Light and frequent firing, especially with water-tube boilers, is now, however, the rule where the best results are to be obtained. The furnace door should be opened smartly and kept open only the minimum time needed to get the coal on. Hard coal is spread in as even a layer as possible over the grate. For firing soft coal two methods are available. When firing for coal efficiency, that is to get the most heat out of a pound of fuel, the coal should be first charged in front and coked, and then should be pushed back and burned. When firing for weight efficiency, that is to get the most power out of the boiler, the former method would be too slow and the coal must be spread over the fire and burned without waiting for the separate distillation and combustion of its gases. Where the coal runs irregular in size the large lumps should be broken into pieces not larger than the fist. The thickness of the fires varies with the conditions, from six to ten or twelve inches, or even thicker under a heavy forced draft. With a given speed of fan the air pressure in the ash-pits will vary widely with the thickness of the fire, rising as it is thicker, and falling as it is thinner and the air finds more ready passage through. With a thick fire it will therefore be easy to get a strong draft pressure in the ash-pits, while with a thin fire it will be perhaps impossible, even with a much higher speed of fan. A strong draft pressure will not, however, produce the corresponding rate of combustion unless the thickness and condition of the fire are such that the pressure is able to drive through it the necessary amount of air. For the best combustion the thickness of the fire should be so adjusted to the draft pressure that the latter is able to drive the necessary air through, and keep it in a state of active combustion throughout from fire grate to upper surface. Care must be taken to keep the grates evenly covered, especially at the back, and to prevent the formation of relatively thin or bare spots. A spot which is relatively thin allows of the passage of relatively more air. This further increases the combustion at that point and the spot becomes still thinner, thus allowing more and more air to escape freely instead of passing through the remainder of the fire as it should.

In the intervals of firing the pricker and slice bars may be used to clear away the ashes and clinker, if such is forming. Care should be taken to prevent the formation of dull or dead spots due to the accumulation of ashes or clinker, especially at the corners of the grate. Among old firemen a familiar saying relating to this point is: "Take care of the corners and sides of the fire and the middle will take care of itself." The ash-pits should also be kept clear of ashes, for if allowed to accumulate they will prevent the passage of air to the grate, especially at the back. If the passage of air is thus interfered with to any considerable extent there will be also danger of overheating the grates and of bringing them down into the ash-pits.

In connection with the use of the slice bar, it must not be forgotten that every opening of the furnace door means an inrush of cold air into the furnace, a checking of the draft, a disturbance of the combustion, and often severe strains on the structure of the boiler, due to the sudden chilling and contraction which the heating surfaces undergo. If shaking grates are fitted much of this cleaning may be done without opening the door, though no form of grate is quite able to deal satisfactorily with coal showing a decided tendency to form clinker.

In thus working the fires a certain amount of fine unburned or partly burned coal will be shaken down into the ash-pit. In some cases this forms so large a part of what comes through the grate, that it may be immediately thrown onto the fire and burned over again. In most cases a sifting or washing of the ashes and separation of the combustible from the non-combustible would show a surprisingly large per cent. to be available as fuel, and some saving could usually be effected in this way. It is rare, however, that anything of the kind is attempted, as with present prices of coal it may be doubted whether the additional appliances and labor would be paid for by the saving effected.

Water-tending. The care of the water is the most important and responsible of the duties in the fire-room. The ideal is to keep the water regularly flowing inward through the check-valves at about the same rate as it is flowing outward as steam through the steam pipe. This requires constant watch and adjustment of the valves, closing down where it is entering too rapidly and opening up where it is entering too slowly. Instead of this method it is often the custom to put the feed on strong first to one boiler and then another, in order, according to the firing, feeding the boiler up when the fire is at its best, and shutting down when it is freshly coaled. The steady and uniform feed is, however, better because it approaches more nearly to a uniform condition of the boiler, especially on the water side.

The position of the water is determined, of course, from the water gauge and cocks. It is necessary, of course, that the gauge and its connections be clear of any obstruction in order that the height of the water may be properly indicated. To make sure that everything is clear the gauge glass and connections are blown through by the "double shut off" method as follows: In Fig. 203, *G* represents the glass, *A* the drip cock, *B* and *C* the cocks connecting to the stand, and *D* and *E* those connecting the stand to the boiler. First, *B* and *E* are closed, and *A* is opened. If steam blows through it shows that *A G C D* are clear. Second, *C* and *D* are closed and *A* is opened. If water blows through it shows that *A B E* are clear. The action of the water in the glass will usually show to an experienced eye whether or not the connections are clear. If the water is lively and follows the rolling of the ship it is a good indication that the passages are clear. Otherwise it indicates that an obstruction exists which must be sought out without delay. In the meantime the water cocks are relied upon, and in fact many experienced water tenders prefer the indications of the cocks to those of the glass, while they should in all cases be freely used as a check on the glass. To those without experience, however, the glass is less apt to be misleading. The indications of the

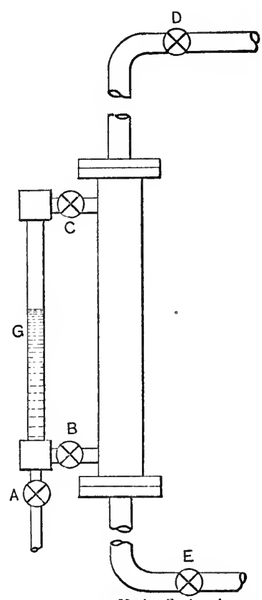


Fig. 203. Test for Water Gauge and Glass.

water cocks are sometimes difficult to interpret, because frequently it is not easy to tell whether water or steam is blowing off. With high pressure steam especially, a jet of water issuing at a temperature of 350 degrees to 400 degrees is instantly surrounded with a shell of vapor formed by the vaporization of part of the jet. Furthermore, if the water in the boiler is in active ebullition near to the surface so that the jet would be drawn from a mixture of steam and water, then on issuing it becomes practically a jet of moist steam. On the other hand if the water is well below the cock so that the jet would be drawn from steam alone or from moist steam, then on issuing it will become dry and usually super-heated. It is also a fact, especially with water-tube boilers, that due to a kind of lifting action, a cock will often discharge moist steam or a mixture of water and steam, even if the water level is somewhat below the mouth of the cock. It is hence readily seen that the indications from the cocks must be interpreted with judgment, and that some experience is necessary in order to always draw correct conclusions from them. It is often difficult to distinguish between an empty glass and one entirely full. In order to make sure close the cock *B*, Fig. 203, and slightly open *A*. If the gauge is full of water the surface will gradually descend, first coming into view in the top of the glass and then passing out of view at the bottom. If then *A* is closed and *B* is slightly opened, the water will rise again in the glass and pass out of view at the top.

Blowing off. Blowing off boilers to reduce the concentration or density of the water is now rare in good practice. Instead of reducing density by introducing sea-water for feed make up, evaporators are installed for providing fresh water make up, or for short runs additional fresh water is often carried in double bottoms or spare tanks provided for the purpose.

In modern practice the purpose of blowing off is (1) to get rid of mud or slush in the bottom of the boiler or in the special mud drums of a water-tube boiler, and (2) to get rid of oil and scum at and near the surface of the water. For the former a bottom blow or special mud cock is required, while for the latter the surface blow is used. In ordinary experience on deep sea voyages where evaporators or other fresh water make up is provided, the use of the surface blow is all that is needed to keep the water in good working condition. It must not be forgotten that the use of the blow-off means a direct loss of heat, and hence it should be used with discretion, and no more frequently than is needed for the purpose in view. An idea of the condition of the water in the boiler near the surface may be obtained by drawing off a little water from a cock fitted into the surface blow pipe, or from a gauge cock fitted directly to the boiler. The water being allowed to cool and settle, is then poured into a glass jar, when its condition is readily noted, and the need of using the blow determined.

It may be well to speak at this point of the proper method of testing, from the outside, the correct position of a plug cock handle for "closed" and for "open." Instances have been known where there was no mark on the head of the plug, and the handle becoming bent or being wrongly placed, the cock was left shut when it was supposed to be open, or open when supposed to be shut, with the possibility of most serious consequences, especially in the latter case.

A careful examination of the cock, aided if need be by placing the ear to the chamber, will suffice to tell whether or not the cock is open and water or steam passing through. Then the cock being open let it be turned in one direction until it is just closed, and then back in the other direction until it is closed again. Half way between these two positions it will be wide open, and at right angles to the latter position it will be full shut.

Taking the Saturation or Testing the Density of the Water.—The density of the water is determined by the use of a hydrometer or salinometer as it is often termed. Under modern conditions where evaporators provide fresh water make up, the density rises but slowly, and it is usually only necessary to observe its value once or twice a day. It is usually not allowed to rise above two or two and one-half. See Sec. 17 [10].

Disposal of Ashes.—For the disposal of ashes two chief means are in use. According to the older method they are sent up and disposed of through an ash chute leading overboard and down the side of the ship to the water, and this method is still extensively used in large and deep ships. In the more modern system they are disposed of from the fire-room direct by means of an ash ejector. In either system it is usually sufficient to dispose of the ashes once in a watch, and they are collected, wet down and either hoisted in buckets or shoveled direct into the ash hopper usually between 6 and 7 bells.

Cleaning of Fires.—The routine working of fires spoken of above will suffice to keep them in fairly good condition for several hours, provided the coal is of fairly good quality. It usually becomes necessary, however, to give to each fire from time to time a more thorough cleaning than is possible in the manner previously referred to. To this end the fire should be taken when partly burned down, but not too far lest there be nothing left after cleaning on which to build up again. One side may be cleaned first, working the good coal over to the other side, separating out the clinker and ashes, and hauling out the latter. Then similarly with the other side, working the good coal over to the side first cleaned and pulling out the clinker and ash. The live coal is then spread over the grates, fired lightly, and so brought up again into regular conditions. In some cases there is so little left after a thorough cleaning that live coal must be borrowed from another furnace to save the fire. Only judgment and experience can determine the best point at which to clean a fire so as to insure the minimum loss of heat, and at the same time have enough coal left to nicely build on again. Some engineers prefer to burn the fire almost completely down to the ashes and clinker, and then pull the entire contents of the grate out and start afresh. This method, however, chills the furnace and more seriously interferes with the operation of the boiler, and is not to be recommended. It must of course not be forgotten that heat is lost with the clinker and ashes withdrawn, and the general operation should be so conducted as to keep this loss down to the minimum possible.

Under usual conditions the fires will need cleaning in this way at intervals of from 12 to 16 hours, or at least once a day.

Sweeping Tubes.—In addition to the cleaning of fires the tubes will require cleaning from time to time, dependent on the character of the coal and other circumstances. With soft coal and moderate draft they will soon become partly filled with soot and ashes, thus choking the draft still further, and preventing the transfer of heat to the water through the metal of the tube.

To prepare for sweeping tubes the draft is checked, ash-pit doors put up, furnace doors opened and front connection doors raised. Care should be taken to wait until the fire is burned partly down before doing this, so that the circulation of air through the grates may not be shut off while the fires are too heavy, thus endangering the grate bars. For cleaning the tubes the ordinary wire tube brush may be used. This consists of a mounting carrying wire bristles and fitted usually with a jointed handle by means of which it is pushed and pulled through the tubes, thus cleaning out the soot and ashes collected there. A more modern method consists in blowing through the tubes with a steam jet. The mounting of this appliance consists of a flange or conical ring fitting closely to the end of the tube and provided with a steam nozzle directed along the center of the tube. A handle is provided for holding and guiding the apparatus, and steam is led to it by means of a flexible hose. By this means the ashes and soot are driven out of the tubes into the combustion chamber. By still another form of apparatus the jet is not directed into the tube but across the front end producing a suction, and thus drawing the ashes and soot to the front connection and discharging them up the funnel.

The operation of sweeping tubes is one that is necessary to maintain the continued efficient operation of the boiler, but it must not be forgotten that it involves a serious disturbance to the draft of the whole battery, that the chilling of the heating surfaces and interruption to the regular routine are hard on the boiler itself, and that hence, it should only be done when necessary and then as quickly as possible.

Stopping Suddenly.—With everything going along its regular schedule, suppose that the engine is suddenly stopped. The dispositions to be taken will depend on whether the stoppage is momentary or whether it is expected to last for some time. Here again the caution regarding a sudden change in the conditions must be kept in mind. If the stop is but momentary it will probably be sufficient to shut off the draft, close the dampers and put on the feed strong, standing by to ease open the safety valves in case the pressure rises too near the point of blowing off. If the stop is to be longer it may be necessary to still further check the fires by putting up the ash pit doors and opening the furnace doors. Caution must be exercised in thus checking the flow of air through the grates lest there be danger of overheating the bars, or even of bringing them down into the ash pit. Of these various steps for checking the fires the opening of the furnace doors and the sudden chilling of the heating surfaces is the most objectionable and should not be resorted to unless necessary. As an additional means the fires may be freshly coaled, especially with dampened coal. This will check the formation of steam and provide fuel for bringing them into good condition for the next start. A period of stoppage like this may also be taken advantage of to clean such fires as may be in need of it.

In addition to checking the formation of steam, that which is formed may often be used in a variety of ways. If evaporators are provided it may be turned on to them and thus go toward increasing the store of fresh feed water. The bilge pump may also be put on strong, and if its exhaust is saved there will be no loss of fresh water. In some cases with independent air and circulating pumps a bleeder is provided for taking the steam direct from the main steam pipe to the condenser. Here it is condensed and then sent by the feed pumps back to the boilers, thus avoiding blowing off at the safety valves and the loss of fresh water, and allowing the fires to be gradually reduced to the condition desired for the period of stoppage.

Here again in all of these operations general principles are often worth more than a multitude of minor directions. These principles are (1) *Sudden* chilling of the boiler heating surfaces must be avoided as far as possible. (2) Fresh water in the form of steam should not be wasted, and (3) Care must be taken not to allow the grate bars to melt down.

So far as relates to the general securing of the machinery and gear in the fire-room, the hints given in connection with getting under way will be a sufficient guide in reversing the process.

Supplementary Hints Relating to Water-Tube Boilers.—In water-tube boilers the circulation is usually more nearly natural than in fire-tube boilers, and circulating devices are not, therefore, required. Steam may be raised in such boilers in from twenty minutes to one hour, depending on the type, character of the draft, etc. With this type of boiler it is especially necessary that for the best results the firing be light, often and regular, and that the fires be kept as nearly as possible in a uniform condition. It is also necessary that the feed be regular, and the water must be carefully watched, since from the small amount contained, any lack of feed in a given boiler will be followed by rapid lowering of the level, and by a rapidly increasing likelihood of danger to the tubes. In water-tube boilers it is especially necessary that nothing but fresh water be used as feed, and great care must be taken to keep the condenser tight and the fresh water make up ample in quantity.

The tubes of water-tube boilers become coated with soot and ashes on the outer or fire side, and it is usually a very difficult matter to satisfactorily clean them without the use of a steam jet. In continued steaming for long periods, it will usually be found necessary from time to time to let the fires die down somewhat and to use what methods are available for blowing off and dislodging the soot from the tubes.

When stopping or standing by, the same general means may be adopted as in fire-tube boilers. As regards injury through sudden change of temperature, the water-tube boiler is somewhat less liable than the fire-tube. This is due to the nature of the construction which, especially with curved or built up elements is much more elastic than in the fire-tube boilers. It is always better, however, to avoid sudden temperature changes where possible, and the same principles may be properly applied here as previously discussed in reference to the other type of boiler.

Coming Into Port.—When coming into port notice will usually be given some hours in advance, so that the fires may be worked into a condition in accordance with their expected disposition after arrival. If they are to be drawn and the boilers opened up for examination and repairs, they will be allowed to burn down as low as possible so as to use no more fuel than necessary, and to leave as small an amount as possible to finally haul, while at the same time sufficient steam must be maintained to bring the ship safely to her anchorage or dock. If, on the other hand, the fires are to be banked, they will not be allowed to burn so low. It may be recommended to bank fires on the front of the grates, as in such case the air is heated as soon as it enters the furnace and the boiler is kept at a more nearly even temperature than if they are banked at the back of the grate. As the fires are banked they should be cleaned and enough fresh coal put on to hold them in the condition desired. If the fires are properly managed there will be little extra steam after the engines are stopped, and this may be readily disposed of by means of the evaporator, bilge pump, bleeder, safety valve, etc. Loss of fresh water at this time is of course less objectionable than when on the voyage, and if desired the steam may all be blown off through the safety valves. Many engineers, however, object to using the safety valves and escape pipe for this purpose except as a last resort, and prefer other means as mentioned. In passenger vessels the noise occasioned is usually considered objectionable, though to obviate this a muffler is frequently fitted in the escape pipe.

If the boilers are to be opened fires are allowed to die out or are hauled immediately. If time permits the former plan is preferable, as the change in the condition of the boiler is more gradual. When the fires are finally hauled and the furnaces, back connections and tubes cleaned out, the ashes, soot and clinker are wet down and piled away until they can be disposed of to the ash barge, as few harbor regulations allow the dumping of ashes overboard. In wetting down the fires after they are hauled out on the fire room floor, or in wetting down ashes at any time, care should be taken not to wet the fronts of the boilers or the mouths of the ash pits. The local chilling will not improve the quality of the steel, and the alternate wetting and drying will increase the opportunities for surface corrosion. For the same reason damp ashes should never be piled up in contact with the boiler or furnace plates, as in many instances serious corrosion has resulted from a neglect of this precaution.

The fires being burned out or hauled, some engineers proceed to blow the boilers down immediately. This plan, however, cannot be recommended and should not be adopted unless the time available for examination and repairs is so short as to make it absolutely necessary. It is far better to let the steam condense and the water gradually cool, and then draw it out by means of a pump, or in some cases run it into the bilge. In this way the boiler cools more gradually and the structure is left in better condition, while on the water side the scale and incrustation will usually be made softer and more easily removed than when the boiler is blown down with steam on.

[3] Emergencies and Casualties.

(1) *Foaming and Priming.*—These terms refer to a disturbed condition of the water in the boiler, of such a nature that the water level is more or less uncertain in location, and the steam space is partially filled with foam or a mixture of foam and water. In severe cases of foaming, steam seems to be given off from almost the entire mass of water in the boiler, causing it to rise bodily as foam and water and fill the whole steam and water space, thence entering the steam pipe and passing on to the engine. In other cases the water seems occasionally to rise in gulps, nearly unmixed with steam, and entering the steam pipe pass on to the engine. The terms *foaming* and *priming* are often used as meaning practically the same thing. Where a difference is implied, foaming is understood to apply more especially to the uplifting of the mixed steam and water as foam, while priming may refer more particularly to the lifting of water as such, and its passage over into the engine. There is, however, no clear line of distinction between the two kinds of disturbances, and there are all grades intermediate between the extremes.

Foaming may be due to the presence of certain forms of oil or grease, or to the excess of soda used for scale prevention, or to other impurities in the water, or to the demand for steam too large in proportion to the steam space in the boilers. A sudden change in the character of the feed water may also produce foaming. In former days when jet condensers were in common use, boilers were liable to foam in passing from sea water to fresh water, especially if the latter was muddy, and again in passing from fresh water back to sea water. In modern practice foaming is due either to the presence of oil, or to the extreme demand for steam from the boiler. In the former case the oil must be removed from the boiler by a free use of the surface blow, and kept out by a proper filter. In the latter case the engine must be slowed and the demand for steam reduced to an amount which the boiler can supply without the danger of such disturbance.

As a result of foaming the engine slows down, power and speed are lost, while due to the possible inability of the relief valves to handle all of the water coming into the cylinders, there may be serious danger of breakdown. There is also danger to the boiler in foaming, because the water level cannot be known with certainty, and plates or tubes may become overheated, with danger of collapse and rupture. The tendency to foam is, therefore, a symptom of serious import, and no steps should be neglected to discover and, if possible, to remove the cause.

(2) *Feed Pump.*—In modern practice an auxiliary feed-pump is always provided except perhaps in very small craft. If then the main feed-pump refuses to work, the auxiliary pump must be brought into use while the other is under examination.

The chief causes which may disturb the operation of a feed-pump are the following:

(a) Jamming of check valve or other closure in the delivery pipe.

(b) Water in the steam cylinder.

(c) Derangement or sticking of the steam valves.

(d) Jamming or sticking of the water or steam plunger in its cylinder.

In addition to these causes which may affect or prevent the motion of the pump, the following causes may prevent it from throwing water into the boiler, even though its movement may be entirely regular:

(e) Split in the feed-pipe, or valve open, allowing the escape of the water at some unexpected point.

(f) Excessive wear of water plunger.

(g) Split or leak admitting air on the suction side or into the suction pipe.

(h) An excessively high temperature of the feed water.

(i) Derangement of the suction or discharge valves.

To make sure that the delivery pipe is free, one or more feed-checks and the air-cock on the pump may be opened. If there is no movement of the pump or no discharge from the cock it may be concluded that the trouble is located elsewhere.

The drain valves in the steam cylinder should then be blown out freely, and if there is still no inclination to start, the trouble is presumably with the valve gear.

A fresh supply of oil should be admitted to the valve-chest, and an attempt may be made to work the tappets or other valve mechanism by hand. In many cases this will suffice to start the pump off at its regular gait. If it does not, the chances are that the trouble is more serious, involving the stopping up or clogging of some of the auxiliary ports or passages, or the sticking, jamming or excessive wear of some part of the valve-gear. A removal of the bonnets and complete overhaul can alone lead to a discovery of the difficulty in such cases.

The jamming or rusting of the steam or water plungers in their cylinders could only result from long disuse and gross neglect, and can hardly be considered as of likely occurrence in routine work.

If the feed-pump is working properly and throwing water into the boiler, the chamber of the check-valve will be relatively cool, there will be a click as the valve rises and falls, the air cock at the pump will show a stream, and the water will rise in the boiler gauge glass. At the same time an experienced eye and ear will detect by the manner of the pump, by the way it moves and by the character of the sounds, whether or not it is throwing water. If then the pump works, but does not seem to be throwing water, we must have resources to causes such as those mentioned in (e)-(i).

There are here two chief questions to be answered. First, is the pump getting water? and second, if it is, where is it going? The air cock will usually serve to answer the first question. If water appears here, and if the pump shows by its action that it is handling water, it is evident that there must be escape at some unexpected point. The feed-pipe must then be carefully examined for leaks, and all valves or connections leading to or from it should be examined to make sure that the water is not escaping in some such way. In one case coming under the author's notice the main feed-pipe was fitted with a small branch leading to the forward tank. This branch was closed off from the feed-pipe by a globe valve. Due to the ignorance or carelessness of some attendant, this valve was jammed wide open instead of being jammed hard shut. At low or moderate pressure the feed-pump would throw enough water to feed the boiler in spite of this leak. The trouble was therefore not discovered until a full power run being started, the demand for water was greater and the leakage as well, so that the boiler was soon short of water, and the run was lost.

If no trouble is found in the feed-pipe, the difficulty may be sought in a very loosely fitting or badly worn water plunger. Such a plunger will discharge water into the air or even against a low boiler pressure, but may not be able to force it against the regular pressure in the boiler.

If from the evidence of the air cock and general behavior of the pump it is evident that no water is being handled, the suction pipe and plunger rod packing should be examined for air leaks.

Where there is some considerable lift from the hot well to the feed-pump, an unusually high temperature of feed-water, on account of the vapor formed, will sometimes prevent the pump from taking water. In good practice, of course, the hot-well is above, or at least not below the feed-pump, so that this difficulty is not likely to arise. If such should prove to be the trouble the feed-water must be cooled and the difficulty will be removed.

If none of these causes seem to explain the failure to draw or discharge water, then the trouble is probably to be found in the suction or discharge valves, and the necessary bonnets or covers must be removed to allow of their examination.

In any search for the cause of the trouble in the feed-pump, the details may, of course, be modified according to the circumstances, and the above suggestions are more especially intended to illustrate the principle that in such a search the trouble has often to be found by a continued elimination of one thing after another, taking those which are most readily examined, and thus localizing the difficulty as quickly and as readily as possible.

(3) *Check-Valve Jammed.*—If the feed-pump seems to be in proper condition except that it slows down and stops when outlets excepting a particular check-valve are closed, if furthermore the check-valve chamber is hot, there is no click, and the water does not rise in the glass, we may conclude that the check-valve is jammed on its seat. In former years this was sometimes due to unequal expansion of the valve and seat, and nipping of the former by the latter. In modern practice with good design and an angle of valve seat not steeper than 45 degrees, such an occurrence is very rare. The valve may also become jammed by the bending or other derangement of the stem or wing guides, or by the lodging of some foreign body within the chamber. If the trouble is due simply to unequal expansion the usual treatment is to wrap the valve chamber in waste or clothes, and then to pour on cool water, thus reducing the temperature. At the same time the chamber may be tapped near the valve seat with a hammer or bar. In any ordinary case of sticking due to unequal expansion the result will be to free the valve from its seat. If this does not avail, then the stop-valve between the check-valve and boiler must be closed and the check-valve cover removed, so as to allow an examinaton of the interior. In good, modern practice at least two check-valves, main and auxiliary, are provided for each boiler, so that there should be no danger of low water. If, however, only one check is provided, or if there is any question of shortness of water while the necessary repairs are being made, the stop-valve should be closed to stop the draft of steam and the usual precautions taken when stopping suddenly. (See [2] above).

(4) *Bursting of Water Gauge Glass.*—This occurrence is by no means uncommon, and under ordinary circumstances is of relatively small importance. With the type of gauge glass fitting having self-closing valves, as described in Sec. 17 [8], the flow of water and steam is automatically shut off and the new glass is readily put in. Without such provision the shut-off valves must be shielded from the discharge in the manner most readily effective, and then quickly closed. In setting a new gauge glass care should be taken to see that the fittings are well lined up, so that when screwed down there will be no bending strain on the glass. If any pronounced strain is thus set up on the glass by the fitting, it will be almost sure to break in a short time.

The packing rings should also be screwed down no tighter than barely sufficient to keep the joint tight. To make the closing of the supply valves as short a job as possible, in case it should become necessary, it may be recommended to open them only enough to insure a good supply of steam and water to the glass, rather than wide open. Two turns of the valve will be usually sufficient, and it is then much more quickly closed than when open five or six turns.

(5) *Low Water.*—The occurrence of low water or the absence of water from the gauge glass is one of the most serious emergencies which can arise in the fire-room. Immediate action is called for, and the most serious consequences may result from a mistake, or indeed may result in spite of whatever may be done.

If there is reason to believe that the water has but just disappeared from the glass and the lower gauge cock gives indications of water or of very moist steam, it may be fairly assumed that the level of the water is not below the tubes or combustion chamber tops, and in such case there will be no immediate danger of overheating or collapse, at least so far as the level of the water is concerned. The feed may therefore be put on strong without hesitation, and if the diagnosis has been correct the water will soon reappear in the glass and the incident is at an end. It may be considered prudent, however, to check at the same time the draft of steam from the boiler, and to deaden the fires to some extent by some of the methods referred to below. When the water reappears the boiler can then be put on its regular work as before.

In the more serious case when the location of the water is quite unknown and the gauge cocks and glass give no indications, there is some diversity of opinion as to the best procedure. The chief point of difference relates to the propriety of immediately putting on a strong feed. It has been claimed that if the plates were red hot, so much steam would be suddenly generated as to rapidly increase the pressure, and burst the boiler. On the other hand, it has been pointed out that the amount of steam which can be thus formed is in reality comparatively small, and that its formation cannot be instantaneous, nor even especially rapid, since its formation will extend over the period while the water is rising over the heated surfaces. In no way then could any great amount of steam be generated with especial rapidity, and it is hard to see how its formation could take place more rapidly than would be provided for by the natural outflow to the engine, and by the safety-valve, if need arose. To obtain some definite information on these points, experiments were carried out by a Steam Boiler Insurance Company some few years ago, in which the plan of putting in cold feed upon overheated plates was followed. The boiler could not be burst by the operation, and no very pronounced elevation of pressure was produced. So far as the results of these experiments went, it would seem to be safe to put on the feed immediately in such an emergency as we are now discussing. This conclusion seems also to be borne out by the results of such practical experience as is obtainable. There are, however, differences of opinion on this point, and while the author would follow this plan in such a case, it should not be denied that many good authorities would consider it unwise, at least in the earliest stages of the measures, to be taken.

Aside from getting the feed into the boiler as soon as prudence will permit, the other great point is to deaden the fire. If the heating surfaces have not collapsed when the condition is discovered, it is hardly likely that the plates can be at more than a very dull red, and if the supply of heat can be effectually checked, further trouble may be averted.

To this end a plan often followed, especially in former years, was to haul the fires. This, however, seems very unwise indeed. While being hauled the fires will burn up all the more fiercely, and for a few moments the heat supply will be increased rather than decreased. Instead of hauling directly, some engineers prefer to have the fires dumped into the ash-pits by dislodging a few grate-bars, and then to haul from there. This plan seems but little better than the other, and in any event the immediate result will be to increase the amount of heat given off at the very time when it should be decreased.

A far better plan seems to be to deaden the fire with either moist ashes or coal. If a pile of wet ashes is at hand they should be thrown immediately on the fire, and will be found a most effective means of deadening the burning coals. In default of wet ashes, wet or even dry coal may be thrown on and the fire simply smothered. At the same time the dampers should be put up and furnace doors left open, thus checking all draft and stopping the formation of heat.

As between these two operations, the deadening of the fire and the getting in of feed-water, the author is of the opinion that the deadening of the fire is of the more immediate importance, and should be attended to first, because it will produce the most immediate effect over the whole heating surface, and will serve most effectually to rapidly check the further heating of the plates. Putting in the feed water will then complete the cooling, but the direct operation is slower, and more local in its influence, and is therefore of relatively less importance.

In all such emergencies much, of course, will depend on the special circumstances, but if the two principles be kept in view that the fire must be deadened and the water restored, the details may be left to good engineering judgment to execute.

After the water has again appeared in the glass, and if no ill effects seem to have resulted to the boiler, the fire may be again gotten into condition and the boiler put on its regular routine. If, however, there be any doubt whatever regarding the possible results to the boiler, no chances should be taken, but the boiler should be disconnected from the others, the safety-valve raised and the steam blown down, while the fires should be hauled or allowed to die out and the boiler allowed to cool down. It should then be carefully examined for symptoms of distress or collapse, and if any such are found they must receive proper attention before the boiler is again set to work.

(6) *Collapse of Furnace Crowns or Combustion Chamber Plates.*—The collapse of a part of the heating surface is due either to an overheating of the plates or tubes, or to a gross error in the design. We may dismiss the latter as not liable to occur in good practice. Overheating may result from either low water, or from a coating of scale or of oil and scale on the water side. In the former case with no water to absorb the heat the natural result is an overheating of the plate until it becomes red hot, followed by its collapse and rupture. When the overheating is due to the presence of a coating of scale, the gradual bulge or start toward a collapse may result in cracking off the coating and in letting in the water to the plate. This will cool the metal, restore its strength, and thus put an end to the operation.

In some cases when the covering is a mixture of scale and oil, the overheating of the plate will result in burning off or volatilizing the oil, leaving the scale as a fine powdery deposit. This readily admits the water to the plate, and thus the metal may be cooled and restored in strength, and further bulging prevented. This nice adjustment of heating, burning or cracking off, cooling and re-strengthening before final rupture, does not, however, always occur. Before the re-cooling is effected rupture only too often results, and the contents of the boiler are more or less completely emptied into the fire-room, with consequences always severe and sometimes fatal.

If it is discovered that the furnace crowns have come down, but without final rupture, or if any portion of the heating surface has suffered collapse or bulging, but without final rupture, it may be assumed that the overheating was due to scale or oil, and that the change of form or the overheating has resulted in getting rid of the coating, and in readmitting the water to the plate. So that if rupture has not yet occurred, it is probable that the plate is safe for the time being, and in such case the fire may be first deadened and then hauled, the boiler shut off from the others and allowed to cool down, and then examined as to the nature and extent of the injury sustained.

(7) *Collapse and Rupture of Furnace Crowns or Combustion Chamber Plates.*—In the event of rupture following upon collapse, the chief thought after the safety of human life must be for the remaining boilers while the fire-room is in such condition that it cannot be entered.

The general nature of the steps which may be taken in such case are discussed in the following section, and while naturally judgment must be depended on for many details, it is readily seen that the two main points are as follows:

(1) To isolate the injured boiler.

(2) To safeguard the remaining boilers from injury due to low water.

(8) *Serious Leakage in Boiler Tubes.*—A serious leak may suddenly develop in one or more of the tubes. A split in the metal, a collapse due to overheating, or perforation due to deep pitting or general corrosion may give rise to such an occurrence. If the hole or holes are not too large the immediate consequence will not extend beyond a more or less complete filling of the fire side of the boiler with water and steam, and a more or less pronounced checking or deadening of the fire. In such case the feed should be looked after to see that the water does not get low in the boiler, while preparations are made for plugging the tubes. For this purpose a tube *stopper* or *plug* is used,

of which there are many varieties. A standard type of stopper consists of two heads or tapered plugs which make a joint within or against the ends of the tube, and are held in place by a rod running through the tube, threaded at the ends and provided with nuts for holding them up to their position against the ends of the tube. To fit this stopper it is necessary to enter the combustion chamber to adjust the back end, but once properly fitted, it may be depended upon to fulfil its purpose. There are also special forms of tube stoppers which may be inserted and adjusted from the front end only. It is more difficult to make a tight joint with the latter than with the former stopper, but they can be fitted without drawing the fire, and therefore in an emergency may prove of great value. Plugs of soft pine are also used for temporary purposes. These are pushed in from the front until they cover the leak, and the expansion due to soaking with hot water is depended on to make a tight joint. See also Sec. 42 [7].

If, however, the holes in the tubes are of considerable size, so great a quantity of water and steam may be liberated as to make it impossible to remain in the fire-room. In such case the fire will usually be put out as well, or at least so deadened that no further danger of collapse due to overheating need be feared, even should the water become low in the boiler. The general safety of the boiler itself is thus secured, but the other boilers, connected through the main steam pipe, will continue pouring out their steam through this leak, and as long as this condition continues it will thus remain impossible to return to the fire-room to attend to the other boilers. The first care after escaping from the fire-room, or before effecting escape, if possible, should be to close the stop valve which connects the boiler to the others, and thus to localize the trouble to the one boiler. If the stop valves are arranged so as to be worked from the deck above, as is frequent in good modern practice, this may be readily accomplished. If at the same time the safety valve on the injured boiler can be opened, the pressure will soon be blown down, and with good ventilation from the fire-room it will be possible in a short time to re-enter it and give the needed attention to the other boilers. In the meantime also, the engines may be slowed down so as to reduce the demand for steam.

Where the feed-pump is located in the engine room, it will be possible, if the checks have been left open on the other boilers and closed on the injured boiler, to feed by judgment at a rate which will keep these boilers safe from all danger of low water. The object of these various steps is, of course, to isolate the injured boiler and safeguard it from anything more serious, and to provide for the safety of those remaining; and while circumstances may alter the details of the measures which should be taken, the above suggestions will serve to illustrate the main points to be looked after.

When return to the fire-room is possible, and after the other boilers have received the necessary care, attention may be given to the injured boiler; the fires may be hauled, and after cooling down, the nature and extent of the damage investigated.

(9) *Rupture of Steam Pipe.*—In the case of a ruptured steam pipe the valves controlling the flow of steam to the point of rupture each way should be closed at the earliest possible moment, as the escape of steam will soon make it impossible to remain in the fire-room with safety. If it is arranged to work the stop-valves from the deck above, this is readily effected, and the trouble thus gotten under control. Self-closing valves are also often provided in modern practice, as referred to in Section 17 [3]. It is not possible, however, to so fit these that they will always act, or that they will shut off the steam in more than one direction. In one way or another, however, the first attempt must be to shut off the escape of steam. The next thought must be for the boilers, to safeguard those which may have been shut off from the engine from danger due to increase of pressure, and those which still remain connected to the engine from danger due to low water. The particular steps suitable for attaining these objects have been already sufficiently discussed, and no further mention will be here necessary.

(10) *Casualties With Water-Tube Boilers.*—With water-tube boilers the same general emergencies are liable to arise as with fire-tube boilers, and may be met in the same general way. It should be remembered, however, that, due to the small amount of water carried, the results due to shortness of water will come much more rapidly than with fire-tube boilers, and promptness in action is all the more necessary. In such boilers the tubes are most liable to suffer through shortness of water, and any considerable overheating is likely to result in their rupture. With the small tube type the most serious results of such an

accident are usually confined to the emptying of the contents of the boiler into the fire, and the effectual deadening or extinction of the latter. Here, as before, however, with more than one boiler the trouble must be localized by shutting off the boiler with the ruptured tubes from connection with the others. In other cases, however, with large tube types especially, or with the rupture of steam or water drums, the consequences may be more serious, resulting in driving every one from the fire-room, or even in loss of life. The same general principles apply here, however, as with fire-tube boilers, and good judgment must be depended on for the details suitable to the occasion.

Sec. 39. ENGINE-ROOM ROUTINE AND MANAGEMENT.

[1] Getting Under Way.

In the engine room the same as in the fire-room, a general inspection is first in order, more or less detailed and extensive, according to the time the machinery has been out of use, and the degree of acquaintance with its various features and peculiarities. If the engine has been laid up for any length of time a detailed examination will of course be required similar to that referred to at a later point in Sec. 43. We here assume, however, that no such general overhauling is needed, and that the machinery is to be supposed in a working condition.

A good general idea should first be obtained of the lead of the principal piping systems as noted in Sec. 25, and especially of the main and auxiliary steam and feed systems. These lines of piping should be looked over, the location of the valves noted, and where permissible the valves should be opened or closed to insure their being in working order, and then left in the condition desired for getting up steam.

The various parts of the main engine will be looked over so as to insure, so far as an external examination can, that everything is in proper working order.

The various auxiliaries will be looked over in the same way, and the results of this general examination being satisfactory, steps may be taken to test the various parts of the machinery under steam as soon as it is ready.

We have already in Sec. 38 pointed out the importance of trying the feed-pumps and getting them into working order as soon as possible, in order to insure the proper supply of feed-water to the boiler. Next in order may come the circulating pump, the engine of which is started at moderate speed, and the main injection and discharge valves opened. In starting all of this auxiliary machinery, proper precautions must, of course, be observed in regard to freeing the steam cylinders of condensed water by means of the relief valves, as noted in Sec. 38 [3] in connection with the feed-pump. The circulating pump being usually below the level of the water outside the ship, it naturally floods itself so that no trouble should be met with in getting it to take water. Assuming the air-pump independent, this may be started next and put at a moderate pace, or sufficient to maintain a vacuum of 15 to 20 inches.

The electric light engines, if not in operation from the donkey boiler, will also be looked after and started in due time, as well as any other auxiliary machinery whose operation may be required for getting under way.

In case the main engine since the last time used has been subjected to any adjustment or overhauling, it will be well to turn it completely over with the turning engine once or twice in order to make sure that everything is clear and in running condition.

In the meantime, while the auxiliaries are being gotten into operation, the steam will have been admitted to the jackets, if there are any, and to the cylinders through the main stop and throttle valves. Here, as noted in Sec. 38 [1], the object in view is to avoid any *sudden* change in the temperature or heat condition of the machinery. A good method of gradually warming up the engine is to just unseat the main stop and throttle valves, and then with the links in the ahead gear, say, to slowly turn the engine over ahead with the turning engine. This will allow the steam to work its way through the engine, warming up the entire series of cylinders and bringing them practically to their working temperatures. The water condensed must, of course, be allowed to escape by opening the relief and drain valves. During this period the reversing gear will also be warmed up and tried under steam until it works properly and throws the links smoothly from one side to the other.

When the engine has thus become well warmed up, the turning gear will be disconnected and locked out of gear, and immediate preparations made for turning over under steam. At this point the question of lubrication must be borne in mind, and while the regular schedule of oiling, etc., need not be started until the ship is fairly under way, still a moderate provision of oil may be made to the more important bearings, and if the engine has been out of use some little time, it will be well to work oil into the principal bearings during the preceding operation of the main engine with the turning gear as suggested above.

Before turning over under steam the deck officer should be notified in order that the hawsers securing the ship to the dock may be looked to if necessary, or the presence of anything about the stern which might foul or jam the propeller may be reported back to the engine room. Everything being in readiness, the main stop valve is opened slowly to full opening, and steam is turned on the reversing gear. Then the main throttle being still closed the links are thrown back and forth a few times, the passover or starting valves opened, and the throttle opened moderately. If the engine does not start off in one direction the links are thrown into the other gear, and if everything is in the proper condition the engine will start in one direction or the other after a few see-saws of the links. The hand relief valves are, of course, operated at the same time in order to aid in freeing the cylinders of any water which may collect there, or which may enter with the steam. Often the engine will move a little way, but the high pressure, or one of the other pistons will not pass the center. This is on account of the water in the cylinders, and is especially liable to occur if the engine has not been well warmed up, or if the steam pipe has not been properly drained. In such case the water must be worked out through the relief and drain valves, the links in the meantime being moved back and forth. In answer to this the piston will see-saw up and down, getting gradually nearer the center, and finally when the water is sufficiently cleared out, passing over and continuing the revolution. As the main engine is thus started the circulating pump and air pump, if independent, will be started up at the increased pace suitable to the amount of steam passing through the engine and into the condenser. After thus running for a few minutes, or until everything seems to be in proper running order, the engine is stopped, and the signal for the regular start is awaited. The object of thus turning over under steam is simply to make sure that everything is free and in working condition. Little of course, can be told regarding the adjustment of the various bearings, etc., or their liability to heat or pound. If the machinery is new or has undergone any considerable readjustment, or has been out of use a long time, it should have a dock trial of some considerable time, in order to determine the various points, and to bring out any defects liable to present themselves in the course of a continuous run.

Naturally the time when the ship is to start will be known to the engineer, and these various preparations will be so timed that soon after the final turning over under steam, the signal for the regular start may be expected.

PASSAGE OF STEAM THROUGH THE ENGINE.

In connection with the operation of a marine engine it will be instructive to note in order the names of the various parts through which the steam passes from the boiler until it returns again as feed water to its starting point. Starting, then, with its formation in the boiler, we have the following route for the case of an ordinary triple-expansion engine:

Dry-pipe—Safety valve chamber—Boiler stop-valve—Boiler steam-pipe—Main steam-pipe—Main stop-valve—Main throttle-valve—High pressure valve-chest—Steam ports and passages—High pressure cylinders—Steam passages and ports—Exhaust side of valve—Exhaust passage—Exhaust pipe to intermediate valve-chest and cylinder as above for the high pressure—Exhaust pipe to low pressure valve-chest and cylinder as above for the high pressure—Exhaust pipe to condenser—Condenser—Air-pump suction—Foot-valves—Air-pump chamber—Bucket-valves —Delivery-valves — Hot-well—Feed pump suction — Induction valves — Feed-pump barrel — Discharge valves—Feed-pipe—Check-valve—Boiler.

In addition a separator may appear between the boiler and the engine, and the filter and feed water heater between the hot-well and the boiler.

[2] Routine Operation.

In the routine operation of the main engine and of the other machinery in the engine room, the following are the points requiring chief consideration:

(a) The proper provision of oil and other lubricant in suitable quantities and at proper intervals, or continuously, according to the nature of the oiling gear in use.

(b) A constant watch over the general conditions of operation of the machinery in order that any symptom or sign of derangement or disturbance may be noted, and the proper steps taken for its control or removal.

The chief points relating to lubrication have been already discussed in Sec. 24 [12]. The watch over the general conditions extends to all features and depends to such an extent upon the special circumstances that only a few general hints can be given.

First, regarding the sounds which accompany the operation of the machinery and the part the ear may take in detecting symptoms of disturbance. The operation of the main engine and of the various parts of the machinery individually is accompanied by more or less plainly marked sounds or noise or combination of sounds. These in the end tend to combine themselves into a kind of resultant rumble, click, and rattle, which often remains quite constant in character, and so comes to have a kind of individuality of its own. To a person accustomed to the regular sounds of the engine room, the ear is often a delicate means of detecting any departure from the regular routine, and often the first indication of some disturbance will be furnished by a change in the character of the sounds produced. In particular, any unusual pound, jar, squeak or rattle should be located as soon as possible, and its cause investigated.

In some cases assistance in the detection or location of a pound or knock may be gained by the use of a convenient piece of metal, such as a spanner, or bit of pipe, one end being placed to the ear and the other against the cylinder, valve-chest, or other point nearest to where trouble is expected. At the same time too much reliance must not be placed on the ear to the neglect of other means of observation. In fact in the modern engine room all of the available senses keenly on the alert will be found none too many for the proper watch and care over the machinery in use.

The danger of heating, due to insufficient lubrication, poor adjustment or bad condition of bearing, is one which the ear will often aid in detecting, but the chief reliance must be placed on the sense of feeling and on the nose. With the necessary skill most of the important bearings may be felt by the hand. Caution must be used, however, so that the hand may not be caught or jammed. This part of an engineer's training is one which can be learned only by observation and cautious trial. If the heating of the bearing passes beyond a moderate elevation of temperature, the oil will become correspondingly heated and will give off a burnt odor, or perhaps will smoke freely, thus showing plainly the existence of trouble. The nose and eye will thus come in as factors in detecting trouble of this character.

Small steam leaks at joints, stuffing-boxes, etc., will make themselves plainly visible, and should receive such treatment as the circumstances may require, in order that they may be closed up. It must not be forgotten that every steam leak means a loss of both heat and fresh water.

The vacuum in the condenser will depend not only on the proper operation of the air-pump, but also on the reduction of all possible air leaks which might admit air to the low pressure cylinder during the exhaust, or to the steam side of the condenser. All such stuffing-boxes, joints, etc., must therefore receive careful attention, especially if the vacuum is not what it should be.

Water coming over into the cylinders from the boilers produces a crackling or snapping noise, which is readily recognized. The automatic relief valves may, of course, be depended on, or the hand gear may be operated to aid in removing the disturbing cause.

In stopping momentarily the throttle is closed and the links are run to mid-gear, no other change being made, and everything remaining ready to start again at an instant's notice. In stopping for a known period of time of any considerable duration, means should be taken to stop the flow of oil to the bearings by the closure of the feeding valves or the withdrawal of wicks, according to the means in use. If the stop is only temporary, and the engines are to be kept in readiness for starting again, the further steps taken will be only such as will serve to bring the machinery into its condition just previous to getting under way. That is, the circulating and air-pumps, so far as independent, may be slowed somewhat, while by the aid of the steam jackets, if fitted, and steam which is allowed to flow past the stop and throttle valves, the main engine is kept warmed up and ready for operation at short notice.

If the stop is to be of longer duration and the steam is to be shut off the engine, the stop and throttle valves will be closed, the air and circulating pumps shut down, and steam shut off the reversing engine, jackets, etc. The various drain valves and drips will be left open so as to free the machinery, as far as possible, of all the water formed by the gradual condensation of the steam. At the same time the flow of oil to the bearings will be shut off and such measures taken with respect to the oiling gear as the circumstances may require.

[3] Minor Emergencies and Troubles.

We will now consider briefly the steps to be taken in the event of the more commonly occurring troubles, some of which have been mentioned in the preceding paragraph.

(1) *Derangement in the Oiling Gear.*—In sight-feed apparatus this is readily detected, and without loss of time the trouble must be located and remedied, the oil in the meantime being supplied to the bearing in question by hand. The trouble in such cases usually arises from a clogging up of some of the pipes or passages, and as noted in Sec. 24 [12] all such pipes should be put up with union joints so that they may be readily taken down, cleaned and replaced.

(2) *Hot Bearing.*—This is one of the most important of the minor troubles which may arise in the engine room, and one which may lead to serious consequences in case the proper steps for its control are not taken in time. A hot bearing may arise from a variety of causes, among which the following are the more important.

(a) Lack of lubrication.

(b) Lubricant too thin so that it will not remain in place in the bearing and sustain the load.

(c) Improper adjustment, the amount of clearance between journal and brass being too small.

(d) Lack of alignment in the machinery, as a result of which the bearing is excessively severe on certain parts, thus forcing out the lubricant and causing the surface to nip and abrade.

(e) Bearing surface not of sufficient area to carry the load or take the work put upon it without an undue rise in temperature. This means, of course, either that the design is faulty or that the machinery is worked beyond the loads for which it was intended.

(f) Bearing surfaces rough and uneven, due to the poor workmanship, or as a result of serious heating on a previous occasion.

If the trouble is due to a lack of lubrication simply, and is discovered in time, an abundant supply of oil will be usually sufficient to control the condition and to gradually bring the bearing back to its normal temperature. If, however, the temperature rises considerably, the journal may expand more than the bearing brasses, so that the clearance will be decreased and the brasses will pinch the journal, thus introducing a further source of trouble as noted in (c) above. If this is not soon relieved, the metal surfaces will nip and the softer of the two will begin to abrade or "cut." This is always the bearing metal, and the resulting condition, in consequence of which the smoothness of the surface is destroyed, will tend simply to make matters still worse, to generate more heat, expand the parts still more, perhaps nip the surfaces still more tightly, and so cut the worse, until the bearing metal melts and runs out.

The treatment of a heated bearing involves two chief items, viz., the removal of the cause and the restoration of the bearing to its normal condition.

We may remove the cause entirely, of course, by stopping the engine, and in an advanced case of trouble, such as just described, this may be necessary. Otherwise we may reduce the cause by slowing down somewhat, and thus decreasing the amount of work thrown on the bearing. We may further decrease the cause by easing up the bearing cap and thus increasing the clearance between journal and bearing surface. This, however, can only be done to a slight extent, else trouble will be met with from too great clearance and the consequent pounding in the joint.

A plentiful supply of oil, or other lubricant, will also aid in decreasing the cause and in restoring matters to their proper condition.

A decrease in temperature will also usually aid in removing the cause, and is, furthermore, of course, one of the chief steps in bringing the bearing back to its normal condition.

To this end in the extreme case, it may be considered necessary to turn a stream of water on the bearing, thus to absorb and carry away the heat, and in many cases full power trials are run with streams of water playing for a considerable part of the time on various parts of the machinery in order to carry off the heat and so control the temperature. Water, however, is doubtless used far more than is absolutely necessary, and far more than good engineering would authorize. If sprayed or run from a hose on the bearings it is almost certain to find its way in and on to the bearing surfaces, where it will prevent action of the lubricant. For this reason its use once begun, it may be necessary to continue, simply because the lubricant cannot lubricate in the presence of water. In the best modern practice, as indicated in Sec. 21 [11], provision is made for circulating water through hollow bearing blocks, and thus in the most effective way the water is able to remove the heat generated without coming into contact with the bearing surface itself.

In case the machinery is not properly lined up, or the bearings are of insufficient area, or not in proper condition, only temporary relief can be looked for from the various means suggested above, the most effective of which will presumably be the operation of the machinery at a low or moderate power until such time as the needed readjustments, changes or repairs can be effected.

To sum up the treatment for a hot bearing, the measures taken may be selected according to judgment and the special circumstances from the following:

Lubrication.

Easing up bearing caps.

Slowing down and consequent reduction of load.

Application of water.

(3) *Pounding.*—This condition may arise from several causes, chief among which are the following:

(a) Bearings not in proper adjustment, too much clearance being allowed between journal and bearing metal. (See Sec. 24 [12].)

(b) Lubricant too thin and thus unable to retain its place in the bearing.

(c) Valve events not properly adjusted, especially the exhaust closure and following compression.

Furthermore, an engine will often show at different speeds a marked difference in this respect, such difference being chiefly due to the increasing effect of the inertia forces with increase in the revolutions.

If the trouble arises from the nature of the lubricant in use a change to a heavier oil may show an improvement. If, however, as is more commonly the case, it is due to faulty adjustment, either of bearing or valve gear, or both, but little can be done while the engine is in operation, and the first opportunity for overhauling and readjustment must be taken for a study of the conditions, both as regards the bearings themselves, and the possibility of improvement by an adjustment of the compression. If the pounding becomes very severe, it may become necessary to slow down the engine and operate under less than the regular or full power until the proper examination and readjustment can be made.

(4) *Priming or Lifting Water.*—This emergency has been more particularly referred to in Sec. 38 [3]. In small quantities water produces a crackling or snapping sound in the cylinders, and the automatic relief valves may be allowed to take care of the situation, of if desired, the hand reliefs may be operated as well. If, however, the water comes over in large quantities the engine will slow down and work with an irregular and labored motion, which may be readily recognized as denoting this condition. In such case the throttle or main stop valve should be partially closed and the water gotten rid of as quickly as possible by the use of the relief valves. The engine will then operate at the reduced speed permitted by the partially closed valve, presumably without further trouble. If on opening out again the tendency to lift water at full or ordinary power is persistent, the power must be reduced until the trouble is removed and the engine will operate continuously without disturbance of this character.

(5) *Vacuum Falls and Becomes Poor While the Condenser Becomes Hot.*—Following are the chief causes which may lead to such a condition:

(a) Insufficient condensing water from any cause.

(b) Division plate in condenser head carried away so that water goes directly from inflow to outflow without going through the tubes.

(c) Excessive inflow of steam caused by leakage either past low pressure piston or slide valve, or possibly in the "bleeder" of "silent blow" if such is fitted.

(6) *Vacuum Falls and the Condenser Remains Cool.*—In such case the indications are that this condition is due to the presence of air not removed by the air pump, as may be caused by any one or a combination of any of the following:

(a) Air-pump valves defective.

(b) Leak in the condenser, either at the head joints or through a crack.

(c) Soda or drain cocks open or leaking.

(d) Low pressure piston rod stuffing box leaking air inward during exhaust stroke.

(e) With "inside" piston valves on the low pressure cylinder, leaky valve stem stuffing boxes.

(f) Leak or obstruction in the pipe leading to the vacuum gauge.

Sec. 40. BOILER CORROSION.

No sooner has a boiler been completed than the various corrosive and destroying influences with which it is surrounded set to work on its destruction. We may conveniently consider corrosion as of two kinds, that due to oxygen and that due to an acid. These two are, however, by no means independent, and are often combined in very complex ways. The process by which oxygen combines with another substance is called *oxidation,* and the product of the operation an *oxide.* In the case of iron and steel the typical product is the ordinary red iron rust, or ferric oxide (Fe_2O_3), consisting of about 56 parts by weight of iron and 24 of oxygen. In order that oxidation or rusting of iron may continuously proceed at ordinary temperatures, however, it is not enough that oxygen and iron shall be in contact. It requires the additional presence of moisture and carbon dioxide (CO_2), small proportions of which are always present in the atmosphere. Oxygen and moisture alone act feebly and very slowly on iron, but when the four substances, iron, oxygen, moisture, and carbon dioxide, are all present together, the operation of rusting proceeds continuously and with vigor. Oxide is first formed, and this is reduced by the carbon dioxide to a carbonate, and this in turn breaks up, forming hydrated oxide ($FeHO_2$), setting free the carbon dioxide to continue the process. The hydrated oxide thus formed is furthermore electro-chemically negative to iron, and thus helps on the operation as explained at a later point. If either the moisture or the carbon dioxide is absent the oxygen will have little or no effect, and the iron will be protected. This is shown by the non-rusting of iron in perfectly dry air, even though there may be some carbon dioxide present; or again, by its preservation in a weak alkaline liquid, as lime water, in which there can be no free carbon dioxide. The piano wire used in certain forms of deep sea sounding apparatus, for example, is thus kept from corrosion under conditions which would naturally soon destroy its regularity and value for the purpose used.

Acid corrosion means the attacking of a substance by an acid, the breaking up of the latter, and the formation of a new substance known as a *salt*, and composed of a part of the acid and of the substance attacked. Thus hydrochloric or *muriatic* acid (HCl), as it is commonly called, is sometimes present in boilers. This is composed of hydrogen and chlorine. When it is brought into the presence of iron or steel the chlorine leaves the acid, and joining with the iron, forms a salt known as ferrous chloride, or chloride of iron ($FeCl_2$). With iron rust and muriatic acid the result would be similar, the chlorine would join with the iron and form ferrous chloride, while the hydrogen of the acid would join with the oxygen of the oxide and form water.

As before stated, acid corrosion and oxidation are very commonly both present, especially in the latter operation, and in fact the continued progress of oxidation with iron, moisture and carbon dioxide is dependent on the combined action of both operations. We shall not, however, deal further with the chemical details of corrosion in general, but proceed to a brief consideration of the causes, effects and remedies as related to corrosion in marine boilers.

Taking first the exterior of boilers and of all exposed iron and steel work in general, it is clear that the conditions for continued rusting are all present on board ship. The air is moist and there is likely to be present carbon dioxide in abundance. The only safe protection is, therefore, a covering which shall keep the air, moisture and carbon dioxide from contact with the iron. To this end metal paint or other equivalent coating is used wherever possible. Many small fittings, especially about the deck, are of galvanized iron, that is, iron covered with a thin coating of zinc. The latter metal is but slightly affected by the process of oxidation, and it, therefore, forms an efficient protection for the iron. Brass, bronze and copper are also oxidized but slightly, and the oxide formed serves as a protective covering to the metal underneath. For this reason, among others, many of the fittings about boilers and elsewhere are, as we have already seen, made of these metals.

Passing in now to the fire side of the boiler, we find the application of paint or other protective coating impracticable.

Here we must depend on the heat, which will so dry the air that it is no longer moist. That is, while water vapor may still be present in the air, there is so little compared with the amount the air could naturally contain at that temperature, that it is held by the air and is no longer free to enter as a factor into the operation of oxidation. Rusting in the usual way is, therefore, very much retarded or prevented. To this fact we owe the general preservation of the furnaces, ash-pits, etc., from serious and continued corrosion. We here, however, run into another danger in the extreme case when oxygen is present in excess, and both the oxygen and iron are very hot. The oxygen in such cases enters more readily into union with the iron, and if the temperatures should be sufficient, a different kind of oxide is formed, the black, or magnetic oxide (Fe_3O_4), the same as the mill scale or forge scale, which forms when iron is worked at a red heat. The oxide thus formed may presumably be swept away by the scouring action of the draft, thus exposing a fresh surface to renewed attack. The back ends of the tubes seem especially liable to attack in this way, and particularly with hard forced draft. The cure for this trouble is found in the use of cast iron ferrules, as previously described.

These ferrules protect the tube ends from the extremes of temperature, and also provide something for the hot oxygen to attack, while they are readily renewed.

Turning now to the water side of the boiler, we find more serious trouble than with the fire side. There is likely to be more or less air in the feed water, either entering with the make-up feed, or occasionally drawn into the feed-pump and sent on to the boiler. There may also be free carbon dioxide liberated from the salts entering with the make up feed, and thus all the conditions for continuous rusting may be present. Even if free carbon dioxide is not present the formation of iron oxide, combined with electro-chemical reactions, as referred to later, may result in serious local corrosion. Furthermore, as the feed-water is heated the air is liberated, and the oxygen just at the instant of liberation seems to be especially active chemically, and is thus all the more likely to attack exposed places than if allowed to remain in solution in the water, as at ordinary temperatures.

Turning next to acid corrosion, mention may first be made of the serious trouble formerly experienced from the use of animal and vegetable oils for cylinder lubrication. Such an oil is a compound of a fatty acid and glycerine. When exposed to a high temperature the fatty acid and the glycerine become separated. If a substance such as soda or potash is present, the fatty acid combines with it and forms soap. This process is called saponification. If, however, no such substance is present the acid will be free to attack other substances as it may be able. Fatty acids attack iron feebly, but if long continued the result may be a serious corrosion, resulting in the formation of what is known as an iron soap. The temperature within the cylinders and boilers was quite sufficient to thus decompose the oil, and there would, under such circumstances, be set free in the boilers an amount of fatty acid depending on the amount of oil used in the cylinders and finding its way into the condenser and feed-water. There were thus present all the conditions necessary for the corrosion of the interior of boilers by fatty acids, and many serious cases were laid, in part at least, to this cause. These troubles appeared especially with the introduction of the surface condenser, and the part which fatty acids might play being understood, the use of animal and vegetable oils for the lubrication of the cylinders was abandoned, and in their place hydrocarbon or mineral oils are now used. Such oils are derived as one of the constituents of crude petroleum, and are not compounds of a fatty acid and glycerine. They are compounds of carbon and hydrogen, and belong to an entirely different class of chemical substances. They do not produce a fatty acid on being heated, and cannot, at least directly, take part in the process of boiler corrosion.

In modern practice, therefore, nothing but the best hydrocarbon oil, entirely free from animal or vegetable admixture, should be used for cylinder lubrication. With lubricant of this character modern boilers should be free from corrosion chargeable to the action of fatty acids.

These are, however, not the only acids which have given trouble in the way of boiler corrosion. Under certain circumstances free hydrochloric or muriatic acid is found in boilers. This is presumably due to the breaking up of magnesium chloride, forming hydrochloric acid and magnesium hydrate. The most dangerous feature of the corrosion due to hydrochloric acid is that under conditions which may exist within steam boilers the chloride of iron first formed may become broken up, giving rise to other neutral compounds of iron, and setting free the acid to continue its ravages.

There are also possibilities of the development of nitric acid from the organic matter which in small quantities may occasionally find its way into steam boilers.

Except as it may be modified by electro-chemical action, the presence of such an acid usually results in a general surface corrosion, at least of all surfaces not protected by a sufficient layer of lime scale.

The most troublesome feature of boiler corrosion has not been, however, a general or more or less uniformly distributed effect, such as would naturally be charged to the action of an acid diffused throughout the boiler. It has been rather in the so-called *pitting*. This term refers to the formation of small pits or depressions from the size of a pin head upward, and conical or cup-shaped in form. The depth of such pits may be anything from a slight depression to a hole cut entirely through a boiler tube. They are found in no fixed locality, though more commonly on the tubes, furnaces, and combustion chambers than elsewhere. When found they are usually filled with a blackish or brownish pasty mass, consisting chiefly of iron oxide with a slight admixture of lime salts, oily matter, and other substances. This deposit within the pits is often covered with a skin of somewhat different composition, consisting of lime salts and iron oxide in more nearly equal proportions.

To account for the formation of these pits, various explanations have been suggested, most of them involving *electro-chemical* action as a more or less pronounced feature. To understand the nature of this action a few explanations must first be given.

Nearly all substances are in a different electrical condition, or at a different electrical *potential*, as it is called. This difference is found not only between substances of different kinds, but also between similar substances at different temperatures, or in different physical conditions, as, for example, between two pieces of iron or steel, one of which has been hammered or worked more than the other. Due to this difference of electrical potential there is a tendency to set up a flow of electricity from one to the other, and as a further result to so change the two substances as to bring them into electrical equilibrium. In other words, the result of such a difference of electrical condition is always to bring about changes which will cause the difference to disappear, and so bring the two substances into equilibrium. These chemical changes of the two substances, which tend toward electrical equilibrium, may be much helped or hindered by the medium in which the bodies are immersed. If they are in dry air, for example, no such activity takes place, and the difference of electrical condition continues unchanged. If, however, they are immersed in water, or especially in salt or slightly acid water, the operation will usually be much assisted by the activity of the medium for the substances. It may also happen that the medium and substances are so related as to bring about a series of chemical changes, of which the first are those which would naturally be associated with the transfer of electricity and the development of equilibrium, while the second counteract these changes chemically, and bring the substances back to their original condition, and so keep them constantly in the condition of electrical difference. There is as constantly the attempt to restore equilibrium, and hence so long as these conditions continue there will result this continued series of chemical actions, accompanied by a constant flow of electricity from one substance to the other. In order, however, that this flow of electricity may be thus constant and so constitute a *current* of electricity, as it is termed, there must be a path for a complete circuit or flow in one direction through the medium which produces the chemical changes, and in the other direction outside of this medium. The substance from which the current flows in the medium is known as the electro-positive element, and the other the electro-negative. The chemical activity proceeds and the current is formed, in general, at the expense of the electro-positive element.

These operations are illustrated in the ordinary voltaic cell or battery, such as those used for ringing bells, etc. In most of these batteries, however, the action is not self-sustaining, and if allowed to continue for a little time, a condition of electrical equilibrium is reached, or, as ordinarily stated, the battery is run down. In others used for telegraphy and other purposes the operations are self-sustaining and continuous until the chemical substances are exhausted.

In a boiler these conditions for a more or less continued electro-chemical action may be fulfilled in a variety of ways. Parts of the structure of widely differing temperatures or of different physical or chemical compositions may provide the elements in a different electrical condition. Still more likely is such a difference to be found between iron and its oxides, especially the magnetic oxide or mill scale ($Fe_3 O_4$), or between a particle of carbon in the steel and the surrounding metal, or between a place in the steel where the proportion of carbon is much greater than the average and the surrounding metal, or between a bit of slag or other impurity in wrought iron and the surrounding metal. Copper, either in the form of oxide, or especially in the metallic form, would also supply a substance differing strongly from the iron. The exciting liquid is the water in the boiler, and its action will be more vigorous according as it is more acid in reaction, higher in temperature, and denser in concentration. With a high pressure boiler, water of high density and quite acid in character, and with the usual lack of homogenity or uniformity in the structure of the boiler, we should, therefore, expect the effects of electro-chemical action to be shown in marked degree. It happens, furthermore, that iron is electro-positive to copper, to carbon, and to its own oxides, so that in all cases likely to occur the operation will proceed at the expense of the iron.

From the very nature of these electro-chemical actions their effects are necessarily local in character, and so far as understood they seem to provide a fairly good explanation of the formation of pits as already described. It is not unlikely, however, that in some cases they are due rather to simple chemical action, and that their localization to a small spot is due to special or accidental causes, such as the protection of the surrounding metal by lime scale, or a peculiar weakness against chemical attack at that point, due to peculiarities in chemical or physical structure.

The possibility of deposits of copper on boiler surfaces has been already mentioned. These were first noted in connection with the corrosion accompanying the general introduction of the surface condenser. It was believed that the copper of the condenser tubes was attacked by the pure water resulting from the condensation of the steam, or by the fatty acids formed as above explained, and was then carried over into the boiler and deposited on the surfaces. To prevent such action the condenser tubes were tinned, thus covering the copper from the action of the water or the fatty acids. Neither this step nor the substitution of hydrocarbon oil for that containing fatty acids has made any very marked difference in boiler pitting, and at the most the presence of the copper can have been only one among a number of causes as suggested.

There has been much difference of opinion and difference in experience regarding the question whether wrought iron or steel boiler tubes were the more liable to corrosion. It was pointed out that wrought iron was less homogeneous than steel, and therefore the latter should be the better. The early experience with steel hardly bore out this claim, and in fact the general opinion seems to have been that wrought iron tubes were found to corrode less readily than steel. In explanation of this, it may be said that while wrought iron was less homogeneous physically, the steel was perhaps less homogeneous chemically, and in any event contained a larger proportion of carbon than the iron, so that it would by no means follow that it would necessarily be less subject to electro-chemical action. The latest and best products of the steel makers for such purposes, however, are extraordinarily low in carbon and very homogeneous, and experience with such grades of material seems to show them superior to wrought iron in this respect.

We have thus developed in some detail the causes of corrosion on the water side of steam boilers, so far as they are understood. For the prevention of such effects their causes must be removed or counteracted.

For reducing the amount of oxidation and the possible results due to electro-chemical action, the presence of air in the feed water must be avoided by preventing as far as possible the entrance of water from overboard into the feed. The hot-well or feed-tank should also be of good size and kept full, so that there may be no danger of its getting too low from time to time, and thus allowing the pump to take air. The piston rod on the low-pressure cylinder should be kept well packed, so as to prevent the entrance of air during the exhaust part of the stroke. The feed pump rods on the water end should be kept well packed for the same reason.

To prevent acid corrosion the formation of the acids must be prevented as far as possible, and such as may form must be neutralized within the boiler. The prevention of the formation of fatty acids has been considered above. We have also seen that the formation of other acids is due chiefly to the presence of salts contained in sea water, or to organic substances. We have therefore simply an additional reason for keeping all such substances out of the boiler as far as possible. To neutralize such acid as may form, bicarbonate of soda, or soda-ash, as it is known in the trade, may be used from time to time, and in such quantities as may be found necessary. To test the water for acidity the litmus test is used. Blue litmus paper turns red when dipped in water slightly acid, while if the water is alkaline it remains blue, or the red color caused by an acid is changed to blue. By this means the condition of the water may be tested from time to time and soda used accordingly. Care must be taken not to use it in too great excess, as it may cause foaming. The soda is introduced by means of a soda cock on the condenser. Instead of keeping the water alkaline by the use of soda, dependence is often placed on the zinc slabs used to prevent electro-chemical corrosion. These are gradually dissolved, forming zinc chloride, and this will undoubtedly tend to neutralize free acids and to keep the water alkaline. Whether sufficient or not, can of course be readily determined by the litmus test before referred to.

For the prevention of electro-chemical action the causes must also be removed or neutralized as far as possible. This cannot be realized entirely, but it is clear that the results will be the better, as the following conditions are the more nearly fulfilled:

(1) The structure of the boiler should be of material as homogeneous as possible in its chemical constitution and physical condition.

(2) Causes liable to produce oxidation or the presence of foreign substances should be kept out of the boiler as far as possible.

(3) The water in the boiler should be made as nearly neutral or non-exciting relative to the iron as possible. This in a general way will be attained by keeping it slightly alkaline rather than acid, and by avoiding very high densities.

In addition to these means for reducing the causes, there remains one further step, and that is:

(4) The provision of a substance which shall be electropositive to iron, and readily attacked, so that the activity will be diverted from the iron to the protecting substance, and the operation will proceed at the expense of the latter rather than

of the former. Such a substance we find in zinc, and its use for this purpose is very general and seemingly beneficial.

It may be also noted that the formation of zinc chloride as referred to in the foregoing will aid in keeping the water alkaline in reaction, thus reducing its natural activity, and contributing further to the general decrease of electro-chemical action.

In order to be effective as a protection to the iron in the manner described, the zinc must be in actual metallic contact with the structure of the boiler. It is usually in the form of rolled or cast slabs, weighing 8 to 12 lbs. each. These are often placed in perforated sheet metal baskets hung from the stays or attached to other portions of the boiler. Where the basket is attached to the boiler there should be *bright metal* contact, and the attachment should be by screwed joint or other equivalent means, so that the separation of the two surfaces by the formation of scale or corrosion between them may be prevented. The zinc should also be connected to the basket by through bolts or other means which will insure continuous metallic contact. In some cases the zincs are hung by a through bolt without other means of support. In such case, as the zinc becomes used it may fall apart and the pieces may lodge where they will obstruct the circulation, or be otherwise undesirable. In any event, they will no longer protect the part of the boiler confided to their care, and their period of usefulness may therefore be less than when supported and connected to a basket, as described above. The number of zincs fitted varies greatly, according to the judgment of different engineers. In some cases not more than 10 or 12 would be assigned to the protection of a large double-end boiler, while in others as many as 40 or 50 would be used. The latter number is the better representative of good modern practice. In any case they should be distributed as nearly uniformly as possible throughout the boiler, in order that the latter may be thus subdivided into parts, each more especially under the influence of a given slab.

In connection with the use of zinc it may be noted that for such boilers as may be used for distilling purposes, that is, for the provision of fresh water for drinking and cooking, the zincs should be omitted, as the presence of any considerable amount of zinc chloride will render the water unsuitable for such uses.

Instead of depending on zinc to prevent or divert electro-chemical action, as above described, some engineers prefer to depend simply on reducing the activity of the water by keeping it alkaline by the use of soda, introduced as the litmus test may show to be necessary.

When spots are found in a boiler, showing the presence of pronounced corrosion, they should be cleaned off thoroughly, washed with soda solution, and, if not on a heating surface, covered with a thin wash of Portland cement. This will attach itself to the iron and protect it in a manner similar to the lime scale.

The beneficial effect of scale in thus protecting the surfaces of boilers from corrosion is well recognized, and there is no doubt that its presence as a thin wash or layer is of great value. In order to be effective, however, it must be so firmly and closely attached to the iron as to prevent contact of the water with the surface, else the corrosive action may proceed under the scale and result all the more seriously because it is protected from inspection until the scale is thoroughly removed. On the tubes and other heating surfaces of the boiler, with their changes of temperature and consequent expansions and contractions, the scale is especially liable to be cracked off or partially separated from the iron, with possible results, as here noted. This is still more likely to be the case as the scale becomes thicker and the metal more liable to become overheated. It has also been suggested that a very heavy scale may result in an overheating of the metal sufficient to decompose the moisture present, thus liberating oxygen and forming the magnetic oxide of iron or black mill scale (Fe_3O_4). This is highly electro-negative to iron, and thus it may give rise to harmful electro-chemical reactions.

Laying Up Boilers.—When boilers are to be laid up, the principles already explained will indicate the nature of the means suitable for preventing corrosion.

On the outside, paint or other like coating may be used, as already noted. On the fire side of water-tube boilers protection is sometimes gained by building a slow fire of tar or resinous material, the tarry smoke from which condenses on the tubes and furnishes protection from the air with its moisture and carbon dioxide. Use is also made of quicklime in trays renewed from time to time. This absorbs the moisture and so keeps the air dry.

On the inside, all boilers when laid up should be either empty or entirely full. If a boiler stands for any considerable length of time partly full, corrosion is likely to occur about the water line. If they are to be out of use for a short time only, they may be filled full of water made slightly alkaline by the addition of soda, the condition of the water being determined by the litmus test already referred to. If they are to be laid up for a longer time it is better to lay them up dry. To this end the water is removed, the manhole-plates taken off and the interior thoroughly dried out by the introduction of trays of burning charcoal or coke. The boiler is then closed up, except a lower manhole, through which a tray of freshly burning charcoal is introduced, and the manhole cover is put on. The charcoal will consume most of the remaining oxygen, and the boiler will thus be protected. Instead of the final introduction of a tray of charcoal, trays of quicklime may be used to insure the absence of all moisture, and the boiler then closed as before.

It is readily seen that these various methods are simply ways of carrying out the necessary conditions for preventing oxidation, as already discussed, and if these principles are kept clearly in view the means most conveniently at hand may be suitably adapted to provide the protection desired.

Sec. 41. BOILER SCALE.

It is well known that sea water contains in solution a certain amount of solid matter, while even ordinary fresh water is not wholly free from similar substances. As long as the water remains in its natural condition these solids remain in solution; but under the change of condition to which the water in a steam boiler is subjected, they are liable, as explained later in detail, to separate out from the water and thus to form scale or sludge, according as the circumstances may determine.

The proportion of the solid matter in ordinary sea water is about (by weight) 1 part in 32, or 1-32. This is the same as about 5 oz. per gallon, or 2 lbs. per cu. ft. The solid matter consists chiefly of chloride of sodium or common salt, with small quantities of calcium sulphate and carbonate, magnesium sulphate and chloride, with smaller quantities of other substances. An average composition of this solid matter is about as follows:

Chloride of Sodium (common salt)	76	per cent.
Chloride of Magnesium	10	"
Sulphate of Magnesium	6	"
Sulphate of Calcium (gypsum)	5	"

The remaining 3 per cent consists of small quantities of other salts with a little organic matter.

The proportion of the solid matter in river and lake water is quite variable with the locality, and no representative or average analysis can be given. The amount held in solution may vary from perhaps 10 to 250 parts in 100,000 or from .015 oz. to .30 oz. per gallon, or .1 oz. to 2.5 oz. per cu. ft. It is composed chiefly of the carbonates of calcium and magnesium with smaller quantities of the sulphates of calcium and magnesium, and other substances. In addition to the substances in solution, quantities of sand, mud, organic matter, etc., may be carried in suspension, dependent entirely on the locality and special circumstances.

Boiler scale from sea water is composed chiefly of calcium sulphate or sulphate of lime, as it is commonly called, while that from fresh river or lake water is composed chiefly of calcium carbonate or carbonate of lime, as commonly called. With brackish water, as we might expect, the proportions of the two are more nearly the same. Following are analyses of boiler scale by Professor Lewes which may be considered as typical of the incrustations formed by river water, brackish water and sea water, respectively:

CONSTITUENTS.	RIVER.	BRACKISH.	SEA.
Calcium Carbonate	75.85	43.65	0.97
Calcium Sulphate	3.68	34.78	85.53
Magnesium Hydrate	2.56	4.34	3.39
Sodium Chloride	0.45	0.56	2.79
Silica	7.66	7.52	1.10
Oxides of Iron and Alumina	2.96	3.44	0.32
Organic Matter	3.64	1.55	trace
Moisture	3.20	4.16	5.90
	100.00	100.00	100.00

It thus appears that scale from river water may be looked on as an impure calcium carbonate, that from sea water as an impure calcium sulphate, while that from brackish water, as we should expect, is a mixture of the two in more nearly equal proportions.

Sodium chloride or common salt is soluble in water until the proportion exceeds some 25 or 30 per cent. This corresponds to a density of 8 or 10 on the usual hydrometer, and is far greater than that reached by the water in marine boilers. This substance therefore gives no trouble so far as helping to form scale is concerned, and the small amount found in analysis of boiler scale is probably due to the shutting in, so to speak, of a small amount of water during the formation of the scale. In discussing the formation of boiler scale for our present purposes, it will be sufficient to refer to the behavior of the salts of calcium and magnesium.

Calcium carbonate ($CaCO_3$) is practically insoluble in water, while calcium bicarbonate ($CaC_2 O_5$) is quite soluble, and it is in this form that the substance exists in solution in water. If now the water is heated to the boiling point carbonic acid (CO_2) is driven away from the bicarbonate, it becomes reduced to the simple carbonate, and being now insoluble it separates out as a more or less powdery deposit. Mixed with other salts, however, especially calcium sulphate, or if there is a little sulphuric acid in the water, it may collect on the heating surfaces and form a hard and closely adhering scale. Magnesium bicarbonate is in a similar manner reduced to the simple carbonate, which is insoluble, and is then deposited in the same fashion.

Calcium sulphate is soluble in cold water to a slight extent, as found in sea water. As the water is heated, however, or as the density becomes greater, the proportion of sulphate which it can retain in solution becomes less and less. When the temperature rises to 280° or 290° (corresponding to from 35 to 45 lbs. gauge pressure) the water can no longer retain any of the sulphate in solution, and it is all deposited. It is also largely deposited, even at a temperature of 212°, if the density rises to 3-32 or above. The other sulphates become likewise insoluble and are completely deposited if the temperature rises to about 350° or over, corresponding to about 120 lbs. gauge pressure. These sulphates of lime and magnesium thus deposited tend to attach themselves to the surfaces within the boiler, and to form a very hard and crystalline scale.

As to the effects of this scale, its presence in a *very thin* layer is often considered beneficial as a protection to the surface of the boiler from corrosive influences. On the other hand, however, it is a much poorer conductor of heat than metal, and its presence on the heating surfaces retards the transmission of heat from the fire through to the water. In the extreme case the heat may be so effectually shut off from the water that it simply becomes banked up, so to speak, in the metal, and in this way the tubes and other heating surfaces may become seriously overheated with resulting damage to the boiler. The scale may also in extreme cases become so collected between the tubes or between the combustion chamber and boiler sheets as to impede the circulation of the water and thus lead to overheating and its dangers, as referred to above. In water-tube boilers the accumulation of scale on the inside of the heating tubes is of special danger, as the circulation becomes in such case rapidly obstructed and the danger of overheating and rupture is correspondingly increased. In a similar manner the accumulation of scale in the interior of tubular feed-water heaters rapidly decreases their efficiency as heaters, if no worse results follow due to the burning out of coils, or to the resulting shortness of water in the boilers.

Scale Prevention, Fresh Water.—The only sure way of preventing scale is simply to keep it out of the boiler. If the scale-forming substances find entrance to the boiler it will be found very difficult to prevent its formation, at least to some extent.

On boats navigating inland waters the jet condenser is still for the most part used, the feed is ordinarily taken from the condenser, and therefore practically from overboard. In such boilers, therefore, we may expect the formation of the usual fresh water scale, consisting chiefly of calcium carbonate. For the treatment of fresh water scale a great variety of methods have been proposed. In some cases the substances proposed act chemically, in others mechanically. From the great variation in the character of the solid matter contained in fresh water, it can hardly be expected that any one method of treatment or substance will prove equally good in all cases.

If a feed-water heater is used, and is effective in heating the water, it will be found that most of the scale will be deposited in the heater, especially if it is of sufficient size to allow of proper time. In this way the scale may be kept out of the boiler proper. The heater, however, should be so made as to readily admit of cleaning, especially if the water contains any considerable proportion of scale-forming salts, otherwise it will soon become choked and ineffective.

Among the various substances which have been recommended for the prevention of fresh water scale the following may be mentioned:

Oak and hemlock bark and other like substances which contain tannic acid are more or less effective in waters containing carbonates of calcium or magnesium. The tannic acid, however, will attack the iron of the boiler and may lead to serious corrosion.

Molasses, cane juice, fruits, distillery slops, vinegar and other like substances containing acetic acid have also been used with success where no sulphates are present. The acetic acid, however, is still more injurious to the iron than the tannic acid, and the organic substances will form a scale with sulphates if they are present.

Sal-ammoniac when used with a feed water containing calcium carbonate brings about an exchange between the two substances as a result of which ammonium carbonate and calcium chloride are formed. The former of these is soluble and quite volatile and passes off mostly with the steam. The latter is quite soluble and thus the deposition of the calcium carbonate is avoided. This operation by itself, however, would result in the gradual accumulation of calcium chloride in the boiler, thus raising the density of the water to a point where ultimately it would begin to deposit. This condition may, of course, be controlled by a suitable use of the blow.

Tanate of soda is well recommended for general use, but with water containing sulphates a small amount of soda-ash should be added.

Among the substances which act mechanically, crude petroleum and kerosene oils are probably the most widely used. The latter may be recommended as the better of the two, as the crude oil will sometimes aid in scale formation. They seem to act best in cases where there are some sulphates present, as in slightly brackish water, or in the waters of certain geographical regions. Kerosene seems to act by preventing the particles of scale from sticking closely together or from tightly adhering to the heating surfaces, so that much of the matter will collect as a sludge in the bottom of the boiler, and that on the heating surfaces will be more easily removed.

In all cases where there is reason to expect the accumulation in the bottom of the boiler of deposits thrown down in a loose or powdery form, the bottom blow should be freely used so as to prevent the accumulation of too great a quantity, or opportunity for its hardening into scale.

In spite of all modes of treatment there will be found some scale on the heating surfaces, and provision must be made for entering the boiler and removing it with appropriate tools as the occasion demands and circumstances permit.

In many of our inland waters the amount of scale-forming substances is so small that no special treatment is thought necessary, and little attention is paid to the matter except to remove the accumulation at the periods of regular inspection and overhaul.

Scale Prevention, Salt Water.—Turning now to boilers in which sea water may form a portion of the feed, it will be of interest to first note briefly the historical development of the modern situation.

In the early days of marine engineering, the temperature and pressure of the steam were low, and the jet condenser was in general use. The feed water which was drawn from the mingled condensing water and condensed steam was but slightly fresher than sea water, so that large amounts of solid matter were thus fed into the boiler. In consequence the density would have risen rapidly had it not been kept down by blowing off a part of the water in the boiler of relatively high density and replacing it with the salt feed of lower density. Had the sulphates of calcium and magnesium thus brought into the boiler been completely deposited, enormous quantities of scale would have been formed, and this method of operation would have been quite impracticable. Due, however, to the moderate pressure then in use and to the fact that the density was kept usually between 1 3-4 and 2, the salts were held fairly well in solution, and but a moderate amount of scale was deposited.

As steam pressures advanced, however, beyond 40 or 45 lbs., conditions were reached under which first the calcium sulphate and later magnesium sulphate and other salts are completely deposited. Under such circumstances blowing off to reduce the density of the water will only make matters so much the worse, for the lower the density is to be maintained the greater must be the amount blown off, and hence the greater the amount of extra feed, and the greater the amount of scale forming salts brought into the boiler, all of which will be deposited.

It became therefore necessary to abandon the use of the jet condenser and salt feed. Its place was taken by the modern surface condenser. So long as this condenser is perfectly tight the feed water consists of the condensed steam, and is therefore almost perfectly fresh water. Due, however, to steam leaks at the various joints, seams and glands, to the occasional use of the steam whistle, and to the use of steam in certain auxiliaries from which it is not returned to the condenser, there will be a continual shortage in the feed water which under usual conditions will be found between say 2 and 5 per cent. Until recent years this shortage was made up by the use of sea water obtained usually by opening, as circumstances required, the *salt water cock* connecting the salt water side of the condenser with the steam side. It is very difficult to keep the tubes of a surface condenser packed perfectly tight, and in some cases the condenser was allowed to run a little leaky, simply to make up in this way the salt feed required.

Due to this admixture of salt feed, the scale forming salts of which are all deposited in the boiler, there will be a gradual formation of scale greater or less, according to the length of the run and the proportion of salt feed make up.

In recent years experience has clearly shown that the dangers of overheating and the general bad effects due to the presence of scale are more and more pronounced as the pressures are higher. It has become therefore more and more important to prevent so far as possible the entrance of any sea water into the boiler, and thus avoid the formation of scale with its troubles and dangers. To this end, in modern practice, the make up feed is provided by an evaporator, or in some cases by feeding one boiler with salt feed and thus restricting the scale formation to this boiler, while the condensed steam from all the boilers is returned to the other ones as feed. In all such cases it will be noted that this scheme amounts to a transfer of the use of salt water and the formation of scale from the boilers in general to the evaporator, or to the particular boiler in which it is allowed to accumulate.

For short trips as, for example, those met with in bay, harbor or channel service, or on short coasting voyages, fresh water for make up feed may be carried in tanks instead of providing it by means of an evaporator. By many engineers this is considered the preferable method whenever tanks of sufficient size can be provided, and in some cases with the double bottom style of construction, double bottoms have been utilized to a considerable extent for this purpose.

It is rare that the condenser can be maintained perfectly tight, so that even under the best practicable conditions there is apt to be some passage of sea water into the steam side of the condenser, and thence into the boiler. Under the best conditions the amount of scale formed, however, is so small that commonly no special treatment is attempted, and the scale is allowed to deposit, and is then removed at the regular periods of inspection and overhaul.

Some attempts have been made to prepare sea water by the removal of the calcium sulphate in a separate vessel before entering the boiler. This may be done by the use of sodic fluoride which causes the sulphate to separate out and settle to the bottom as a fine powder. The remaining water is practically free from this substance and may be used for boiler feed without fear of causing scale.

Soda ash and other alkalies have sometimes been used in boilers, with feed water containing sulphate of lime. They act by converting the sulphate into a carbonate, and thus into a somewhat less objectionable form.

Barium chloride acts in a somewhat similar fashion by producing barium sulphate and calcium chloride.

The use of zinc in boilers is also by many believed to prevent to some extent the formation of scale by the reaction of the alkaline zinc chloride on the scale forming salts.

With sea-going, as with inland boilers, the bottom blow should be used occasionally and as the particular circumstances may demand, so as to remove the accumulation of such substances as may be thrown down as a powder or sludge and thus collect in the bottom of the boiler.

However careful the provisions for keeping sea water out of the boilers or no matter what methods may be used to prevent scale formation, it is almost sure to gradually accumulate, and assurance of safety from the troubles and dangers which may result can only be obtained from periodical examination and scaling as may be found necessary. All marine boilers must, of course, be provided with manhole plates for this purpose, and the internal arrangement of tubes, braces, furnaces, etc., should be made, so far as possible, with a view to furthering this necessary operation.

Combinations of Oil and Scale.—We have thus far referred to scale formed simply from the solid matter in the feed water. The combinations which may be formed by the deposited salts and oil from the cylinders as it may enter with the feed water are, however, of even still greater importance, and must now be noted.

Oil coming in thus with the feed water is caught by the circulating currents and distributed more or less throughout the boiler, though by reason of its lesser weight it will tend gradually to rise and accumulate as a scum at the surface of the water. In thus wandering about, a drop may come in contact with a bit of solid matter separated from the water. The two join together, the oil forming a coating about the sulphate, and they journey on meeting and joining with other like particles. The combination of the oil and sulphate may have about the same specific gravity as the water in the boiler, and hence these particles will readily move with the circulating currents, either up or down, as they happen to be flowing. They are thus swept along the heating surfaces, to which they attach themselves all the more readily by reason of their oily covering, and on either the upper or lower side as they happen to be moving with a down or up-flowing current. In this way the coating gradually increases until it has attained a thickness sufficient to seriously interfere with the passage of the heat.

In other cases, when the scale and oil are lighter or the water is denser and heavier, there seems to be formed at the surface of the water in the boiler a kind of oil and scale blanket or layer floating about, and perhaps ultimately by the gradual increase of weight sinking and covering some portion of the heating surface. Especially is this oil "pancake," as it has been called, liable to settle should the density of the water in any way be suddenly decreased. Still otherwise should the boiler be blown down by the bottom blow, such an oil blanket would naturally settle and attach itself to some part of the heating surface. Should the boiler be then filled again, the coating would remain where attached. This shows that under such circumstances a boiler should never be blown down with the bottom blow without first using thoroughly the surface blow to remove as far as possible all such accumulations of oil or of oil and scale from the surface of the water.

The danger to be feared from this combination of scale and oil is not in its close adherence to the surfaces, but in its non-conductivity for heat. Experiments show that 1-16 to 1-8 inch of such a covering is far worse in this respect than perhaps 1-2 inch or more of scale alone. The danger to be feared is therefore overheating and collapse, and not a few cases of the collapse of furnaces and other parts of marine boilers are believed to be due to this cause.

So far as these effects are concerned it is seen that it is better to carry a high density in the boilers than a low one, so as to keep such oil and scale combinations at the surface of the water, where they may be disposed of by the surface blow.

As it is practically impossible to prevent the entrance of some scale-forming materials into the boiler, the danger of trouble with oil and scale combinations is most surely prevented by keeping the oil out. To this end a cylinder oil should be used having a high point of vaporization, as the higher this point the smaller the amount carried into the condenser. Of this oil the minimum amount necessary should be used in the cylinders, and the feed water should be filtered to remove whatever oil it may contain.

Sec. 42. BOILER OVERHAULING AND REPAIRS.

[1] Inspection and Test.

We will suppose that after some considerable term of service, and preparatory to a general overhauling, a battery of boilers are to be carefully and thoroughly examined. The more important points may now be considered.

(1) *Furnace Fronts.*—The furnace fronts and doors may be found warped and cracked, and if this is the case to such an extent as to interfere with the proper closure of the furnaces, or with the proper and convenient care of the fires, the necessary repairs or renewals should be made.

(2) *Grates and Bearers.*—The grate bars will often be found warped and twisted, or badly burned, and at various points sunken below or sprung above their proper level. Such irregularities in the grate may occasion loss of coal at some points, while they will further the accumulation of ash and clinker at others, and will make it almost impossible to give to a fire the proper attention, or to get from a square foot of grate surface the power which it should be able to give. The bearers may also be the cause of trouble by warping or settling from having been overheated, and all of these points must be attended to before the boiler can be considered again ready for proper service.

(3) *Bridge-wall.*—The bridge-wall is liable to be found more or less burnt out and dilapidated, while on the front side clinker and bits of brick may be found fused together in irregular masses. All of this must be removed and the bridges built up again with fresh bricks to the proper height as referred to in Sec. 38 [1].

(4) *Tubes.*—The tubes will, of course, be swept and properly cleaned on the fire side. The existence of small leaks must be carefully looked for, the evidence being the presence of soot and scale burned to the metal where the water has come through and evaporated. The back and front tube sheets and the inside of the tubes must be carefully examined for any such evidence. Signs of especial wear should also be looked for at the back ends of the tubes, and if ferrules are used some will probably be found so worn and burned out as to require renewing.

In water-tube boilers any special warping or change in the shape or curvature of the tubes should be carefully noted, as it may indicate overheating due to faulty circulation caused by a clogging of the tube by scale and sediment. A split or badly ruptured tube will, of course, show itself by the resulting leak, but in a water-tube boiler such leak may be very difficult to locate without the removal of several of the tubes in the vicinity of the one giving the trouble. These points depend entirely on the type and style of construction, and no general rule can be given for definitely and immediately locating such a tube in a water-tube boiler.

(5) *Joints and Seams.*—The joints and seams throughout the boiler, both in the combustion chamber and on the outside, should be carefully examined for small leaks, either between the plates or about the rivets. If the leakage is not serious, caulking will serve as a sufficient remedy. In other cases, however, the removal of old rivets and the insertion of new ones may be found necessary.

(6) *Front Connections and Uptakes.*—The front connections, uptakes and fittings should be examined to make sure that the plates are not warped or broken from their fastenings, and that the dampers and their operating gear are in proper condition.

(7) *Fittings.*—The valves and cocks are likely to be found more or less worn on their seats and leaky in consequence. These will require regrinding and refitting, or replacing by new where necessary. The operating gear, such as valve spindles, wheels, levers, chains, gear-wheels, etc., should be examined for any breakage or derangement of parts. The various joints and fittings about the steam and water pipes must also be examined for signs of leaks, distress, corrosion, or other derangement.

(8) *Bracing.*—The manhole plates will, of course, have been removed to facilitate examination of the interior.

The braces, especially where pin joints and like connections are used, should be carefully examined for defects in the connections and fittings, and also for any symptoms of buckling or distress in the braces themselves.

(9) *Scale.* The scale present in the boiler should be examined as to its amount, distribution and character—whether hard or soft, greasy or otherwise, closely adhering or readily cracked off. Accumulation of scale between the tubes or screw stays, and of scale and sludge in the bottom of the boiler must also be looked for and noted. In some cases an oily or greasy coating with little or no mineral matter and forming a coating over the scale and on the heating surfaces may be observed. This will indicate large quantities of oil in the boiler, and insufficient use of the surface blow.

(10) *Corrosion.*—It is, of course, of the highest importance to examine carefully for signs of corrosion and pitting throughout the interior of the boiler. The following locations, however, are those in which it is most apt to be found:

On the sheets at and near the water line. Occasionally also severe corrosion is found in the steam spaces.

On the braces near the water line.

On the tubes and combustion chamber tops.

On the furnaces near the grate level.

The nature and distribution of this corrosion must be carefully noted, in order that the most suitable steps may be taken for its arrest and prevention in the future. When they can be gotten at, corroded spots may be scraped and scrubbed clean with water made alkaline by the addition of soda or weak lye, and if not on a heating surface, a redistribution of the zincs may prove of service, while in general a more careful attention to the various means suggested in Sec. 40 may be recommended. The zincs and their fittings, as discussed in Sec. 40, must also be carefully looked after. Many of the zincs will probably be found to have wasted away to only a small part of their original size, and to have become changed in physical structure to a blackish or brownish crumbly or brittle mass. In some cases remnants of the slabs may be found lodged between the tubes and screw stays, and often more or less covered or imbedded in deposits of scale.

On the exterior of the boiler the points most liable to corrosion are on the fronts about the bottom where damp ashes may have lain, or about the saddles and on the under side where dampness and water are liable to be formed. Thorough cleaning, followed by a coat of paint, asphaltum varnish, or other like material, is the usual remedy in such cases, at least where its application is practicable.

Before the application of any such coating, the plates should be thoroughly dried, else it will be of little use. The presence of moisture on the plates causes the especial difficulty connected with the effective application of paint in such places, and where convenient the use of a portable sheet-iron drying stove containing burning charcoal or coke may be found of use. This may be placed under the surfaces to be covered so as to furnish an ascending current of warm air, thus aiding in keeping them dry during the application of the paint.

For the structural material in bilges and bunkers a coating of Stockholm tar put on hot and then sprinkled with Portland cement is highly recommended by some engineers.

(11) *Manholes and Covers.*—The faces on which the manhole cover joints are made should be examined for corrosion or scale, or anything which may affect their evenness, or make difficult the fitting of a tight joint.

(12) *Drill Test.*—Where the boiler has seen long service, or where there are evidences of serious corrosion, or doubt exists as to the thickness or quality of the plates, they must be drilled at such points as may be selected. In this way the thickness of the remaining good metal may be ascertained, and the safe pressure to be carried may be fixed in accordance with the evidence thus found.

(13) *Hydraulic Test.*—When the boiler has been overhauled and put in proper condition, at least as far as anything which may affect its strength is concerned, the hydraulic test may be applied. To this end the boiler is filled full of water and pressure is put on, usually by means of a special pump connected for the purpose. The test pressure is usually one and one-half times the working pressure desired. It is considered that this pressure is not sufficient to seriously try or injure the boiler should it be properly constructed, and of suitable factor of safety throughout, while at the same time it will be sufficient to develop small leaks, and should the boiler be unduly weak at any point, the bulging or yielding or distress at such point should become apparent. If no such evidences appear, then it is con-

sidered that the boiler is abundantly strong for the working pressure as desired.

To prepare the boiler for the test the springs should be withdrawn from the safety valves and lengths of pipe of suitable size substituted, so that the valves may be screwed down fast. All stop valves and gauge glass connections should then be tightly closed, as well as the connection to any pressure gauge which will not indicate up to the test pressure.

While the test is under way the boiler is subjected to the most careful examination, both inside and out. The furnaces, combustion chambers and back tube-sheets are examined from the inside, and the shell and its various joints and seams from the outside. Small leaks are watched for and stopped by caulking, if possible, or if about the tube ends, by re-expanding. Especial care must also be had in watching for any signs of bulging, buckling or other deformation or distress.

Where the test is to be carried out with especial care, extension and compression gauges are provided in the furnaces and combustion chambers, and at other points, as may be desired. These serve to indicate and to measure the actual amount of distortion which results from the gradually increasing pressure. If the distortion or bulging at any point should become abnormal, the pressure should be relieved by letting off a little water, in order to see if any permanent set has been made. The continuance of the test should then be made to depend upon the behavior of the part showing this relative weakness. For a thoroughly satisfactory test, all gauges should return to the original setting when the pressure is removed, showing no permanent set at these points.

It will usually be found very difficult to so tighten the various valves that there will be no leakage. An idea of the amount of leakage may be obtained by watching the rapidity with which the pointer on the pressure gauge moves backward when the pump is stopped, as well as by the amount of pumping required to maintain the pressure at its full value. The pressure gauge pointer will also frequently indicate by its more or less sudden movement backward, the sudden development of leaks about the riveted joints or tube ends.

It has sometimes been urged against the hydraulic test that it may severely strain some part of the boiler where the yield or distress is difficult to observe, and thus so weaken it that a further yield or rupture may occur under a much smaller load at a later time. The hydraulic test is, however, very generally employed both in the naval and mercantile marines, and if carefully conducted and with a pressure not exceeding one and one-half times the working pressure, it is not likely that harm will result, while the test will develop the small leaks and minor defects, and may be the means of exposing serious faults of workmanship or design.

The same test is, of course, applied to new boilers as a final preliminary to the getting up of steam.

In some cases the water test has been carried out by filling the boiler and then lighting wood fires within the furnaces. The expansion of the water will furnish the increase of pressure desired, which may be eased by the safety or stop valve, as necessary. It has been claimed that the boiler being in this way heated, was more nearly in its regular service condition. While this may be so to a slight extent, the boiler is, nevertheless, far from regular service condition, and the method has the serious disadvantage that it does not allow examination of the furnaces, combustion chambers and back tube sheets while it is under way. It is also under less ready control than the pump method, and is now but rarely employed.

BOILER REPAIRS.

In the following suggestions regarding boiler repairs we shall refer more especially to such as may become necessary at sea or under emergency conditions, rather than to those which may result in the course of a thorough overhauling in port.

(2) Leakage from the Joints of Boiler Mountings.

Such leakage by soaking through the lagging and keeping the plates wet may give rise to surface corrosion on the boiler shell.

The first care must be to stop the leakage by screwing down, re-caulking or re-making the joints, as may be necessary.

If there is reason to suspect corrosion of the boiler as well, the lagging should be removed and the corroded surfaces scraped clean and painted with good metal paint or other suitable covering.

(3) Leakage About Shell Joints.

Usually caulking will be sufficient to stop any ordinary small leak in these joints. If it is serious and caulking gives but little improvement, it may indicate a loose rivet or one with the head gone. In such case leakage about the rivet will usually be present also, and will thus serve to locate the trouble. Such rivet must, of course, be replaced, in order to effectually stop the leak.

Where a rivet has blown out and a quick repair is desired, the hole may be drilled or reamed true and then tapped out. Then fit a bolt with corresponding thread and cut it partly through near the root with a hack saw. Screw this in, knock off the projecting end, rivet down the remainder and the job is complete.

In some cases instead of replacing loose or broken rivets, or where for other reasons caulking seems to be inefficient in stopping the leak, it may be considered desirable to put a patch over the seam, rivets and all. In such case a so-called "soft-patch" is applied. This is illustrated in Fig. 204. The patch is flanged and made with a recess of suitable size and form to accommodate the rivet points. It is then filled with a stiff putty

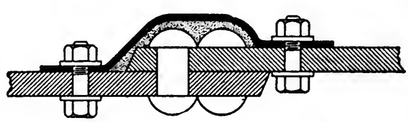


Fig. 204. Patch for Leaky Joint.

of red lead and secured by bolts as shown. Such an application is really a red-lead poultice, kept in place by a suitably formed steel cover, and secured to the shell, as explained. It is unnecessary to make such a patch of metal more than 3-16 or 1/4 inch thick, since it is not intended to add strength to the shell, but simply to keep the red-lead putty in place, and thus stop the jet of leaking steam or water.

The chief difficulty with leaks in the shell seams and with the outside of boilers in general arises from the trouble in subjecting them to the proper examination due to the presence of the lagging. As usually fitted, this covering is difficult of removal, and small leaks thus covered in may continue for long periods of time, keeping the outer surfaces wet and causing rust and corrosion where its existence may not be expected. A form of boiler lagging admitting of ready renewal and replacement in sections is much to be desired, and if full advantage were taken of such a form of covering to keep closer watch of all joints on the outer surface, much trouble might be avoided by taking the first appearances of trouble in time.

(4) Leakage at Internal Joints.

The internal joints, on the whole, give more trouble than those on the outside. This is only to be expected due to the thinner plates, the enormous range of temperature differences which exist, and the resultant expansions and contractions. The difference in expansion between the furnaces and tubes is especially liable to give trouble with the joints connecting the furnace to the combustion chamber, and several varieties of joint have been proposed to reduce this trouble to a minimum. With a joint such as shown in Fig. 43, the greater expansion of the furnace tends directly to open up the joint, while with that shown in Fig. 9 the result is a shear on the rivets but no direct tendency to open the plates. In the latter case, however, the

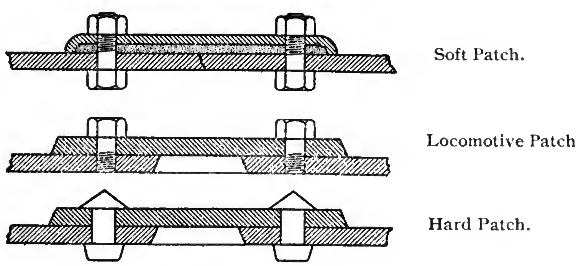


Fig. 205. Different Forms of Patches.

rivets are more directly exposed to the fire than in the former, and thus the points between the two joints are very nearly balanced. The joint of Fig. 9, however, allows a more ready removal of the furnace, and on this account it is often selected rather than that of Fig. 43.

Leaky joints on the combustion chamber should be first carefully re-caulked. This operation, however, cannot be carried on indefinitely, for after caulking to a certain extent, the edge must be chipped off to get a fresh caulking edge. This, if repeated, will leave the metal between the rivets and edge too narrow for safety. If careful and judicious caulking does not remedy leaks in these seams, it is evident that the rivets need renewal, and in carrying this out especial care should be taken to see that the holes are fair in the two plates, and that the rivets fill them completely.

(5) Patches.

We may now turn more especially to the patching of boilers and to the different kinds of patches employed.

We must first remember as a general principle that any thickening of metal on the heating surfaces is undesirable and to be avoided as far as possible, or reduced to the lowest possible extent. If, therefore, a patch on a heating surface is to be considered as a permanent fixture, the faulty metal should be cut out, thus doubling the thickness only over the necessary width for the fastenings. A patch put on in this way with rivets headed up as in regular boiler work is known as a hard patch, and is illustrated in Fig. 205. In some cases the patch must be put on from one side only, or is more temporary in character. In such case either the locomotive or the soft patch is used. The former is a patch put on with tap bolts, as illustrated in Fig. 205, and usually without cutting out the metal. The soft patch has been already referred to. In its more usual form, as shown in Fig. 205, it is made by lipping down a plate of steel so as to contain and hold in place a coating or layer of red-lead putty. It is more suitable for temporary repairs, or where the surfaces are so rough and uneven that a patch of the other forms could not be fitted. The soft patch is sometimes secured with tap bolts, and sometimes with through bolts and nuts, as may be most convenient with the case in hand.

As to whether a patch should be put on the water or fire side, much will depend on location and convenience. Where it is possible the water side may be chosen so that the steam pressure will tend to keep the patch in place. Usually, however, the fire side of the plate is more easily gotten at, and in many cases there is no choice but to put the patch upon this side. In any event with either the hard or locomotive patches, the edge will require careful caulking as the final closure and making of the joint. With the soft patch, caulking the edge is not necessary, as the putty is depended upon to stop the leak, and the office of the patch is merely to hold it in place.

(6) Cracks and Holes.

A small crack is usually treated by drilling a hole at each end to prevent its extension, and then covering with a patch, according to location and convenience. Very small cracks are sometimes drilled and tapped out as close together as the holes will stand, and then filled with soft iron or steel bolts, riveted down so as to overlap and thus completely close the crack.

Small isolated holes not accompanied by a general thinning of the metal may be treated in a similar fashion by drilling and tapping out the hole and riveting in a bit of soft iron or steel bolt. Larger holes, or where many are located near each other, or where they are accompanied by pronounced thinning of the metal must be treated by patching.

In general it may be noted that plugging as above described, is only suitable for the mere stopping of a leak, and that it adds nothing whatever to the strength of the plate. If, then, the conditions are such as to make additional strength desirable, a patch must be fitted.

Where a crack is found in a tube sheet it usually extends from tube to tube. Such a crack may be covered by a patch ex-

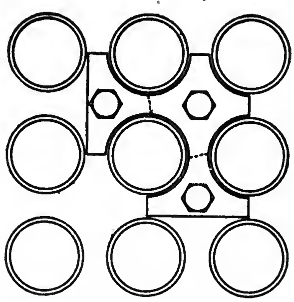


Fig. 206. Patch for Boiler Tube Sheet.

tending over the crack and taking enough good metal to obtain a secure hold. The patch must have holes cut in it, of course, to correspond to the tube ends as shown in Fig. 206.

[7] Blisters and Laminations.

With modern boiler material these defects are happily rare. In former years, and especially with iron plates, they were only too frequently met with as referred to in Sec. 5. The chief danger from these defects is due to the weakening of the plates and the liability of overheating, due to poorer conductivity for heat. Such defects if very small are often left undisturbed, with careful watching and measurement from time to time. If small and the metal quite thin on one side, the thinner part was cut away, leaving the thicker side to do duty for both. In some cases also a dog and supporting bolt was fitted to support the remaining metal. For some serious cases, however, it was usually considered preferable to cut out the metal thus affected, and cover the hole with a patch.

In all cases where patches are put on, or in general where new material is put into the boiler, it is well, if convenient, to select it of stock as nearly like the boiler as possible in physical and chemical constitution. This will tend to decrease the possible sources of electro-chemical action as discussed in Sec. 40.

[8] Tubes.

The repairs to boiler tubes comprise re-expanding, plugging, and renewal.

Expanding has been explained in Sec. 16, and a repetition of the process may be required from time to time, to keep the tube ends tight. The immediate cause of the leakage of boiler tubes is often the accumulation of dirt and scale about the tube ends and on the tube sheet. These points should therefore be carefully looked after, or the re-expanding will be of little use. Care must be taken that the operation of re-expanding is not repeated too often, or the metal of the tube end may become so thinned and hardened that there will be danger of weakness or brittleness at this point.

For stopping a tube which has split or in any way developed a serious leak, a tube stopper or plug is used. Temporary stoppers of pine wood are often employed. Such a plug closely fitting the tube and twelve or fifteen inches long may be forced in from the front end to a point where it covers a split or hole, and thus provides a temporary repair. The swelling of such a plug caused by the action of the water and steam will cause it to stick closely in place and thus more effectually stop the leak than would a metal plug in the same location. For more permanently plugging a tube, tapered cast iron plugs are used, one at each end, driven in to a tight fit and held in place by a rod passing through them from one end of the tube to the other and set up with thread and nut. The plugs and nuts should be faced so that when set up with a copper washer or turn of copper wire underneath, a steam tight joint between the plug and nut may be made. To plug a tube in this manner the fire must, of course, be drawn and the back connection entered in order to insert the back plug in place and adjust the rod and nut.

When a tube is to be renewed the old one must first be drawn. To this end the back end is closed down so as to readily pass through the hole in the tube sheet. A rod is then passed through the tube and a nut and washer are fitted to the back end, the washer being of such size that it will bear on the tube end and at the same time will pass through the hole in the tube sheet. The front end of the rod passes freely through a dog or strong-back whose feet rest on the tube sheet, and is provided with a nut bearing on the face of the dog. The nut being then forced down, the rod is withdrawn and with it the tube. To facilitate withdrawal the front end of the tube is usually made of slightly larger diameter than the back end, and the tube once started is readily withdrawn the remainder of the way. Before inserting the new tube the metal of the tube sheets about the holes should be carefully examined to see that the holes are smooth and fair and that no reason exists why the expansion of the new tube may not make a steam tight joint. The new tube may then be inserted, the ends expanded, beaded over at the back end or at the front and back ends if no stay tubes are fitted, and the operation is complete.

[9] Leakage About Stays and Braces.

Leakage at these points may be due to corrosion about the joint, or to loosening due to repeated expansion and contraction, or to bending or distortion of the plate or stay caused by bulging or partial collapse of the plate. If the leak is not serious, setting up on the nut, or caulking about the joint between stay and plate may prove sufficient. If not it will usually be necessary to remove and refit the stay. For screw-stay bolts this will usually require the reaming out and retapping of the holes for the next larger size. For braces fitted with nuts or nuts and washers, the same size can usually be replaced, but especial care should be taken to insure the proper smoothness and fairness of surface about the holes, as well as the proper fit and adjustment of the nuts and washers, so that the usual fitting will provide tightness of joint.

[10] Bulging or Partial Collapse of Furnace or Combustion Chamber Plates.

We have already discussed in Sec. 38 the causes of bulging or collapse, and the steps most suitable to insure immediate safety. When the time comes for examination and repair the following steps may be taken.

If the bulge is quite small it may be decided to leave it undisturbed, assuming that its strength is practically as great as before. In such case, however, a template should be fitted to the bulge, and this should be applied from time to time in order to detect any signs of further yielding at this point.

In other cases special girder or through braces may be fitted to support the bulged portion, the details of the arrangement depending, of course, wholly on the circumstances of the injury.

In other cases the bulged part may be more or less completely forced back into place. This, however, is an operation requiring both skill and care, and should not be undertaken without making the preparations necessary to carry it out in the proper manner.

A portable grate or furnace for burning coke or charcoal is first provided of such shape and with such arrangements that the burning fuel may be brought close up to the bulged surface. An artificial draft may be provided by means of a blower from a hand forge fitted with a suitable conduit, or otherwise as may be most convenient. In the meantime a cast-iron former block should be provided, shaped on its face to correspond to the surface of the plate when forced back into position. A hydraulic jack or other like appliance must next be provided, taking especial care to arrange for the distribution of the load on the base over a considerable area of plate so that no harm may be done by the reaction from the head. To this end a support of heavy timbers is usually the most convenient to arrange. These various appliances being in readiness the bulged plate is heated to a low red, the former-block and jack are adjusted in position, and the plate is carefully forced back into shape. Several applications of the heat and of the jack may become necessary before the plate is restored to its original form.

It is thought by many that the partial heating of a steel plate in this way with no later opportunity for general annealing is liable to injure its homogeneity and toughness, especially if the operations are carried on at too low a temperature, or approaching what is known as blue heat. This was undoubtedly true for much of the earlier products of the steel makers, and there is no doubt that the homogeneity of the metal is thus somewhat disturbed. With the latest and best grades of boiler plate, however, little danger as regards strength at least, need be feared on this score, and but little hesitation is now felt in reducing to shape a bulge of moderate depth.

[11] Split in Feed-Pipe.

A small split in the feed-pipe may sometimes be temporarily repaired by wrapping with heavy canvas and marline, or copper wire. It is, however, difficult to make a tight joint with hand wrapping, especially with modern high pressures, and a more effective plan is to form up a patch of sheet copper and secure it in place by bolted strap clamps, with a sheet of good elastic packing, or a thin layer of stiff putty between the patch and the pipe to make the joint.

A small hole in the feed-pipe may, if the metal is thick enough, be stopped by drilling and tapping out and riveting in a small screw plug. Soft solder when applied with skill may also be used to stop pin-holes, or to aid in securing a suitable plug in a larger hole. If, however, the hole is of any considerable size, or if the metal is thin about its edges, some form of patch, as described above, must be made use of.

All such repairs are, of course, only temporary, and at the earliest convenient opportunity the damaged length of pipe should be replaced with new.

In some cases with copper pipes, however, where the necessary materials and skill are available, it may be desired to undertake a more permanent repair by brazing a patch over the hole or split. To this end the metal about the defective spot is first cleaned with file, emery cloth and acid. The patch similarly cleaned is cut from sheet copper of about the same thickness as that of the pipe, formed to fit over the defective spot and wired in place. A clear coke or charcoal fire is then prepared on a forge and the pipe placed in position. The spelter or hard solder, mixed with borax as the flux, is then placed in position on the inside of the pipe and the whole carefully heated. At the proper temperature the spelter will melt and run in between the patch and pipe, thus forming a joint between the two. Care must be exercised in this operation in order to avoid danger of overheating or "burning" the copper. The resulting loss of strength has been referred to in Section 3.

Sec. 43. ENGINE OVERHAULING, ADJUSTMENT AND REPAIRS.

We shall undertake in this section only a few hints regarding the more common operations involved in the overhauling, adjustment and repair of marine machinery.

[1] Cylinders.

First, in the main engine, the cylinder covers must be removed at proper intervals, and attention given to the condition of the wearing surfaces and of the piston springs. The troubles most liable to be found are cutting and scoring of these surfaces, and derangement or breakage of the springs. The nuts of the follower studs and all other forms of screwed fastenings should also be examined, in order that any tendency toward loosening up or backing off may be noted and checked.

To examine the condition of the piston the follower plate is lifted and the springs and packing rings removed. The latter, if wearing properly, will be of uniform thickness around the entire circumference, and of uniform polish on the outer surface. If the piston-rod is bent or cylinder bore not quite in line with the motion of the rod, the ring will wear wedge-shaped in cross section. In replacing, care should be taken to note the degree of tightness of the ring when set out with its springs. This should be such as to support the ring anywhere along the bore, but not so much that the ring cannot be pushed along by hand, using moderate force. In closing up the cylinder or valve-chests, especial care should be taken to see that all nuts, split pins, etc., are properly secured in place, and that no tools, waste or other foreign substance are left within. Neglect of this latter point has often been the cause of serious trouble, resulting in broken cylinder-heads, bent piston-rods, broken valves, port metal, etc.

To test the tightness of the joints of the cylinder liners, the cylinder being open, steam is admitted to the jacket and the joints are carefully examined top and bottom. Where there is no manhole on the lower head, as is common with small cylinders, a leak of any significance will become evident by opening the lower indicator cock.

In order to test the tightness of the piston under steam, the cylinder-head being in place, we may proceed as follows: Put the engine and links in such position that a little steam can be admitted on top of the piston and then open the bottom indicator cock. Then admit the steam, usually through the starting valve, and if it blows through the lower cock the piston is thus shown to be leaky. For a thorough test it will be well to try the leakage in both directions; that is, from top to bottom, as above, and similarly, from bottom to top.

[2] Pin Joints and Bearings.

The various pin joints and cylindrical bearings will need attention according to the special circumstances of the case. The manner in which a joint or bearing has been working, both as to noise and temperature, will often serve as a guide to those which the most require attention.

To examine the crosshead bearings the main points of the operation may be outlined as follows:

(1) The crosshead with piston-rod and piston must be secured near or at the top of the stroke. This is sometimes done by inserting pins or bolts in the upper edge of the slide through the holes for securing the slipper, and allowing such to project over the top of the guide. Or otherwise it may be done by shoring in such way as may best be suited to the details of the case in hand.

(2) The outer caps and brasses are removed.

(3) A wooden chock lashed to the connecting rod is fitted between its upper end and the guide. This will serve to support the upper end when free from the crosshead.

(4) The connecting rod sloping in such way as will bring the support of its upper end upon the chock, the turning engine is used to revolve the crank down until the parts are sufficiently clear to admit of the examination desired.

To remove the crank-pin brasses the main points of the operation may be outlined as follows:

(1) Starting with crank near its lower position the bottom cap is removed and landed on a bed suitably prepared for it.

(2) The turning engine is then used to carry the crank up nearly or quite to the top of the stroke, where the crosshead is secured by hanging up or shoring as described above.

(3) The upper brass is then secured and the crank is rotated down until the parts are sufficiently clear for the purpose in view.

For like examinations of other parts, similar means will readily suggest themselves.

For removing the lower brass of the main pillow-block bearings where it is in the form of a half cylindrical shell, it needs simply to be rotated out as noted already in Sec. 21 [11]. A method sometimes used for this purpose is to clamp to the crank-arm a bar of steel carrying a projecting pin or bolt so placed that it will engage with the top face of the brass when rotated around. The upper cap and brass being then removed, the shaft is rotated carefully by means of the turning engine, and in this way the lower brass is forced around and up until free from its bed.

Bearings and journals of this character may simply require readjustment, or refitting and adjustment as well. When the wear has been considerable, the liners or chock pieces between the brasses will need thinning, in order to reduce the clearance between the journal and the bearing to the propor amount. In order to measure the clearance under any given adjustment, a piece of lead wire is employed, of somewhat greater diameter than the clearance is to be. This wire is placed on the journal and the cap is screwed down hard, thus compressing it between the journal and the brass. The cap is then removed, and from the resulting thickness of the wire the clearance at any point between the journal and the brass may be readily measured. Such an operation is called taking a lead. Several leads, if desired, may be taken at once from a bearing, and will thus serve to map out quite satisfactorily the distribution of clearance, and thus to show when the proper adjustment has been made. The proper amount of clearance in any given case is somewhat a matter of judgment, and will, of course, vary with the size of the journal. In ordinary cases it may be made about .002 of the diamater.

If time is limited the adjustment may be effected without the taking of leads, as follows: The chock pieces or liners are taken out and the nuts are tightened up till the brasses bear full on the journal. Their positions are then marked, and they are then backed off an amount determined by judgment or experience with the particular circumstances of the case, and the liners are stripped to fit this adjustment. While this method is not as satisfactory as with leads, it is much quicker, and with experience good results may be obtained.

In connection with the marking of the nuts for such purposes it may be recommended as a good plan to mark permanently with a line and a numeral 1, 2, 3, 4, 5, 6, the faces of all the large nuts likely to be used in making adjustments. An adjacent reference line on the bolt or on the metal of the bearing caps will then furnish means for making a record of each adjustment, or if no permanent record is desired, such means will greatly facilitate making the adjustment in any given case.

The refitting of bearings and journals on shipboard does not usually extend beyond removing or smoothing rough spots caused by overheating and scoring. This may be done by filing followed by "lapping" with oil stone powder or dressing with an oil stone. In some cases emery is used, but great care is then necessary in order to remove all particles, as if allowed to remain they will give trouble by continued cutting and grinding in the bearings. The brasses are similarly redressed, usually by scraping. The nature of the contact between the brass and the journal is tested by lightly smearing the latter with red lead and then applying the brass in place and lightly rotating and forth. The high spots will then be shown by the red lead and may be further dressed down till a satisfactory fit is obtained. It may be noted that where brasses are thus fitted up each one may show a satisfactory fit when tried separately and without external constraint, and yet when in place they may be unable to come to the same relative positions on the journal and make satisfactory contact with it. For this reason it is preferable to test the contact with the brasses together and regularly secured in place. This is not always possible, however, by reason of the additional time required, and judgment in all such cases must be used, having in view the various circumstances of the case in hand.

[3] Cross Head Guides.

The main guides and cross head slides must receive attention, both as regards cutting or uneven wear, and as regards the question of adjustment. The result of wear is to throw a cross-breaking stress on the piston-rod at each stroke, as the cross head is forced over by the oblique action of the connecting rod to a bearing on the guides. Care must therefore be taken that when the engine is turned without a load, the guide surface remains in full contact with the face of the slide, and therefore in a condition to support and guide the lower end of the piston-rod in a path consistent with the movement of the rod in the axis of the cylinder. If this condition is not fulfilled the necessary adjustments must be made in the manner best suited to the structural arrangements of the case in hand.

To remove the slipper or bearing piece for examination or refitting, screw eyes may be screwed into holes in its upper edge made for the purpose, and from these the slipper may be supported by means of wire rope or stout wire wound around a bar suitably supported and secured above the slipper. The bolts holding the slipper to the crosshead are then removed and the crosshead forced over by screw or hydraulic jack or other convenient means, sufficient to ease the pressure between the slipper and the guide. The slipper may then be lowered by using the bar as an axle and the rope or wire will readily follow down between the crosshead and guide surface.

[4] Crosshead Marks.

In connection with the adjustment of the moving parts of the engine it is well to have a mark on the crosshead and corresponding marks on the guide, showing the extreme positions of the piston when in contact with the cylinder heads, top and bottom; also two marks placed slightly within the latter and showing the ends of the natural stroke, and a mark placed midway between the two latter, showing the location of the piston when in midstroke. The distance from the extreme marks to those showing the ends of the stroke, shows the amount of clearance proper between the piston and the cylinder heads when the former is at the ends of the stroke.

This will vary with the size of the engine and character of the workmanship, but it is usually found between ¼ and ⅝ or ¾ inches, being a little more at the bottom than at the top, to allow for the general tendency of the parts to lower rather than to rise through the effect of wear.

To determine these points a convenient reference mark is first placed on the crosshead. The connecting rod may then be disconnected and the parts hoisted up as far as they will go, or until there is contact between piston and head. A mark is then made on the guide corresponding to that on the crosshead. The parts are then lowered down as far as they will go, or until there is contact between the piston and the lower head, and another mark is made on the guide corresponding to that on the crosshead. The distance between these is then taken, and from it is subtracted the length of stroke. The remainder is then divided between the two clearances, top and bottom. Midway between the two inner points a point may be placed to indicate the location for mid or half stroke.

Thus, if the stroke is 36 inches and the distance found as above is 37 inches, the 1 inch difference is to be divided between the two clearances, giving to the upper, say, 7-16, and to the lower 9-16 inch. These differences are then laid off within the outside marks, and the points thus given will serve at any time as a guide for the adjustment regarding clearance proper, while the movement of the piston may be readily brought to conform to these limits by suitable adjustment of the liners or chock pieces in the joints and bearings of the connecting rod and crank-shaft. The 36 inches may then be divided equally and the mark placed to show mid stroke, such a point being sometimes of use in connection with the setting of the valve.

Another method of determining the clearance which is available when the cylinders have manholes is as follows: The manholes are removed and a number of balls of stiff red lead or other putty and faced with plumbago are distributed on the top of the piston and on the inside of the lower head. The engine is then given a revolution by means of the turning engine and the balls are collected. This method serves to show just how the clearance is distributed, and is therefore a valuable test for a bent piston-rod, a condition which will throw the piston out of position and give greater clearance on one side than on the other.

[5] Lining Up.

An important feature, both of the original setting up of machinery and of its overhauling and adjustment, is the determination or correction of the alignment of the various moving parts. It is clear, of course, what the condition of proper alignment is. For a marine engine it may be briefly stated as follows:

(1) The centers or axes of the main pillow block bearings should all be in one straight line, which will coincide with the crank-shaft axis, and which we will take as a standard or line of reference.

(2) The axes or center lines of the cylinders should all be in the same vertical plane containing the center line of (1), and they must also be at right angles to this line.

(3) The same vertical plane should also contain the axes or center lines of all the crosshead pins.

(4) The axes or center lines of the crank-pins or crank-pin bearings in the connecting rods must also be parallel to the center line in (1).

(5) The surfaces of the main guides must be parallel to the plane in (2).

(6) The line shaft must be in line with itself, and unless a

flexible coupling is provided between this and the crank-shaft it must also be in line with the crank-shaft, as determined by the line in (1).

The same general principles, of course, control the alignment of the valve gear and the various other moving parts, such as the starting, handling and drain gear, attached pumps, etc., but into these we need not go in further detail.

The implements used in establishing the relation between these various lines and planes will naturally vary with the circumstances, but are usually found among the following:

The level, plumb line and square.

The straight edge.

The stretched wire or cord.

Provision for using a line of sight.

A straight line, such as that for a line of shafting, may be determined by either of the latter. The sag in a piano wire of known size and length and stretched with a known weight over a pulley is a matter which may be found by a computation, into which we cannot enter here, or, better still, it may be determined in the open air by the aid of a surveying instrument with the usual cross hairs. A table giving the sag at various points between the ends for known lengths and for a known stretching weight will then give a ready means of establishing a straight line on board ship, by stretching the wire under the same conditions, and then setting upward at the various points the amount of the sag. A series of levels may thus be found which will give a true, straight line within the limit of the error in measurement.

When a line of sight is used the following method may be employed: A board is fitted to the bearings at the extreme ends of the line to be run, a hole of some considerable size being made in the board at about the center. This hole is then covered with a piece of thin sheet metal having a small hole, say, 1-16 to ⅛ inch in diameter. These sheets of metal are adjusted by measurement until the small hole, as accurately as may be, is brought to the center of the bearing. Similar boards are then prepared for the other bearings or points at which it is desired to establish the line. A light is then placed at one end beyond the further hole, and the eye at the other end. An assistant then adjusts the intermediate pieces of sheet metal until the light reaches the eye through the entire series of holes. The centers of these holes will then serve to establish a series of levels which may be marked on the pedestals, bulkheads or other convenient points, and which will serve to establish the line as desired.

In lining up and adjusting the engine itself so that the various conditions (1)-(5) above are fulfilled, very much will depend on the accuracy and care with which the various parts have been machined in the shops. The gaps in the bed plate, which receive the main bearing boxes, should be planed out so that they are all in line. This is a matter which may be tested by a straight edge, or by measurements from a stretched wire. If the bottom brasses are then adjusted, all to the same thickness, they will evidently support the crank-shaft in line. This may also be further tested by reference to a line of levels, each of which is obtained by measuring the same distance upward from the bottom of each gap, and locating on the bed plate at any convenient point the level thus determined. This adjustment, it may be noted, will bring the axis of the crank-shaft parallel with the bottom of the gaps.

Passing now to the columns, the seatings for their feet should have been planed at the same time as the gaps. This will bring the bottom of the feet in a plane parallel with the bottom of the gaps, and hence parallel with the axis of the crank-shaft. The columns and cylinders may now be erected and temporarily secured in place. The center line for each set of moving parts may be next determined as follows: A piece of board is placed across the top of the cylinder and secured at each end and over one or more stud bolts. Then, by careful measurement or the use of a beam compass, the center of the bore at the top of the cylinder is located on the board. A small hole is then bored at this point and a fine wire is passed through and attached to a little frame, which will allow a few inches of the wire to show above the board. The lower end of the wire is then located in the crank-shaft axis by suitable measurement from the faces of the bed plate. The wire is then stretched between the two points, and the adjustment of the upper end verified by renewed measurement. If these parts come accurately to place we shall find that the wire will be truly central relative to the opening left in the lower head for the stuffing box, and hence it may be taken as the true center line for the cylinder as a whole. We shall also find the wire at a constant distance from the guide surface on the column, thus proving the latter parallel to the center line, as required by (5) above. We shall also find the wire at right angles to the crank-shaft axis, as required by (2) above, and, of course, in a general way, at right angles to the bed plate transversely.

Supposing these various conditions fulfilled for each center line separately, then we must see if they are all in the same fore and aft plane as referred to in (2) above. To examine this point we may try to look the wires "out of wind," as the expression is, by standing aft or forward and trying, by sighting fore and aft, to bring all the wires accurately behind the nearest one. If these various conditions, for each wire separately and for all collectively, are not fulfilled, then the necessary adjustments must be made until they are brought into the required relations, as above specified. As a further adjustment to be made at this time, a wire may be stretched fore and aft along or near the guide surfaces, and at the same horizontal distance from each vertical center line. This will serve to show whether the guide surfaces stand in the right direction fore and aft, and hence, whether the plane of each is parallel to the central plane defined in (2) above.

In setting up attached auxiliaries, such as air, circulating or feed-pumps, the principles used will be similar to those already discussed, while the methods used will depend upon the particular circumstances of the case. If the seatings are located on the bed plate they will probably be planed at the same time as the main bearing gaps, and if the pump bases are faced at the same time they are bored, their axes will come at right angles to the seatings, and hence parallel to the axes of the main cylinders, and thus properly in line.

In order to test the line of the crosshead pins the connecting rod may be hung from the upper end and allowed to swing partially free at the bottom. Then, by turning the engine into various positions in the revolution and measuring the fore and aft clearance between the crank webs and the faces of the lower end of the rod when thus relieved of constraint, the accuracy of the adjustment can be determined. This operation may be carried out in the following manner: Assuming that it is desired to test the adjustment in the case of an engine completely set up, the cap on the lower end of the connecting rod is removed and the cross head is shored up in any convenient manner. The engine is then turned slightly by hydraulic jack or turning engine in such direction as to carry the crank-pin away from the rod and thus leave the latter free in a fore and aft direction, so far as direct contact with the pin is concerned. Measurements are then taken, and the engine is next moved around so as to bring the pin against its bearing again, and then on to a new position. Here the crosshead is again shored, the pin backed off measurements taken as before, and so on. In this way the piston, crosshead and connecting rod are carried around by resting on the crank-pin, and then in each position the connecting rod is freed by shoring the crosshead and backing off the crank-pin. In case the clearance between the crank-pin brass and the faces of the webs is too small to allow a possible irregularity of movements to show itself, the regular brass may be taken out and a dummy brass or wooden block, with sufficient clearance for all possible motion, may be fitted in its place.

In locating a line shaft, a wire or line of sight may be used in the general manner above described. After locating the center of the stern tube at the stern post, the line is run to the after end of the crank-shaft or on through to the forward end, as may be desired, such point being located by measurement according to the drawings of the ship, engine seating and bed plate.

In twin screw ships the centers at the after struts are located by measuring at the proper height the necessary distance outward from the center line of the stern post, at the same time squaring from this center line so as to bring both centers at the same height above the keel. The two lines are then set at the forward end at equal distances from the keel at the proper height, in accordance with the drawings. In most cases the shafts are not parallel with the keel, either in a vertical or horizontal direction, usually inclining outward from forward aft and either upward or downward, according to the size of the ship and other circumstances of the case.

In the examination of the adjustment of machinery which has been already in operation, as in the routine care of marine engines, it is not to be supposed that the preceding operations in all their details will be necessary. Judgment must be used, having in view the particular points to be examined, and the best means of effecting such examination in acordance with the general principles above discussed.

Thus the fairness of the line shafting must be tested occasionally, lest, as time goes on, wear in the bearings or changes in the structure of the ship may throw it seriously out of line. For this purpose the following method may be used: Three or more laths or battens are provided with a straight edge on one side. These are then placed across the shaft on the upper side, as nearly horizontal as judgment may indicate, being located at successive points along the line of shafting to be examined. A line of sight is then taken along the projecting lower edges of the battens and by adjustment they are brought parallel to each other. Then, if they can all be brought into one line the shaft itself is in line. If not, the shaft is out of line in the vertical direction, and, by blocking or other similar adjustment, until the battens are all brought into one line the amount by which the shaft is out at one point relative to two others is readily determined.

In a similar manner the line in the horizontal direction may be tested by holding the battens vertical against the side of the shafting. This method presupposes, of course, that the shafting throughout the part tested is all of the same size.

Instead of placing the battens on the shafting they may be placed on the couplings quite as well, supposing, of course, that the latter are all of the same size.

Another test which is sometimes used consists in slacking the coupling bolts at a coupling near which it is suspected that the shaft is not in line, and noting whether there is any tendency for the two parts of the coupling to open out on one side or another. Such an opening out then shows an unfairness in the line in one way or another, according to the side on which it appears.

[6] Valve Gear.

In the routine examination of the valve gear the points to be looked after are wear and lost motion in the excentrics and excentric straps, in the link block brasses of the Stephenson link, or in like parts of other forms of valve gear, in the various pin joints and connections, and abrasion or uneven wear in the valve faces or seats. In the case of the excentrics and straps and various pin joints, the wear may be taken up by the adjustments usually provided and in the general manner already outlined for other similar parts. Excessive or irregular wear in a flat slide valve or seat may require either resurfacing or renewal, according to the circumstances of the case. With piston valves, especially if not fitted with rings, excessive wear in either valve or seat will mean a serious steam leak, and will usually call for new parts, either for valve or seat or both.

[7] Thrust Bearings.

The most common trouble with thrust bearings is scoring or irregular wear of the bearing surfaces, or a general wear which may allow the thrust shaft to move forward sufficiently to put a pronounced thrust on the line and crank-shafts. This will result in heating and wear, and may throw the crank-shaft bearings out of line. A little clearance is usually allowed in the shaft couplings and in the fitting-up of the crank-shaft and crank-pin bearings, so as to insure freedom from end thrust for the crank-shaft as a whole. If the wear at the thrust collars should exceed this, then a part of the thrust would be transmitted to the crank-shaft, with effects as above noted.

With adjustable or horse-shoe collars this trouble is readily adjusted by moving the collars back by means of the adjusting nuts provided. With the plain type of thrust bearing as in Fig. 149 the bearing as a whole must be moved slightly aft by means of the screws provided for the purpose.

[8] Circulating Pump.

The engines for operating such pumps require the care necessary in all machinery of such type, the principles of which have been already discussed. The runner simply needs examination from time to time to make sure that it is running freely, but without undue clearance on either side in its casing. Trouble is often met with in the maintenance of circulating pumps by the choking of the inlet passage, valve or strainer by marine growth of one form or another. To aid in freeing the strainer and inlet passage a connection is often made with the delivery of one of the fire or other auxiliary pumps by means of which they may often be cleared without the need of regular overhauling.

[9] Condensers.

The chief troubles to be expected with condensers are as follows:

(1) Leakage about the tube ends through the packings from the salt water to the steam side.

(2) Corrosion and pitting of the tubes, resulting ultimately in the development of holes or the breaking of the tubes and similar leakage, as in (1).

(3) Fouling of the tubes with oil deposit on the condensing surface, thus decreasing the heat conductivity of the metal and the efficiency of condensation.

The condenser heads or bonnets must therefore be taken off from time to time and the condition of the tube ends and packings examined with reference to the matter of leakage and general condition.

To test the condenser for leakage the main inlet and outboard valves should be tightly closed, and the connections to the low pressure cylinder and air pump closed by blank flanges. The condenser heads may then be taken off and the steam side filled with water while a watch is kept for leaks on the tube sheets at each end. It is also desirable to be able to put, by means of a hand-pump or otherwise, a pressure of 15 or 20 pounds per square inch on the contained water, thus making the development of the leak more certain. It is sometimes desirable to be able to identify both ends of the same tube. This may be readily done by passing a wire through from one end to the other. In some cases a lamp held at one end will serve the same purpose.

When provided with bonnets for the purpose, the steam side of the condenser must also be occasionally opened and the tubes and interior surfaces examined, with reference to grease coating and general condition. Where there may be any doubt as to the condition of the tubes in the interior, a few should be drawn when the heads or head bonnets are removed, and their condition determined.

So far as the accumulation of grease is concerned, the condenser may be cleaned by the use of hot soda or lye water, care being taken to wash it out thoroughly so as to remove any excess of the alkali. When soda is used for the boilers it is sometimes introduced into the condenser, there entering the feed water and then passing on to the boiler. It may be questioned, however, whether this is the best plan, as accumulations of grease only partly converted to soap may thus be carried into the boiler, there giving trouble, as already referred to under that head.

Zinc plates are often used on the salt water side in condenser heads to protect against corrosion, in a similar manner as explained for boilers in Sec. 40. The condition of these plates and of their attachment to the shell should be carefully noted when the condenser is opened, and such repairs made as may be required.

[10] Air Pumps.

The head, foot and bucket valves require the most frequent and careful examination. They often tend to become coated by an accumulation of a black, greasy paste formed from the cylinder oil and the material of the wasted zinc plates, if such are used. This accumulation may prevent their proper working, and they should be carefully cleaned as occasion may offer. With air pumps attached to the main engine, and where the number of strokes is usually greater than with the independent pump, the valves sometimes give trouble by severe pounding against their seats and guards. This is due to their inertia, and is likely to be more severe as the valves are heavier and have more lift. Light metal valves of sufficient number and size to allow of moderate lift are therefore to be preferred for all such purposes. (See Figs. 185, 186).

[11] Pumps in General.

The chief troubles to be expected in the operation of the various forms of independent pumps found on shipboard are as follows:

(1) In the water end the plunger rings or packing or the barrel may become worn, thus allowing considerable leakage or "slip," especially under high pressure, as in boiler feed pumps.

(2) The water valves, either inflow or delivery, may become worn or deranged, and thus fail to hold the water as they should, so preventing the proper operation of the pump.

(3) In the steam end the piston rings or cylinder barrel may become worn, allowing steam to blow from one side to the other and decreasing the effective steam load on the piston.

(4) The main steam valve or more often some part of the auxiliary steam operating gear may become worn or deranged so that the pump can no longer properly operate under steam.

These various troubles must be guarded against by periodical examination of the points mentioned with a view to wear or derangement of any kind whatever.

[12] Piping.

Steam piping of copper if properly constrained may become brittle and weakened in spots by long continued expansion and contraction. An indication of this may often be found in the wavy or irregular condition of the surface. Such pipe should, of course, be replaced at the first opportunity. A repair may, however, be made by banding with screw clamps made of strap iron or steel and closely spaced over the suspected part.

Small holes which may sometimes develop may be treated with a soft patch held in place by a screw clamp, by filling with solder, or by a patch as in Sec. 42 [2] (10), according to the circumstances of the case. Leakage and like trouble with steel piping may be treated by the same general means as for boilers, and as discussed in the preceding section.

Sec. 44. SPARE PARTS.

In order to provide for the results of regular wear, and for the possibilities of accident it is customary to carry a certain number of spare parts, especially of those most likely to require replacement either as the result of wear or accident. The pieces carried and their number will depend entirely of course on the extent to which it may be necessary or desirable to fit out the ship with provision for such wear and emergency. No attempt will therefore be made to give any complete list of such parts, but among those more commonly carried the following may be mentioned: Grate bars, bearers and dead-plates, furnace and ash-pit doors, boiler tubes, manhole and handhole plates with fittings, safety valve springs, boiler gauge cocks, fittings for boiler water gauges, feed check valves, bottom blow valve, surface blow valve, piston and pump rods for the various pumps, valves, valve guards and springs for the various pumps, follower bolts, nuts and springs for the various steam pistons, brasses for the various bearings, horseshoes for the thrust bearing, propeller blades, valve stems, metallic packing for the various stuffing boxes where it is used, shaft coupling bolts, emergency shaft coupling, condenser tubes and glands, evaporator and distiller tubes, evaporator coils, one section of crank-shaft, if made in sections.

Sec. 45. LAYING UP MARINE MACHINERY.

The chief dangers to marine machinery when not in use arise from the likelihood of rust and corrosion. Fundamental principles relating to these chemical changes have been discussed in Sec. 40, and by reference to that point it will be seen that the chief points to be attended to relate to the protection of the surface from moisture and corroding acids. Where applicable a good metallic paint well laid on will be found the most efficient and satisfactory. In such case the surfaces must be dry and well cleaned in accordance with the principles discussed in Sec. 42 [1] (10). For finished surfaces or bright work generally, where paint would not be suitable, a coating of heavy cylinder or other like oil may be used, or perhaps even more commonly a mixture of white lead and tallow in about equal proportions. Either of these will efficiently protect the surfaces and will remain for a long period of time without becoming too hard to admit of ready removal, especially with the aid of a little kerosene or other light oil.

One of the most important features connected with the laying up of marine machinery is the getting rid of water contained in the various cylinders, pipes, bends, valve chambers, etc. The draining off of the water is, of course, of importance relative to the question of rust and corrosion, but it may be of even still greater importance relative to the question of freezing and possible rupture of the chamber, casing, or pipe containing the water. Many a cracked cylinder or valve chest or globe valve chamber, or split in pipe elbow or bend, or in boiler or condenser tube, etc., has been due to incomplete drainage of water and subsequent freezing. In laying up marine machinery, therefore, where there is any liability of freezing, a systematic study must be made of the piping systems, pockets, etc., where water might collect and by freezing result in damage. These remarks apply especially to piping and fittings, to small auxiliaries, to the condenser, and to water-tube boilers. If the proper drains are not fitted and the water cannot be gotten out in any other way, then the necessary joints should be broken and the water removed in this manner.

CHAPTER VII.

Valves and Valve Gears.

Sec. 46. SLIDE VALVES.

[1] Simple Slide Valve.

In Fig. 207 VW represents a simple slide valve, supposed to be surrounded by live steam in the steam chest C C. P and Q

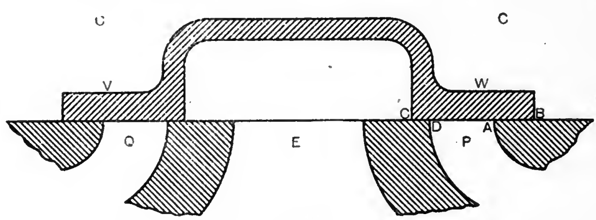


Fig. 207 Plain Slide Valve, Mid Position.

are the ports leading to opposite ends of the cylinder as shown, while E is the exhaust port or passage leading to the condenser or to the air, as the engine is condensing or non-condensing. It is the business of the valve, as we shall explain later, to move

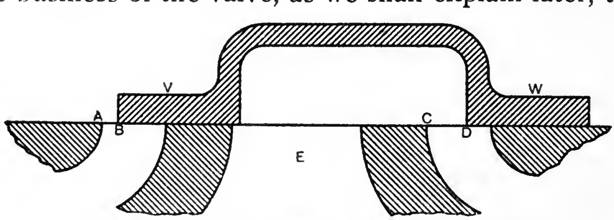


Fig. 208. Plain Slide Valve, Position for End of Stroke.

back and forth, thus alternately uncovering the ports P and Q and admitting steam from the chest to the ends of the cylinder. While the steam is thus being admitted at one end of the cylinder, it must be allowed to escape from the other to the exhaust passage E, and thus the piston is moved to and fro in the cylinder and the operation of the engine becomes continuous.

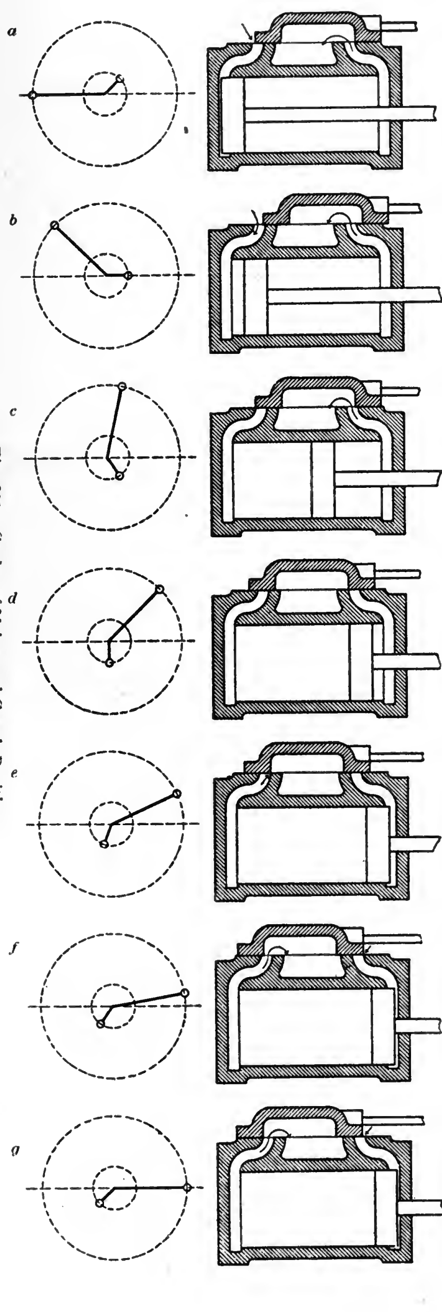


Fig 209. Series of Valve and Crank Positions.

If we suppose in Fig. 207 that the valve is in the middle of its travel back and forth, or in *midposition* as it is commonly called, then the distance A B by which the edge of the valve on the steam side extends over the edge of the port is called the *steam lap*. Similarly the distance CD by which the edge of the valve extends over the port on the exhaust side is called the *exhaust lap*. In Fig. 208 let the piston be at the end of the stroke and the valve in the position shown. Then the distance A B by which the port is uncovered to steam is called the *steam lead*, while the distance CD by which the other port is uncovered for exhaust is called the *exhaust lead*. In general, therefore, *lead* is the amount by which the valve is open when the piston is at the end of the stroke.

In Fig. 209 are shown a series of corresponding positions for valve and piston during a single stroke of the latter. The position *a* is the same as that of Fig. 208 and the port is open for the admission of steam on the left and for exhaust on the right. The piston is at the end of the stroke and just about to begin the stroke to the right. In *b* the piston has advanced about 10 per cent. of the stroke, and the valve has moved so that the port is nearly wide open. Up to this point the piston and valve have been moving in the same direction. In *c* the piston has advanced to about 60 per cent. of the stroke, while the valve has come back and has just closed the port to the entrance of steam. This is called the point of *cut-off*. Between *b* and *c* the piston has been moving on, but the valve has been moving back in the opposite direction. In *d* the piston has moved still further on while the valve is still moving to the left, and has just reached the point where the exhaust opening on the right is closed. This is called the point of *exhaust closure*, and the operation of compressing the steam in the end of the cylinder from this point to the end of the stroke is called *compression* or *cushion*. In *e* the piston is still nearer the end of the stroke on the right, while the valve has moved farther to the left, and is just about to open the port on the left for exhaust, thus allowing the escape of the steam which entered during the early part of the stroke. In *f* the piston has nearly reached the end of the stroke. The exhaust opening on the left is open still wider, while the port on the right is just about to open for steam. In *g* the piston is at the end of the stroke, the valve has moved so as to make the steam and exhaust openings still wider, and the return stroke is about to begin. This completes the history of the stroke, and the next or return stroke follows after it with like series of events, and so on continuously.

[2] Double Ported Slide Valve.

The valve shown in the above diagrams is called *single ported* because it covers but one set of ports or openings. A *double ported* valve is shown in Fig. 210. This is a form of valve within a valve. Thus taking one end of the valve we have at AB one set of edges respectively for steam and exhaust, and at A_1 B_1 another like set. Steam surrounds the outside of the valve and is therefore ready to enter the port P past the edge A when the valve moves sufficiently to the left. Steam likewise enters freely at the side into the passage S, and is therefore ready to enter the port P_1 past the edge A_1, as the valve moves to the left. Similarly as the valve moves to the right the ports P and P_1 are open to exhaust past the edges B and B_1, respectively. The passage E_1 leads over the transverse passage S, and thus the entire exhaust finds its way into E the outlet passage.

With a double ported valve the area of the port opening required may be obtained with a travel of valve only one-half that for a single ported valve, or with the same valve travel, twice the area of port opening may be obtained. It is this feature which often leads to the use of a double ported valve where it is desired to obtain a relatively large opening with small travel of valve.

[3] Piston Valve.

The face of the valves in the types so far noted is a plane, or in other words, they are *flat* slide valves. If now we can imagine such a valve wrapped up so as to form a cylinder with the valve

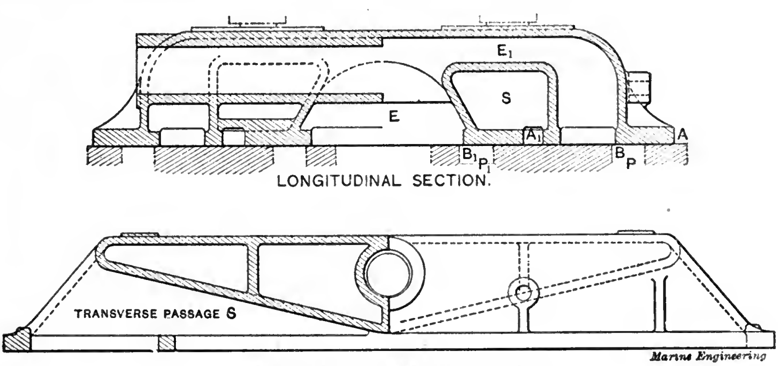


Fig. 210. Double Ported Valve.

stem for axis, we shall have a cylindrical or *piston* valve as shown in Figs. 211, 214, 215. This consists essentially of two heads connected by an intermediate body, as shown. The steam enters past the outside edges, for example, and exhausts past the inside edges to the exhaust passage, in a manner entirely similar to that for the plain slide valve as above described. Otherwise the steam may enter past the inside edges and exhaust past the outside as described below for the *inside* valve. The steam port and passage consists of an annular channel surrounding the valve and connecting with a passage leading to the end of the cylinder in the manner shown. The valve is placed somewhat out of center with reference to this annular passage, so that it is quite shallow on the side opposite the cylinder, and gradually increases in depth toward the side nearest the cylinder. This arrangement, which is shown in Fig. 212, gives a cross-sectional area of passage varying in proportion to the amount of steam flowing through it, as may be seen by noting in the figure the natural direction of flow of the steam, radially outward through the

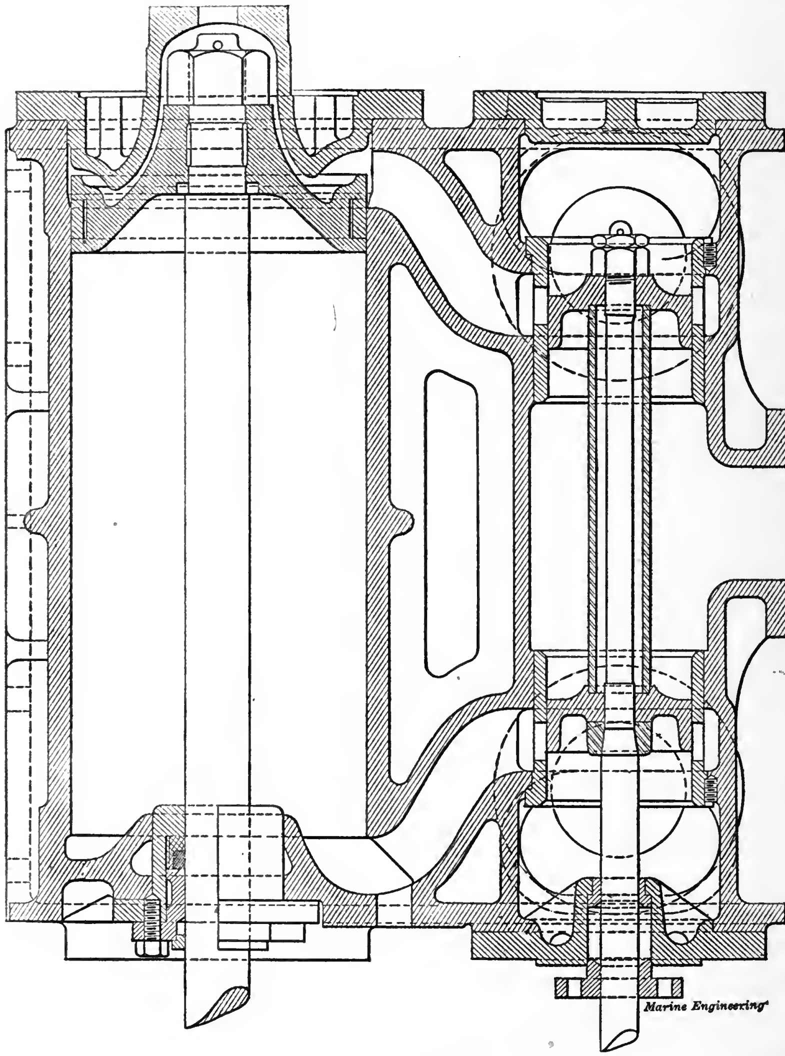


Fig. 211. Piston Valve.

port opening, and then curving around to flow toward the cylinder.

Comparing the two forms of valve it is readily seen that in the piston valve the outer circumference represents the active part of the valve, and corresponds to the plate AB of the flat slide, as shown in Fig. 210.

The great advantage of the piston valve lies in the fact that it is perfectly balanced as regards the steam pressures which act upon it. It is readily seen that the flat slide valve is forced against its seat by the excess of the pressure on its back over that on its face, which excess will in the usual case be large, and will give rise to a heavy frictional load to be overcome by the excentric acting through the valve stem. The piston valve, on the contrary, is forced equally in all directions, and hence moves

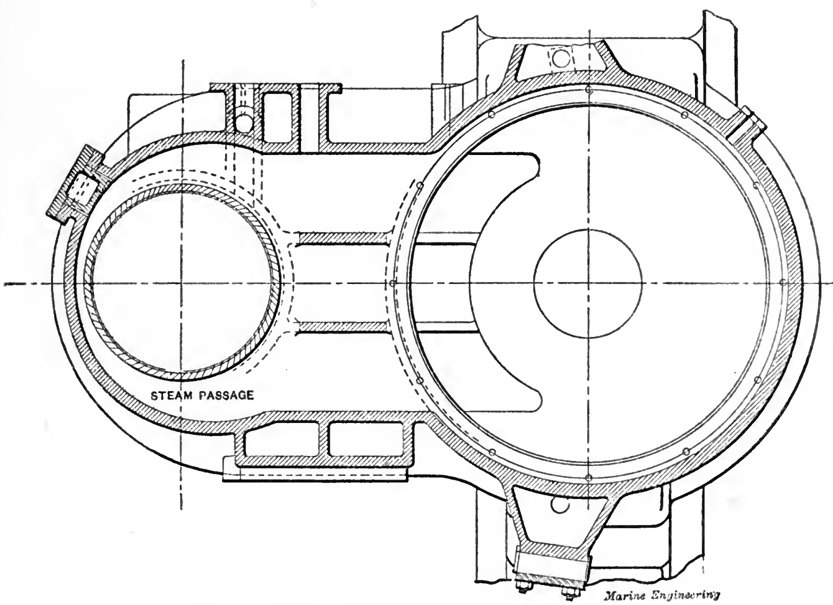


Fig. 212. Section Through Valve Chest and Cylinder.

freely so far as the steam forces are concerned, and with only such frictional resistance as may be necessary to insure tightness against steam leaks.

In order to keep the heads of the valve steam tight they have been very commonly provided with one or more packing rings of similar character to those used on the main piston, but usually without auxiliary steel springs to force them outward. Such an arrangement is shown in Fig. 215. In the latest practice, however, especially for quick-moving engines, the special rings are very commonly omitted entirely, dependence being placed on a good working fit at the start and on the rapid reversals of motion, to reduce the leakage to a negligible amount. Such arrangements are shown in Figs. 211 and 214. In the latter a solid working ring is fitted as shown, in such manner that it may be readily removed and replaced with a new one as occasion may require.

The valve seat is usually a separate piece of hard and fine grained cast-iron, fitted as shown in Fig. 211. The ports in this seat instead of being continuous all around, thus dividing the seat into separate parts, are usually bridged over at several points distributed about the circle. The head of the valve is thus carried across from one side to the other, and is prevented from catching or jamming, as would very likely occur without such bridging. Where valve packing rings are used it is especially necessary to provide such bridging in order to prevent the ring from springing out into the port opening, and thus jamming the valve, or causing other damage. In Fig. 213 is seen the development or lay-out of the port for the valve shown in Fig. 215.

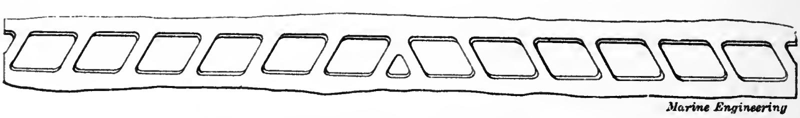


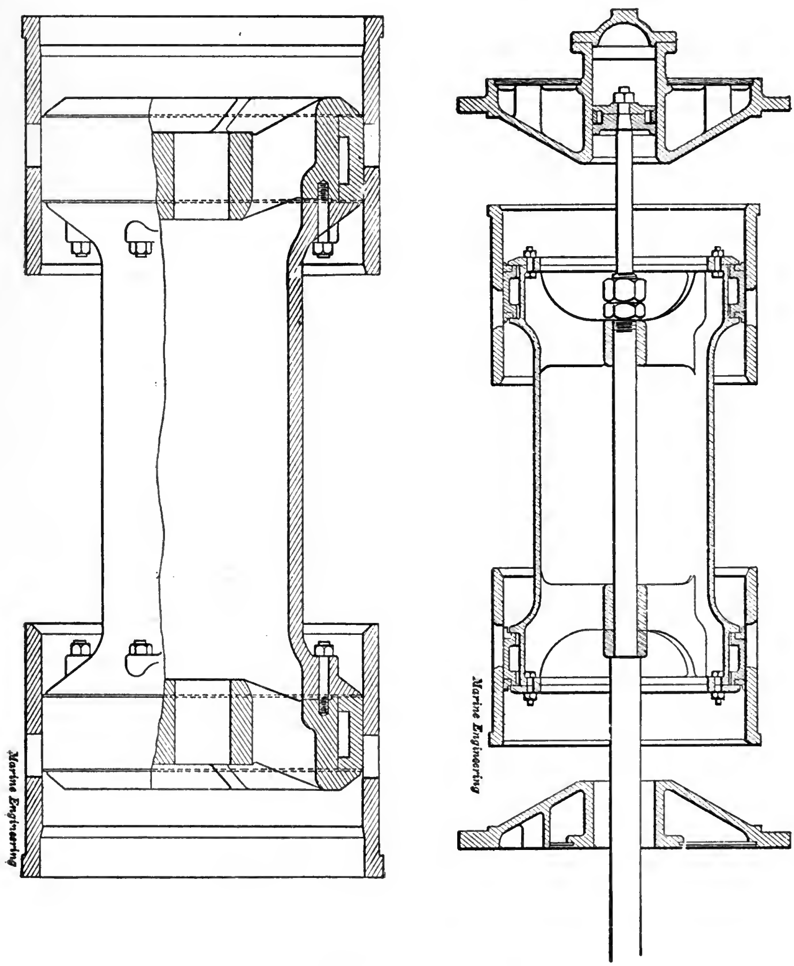
Fig. 213. Development of Piston Valve Port.

The bridges are placed on the slant so as to distribute as much as possible the wear on the valve rings.

In the case of wear on the valve seat or rings, they may be replaced with new. When rings are not used the wear is usually less rapid, thus furnishing a further reason for their omission. The valve itself, however, will slowly wear, and as necessity may require a new head or new valve entire may be fitted.

In some cases where a piston valve takes steam on the outside, it is desired to lead the steam to the chest at one end only, and then to pass it down to the other end through the inside of the valve. In such case as shown in Fig. 215 the body of the valve is made hollow and as large as possible, thus connecting the steam chest at top and bottom as desired.

The valve stem passes through the center of the valve and is usually secured with nuts at top and bottom, as shown in the figures. In case the valve is hollow for the passage of steam between the two ends, the stem must be carried in bosses supported by radial arms connected with the valve heads, and as small as possible in order to present the least resistance to the flow in either direction.



Figs. 214, 215. Piston Valves.

[4] **Equilibrium Piston.**

The work of moving the valve up and down which is thrown on the excentric may be much decreased by fitting an equilibrium piston as shown in Fig. 215 The cylinder in which this is fitted is open at the bottom to the valve chest, and hence the full pressure of the chest acts constantly on the lower side of the piston. This may be so proportioned as to carry practically about all the direct weight of the valve, which thus floats on the steam, requiring comparatively small effort on the part of the excentric to move it back and forth.

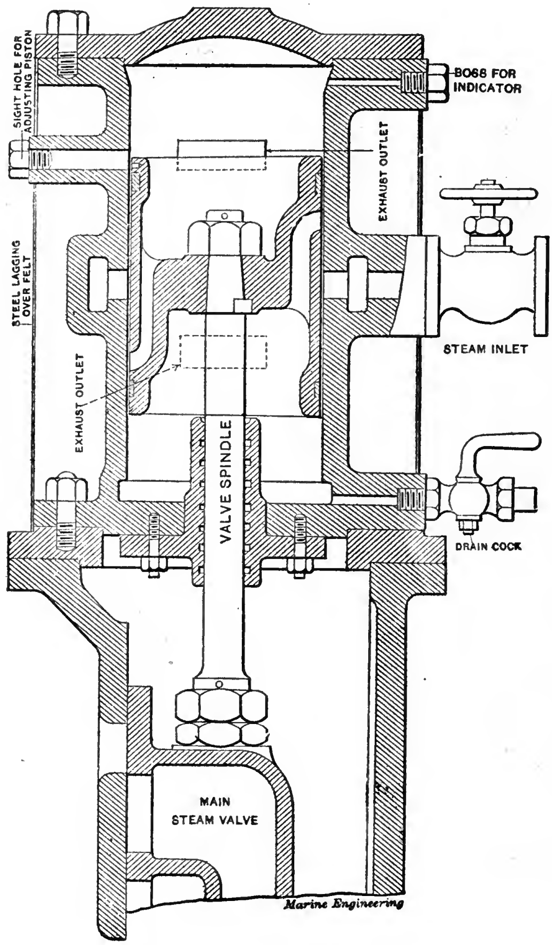


Fig. 216. Joy Assistant Cylinder.

[5] Joy's Assistant Cylinder.

A further development of the equilibrium piston is found in Joy's assistant cylinder, as shown in Fig. 216. The purpose of this fitting is to more perfectly carry the weight of the valve and relieve the excentric and valve gear of the work required to move the valve. The cylinder is provided with steam and exhaust ports, and in fact with its piston forms a small steam engine. The form of the piston as shown is such that it forms its own valve, thus simplifying the number of parts required. Power for assisting the main valve gear is thus obtained by the admission of live steam to the assistant cylinder, and the ports are so arranged that cushioning at the ends of the stroke in this cylinder absorbs the forces due to inertia. An important advantage claimed for such types of assistant cylinder over the ordinary equilibrium piston as in Fig. 215 is a considerable decrease in weight, the greater efficiency of the apparatus for the purpose in view allowing of the use of smaller sizes.

Such forms of equilibrium piston may, of course, be fitted to advantage with either the flat slide or piston forms of valve.

[6] Equilibrium Rings.

With a flat slide valve as in Fig. 210 the full steam chest pressure acts constantly on the back of the valve, while a much decreased pressure will act on a part only of the other side. In

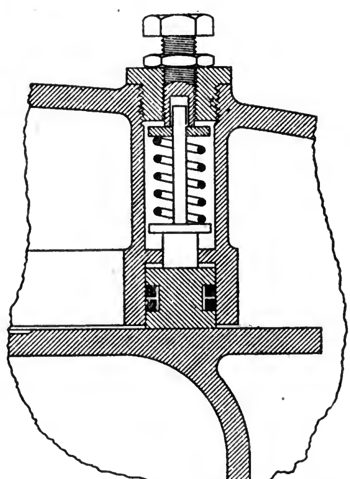


Fig. 217. Section Through Equilibrium Ring.

consequence, the valve is forced with strong pressure against the seat and the frictional force thus developed must be overcome by the excentric. (See Sec. 73). With large valves and where the lubrication is scanty this may become excessive, causing the valve and seat to cut, and throwing a great deal of unnecessary work on the excentric. To relieve this condition, equilibrium or balance rings are often fitted on the back of the valve. Such an arrangement is shown in Fig. 217. The inner face of the valve chest carries a ring of metal within which is cut a groove as shown. Within this groove is a ring which is forced against the back of the valve by springs and screw adjustment as shown. The back of the valve is faced off and thus a joint is made between the two, while the space within the ring and between the back of the valve and face of the cover is shut off from the steam within the valve chest, and hence this part of the valve is relieved from the pressure of the steam. In addition to this, the space is sometimes connected by piping to the steam side of the condenser, thus bringing on this part of the back of the valve only the pressure in the condenser. In this way the load on the back of the valve and the resultant load on the excentric may be much decreased, and the operation of the valves will be correspondingly smoother, and less work will be thrown on the excentrics and valve gear in general.

There are several methods varying in detail for the fitting of the ring in such an arrangement, and in the formation of the joint between the ring and valve chest cover, but the principle is the same in all, and is sufficiently illustrated by the arrangement of Fig. 217.

[7] Outside and Inside Valves.

In the preceding figures for valves the steam enters past the outside edges and exhausts past the inside edges. Such is known as an *outside* valve. In some cases, however, it is convenient to have this relation reversed, and to take steam past the inside edges, and exhaust past the outside edges. Such a valve is shown in Fig. 218, and is known as an *inside* valve.

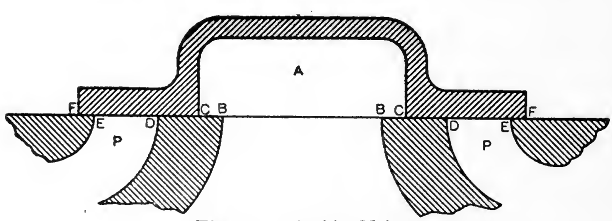


Fig. 218. Inside Valve.

Live steam fills the space A and enters the port P past the edge C, while exhaust occurs past the edge F. The valve as here shown is in midposition and therefore C D is the steam lap and E F the exhaust lap. The only difference in the two forms of valve is in the relative amounts of outside and inside lap. In each case, for reasons which will appear later, it is seen that the steam lap is greater than the exhaust, being in one case on the outside and in the other case on the inside. It is also clear that the outside valve moves with the piston at the beginning of the stroke, and opposite to it during the latter part, while with an inside valve it moves opposite to the piston at the beginning, and with it during the latter part of the stroke.

Sec. 47. MOTION DUE TO SIMPLE EXCENTRIC AND ITS REPRESENTATION BY VALVE DIAGRAMS.

[1] Simple Excentric.

We must now inquire by what means the valve can be given the motion necessary for the proper distribution of the steam as above described. The simplest of such means is the plain excentric as shown in Fig. 219. This consists of a circular disc with center A set on the shaft *excentric* or out of the center. The distance between the two centers is seen to be OA. This is called the *excentricity* or *throw* of the excentric.* About the excentric is a strap ST, and attached to this is a rod RR.

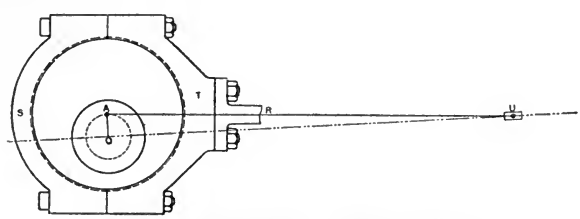


Fig. 219. Plain Excentric, Skeleton of Motion.

As the shaft turns, the excentric turns with it, and thus gives to the rod a to and fro movement exactly as though it were a connecting rod attached to a crank OA. This principle of the equivalence between an excentric and a crank of equal throw is very important, and the motion should be studied until it is quite clear that the operation of an excentric is exactly the same as that of a small crank of throw equal to that of the excentric, or to the distance from the center of the excentric to the center of

*Some writers understand by the term *throw*, twice the above distance. In the present work we understand the term to refer to the distance OA as stated.

the shaft. For purposes of illustration therefore we may represent the excentric by a crank of equal throw.

With this understanding let us again examine Fig. 209, on the left, when it is readily seen that the series of movements desired is exactly such as would be given by a small crank located at an angle somewhat more than 90 degrees ahead of the main crank. Or, in other words, with the valve stem attached to such a small crank or excentric, the valve will be so moved that the piston will be forced to and fro in the manner described in Sec. 46 [1], and the main crank will follow the motion of the excentric. This is what is meant by saying that with a slide valve connected as in the diagrams, the excentric *leads* the crank.

In regard to the angle between the crank and excentric it is clear from the diagram of Figs. 207, 209, that starting with the valve in mid position, it must move a distance equal to the lap before the port will begin to open. Hence when the piston is at the end of the stroke the valve must already have moved from its mid position by an amount equal to the lap plus the lead. Suppose now that the excentric is loose on the shaft and may be adjusted as desired. In Fig. 220 let C denote the crank on the center. Then suppose the excentric first located 90° ahead of O C as shown at O A, where O A = O B = excentric throw. Then neglecting the slight effect due to the obliquity of the excentric rod, the valve will be in its mid position, as shown in Fig. 207. Next, move the excentric ahead until the valve has moved a distance equal to the lap plus the lead as in Figs. 208, 220. This gives the final location of the excentric for the proper operation of the valve, as already described. The angle AOB through which it is thus necessary to move the excentric in order to affect this movement of the valve from its mid position, or more exactly the angle between the excentric and the line at right angles with the crank we shall term the *angular advance*. This term is sometimes used in reference to the entire angle between crank and excentric, but we shall prefer to understand by the term the angle as defined above. The angular advance is usually denoted by the letter δ. With the arrangement of gear shown in Fig. 220 the angle between crank and excentric will therefore be $90° + \delta$

In regard to the direction of motion of the crank, it is clear that when the piston is at the end of the stroke and the crank on the center, the latter must start off in such direction as will increase rather than decrease the opening of the port for steam.

If the connections and arrangements of a valve gear are known no matter how complicated, this principle will always furnish an answer to the question as to the direction in which the engine will turn. Thus in the direct connected gear, as in Fig. 220, with the piston at the end of the stroke, the crank moves in the same direction as the excentric, or follows it, simply because that is the direction which opens the port still wider for steam admission.

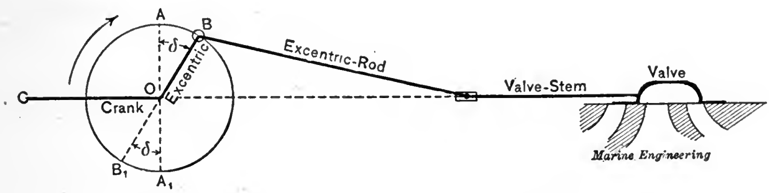


Fig. 220. Diagram Showing Connections for Simple Valve Gear.

With an inside valve the excentric is set 180° behind or directly opposite the position for an outside valve, as shown in Fig. 221, or at OB_1, Fig. 220. In such case the angle between the crank and excentric is $90° - \delta$ or the angle δ is set off *toward* the crank from the 90° position. It is readily seen that this will bring the valve slightly open when the piston is at the end of the stroke, and that the crank will move leading the excentric because this is

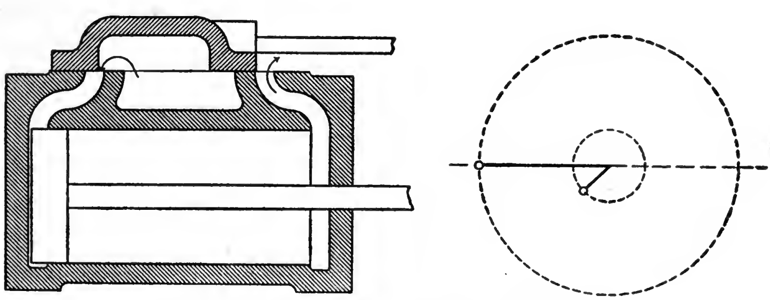


Fig. 221. Inside Valve and Location of Excentric.

the direction which opens the port wider for steam admission. It is also seen that to fulfil the same purpose the piston and valve at the end of the stroke must move in *opposite* directions.

In some cases the valve-rod instead of being directly connected to the excentric rod is worked through a rocker-arm, which reverses the motion as compared with the direct connected gear. See Fig. 222. This provides a second mode of variation

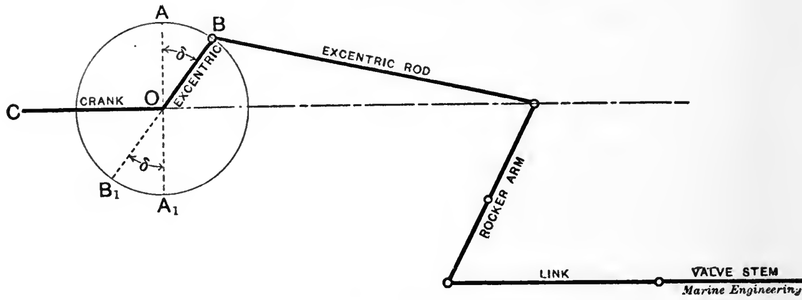


Fig. 222. Diagram for Valve Connections Through a Rocker Arm.

which may affect the arrangement of a valve gear, giving in all four combinations, as shown in the following table:—

Valve.	Connection.	Angle Between Crank and Excentric.	Which Leads.
Outside	Direct	$90 + \delta$	Excentric Leads
Inside	Direct	$90 - \delta$	Crank Leads
Outside	Rocker	$90 - \delta$	Crank Leads
Inside	Rocker	$90 + \delta$	Excentric Leads

[2] Oval Valve Diagram.

We will now proceed to an examination of the effects due to varying the steam and exhaust laps, and the angle of advance [1]. This may be done most conveniently by the aid of a diagram. In Fig. 223 let AB represent to any convenient scale the path of the piston. Then from the various points of AB corresponding to a series of successive piston positions, let the distance of the valve above or below its mid position be laid off, and the points thus found be joined by a continuous line. For a revolution or double stroke the result will be found somewhat as represented by the curved line of the diagram. Thus at the lower end of the stroke, say at B, the valve will be at a distance BT *above* mid position. When the piston has gone on to S it will be at a distance SJ,at R a distance RK, at P a distance PL, at N it will be in mid position, and will then pass *below* and reach a distance AV at the end of the stroke A. On the return stroke the valve will pass through a similar series of locations determined by the distances from AB downward to the curve. With this arrangement of diagram the movement of the valve *above* mid position is laid off *above* the line AB and *vice versa*. With an outside valve this shows *above* the line AB the events for the *up* stroke and below the line the events for the *down* stroke.

With an inside valve the relation is reversed, the events for the *up* stroke being shown *below* the line and for the *down* stroke *above* the line.

We have already seen that starting from mid position the valve must move a distance equal to the steam lap before the port begins to open for steam. Suppose then, that BF is laid off equal to the steam lap on the *lower* end of the valve, and a line FL drawn parallel to BA. Then it is clear that the remaining distances from FL upward to the curve ZTKL give the dis-

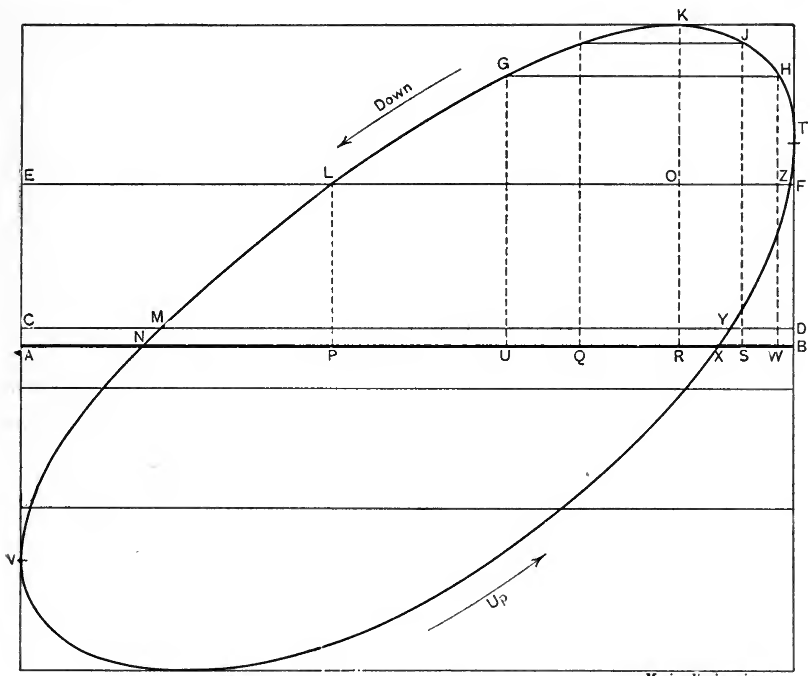


Fig. 223. Oval Valve Diagram.

tances which the edge of the valve travels beyond the first edge of the port, and hence if the port is sufficiently wide, the actual widths of port opening. At R the farthest distance is reached, and the edge of the valve is at a distance OK beyond the first edge of the port. It is clear that the distance RK is the throw of the excentric, and that this is equal to the lap plus the greatest width available for port opening. If the width of port is equal to or greater than OK, then the total movement OK can be utilized for opening and will be its greatest value, while the various distances from FL to ZTKGL will give the entire history of the width of port opening for corresponding positions of the piston. Very often, however, the width of port is less than the distance OK. In such case draw a line IJ parallel to LF and at a distance from it equal to the width of the port. Then it is clear that the widths of port opening will be given by the distances from FL to the lines ZTJIL. Full opening is reached at J or with piston at S. This continues to I, or until the piston reaches Q. The port closes at L or when the piston reaches P. This is known as the point of *cut off* or *steam closure*. Similarly the port opens at Z just before the end of the stroke, and at the end is open a distance FT, which therefore represents the steam lead.

Having thus obtained a general idea of the nature of this diagram let us examine by its aid the effects of a change in the steam lap. This corresponds to raising or lowering the line FL, and it is readily seen that the results of an increase, for example, are as follows: earlier cut-off, steam opening nearer the end of stroke, decreased lead, decreased port opening all the way through. The results of a decrease of lap are of course in the opposite direction respectively.

In a similar manner we may examine the influence of a change in the exhaust lap. This is illustrated in the same figure where M Y is the lap line laid off at a distance B D from the center line, equal to the exhaust lap on the upper end of the valve; that is, on the end which opens the port to exhaust by moving *above* the mid position. It is thus seen that the valve opens to exhaust at Y with the piston a distance Y D from the end of the stroke, while at the end of the stroke it is open a distance D T the exhaust lead. Then during the following stroke the distance of the exhaust edge of the valve from the corresponding edge of the port is given by the distance from D M to the curve T K L M. At M the port closes, and for M C, the remainder of the stroke, the steam undergoes compression in the cylinder. Usually the width of port is somewhat less than the total distance available, so that the full movement of the valve cannot be utilized for actual opening. In such case draw a line G H parallel to M Y and distant from it an amount equal to the width of port. Then full opening is reached at H with piston at W and continues to G with piston at U, the openings for the remaining portions of the stroke being given by the distance from M D to Y T H and G M.

An increase of exhaust lap is thus seen to produce the following results: earlier exhaust closure and longer compression, exhaust opening or *release* later or nearer the end of the stroke, decreased exhaust lead, decreased port openings or decreased time during which the port is wide open. A decrease of exhaust lap will of course, produce results in the opposite direction.

We have thus far been concerned with the influence of an increase or decrease in the steam or exhaust lap. It remains to examine the influence due to a change in the angle δ the angular advance. If this angle is *increased* and the new series of valve movements plotted as in Fig. 223, it will be found that the oval becomes narrower and touches the boundary lines nearer the

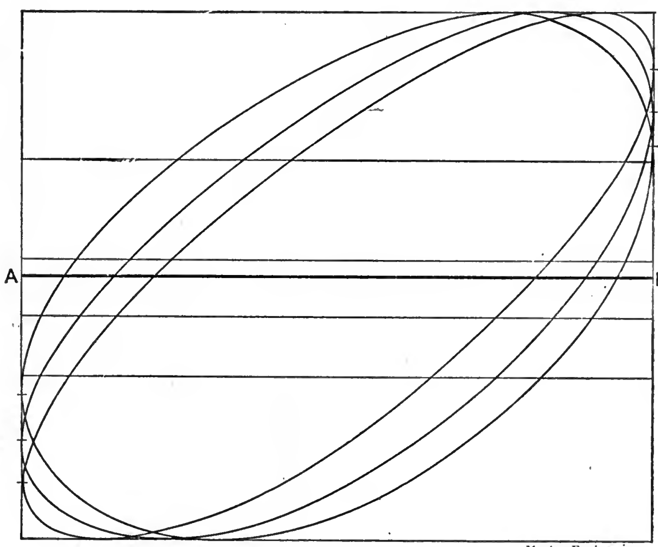


Fig. 224. Oval Valve Diagrams.

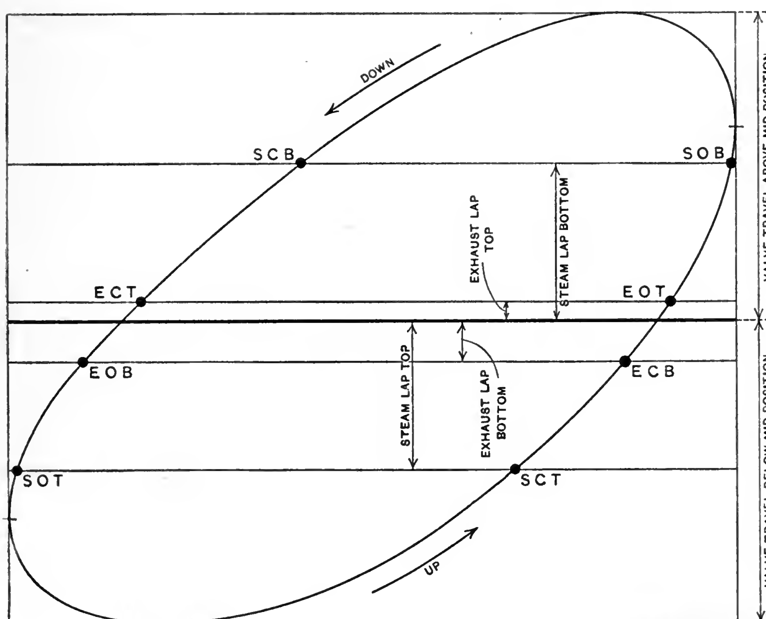
corners, as shown in Fig. 224. Similarly with a smaller value of δ the valve oval will become wider or rounder, and will touch the boundary lines farther from the corners, likewise in the figure as shown. Remembering this and comparing Figs. 223 and 224 it is readily seen with the same values of the lap that changes in the various quantities will take place as shown in the tabular arrangement below. This table shows at a glance the variation in the various events due to change in the three items above the double line, as explained in the foregoing.

Attention may also be called at this point to the diagram of Fig. 225, in which the various quantities for an outside valve are lettered and named in position. A careful study of this figure in connection with the valve positions of Fig. 209 will be of great aid in acquiring a good understanding of the operation of a slide valve, and of its representation by means of such a diagram.

If the measurements for the diagram are taken from an actual engine or from a properly constructed model, it will be found that for the down stroke the curve is more humped or

Angular Advance	Increase	Decrease				
Steam Lap			Increase	Decrease		
Exhaust Lap					Increase	Decrease
Steam Opening	Earlier	Later	Later	Earlier		
Steam Closure	Earlier	Later	Earlier	Later		
Steam Lead	Increase	Decrease	Decrease	Increase		
Exhaust Opening	Earlier	Later			Later	Earlier
Exhaust Closure	Earlier	Later			Earlier	Later
Exhaust Lead	Increase	Decrease			Decrease	Increase
Greatest Port Opening	Same	Same	Decrease	Increase		

rounded than for the up stroke, as shown in the figures above. That is, for the same relative positions of piston the valve will be farther from the center line on the down than on the up stroke. This is an effect due to the angularity of the connecting rod of the engine, and it results that the various points of opening and closure, and the values of lap, lead and port opening cannot be made the same for both strokes. As is shown by the diagram the cut-off is usually later on the down than on the up stroke, and it cannot be equalized without a serious derangement of the other events. Instead of attempting to equalize any two features such as steam lead, cut-off or port-opening, it is usually better to so adjust the steam and exhaust laps above and below that the resulting combination of events shall represent the best compromise possible under the circumstances. If it were not for the effect due to the angularity of the connecting rod, the curve would be an ellipse, and the distribution of the events for both strokes could be made the same.



S O B=Steam Opening Bottom. S C B=Steam Closure Bottom.
E O T=Exhaust Opening Top. E C T=Exhaust Closure Top.
E C B=Exhaust Closure Bottom. E O B=Exhaust Opening Bottom.
S C T=Steam Closure Top. S O T=Steam Opening Top.

Fig. 225. Oval Valve Diagram.

[3] Bilgram Valve Diagram.

In Fig. 226 let a circle be described with OP as radius equal to the throw of the excentric. Then draw the radius OP at an angle δ (the angular advance) with the horizontal or line AB. Then draw CD above AB at a distance LM equal to the lead. Then from the point P as center and with PQ as radius describe a circle tangent to CD. Also on OP as a diameter describe a circle as shown. Then let A_1 be any position of the crank. Draw the radius A_1O and extend it back to cut the circle on OP at E. Then the properties of this diagram are such that the movement of the valve from mid position is given by the distance PE, while the port opening will be less than this by PQ or PF the radius of the circle about P as center. This radius equals the steam lap, and the circle is for that reason known as the *lap circle*. It follows that EF is the port opening, or at least the travel of the edge of the valve beyond the edge of the port. The same construction holds for all other crank positions, so that we have here a means of representing by straight lines and circles the move-

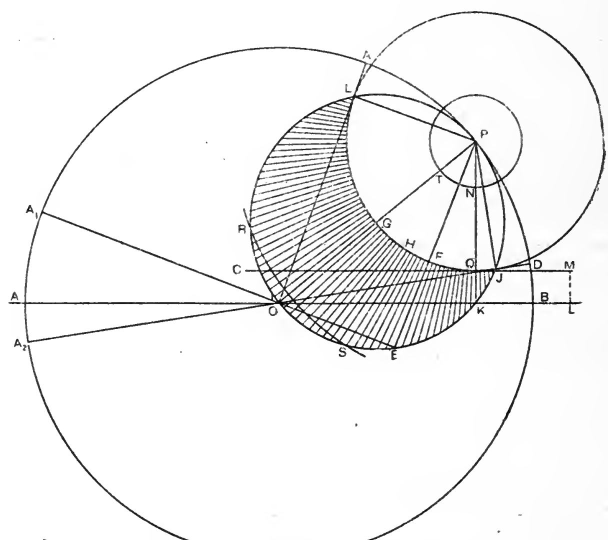


Fig. 226. Bilgram Valve Diagram.

ment of the valve, and of thus determining the various events of the revolution.

It must be understood, however, that while this construction shows with fair accuracy the relation between the movement of the *valve* and of the *crank*, it does not, without a further special construction, connect the movement of the *valve* with that of the *piston*. It is the special property of the oval diagram of [2] to show this latter relation for an actually constructed gear or for a model, where both sets of measurements may be actually made. It is the special property of the Bilgram diagram, and others composed of straight lines and circles, to connect together without actual measurement the movement of the *valve* and *crank* for the ideal case when there is no angularity of excentric-rod. The error here involved is usually small, and since the diagram is so easily constructed, it may be preferred for many purposes of initial design.

It thus appears that the distances of the edge of the valve beyond the edge of the port are given by the intercepts (shown by the shaded part of the diagram in Fig. 226), between the two circles, one on OP as diameter and the other, the lap circle, about P as center. In case the width of the port is less than the distance G O, an arc RS is drawn from P as center, such that the distance HS equals the width of port. The widths of opening are then given by the intercepts between J K S R L and the lap circle with P as center. It is seen that the port opens for steam at J with crank at A_2 and closes at L with crank at A_3, while full opening holds from S to R.

An entirely similar construction throughout with the exhaust lap circle as shown by the small circle about P will give the various features of the movement on the exhaust side.

The results of a change in the steam or exhaust lap, or in the angular advance δ are readily examined by the aid of this diagram, and will be found to agree with the statements of the table in [2].

[4] Zeuner Valve Diagram.

In Fig. 227 let ABCD be a circle described with radius OA equal to the throw of the excentric. Let A denote the angular position of the crank at the end of the stroke, and let OB be drawn perpendicular to AC. Draw OP at the angle δ (the angular advance) with OB, and on OP as diameter describe a circle as shown. Draw also an arc of a circle with OL equal to the steam lap, as radius. Let OA_1, be any position of the crank. Then the properties of this diagram are such that the travel of the valve from mid position is given by the distance OG, while the port opening will of course be given by EG the intercept between this circle and the lap circle MN. A similar construction holds for other locations of the crank, and it thus appears that steam opening will occur at N with the crank in the position ON while closure or cut-off will occur at M with the crank in the position OM. By describing a circle with center at O and with radius equal to the exhaust lap, a similar construction gives the various features of the movement on the exhaust side of the valve. The history of the port opening for steam is thus given by the intercepts between MN and the circle on OP, as shown by the shaded part of the diagram.

In case the width of port is less than the distance LP an arc RQ is drawn from O as center such that the radial distance between MN and RQ equals the width of port. The history of the port opening is then given in the same way as for the corresponding case in Fig. 226, as there explained.

The results of a change in the steam or exhaust lap, or in the angular advance δ, are readily examined by the aid of this

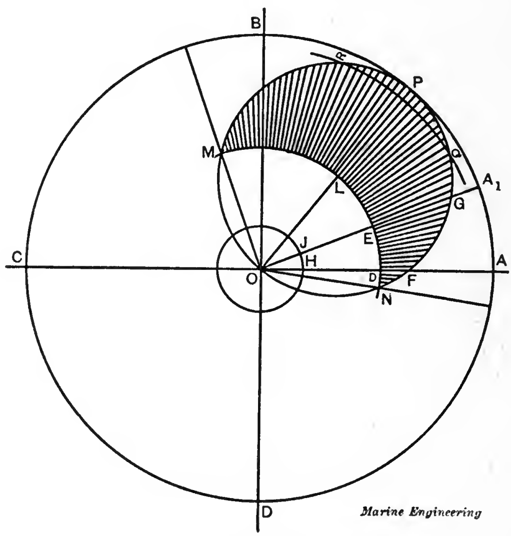


Fig. 227. Zeuner Valve Diagram.

diagram, and will be found in accord with the statement of the table in [2].

Sec. 48. STEPHENSON LINK VALVE GEAR.

We have thus far examined in some detail the operation of a slide valve operated by a single excentric. In the course of the discussion it has appeared that with such a gear the direction of motion of the crank relative to the excentric depends on whether the valve takes steam on the inside or outside, and on the nature of the connection between the excentric rod and valve stem. With any one arrangement, however, motion in one direction only is possible; and it is therefore clear that to enable the engine to reverse—a fundamental requirement of all marine valve gears—some additional features will be necessary.

The general problem of a reversing valve gear is one which has been solved in a great variety of ways, both with and without the use of excentrics, as we shall see later. With the use of excentrics the simplest solution is furnished by the well-known Stephenson link. This is illustrated geometrically in Fig. 228. C represents the crank on the center. A denotes one excentric at an angle COA on one side of the crank and B another at the same or approximately the same angle COB on the other side. AD and BE denote two excentric rods connected by a link DE curved to an arc whose radius is the length of the rod AD or BE. To a block in this link is attached the valve stem DG. For the structural details of this gear reference may be made to Sec. 53.

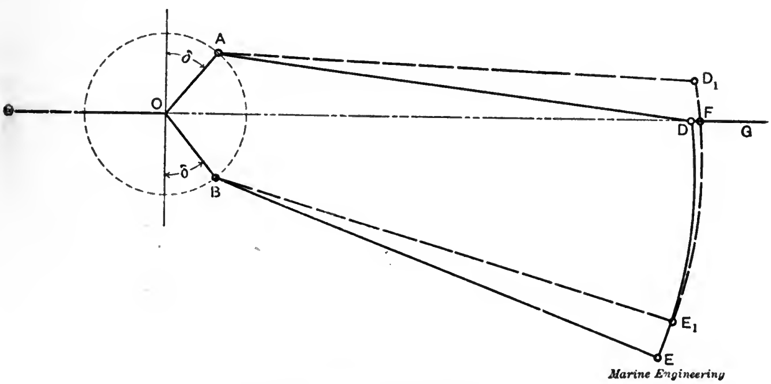


Fig. 228. Skeleton of Stephenson Link.

Now it will be readily seen that with the arrangement of the diagram the valve is under the control of the excentric A alone. The excentric B simply pulls the link DE back and forth, causing it to swing about D, but in no way affecting the movement of the valve. Hence the engine will move entirely under the control of the excentric A, and with an outside valve direct connected, the direction of rotation will be right-handed, or from C toward A. Now to effect the reverse, it is only necessary to move the link over so that E comes to the center line and the valve and engine pass under the control of the excentric B alone. For the reasons already noted in Sec. 47 [1] the motion will now be reversed and the rotation will be left-handed or from C toward B.

IN

We have next to require regarding the motion of the valve stem when the link is only part way over, as shown by the broken lines in the diagram. In such case it is readily seen that the motion of the valve will be derived partly from the excentric A and partly from B, the former giving the principal part of the motion, and the latter exercising a modifying influence. Now without taking up the examination of this question in detail, it will be sufficient to state that within a very small error the resultant motion will be the same as though it were given by a *single excentric* of somewhat decreased throw and increased angular advance as compared with A or B. A simple construction will serve to determine the throw and angular advance of this equivalent single excentric. This may be carried out as follows:

(1) In Fig. 229 lay off the two excentric throws OA and OB, with the proper angular advance as shown, and draw the line AB.

(2) Divide the length of the link DE Fig. 228 by twice the excentric rod AD and multiply the quotient by the length AC Fig. 229.

(3) Lay off the result from C to D, and through the three points A D B pass the arc of a circle, as shown.

Then the arc ADB may be considered as representing the link, and to find the throw and angular advance of the equivalent excentric for any given position of the link-block as F on $D_1 E_1$. Fig. 228, we have only to take a corresponding point F on the arc ADB, and draw the radius OF. The throw is then represented by OF and the angular advance δ by the angle POF.

That is, if the link be put into the position shown by broken lines in Fig. 228 the movement of the valve will be, within a small error, the same as though it were operated by a single excentric of throw OF, Fig. 229 and angular advance POF determined in the manner described. It is readily seen, therefore, that as the link is so moved as to bring the block from the end nearer and nearer the center, the corresponding point F Fig. 229 will move from A nearer and nearer to D, and the throw of the equivalent excentric will continually decrease while the angular advance will increase. As the link passes the center and the block approaches the other end the corresponding point F moves on from D toward B, and the throw again increases, while the angular advance changes to the other side and gradually decreases as B is approached. With the link in full gear at either end, the corresponding point F comes to either A or B, and the equivalent excentric becomes the same as the real excentric, with its throw and angular advance as constructed.

In some cases, especally with the double-bar form (see Sec. 53) the link may be put over so as to bring the block even beyond the points of attachment of the excentric rods. See Fig. 243. In such case the throw and angular advance of an equivalent excentric will be given by extending the arc beyond A and B as shown in Fig. 229, and by then taking a point F corresponding to the relative location of the block and link. In such case it is seen that the throw is increased and the angular advance decreased.

The details of the motion for any given position of the link-block may, of course, be determined by the use of any of the

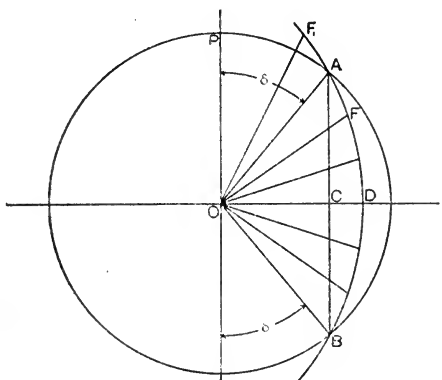


Fig. 229. Construction for Equivalent Excentric—Stephenson Link.

methods given above for the case of a single excentric. It is simply necessary to take the equivalent throw and angular advance determined as in Fig. 229, and use them according to the methods described in Sec. 47. In this way it may be found that as the gear is linked up, or the block approaches the middle, the valve travel and port openings decrease, the lead increases, and the cut-off becomes earlier. This is further illustrated by the two oval diagrams of Fig. 230. These are constructed as explained in Sec. 47 [2], and represent the effect of linking up. The larger diagram represents the movement for full gear position as shown by the full lines of Fig. 228, while the smaller and narrower one represents that for a linked up position as shown by the broken lines of the same diagram.

A gear arranged as in Fig. 228 is known as an *open gear* or gear with *open rods*. That is when the crank is turned *away* from the cylinder the rods are open as shown. If instead of this the rods are crossed as shown in Fig. 231, then it is called a gear with *crossed* rods. It will be noted that with the open gear the rods become crossed when the crank is turned toward the cylinder, while in the same position the rods in the crossed gear become open. It is therefore necessary to note the character of the gear by the appearance of the rods when the crank is turned *away* from the cylinder as stated above. It must now be remembered that the construction given in Fig. 229 and the

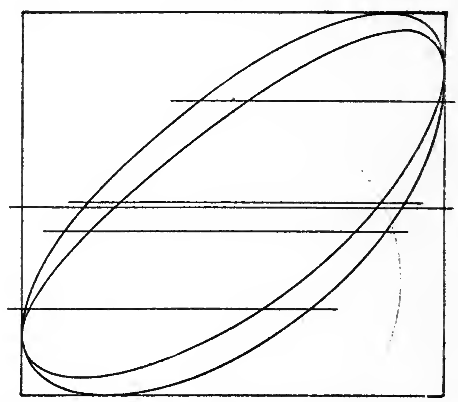


Fig. 230. Oval Valve Diagrams for Stephenson Link.

conclusions drawn from it apply to the gear with open rods only. For the crossed rod gear, however, a similar construction applies as shown in Fig. 232. The distance CD is here laid off toward the center and a like arc is passed through the three points ADB as shown. The diagram thus constructed is used in the same manner as with Fig. 229. It is thus seen as the link-block approaches the center of the link and the corresponding point F approaches D, that the equivalent angular advance increases, the equivalent throw, valve travel and port openings decrease, even more rapidly than with the open gear, while the lead decreases and the cut-off is earlier and earlier.

The principal characteristics of these two types of Stephenson link valve gear with regard to the effect on the various events, etc., due to linking the gear up may be conveniently presented in the following tabular form:

STEPHENSON LINK.

EFFECT OF LINKING UP.

Type of Gear.	Open Rods.	Crossed Rods.
Equivalent Excentric Throw	Decreased	Decreased
Angular Advance	Increased	Increased
Valve Travel	Decreased	Decreased
Port Opening	Decreased	Decreased
Lead	Increased	Decreased
Cut-Off	Earlier	Earlier

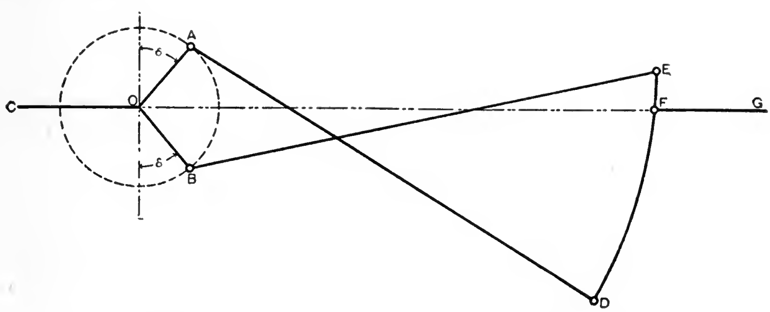


Fig. 231. Skeleton of Stephenson Link, Crossed Rods.

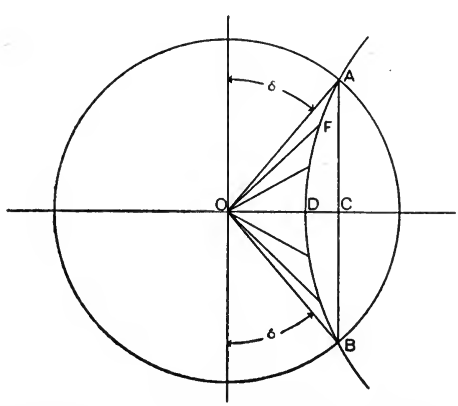


Fig. 232. Construction for Equivalent Excentric with gear of Fig. 231.

The chief point of difference is seen to be in the lead, which increases with open rods and decreases with crossed rods as the gear is linked up. While both types of gear are met with, the open rod gear is more frequently employed. This is due to the fact that as the cut-off is made earlier by linking up, an increase of lead may be preferred to a decrease, and also to the fact that the width of port opening is decreased less rapidly by the open than by the crossed gear.

Sec. 49. BRAEMME-MARSHALL GEAR.

The Stephenson link is by no means the only form of valve gear which will allow of reversal and which will give variable cut-off. Among the other arrangements are several known as "radial valve gears," and of these the more important may be briefly described.

In Fig. 233 let C denote the position of the crank and E the position of a single excentric located directly opposite the crank. Let FD be a slide pivoted on the horizontal line OX, and EB a rod attached to the excentric at E and fitted with a pin joint and block at P so that the block may move in or OUT on the slide FD. Then as the excentric moves around O the end E of the rod describes a circle, the point P describes a straight line FD back and forth, and other points between P and E describe paths intermediate between these two. For a point such as Q this is found to be an inclined oval as shown in the diagram. For points beyond P such as Q′, for example, the path is found to be a somewhat similar oval as shown. Now it is found that the proper motion for a valve can be derived from a point moving as Q or Q′ and that it is simply necessary to connect such point by a proper link to the valve stem as shown for Q.

Instead of placing the excentric at 180 degrees from the crank it may be placed with the crank, in which case in Fig. 233 we should consider the crank at C′. There are thus four arrangements of the gear according as the excentric is with or opposite the crank, and as the point Q is between P and E or beyond P.

In all cases the gear must be so adjusted that when the crank is on either center as C, the point P is on OX, or at the pivotal point of FD. It is readily seen that if this condition is fulfilled for one center it will be likewise for the other.

With regard to the kind of valve to be used (inside or outside) and the direction in which the engine will run for any given arrangement of gear, the same principles may be applied as in the case of the single excentric. Thus it is easily seen from the symmetry of the motion that Q will go as far above the center line as below, and hence that OX contains the middle of its vertical motion. Hence in the arrangement of the figure with the crank on the top center the valve is below its mid position by the distance from Q to the center line OX. Hence to take steam on the top of the piston an outside valve must be employed, and the engine will go in the direction which will open

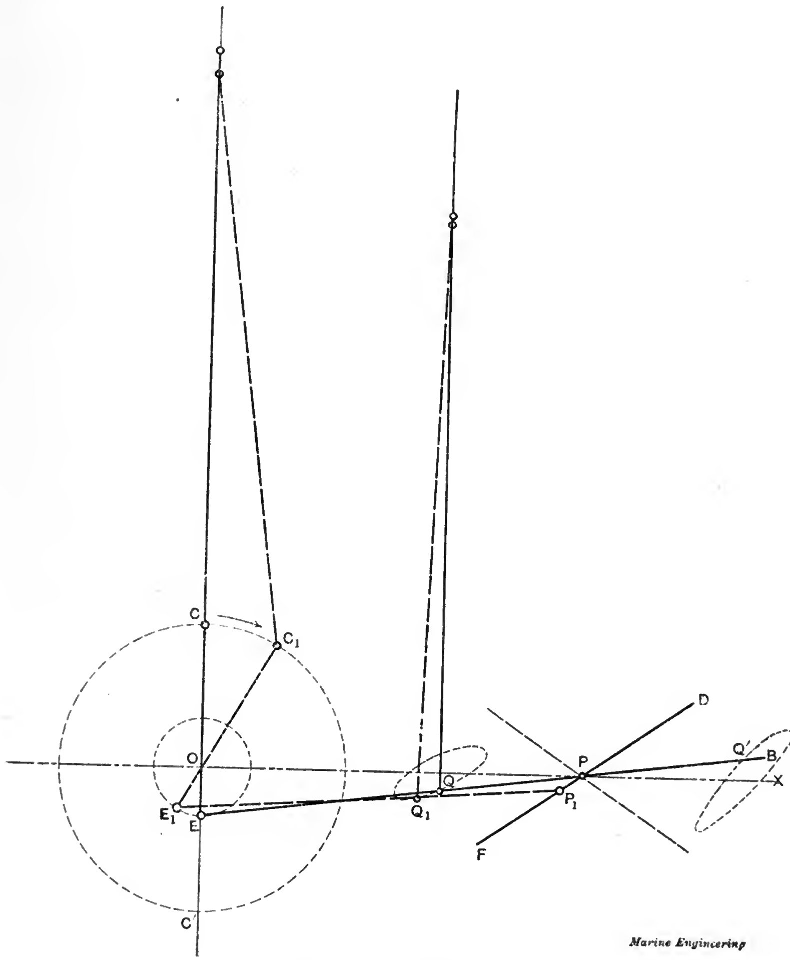


Fig. 233. Braemme Marshall Radial Valve Gear.

the valve still wider. This must be that which will lower the valve, and hence that which will lower P, and hence that which will carry E to the left, or right-hand rotation. If, on the other hand the motion were derived from Q′, the valve will be above its mid position, and hence to take steam on top it must be an inside valve. The engine in this case will turn in the direction which will raise Q′ still further. This requires E to move to the right, and hence the rotation will be left-handed.

If FD be inclined in the other direction, as shown by the dotted line, it will be readily seen by the same rule that the direction of rotation in each case will be reversed. It is also found that as the direction of FD approaches the horizontal or OX, the cut-off becomes earlier and earlier, and it is readily seen that the valve travel and hence the port opening will decrease. We have here therefore an entirely similar action to that which takes place in linking up with the Stephenson link. The means for changing the cut-off and for reversal are therefore furnished by providing a means for changing or reversing the obliquity of the slide FD, and for retaining it in any position desired.

Since the point P comes to the center of the slide or pivot point when the engine is on the center, it follows that in this position the line EB, the point Q and the valve will have exactly the same location no matter what the position of FD, and hence no matter what the point of cut-off. Hence the lead of the valve will be the same for all points of cut-off, or in other words, the lead is not affected by the change in cut-off. This is a feature which, as we shall see, is possessed by the various forms of radial valve gear. With the Stephenson link the lead is variable with the cut-off as described in Sec. 48.

As noted above there are four arrangements of this gear depending on the location of the point Q, and the angle between the excentric and the crank.

Angle between Crank and Excentric.	Location of Q.	Valve.
0	Inside P	Inside.
180°	Inside P	Outside.
0	Outside P	Outside.
180°	Outside P	Inside.

These four arrangements are shown in the above table with the appropriate form of valve. The direction of rotation in each case will depend of course upon the direction of obliquity of the slide FD.

The arrangement of Fig. 233 shows the earlier form of the gear. Another and later form is shown in Fig. 234 in which the slide FD Fig. 233 is replaced by an arm PT_1 pivoted at T_1, and attached by a wrist pin to the rod EB. In this way the point P is caused to move in the arc of a circle F_1 D_1 inclined to the

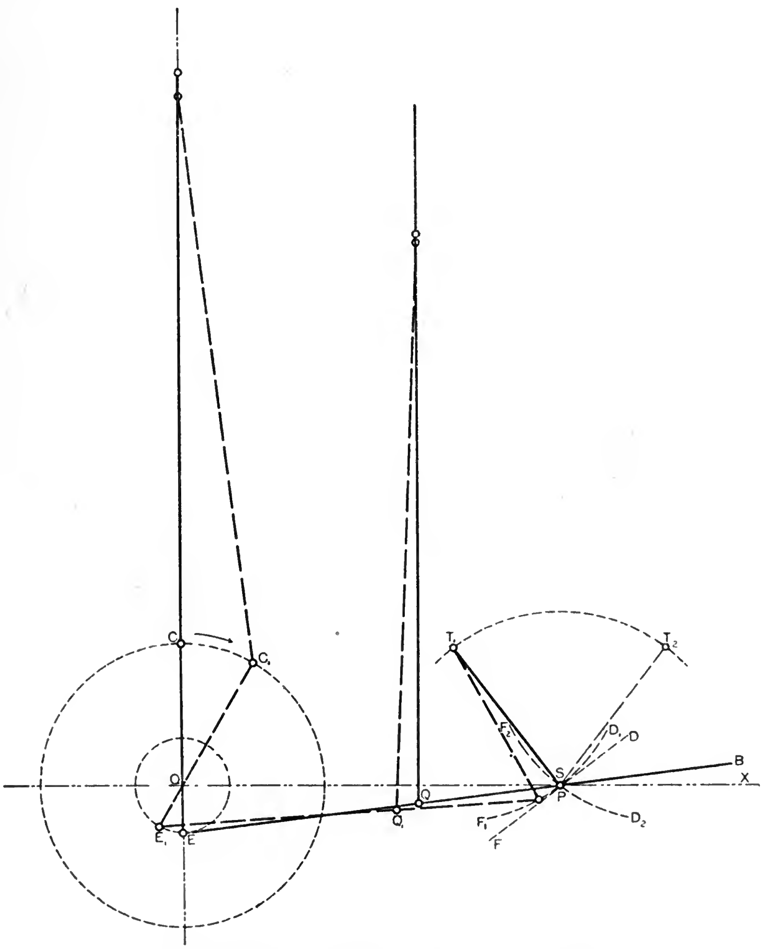


Fig. 234. Braemme Marshall Radial Valve Gear.

horizontal OX as shown. In such case the motion of the valve is very nearly the same as if the point P moved in the line of the tangent F D, and except for secondary modifications it is therefore equivalent to the motion of Fig. 233. The change of cut-off and the reversal are brought about by swinging the pivot T_1

about a center S, the intersection of FD with OX. In this way T_1 is brought to a new position T_2 and the line of motion is brought to F_2 D_2, thus reversing the direction of rotation according to the same principles as applied in the preceding case.

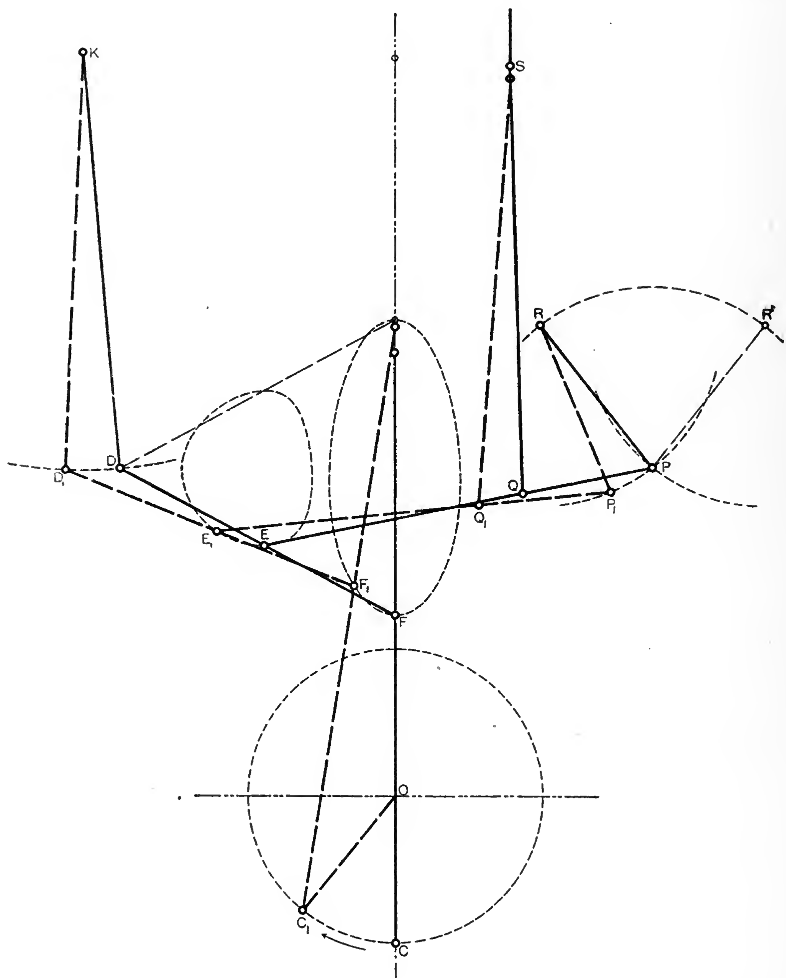


Fig. 235. Joy Radial Valve Gear.

We may evidently have here the same four arrangements as in the other form of the gear, with the same relation between excentric angles and type of valve as given in the table above.

Sec. 50. JOY VALVE GEAR.

In this form of radial valve gear no excentric whatever is required. As shown in Fig. 235 F is a point on the connecting rod to which a link DF is attached pivoted to a swinging or suspension bar DK. The point F moves in an oval path as shown. The point D moves in the arc of a circle as shown. An intermediate point as E will move in a path somewhat as shown. From this point on, the gear is similar to a Braemme-Marshall. Thus EP is a link pivoted at E and with the point P carried by a suspension bar PR exactly the same as PT of Fig. 234. The motion for the valve is similarly taken from a point Q as shown. It is thus clear that the Joy gear as here shown is the gear of Fig. 234 in which, however, the point E instead of moving in a circular path derives its motion from the connections shown in Fig. 235 and moves in a distorted oval path as there shown. It is clear that there will be the same two varieties of gear according as the point Q is taken between E and P or beyond P. The reversal is also effected in the same fashion by swinging PR, or the straight slide may be used as in the gear shown in Fig. 233.

Sec. 51. WALSCHAERT VALVE GEAR.

In this gear as shown in outline in Fig. 236, one excentric E is used. MGK is a curved link pivoted at G. H is a block sliding on or in the link and connected by a radius arm to the valve lever AF. The end F of this lever is connected by a pin joint to the valve rod, and the other end A by a short link to the main crosshead as shown. The valve is thus seen to derive its motion in part from the main crosshead and in part from the excentric. The former part comes through the valve lever which pivots about D and thus communicates motion from C to F. The latter part comes from the curved link which is operated by the excentric, causing it to swing about G and thus through the radius arm and valve lever the motion of the excentric is communicated to the valve rod. The combination of these two motions is found to be such as to give a suitable movement to the valve.

The linking up and reversal are accomplished by swinging the block H and radius rod from one side to the other of the mid-position or pivot G. As the block H is brought nearer to G the cut-off becomes earlier while the valve travel and port-opening are decreased the same as with the Stephenson link above described.

In order that this gear may operate properly, certain adjustments are required as follows:

The radius of the curved link MGK must equal the radius-rod HD.

Place the crank on say the top center, and bring the link

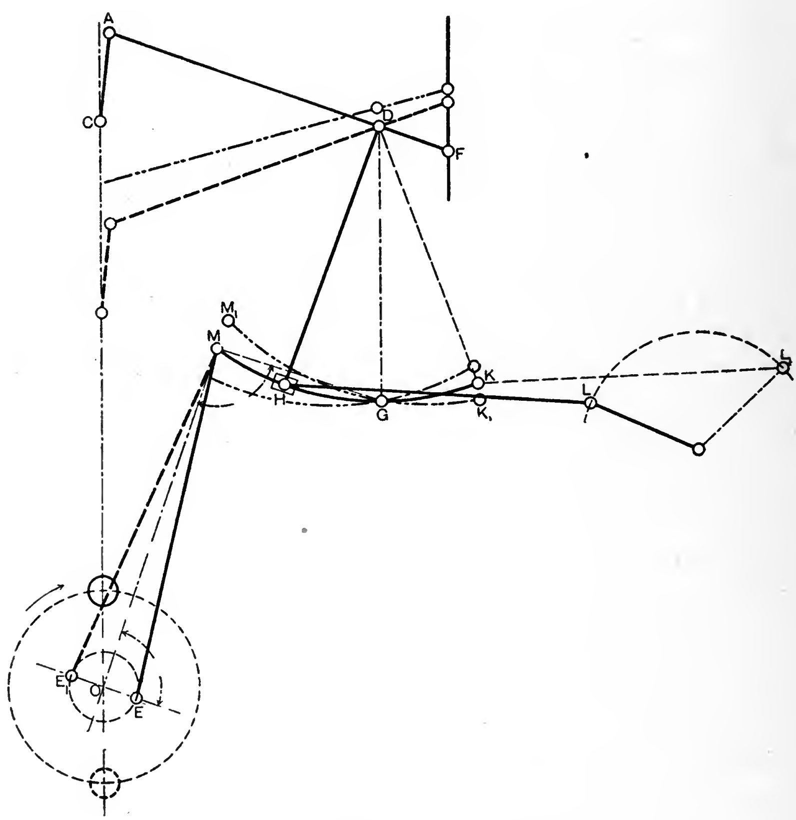


Fig. 236. Walschaert Valve Gear.

MGK into line with an arc struck from D as center with DH as radius. Then M should be so taken that the angle OMG is a right angle. Also the excentric E is placed at a right angle from OM as shown. Then in this position the block H may be drawn to and fro along the link without moving the valve. It is also clear that after the crank has gone 180 degrees the excentric will be at E, and MG will be again in the same position and the block H may be drawn along the link without giving motion to the valve. It follows that the position of the valve when the crank is on the centers will be the same no matter where the block H may be located, and hence that the lead of the valve will be the same for all points of cut-off.

It is clear that the arrangement of Figs. 233-236 will bring the valve chest on the side of the cylinder transversely instead of fore and aft. The cylinders may therefore, so far as valve chests are concerned, be placed nearer together than with the Stephenson link, in which case the valve chests are forward or aft of the cylinders. The length of the engine as a whole may therefore be made somewhat less with the radial types of gear than with the Stephenson link, and it is this feature which gives to them their best claim of advantage. Such gears possess also, as we have seen, the property of giving the same lead for all points of cut-off, while with the Stephenson link the lead varies with the cut-off. With proper design, however, the variation in the latter case is not sufficient to constitute a feature of any importance, and the difference on this point can hardly be considered as forming any noteworthy advantage for the radial gears. The general character of the valve movement and the distribution of events in all cases is best studied by the aid of a diagram such as that of Fig. 223. It will thus be found that the results for these various cases will be quite similar, and that such differences as appear are of relatively small importance. Speaking broadly, it is perhaps fair to say that there is not sufficient difference in the operation of the valve itself to furnish any pronounced claim of advantage for the usual cases arising in marine practice. The choice must therefore be made rather by reason of structural considerations, such as the shortening up of the engine referred to above, or the details of construction of the gear as affecting the questions of breakage, wear and tear, readiness of repair, readjustment, etc. On the whole, the Stephenson link seems to be usually preferred as the best fulfilling the all around requirements for the marine valve gear, and it may be fairly considered as the representative gear in present-day marine practice.

Sec. 52. CRANK VALVE GEAR.

As we have seen in Sec. 47 [1] the action of an excentric is equivalent to that of a simple crank of throw equal to the excentric and set at a corresponding angle with the main crank. It is evident then that a series of cranks will operate a valve and gear in a manner identical with a corresponding series of excentrics. Such cranks, however, on account of their small throw cannot readily be located or formed on the main crank-shaft, and hence where used for operating the valves, are necessarily placed on a special or auxiliary shaft. In Fig. 237 is shown the usual way of arranging the parts of this gear.

S is the center of the main crank-shaft, and ST the main crank. A, B, and C are gear wheels, A attached to the crank-shaft, B an idler, and C attached to the valve shaft. O is the center of this shaft and OP represents the throw and angular location of the small crank for operating the valve. The valve

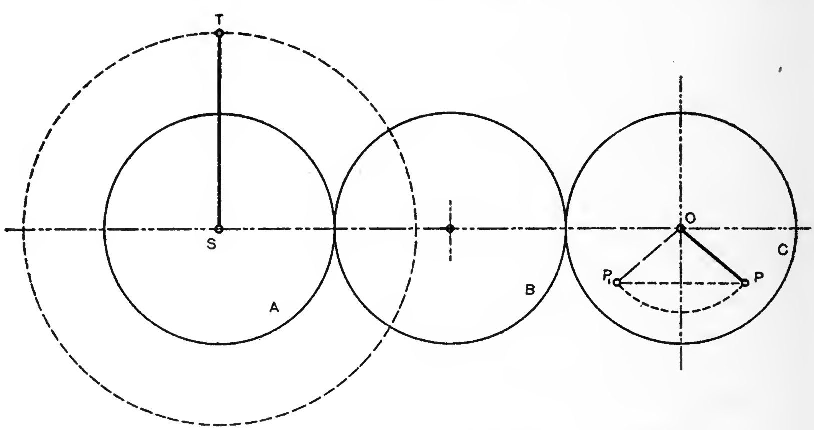


Fig. 237. Crank Valve Gear.

shaft will then turn in the same direction as the main shaft, and as may be readily seen, will operate the valve precisely in the same manner as an ordinary excentric of the same throw and angular location. For reversing with this gear the usual plan is to have but one crank and valve-connecting-rod corresponding to one excentric rod. The angular location of the crank must then be changed from a position such as OP in the figure to OP_1. By comparing this with Sec. 47 [1] it is seen that such a change in the location of the crank will necessarily cause the engine to move in the opposite direction. To bring about this change in the location of the valve crank relative to the main crank, various mechanical devices may be used.

Thus it may be seen that if after the engine has stopped the gear C were slipped out of the mesh with B, turned around through the angle POP_1 and then slipped back into mesh again, the crank would be brought to OP_1 and if the engine were started again it would go in the opposite direction. This, of course, is not a practicable form of reverse, since it cannot be carried out quickly enough, nor when the engine is in motion. It does, however, serve to illustrate the necessary change to be made in the angular location of OP.

In the usual mode of operation, some form of spiral cam is employed, as illustrated in Fig. 238. C is the gear wheel carried on a sleeve AB and connected to it by a key way and feather so that the sleeve may be moved back and forth axially and still remain coupled to the gear C so far as rotary motion is concerned. The gear C is also prevented by suitable stops from being carried out of mesh with the gear B, Fig. 237. This sleeve is carried on the shaft D to which is attached the valve crank,

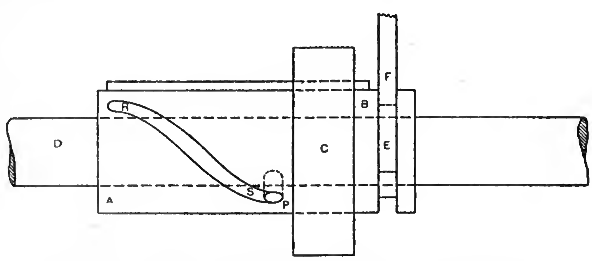


Fig. 238. Crank Valve Gear, Reversing Arrangement.

and is loose on D and only connected with it by a pin P which projects through a spiral groove RS cut in the metal of the sleeve. At E there is a circumferential groove in the sleeve, in which is fitted the end of a controlling lever F, by means of which the sleeve may be moved back and forth longitudinally. In the figure the sleeve is shown pushed in so that the pin is at one extremity of its travel in the spiral groove, and the valve-crank will therefore be in a corresponding position with reference to the gear C and main crank, which we will suppose to be for full gear ahead. If now the sleeve be pulled longitudinally until the other end of the groove contains the pin, it is clear that we shall have changed the angular location of the pin relative to the gear and hence of the valve-crank relative to the main shaft. If then the spiral groove be of suitable extent and location, such a change will serve to move the crank OP, Fig.

237, from its position for full gear ahead, to OP_1, its position for full gear astern. Instead of one pin and spiral groove, two on opposite sides may be used, and other details may vary in many ways, but the arrangement will serve to illustrate the principles involved.

While this form of valve gear is thus efficient for reversing, it is much less suitable for linking up or varying the cut-off than the other forms of gear discussed above. Referring to Sec. 48 it was there shown how to find the equivalent simple excentric for any adjustment of the Stephenson link, as shown by the line A D B, Fig. 229, for an open rod gear. Now it is readily seen that the form of reverse just considered is equivalent to taking the excentric O A and carrying it around from OA to OB, so that for the varying intermediate positions the virtual excentric would be given by drawing a line from O to the arc A B with OA as radius. From the principles discussed in Sec. 47 [2] it is readily seen that the effect on the valve will be as follows:

Linking up will produce:—

Earlier cut-off.

Earlier steam opening.

Greatly increased lead.

Earlier exhaust opening.

Earlier and greater compression.

The same valve travel and opening.

While therefore the port opening is not decreased by linking up, the lead and compression may become excessive, and restrict the practicable range of variable cut-off to narrow limits.

In Fig. 237 we have referred to one cylinder and valve operating crank only, but the same arrangement may be applied, of course, to a series of cylinders for a multiple expansion engine. In such case the valve operating shaft has a series of cranks, one for each cylinder, and set at the proper angle from the corresponding main crank. Then with a reverse gear similar to that described in Fig. 238 all the cranks are moved together and are brought into the proper relation with their main cranks to operate the engine in the reverse direction.

It is also to be noted that this arrangement brings the valve chests at the sides of the cylinders and that on this account it has the same effect in shortening up the engine as the use of a radial valve gear. This type of gear has been used to some extent on launch, yacht and torpedo boat engines, but has not found favor for larger or slower running engines as used in ordinary mercantile and naval practice.

Sec. 53. DETAILS OF STEPHENSON LINK VALVE GEAR.

In the present section we shall give a brief description of the more important details of the Stephenson Link valve gear as a representative gear, and including as it does most of the elements of other forms of gear as described in Sec. 49-52.

[1] Excentric and Strap and Excentric Rod.

The general construction of this part of the gear is shown in Figs. 239-242. The excentric consists of a disc or *sheave* of

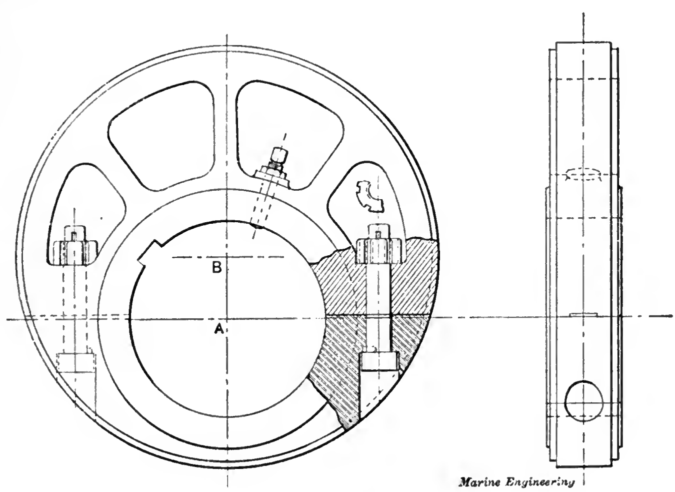


Fig. 239. Excentric, Detail of Construction and Fitting.

circular form, and placed on the shaft excentric or with its center set out from the center of the shaft. As shown in the figure the sheave is made in two unequal halves which join on the center line of the shaft, and are secured by bolts as shown. This brings the center of the disc at the point shown, the distance AB between this and the center of the shaft being the excentric throw as defined in Sec. 47. Once adjusted the excentric is keyed in place and in addition set screws or binding bolts may be fitted, as shown. In the excentric of Fig. 239 the larger part of the sheave is lightened out to save weight. In excentrics of moderate size this is not usually done, and the bolts connecting the two parts are tapped into the larger part and hold the other portion in place by means of a countersunk head. The material of the excentric is either cast-iron or steel, or if small in size, brass is sometimes employed.

The strap which surrounds the excentric as shown in Fig. 240 is also made in two parts bolted together, and the excentric rod is attached by a flanged foot to the upper half as shown in the figure. There are several ways of fitting the surfaces of the strap and excentric together, as shown in Fig. 241. In modern

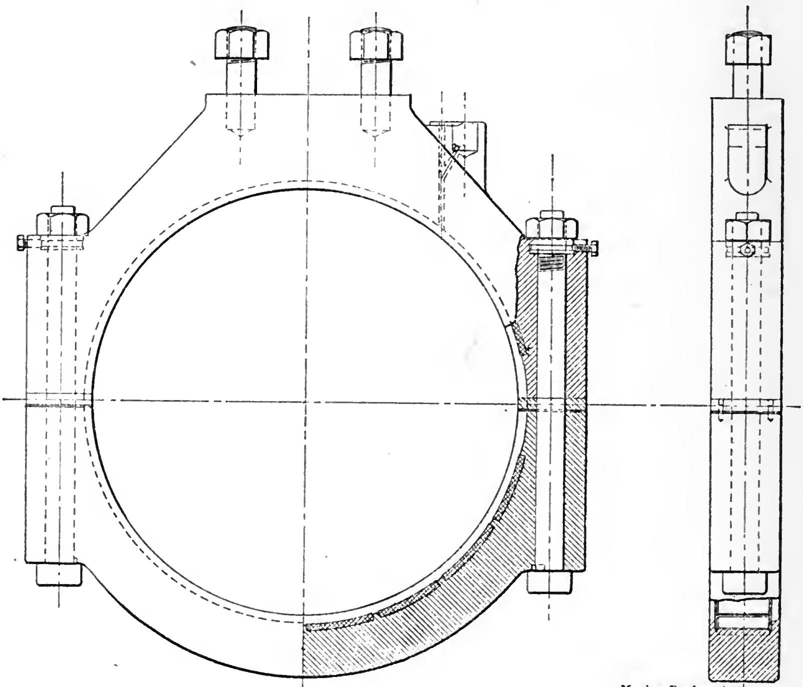


Fig. 240. Excentric Strap, Detail of Construction and Fitting.

practice the strap is usually of cast-steel or brass, lined with white metal for a bearing surface as shown at *a* and described in Sec. 21 [11].

With this arrangement of excentric sheave and strap, it is readily seen that as the former turns with the shaft the center of the sheave turns about the center of the shaft; also that the center line of the excentric rod which will always pass through the center of the sheave, will therefore move exactly as though it were a connecting rod with the distance A B between the two centers as a crank arm. Hence as noted in Sec. 47 [1] the motion communicated to the rod will be exactly the same as would be given by a crank and connecting rod the former of throw equal to that of the excentric. The other excentric strap and rod are

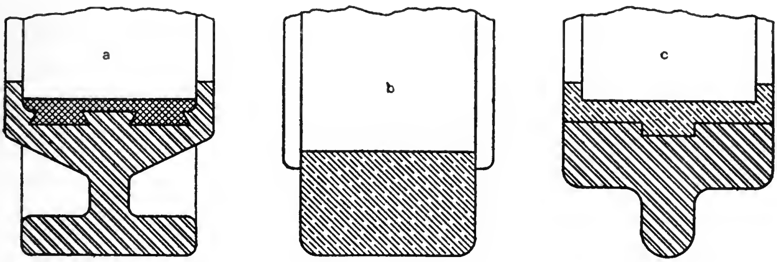


Fig. 241. Different Methods of Fitting Excentric Strap.

fitted up in the same manner and the two rods connect with the ends of the link in the manner shown in Fig. 242. In this figure is shown on the left the lower end of the excentric rod with flange for securing to the upper strap as in Fig. 240. On the

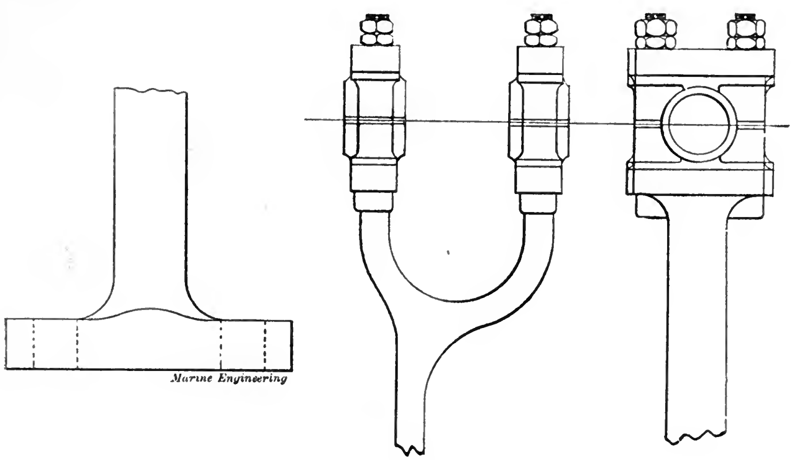


Fig. 242. Excentric Rods, Fittings at Ends.

right is shown the upper end of the rod sometimes known as the excentric rod fork. The form is more properly that of a U, the two sides being fitted with bearing brasses and cap to provide a bearing for the pins by means of which the connection is made with the link.

[2] Link.

In Fig. 243 is shown the usual form of *double bar link* as it is termed. It consists of a pair of bars curved in the arc of a circle of radius equal to the geometrical length of excentric rod; that is, equal to the distance from the center line of the link to the center of the excentric sheave. These bars are connected at the ends by bolts and by a block between so as to maintain the desired distance between them. Near the ends are fitted the pins A, B, C, D, which serve for connecting with the excen-

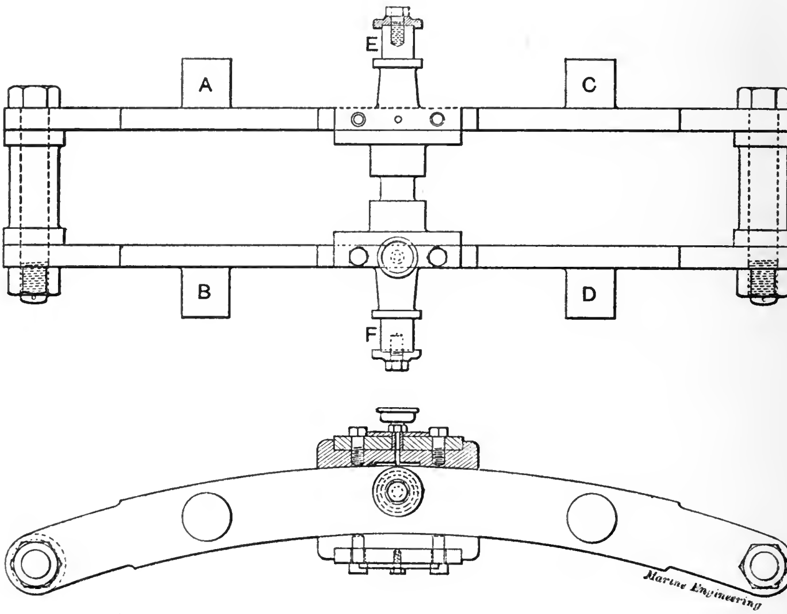


Fig. 243. Stephenson Double Bar Link.

tric rods by means of the bearings in the upper ends, as shown in Fig. 242. Another pair of pins E, F, is fitted, either at the center as shown or as an extension of the pairs at the ends, or at some intermediate point. To these are attached a pair of bars or links usually known as *side* or *bridle rods* as shown in Fig. 244. These lead to the *rock* or *weigh* shaft, and serve to control the gear when linking up or reversing, and to hold it in any desired position.

The rock-shaft or weigh-shaft is usually carried in bearings on the outside of the columns near the top, and is provided with arms, one each for the several links, and one for the connection to the reverse cylinder by means of which it is operated as desired. This is sufficiently illustrated in Figs. 100, 116, 247.

In order to provide an independent adjustment for the valve

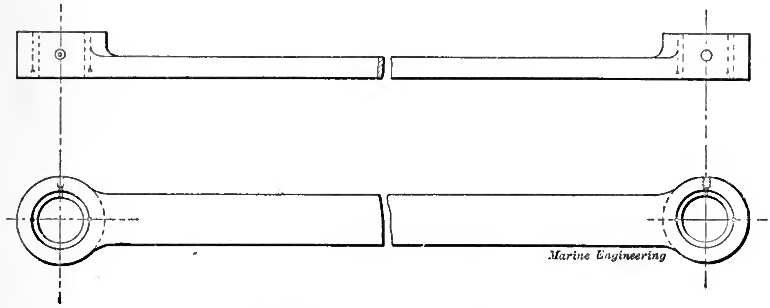


Fig. 244. Side or Bridle Rods.

gear of the different cylinders, the weigh shaft arm as shown in Fig. 245 may be provided with a slot within which moves a block under the control of a hand-wheel and screw as shown. The bridle rods are attached to pins on the sides of this block

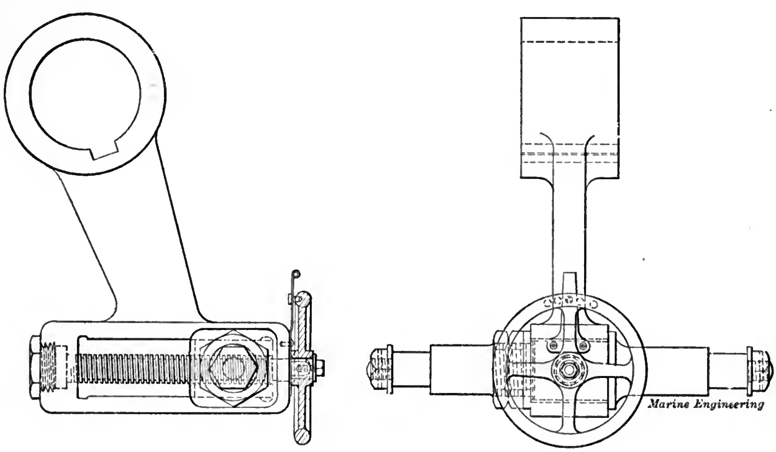


Fig. 245. Independent Adjustment for Cut Off with Stephenson Link.

as shown on the right, and by this means without moving the weigh shaft at all, the links may be given an adjustment within the limits of the motion of the block in the slot. It is customary to so adjust the line of motion of this block that when the gear is in the go-ahead position, it shall lie nearly in the line of the bridle rods so that any movement of the block will be communicated to the link without loss. In the backing position on the other hand, the line of movement of the block will lie across the line of the bridle rods at a considerable angle, and movement of the block back and forth will give but slight motion to the link. See also Figs. 100, 247.

This arrangement is often of use in adjusting the points of cut-off in the separate cylinders so as to divide as equally as possible the power among the different cylinders. See also Sec. 55 [4].

[3] Link Block and Valve Stem.

The connection between the link and the valve stem is made by means of a link-block as shown in Figs. 243, 246. This

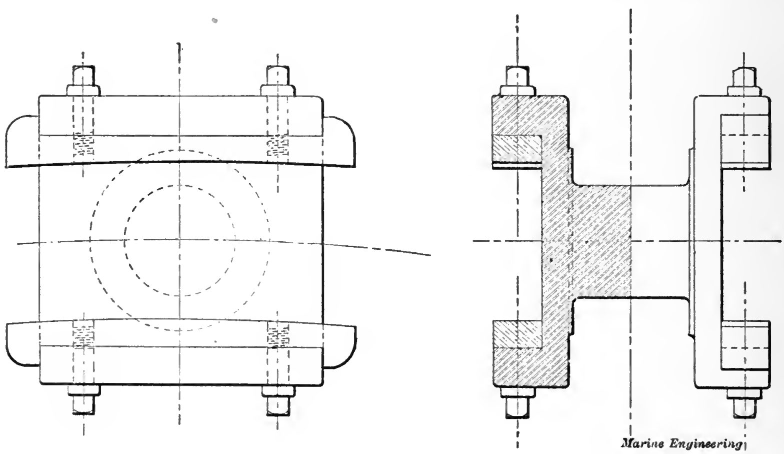


Fig. 246. Link Block for Stephenson Double Bar Link.

consists of a central pin with wing pieces at the ends, as shown in the figure, the latter being fitted with bearing surfaces for connecting with the bars of the link, and permitting sliding motion between the two.

The pin is connected with the lower end of the valve stem, which is formed in the usual manner with brasses and cap. See also Fig. 248. The valve stem is usually guided by means of a special guide or bearing as shown in Figs. 100, 247, which supports it against side stress, especially at the stuffing-box just above. In good practice the valve-rod stuffing-box is usually packed with some form of metallic packing, and is of the same general form and arrangement as the piston-rod stuffing-box described in Sec. 24 [7]. Passing through the stuffing-box the valve-stem is attached to the valve, and thus the chain of connections between the excentric and the valve is completed.

The assemblage of these various parts of a Stephenson valve gear is further illustrated in Fig. 247, showing the upper ends of the excentric rods, the link, link block, valve-stem and guide, bridle rods, rock-shaft arm and brackets for supporting the shaft, and independent cut off control in rock-shaft arm.

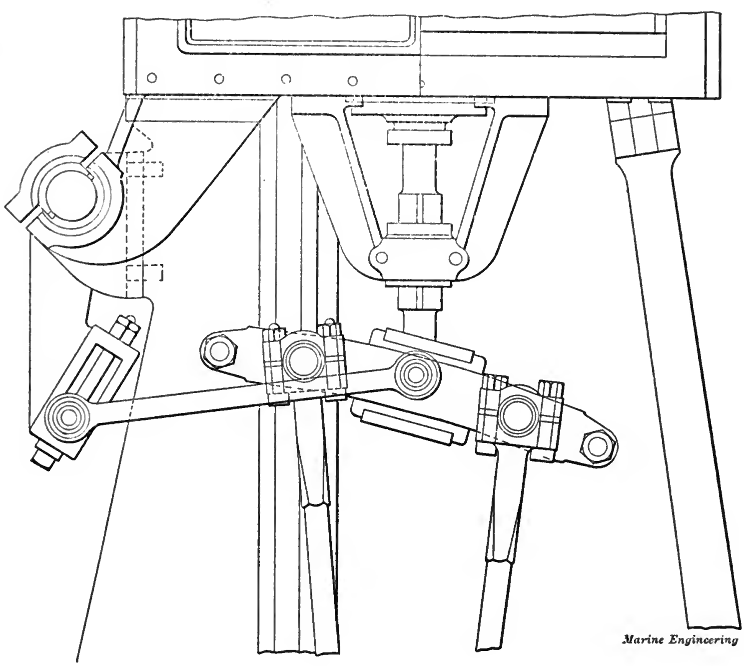


Fig. 247. Arrangement of Stephenson Link and Rock Shaft Connections.

Reference should also be made to Figs. 97, 99, 100 for further general illustrations of this gear.

When two piston valves are driven side by side as is very commonly the case on the L. P. or I. P. cylinders, the two valve-stems are connected across by a yoke as shown in Fig. 248, which in turn is connected to the link block by a form of bearing similar to that for the single stem. In such case the guide is very commonly attached to the yoke, the arrangement consisting of a dovetailed or gibbed slide and guide, the first formed on the yoke, and the second by a vertical plate or bar projecting

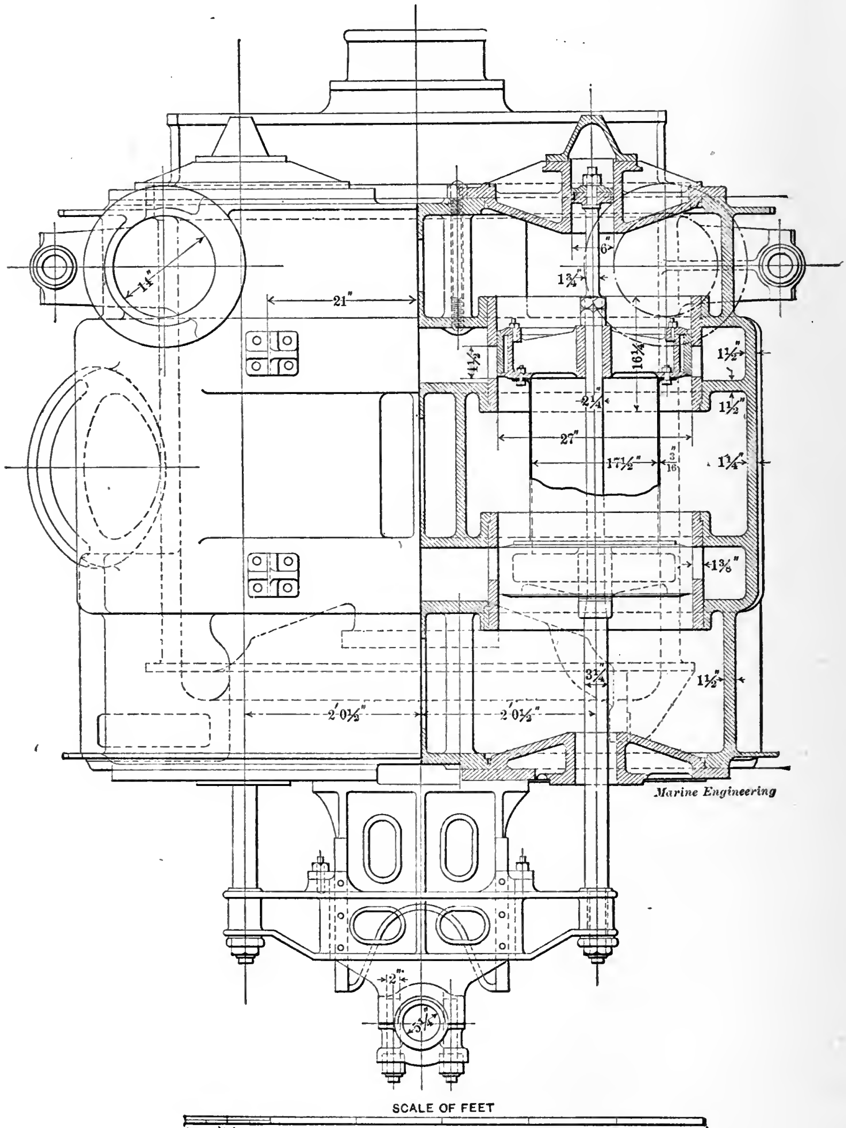


Fig. 248. Yoke and Guide for Driving Double Piston Valves.

downward from the bottom of the cylinder head as shown in the figure.

Sec. 54. VALVE SETTING.

[1] Putting an Engine on the Center.

One of the important features of valve setting is the placing of the engine on the centers or dead-points in order to determine the lead. In a rough way this may be done by turning the engine and watching the cross-head slide as it approaches the dead-points. The slide will move along the guide, more and more slowly, and will finally stop and begin to return. Just as the farthest point is reached, the crank is on the dead-point. By moving the engine back and forth and watching carefully the movement of the slide relative to a light mark or score on the guide, the desired point may be determined with fair accuracy for purposes of valve setting.

The difficulty in making an accurate determination by this method lies in the fact that when near the center the crank may

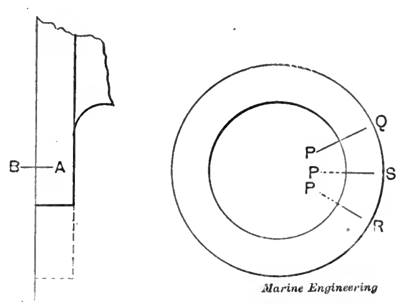


Fig. 249. Putting an Engine on the Center.

be moved to and fro through a sensible angle with hardly a noticeable movement of the slide. Hence while it is possible to determine to a nicety the point on the slide which corresponds to the highest or lowest position of the piston, it is less easy to know just where to set the crank so as to have it accurately correspond to the same location. For more accurate setting the following method may be used.

The engine is placed with the cross-head slide at a small distance from the lowest or highest position. A mark A is then made on the slide and a corresponding mark B on the guide. The distance which the cross-head should be placed from the center when these marks are made depends of course on the size of the engine, but 1 or 2 inches or say 1-20 to 1-10 the stroke will be a suitable distance. See Fig. 249. Another pair of marks P, Q, is next made on the forward end of the shaft and the adjacent brass, or on one of the coupling flanges and an adjacent block or bar set up for the purpose. The object is simply to have two pairs of marks, one on the cross-head slide and its guide, and one on the shaft and its support or guide or an adjacent and fixed object. The engine is then moved around continuously in one direction past the center till the cross-head slide moves back and the mark A again comes opposite B. The point P on the shaft in the meantime will have moved on to a new location, and a corresponding mark R is made on the bearing or stationary part of the engine on which the first mark Q was placed. The angle between these two marks Q R, on the shaft or coupling corresponds to the movement of the cross-head from the position of the first pair on to the end of the stroke and back again an equal distance. A mark S midway between Q and R, will give the proper location for the mark P when the cross-head is at the end of the stroke and the crank on its dead point. In this way by moving the shaft till P is brought opposite S the location of the crank for each dead point may be quite accurately found.

[2] Setting the Valve.

To return to the setting of the valve we first note that the distribution of the steam to a cylinder by means of a slide valve depends on four chief items:

(1) The throw of the excentric.

(2) The angular location of the excentric relative to the crank.

(3) The length of the valve stem.

(4) The steam and exhaust laps.

Now let us assume that the parts of the valve gear are made, and that it simply remains to connect them up and make the proper adjustments. It is seen that we have but two items which may be varied, or which may enter into the question of the *setting* of the valve. These are (2) and (3) above. We should first adjust for (3) and then for (2).

Referring to Sec. 55 [2] (9) it appears that an incorrect length of valve-rod will give an improper balancing up of the events for the top and bottom of the cylinder. Hence the various events for both ends must be examined and compared. To this end the entire gear is connected up according to judgment, the link being placed in the position intended for normal running ahead, and the necessary arrangements made for observing the movement of the valve. If the valve is a flat slide the valve chest cover is left off for this purpose. With piston valves, however, it is necessary to observe the movement of the valve by means of peep holes through the shell of the chest, such holes being fitted with screw plugs or covered by caps when the engine is closed up and ready for operation. In this manner the lead is observed while the engine is on the centers, and the points of cut-off and other items are observed for each end of the cylinder. The location of the valve on the stem is then varied until a fair balance between the two ends is obtained. It will be found that with anything like equal leads the cut-off will be later on the down than on the up stroke, or with an attempt to even the points of cut-off the lead and port opening on top will become too small and the lead on the bottom excessive. It will often be found that with something approaching equal leads on top and bottom, the points of cut-off will vary in the two ends by nearly or quite 10 per cent, or even more.

It is readily seen that similar derangements will result from an attempt to balance the exhaust items. In general it is far better not to attempt to exactly balance any one item in the two ends, but simply to aim for the best all around combination of events which can be obtained in the given case.

When a fair balance is thus obtained, the question of the point of average cut-off, steam-opening, release and compression may be taken up. Changes in these items require a change in the angular location of the excentric relative to the shaft, and it may be shifted according to the relations shown in the table of Sec. 47 [2] until the general character of the various items is made satisfactory.

It thus appears that of the two items, length of valve-rod and location of excentric, the latter really fixes the general character of the various items, while the former makes it possible to approximately even up or balance the various items between the two ends, according to what seems the most desirable average distribution.

If the excentric is shifted through any considerable angle from its first location it will be necessary to again examine the question of balance between the two ends, and to again adjust the length of valve-rod.

If by the adjustment of both valve-rod and excentric, the desired events, openings, etc., cannot be obtained, it means that the trouble lies with one or both of the other items, throw of excentric and steam or exhaust lap, and steps must be taken to modify these features as the conditions may require.

The following table gives an illustration of the balancing up of the various items in the two ends of the cylinder.

	TOP.	BOTTOM.
Steam opening	.7 per cent. before end of stroke	.4 per cent. before end of stroke
Steam closure or cut-off	68 per cent.	59 per cent.
Exhaust opening	90 per cent.	91 per cent.
Exhaust closure	85 per cent.	84 per cent.
Steam lap	2.44 in.	2.40 in.
Exhaust lap	— .12 in.	+.68 in.
Steam lead	.60 in.	.52 in.
Port opening for steam	2.08 in.	2.12 in.
Angle of advance	33 degrees	
Throw of excentric	4 15-16 in.	

The setting of the valve may of course be examined or re-adjusted at any time, as desired, by the use of the same general method.

[3] Valve Setting from the Indicator Card.

The indicator cards interpreted in accordance with the relations given below in Sec. 55 [2] furnish most valuable evidence as to the adjustment of the valve gear, and its suitability for operation under steam. In attempting a readjustment or re-setting by the aid of the indications given by the cards, the question of balance between the two ends as affected by the length of valve-rod should be taken first, next the items depending on the angular location of the excentric, and last the question of lap and excentric throw.

If the cards are pushed over to one side or show differences in the two ends as in Fig. 260 it is evidence that the valve stem is not of the right length, and it must be changed accordingly.

This is done first, cards being taken after each change until the two ends are fairly well balanced up. Attention is next given to the location of the excentric. The points of cut-off, release and compression will show whether the angular advance is too large or too small, and the readjustment is made accordingly. A change in the angle of the excentric for the purpose of adjusting any one item is moreover liable to disturb other items in such way as to require a readjustment of the lap, and is therefore to be avoided unless considered necessary. Thus if the cut-off is too late, for example, and the excentric is turned so as to increase the angular advance, the cut-off will be made earlier and the exhaust and compression as well, while the lead will be increased. If the change necessary to adjust the cut-off produces too great a disturbance in the exhaust and compression, or if in general a suitable and satisfactory arrangement of events cannot be reached by adjustment of the excentric and valve rod only, it means that the lap is at fault or perhaps the throw of excentric. Change in the lap can of course, only be effected by removing the valves and cutting them down if it is to be decreased, or fitting a new valve or head if it is to be increased. Similarly change in the throw of the excentric can only be effected by a removal of the old and fitting a new one of proper throw.

CHAPTER VIII.

Steam Engine Indicators and Indicator Cards.

Sec. 55. INDICATOR CARDS.

[1] Descriptive.

An indicator card is a diagram showing for each point of the stroke in both directions the steam pressure on the piston. Thus Fig. 250 represents an indicator card showing the steam pressure above the piston, say, for both the down and up strokes. RS is the line of zero pressure from which all pressures are measured upward according to the scale of the diagram. This is called the *absolute pressure* line. A is the beginning of the down stroke, B the point of cut-off, C the point of exhaust

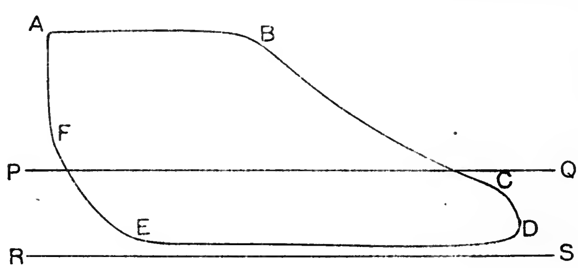


Fig. 250. Indicator Card.

opening, and D the end of the stroke. The line AB is called the *steam* line and shows the steam pressure on the upper side of the piston from the beginning of the stroke to cut-off. The line BC is called the expansion line and shows the decreasing values of the pressure during that part of the stroke. At C the exhaust opens and the pressure drops suddenly as shown by CD. For the return or up stroke, D is the beginning, E the point of exhaust closure or beginning of compression above the piston, and F the point of steam opening just before the beginning of the next down stroke. CDE is the exhaust line and shows the nearly constant pressure during this period. EF is the compression line and shows the increasing pressures on the return stroke after the closure of the exhaust valve. FA is the admission line and shows the sharp jump upward as the steam is opened again just before the beginning of the next down stroke. The line PQ drawn when the space below the indicator piston is shut off from the engine cylinder and connected to the air is called the *atmospheric line*. The distance PR between RS and PQ thus represents the pressure of the atmosphere, 14.7 pounds per square inch, its length depending of course on the scale of the diagram. Thus for the down stroke the varying pressures on the top of the piston are shown by the varying distances from RS to ABCD, while for the up stroke the pressures on the

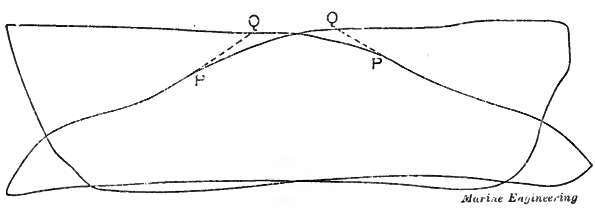


Fig. 251. Pair of Indicator Cards.

same side of the piston are shown by the distances from RS to DEFA.

There will be, of course, a similar diagram for the head end of the cylinder showing the pressures *below* the piston for both the up and down strokes in the same manner as for the diagram described. Such a pair of diagrams taken from actual practice is shown in Fig. 251.

Let us now compare the cards of Figs. 250, 251 with Fig. 252, the latter showing a so-called ideal card; that is, a card which would be given if the valves opened and closed instantaneously, if when closed they were tight against all leakage, if there were no loss of pressure due to friction of steam in the passage, and if the expansion and compression lines were equilateral hyperbolas. Instead of these conditions the valves open and close gradually, even when closed there may be some leakage, there is always some loss of pressure due to friction or resistance to the flow of steam, especially through a gradually closing or opening port and the expansion and compression lines are not true hyperbolas. Added to these we have the inertia of the indicator piston which prevents it from following with absolute exactness all the variations of pressure as they occur.

As a result of these various causes the actual engine and indicator give us the diagrams of Figs. 250 and 251 rather than such as Fig. 252. The gradual opening and closure of the valve rounds off the various corners, while the steam line instead of being horizontal, droops somewhat, due to the loss of pressure through the ports and passages. The piston, of course, moves faster as it approaches mid-stroke and hence the steam must flow in at an increasing velocity to fill up the space behind the advancing piston. The higher the velocity the greater the loss of pressure, and hence there is a continual slope down from the beginning of the stroke as shown in Fig. 251 and often to a far

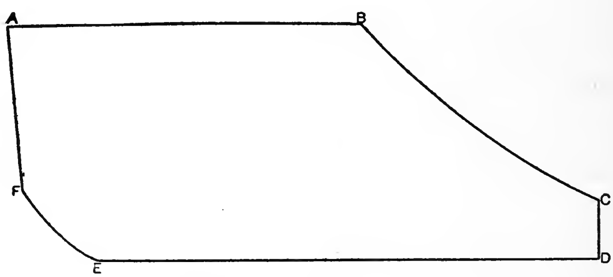


Fig. 252. Ideal Indicator Card.

more pronounced degree. The actual point of cut-off is also not always easy to locate, rounded off as it is by the gradual closure of the valve. We may, however, properly consider that the point of actual final closure is where the curve changes direction of curvature, that is, from convex to concave, as at or near P. Fig. 251. It is sometimes considered that the point of equivalent cut-off is more nearly obtained by continuing the curve back as shown by the dotted line to Q and supposing a sharp cut-off at this point. The result would then be an expansion line from Q similar to that which is obtained by the gradual closure in the actual case.

The steam engine indicator diagram is valuable for two chief purposes.

(a) It enables us to judge of the operation of the valve by noting the various events, steam opening and closure, the location relative to that of the piston, the resulting piston pressure, and to answer various questions relative to the general problem of the distribution of steam to the cylinder.

(b) It enables us to answer all questions which depend on the amount and distribution of steam pressure on the piston and thus to determine the mean pressure, and knowing the revolutions to find the indicated horse power; also the turning effort at the various points of the revolution, and the mean effort for the entire revolution.

[2] The Indicator Card and the Operation of the Valve Gear.

We will now consider briefly the more important derangements which may be met with in the valve gear, and the results as shown by the indicator card.

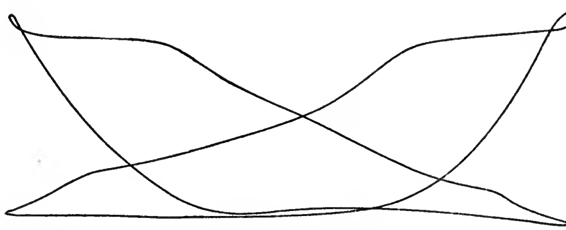


Fig. 253. Indicator Cards with Angular Advance too large.

(1) *Excentric* too far from a line at right angles to the crank; that is, angular advance δ too large (Sec. 47 [1]).

Results: Cut-off too early, steam-lead large, exhaust opening and closure early. In short, the whole round of events is ahead of time. See Fig. 253.

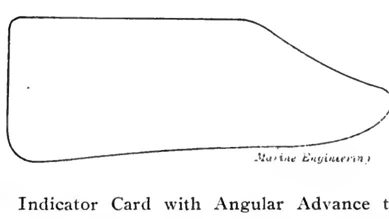


Fig. 254. Indicator Card with Angular Advance too small.

(2) *Excentric* too near a line at right angles to the crank; that is, angular advance δ too small (Sec. 47 [1]).

Results: Cut-off late, steam lead small or even negative, compression small, steam opening late, exhaust opening and closure late. In short, the whole round of events is behind time. See Fig. 254.

(3) *Steam lap* too large.

Results: Cut-off early, steam opening late and lead small or even negative, port opening small with a probable wire drawing of the steam, and drop of pressure on steam line. See Fig. 255.

Fig. 255. Indicator Card with Steam Lap too large.

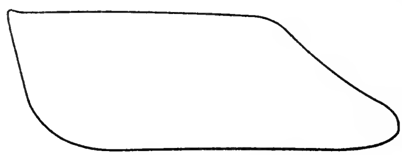


Fig. 256. Indicator Card with Steam Lap too small.

(4) *Steam lap* too small.

Results: Cut-off late, steam opening early and lead large, port opening large. See Fig. 256.

(5) *Exhaust lap* too large.

Results: Exhaust closure early and compression large, exhaust opening late and exhaust lead small. See Fig. 257.

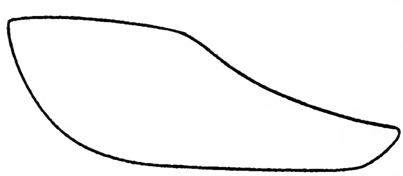


Fig. 257. Indicator Card with Exhaust Lap too large.

Fig. 258. Indicator Card with Exhaust Lap too small.

(6) *Exhaust lap* too small.

Results: Exhaust closure late and compression small, exhaust opening early. See Fig. 258.

(7) *Compression* excessive.

Results: The pressure in the cylinder may be carried above that in the valve chest before the steam valve opens, thus forming a loop as shown in Fig. 253. This may be due to either (1) or (5) above.

(8) *Expansion Excessive.*

Results: The pressure in the cylinder may fall below that in the next receiver or exhaust space beyond, thus forming a loop as shown in Fig. 259.

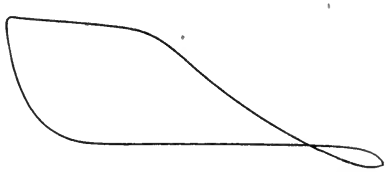


Fig. 259. Indicator Card with Excessive Expansion.

(9) *Valve Stem* too long.

Results: This means that the middle of the stroke of the valve is placed too high relative to the ports. The results for an outside valve will be to give too much steam-lap on top and exhaust lap on the bottom, and too little steam lap on the bottom and exhaust lap on top. Hence we shall have:

Steam opening in top late and small and cut off early.

Steam opening on bottom early and full, and cut off late.

Exhaust opening on top early and full and closure late.

Exhaust opening on bottom late and small and closure early. See Fig. 260.

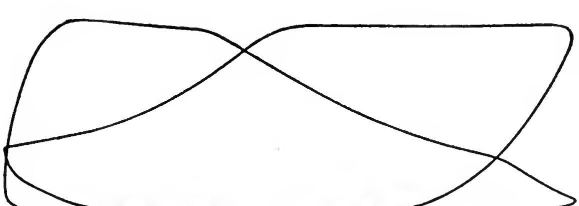


Fig. 260. Indicator Card with Valve Stem too long or too short.

(10) *Valve Stem too short.*

Results: Similar to those for (9) but oppositely related to the ends of the cylinder.

To these we may also add the following.

(11) *Leaky piston* or *piston rod* stuffing-box.

Results: The expansion line will be steeper than it should be. The compression line may also flatten off somewhat near the top.

(12) *Port openings* or *Passages* too small.

Results: Wire drawing or loss of pressure on the steam line and rise of pressure on the exhaust line. See Figs. 251, 255.

It will be noted in the above that different causes may produce similar results, so that in interpreting a given set of cards caution must be used in working back from result to probable cause and remedy. This operation may be aided by the following general hints.

It will be noted that the general effect of a valve-stem too long or too short is to effect the two ends of the cylinder in op-

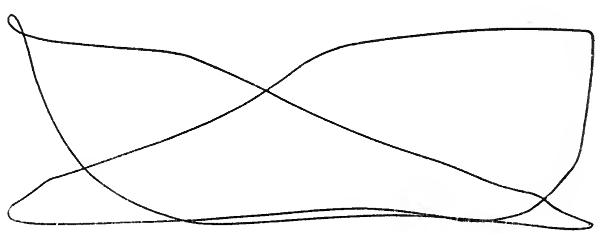


Fig. 261. Indicator Card Showing Combination Effect.

posite directions, thus giving the cards the appearance of having been pushed over in one direction or the other as in Fig. 260. On the other hand, if the valve-stem is of proper length but the excentric is improperly set the results will be of the same kind in both ends of the cylinder as shown in Fig. 253. Various combinations of these may exist in the same engine. Thus a pair of cards as shown in Fig. 261 indicates an incorrect length of valve-stem, an incorrect adjustment of the laps, with perhaps too large an angular advance. The combination nearly corrects certain difficulties and makes others still worse.

Various special features may combine to make the so-called "freak" cards, but we shall not examine this part of the subject further as such freaks are of rare occurrence, and a careful study of the results of the various single derangements as given above in (1) to (12) will usually be sufficient to show the nature of the trouble.

[3] Working Up Indicator Cards for Power.

From the principles of mechanics we know that *work* is the result of a force or effort acting through a distance, and is measured by the product of the force in pounds by the distance in feet. This gives the measure of the work in foot-pounds. *Power* measures the capacity to perform a certain amount of work in a given time. The common unit is the *horse power*, which is 33,000 foot-pounds of *work* done in one minute of *time*. Hence to find the power of an engine we have two chief steps:

(1) To find the foot-pounds of work done per minute.

(2) To reduce this to horse power by dividing by 33,000.

It may be noted here that the term *Indicated Horse Power* means simply the horse power as determined from the *indicator* cards.

Now by definition the foot pounds per minute for the steam engine will be the product of the acting force multiplied by the distance through which it acts in one minute. The acting force equals the mean load on the piston, and this equals the mean effective pressure per square inch multiplied by the area in square inches. The distance acted through per minute must be measured in feet, and equals twice the stroke multiplied by the number of revolutions per minute.

Let p = mean effective pressure in pounds per square inch; A = area of piston in square inches; L = length of stroke in feet; N = revolutions per minute. Then pA = acting force or mean total load on the piston measured in pounds, and $2LN$ = distance moved per minute in feet = piston speed. Hence foot-pounds of work per minute equals product $(pA) \times (2LN)$ or what is the same thing $2pLAN$. Hence we have the formula:

$$\text{Horse power} = \frac{2\,pLAN}{33000}$$

This is the usual formula for finding the indicated horse power, and is commonly employed for working up indicator cards for this purpose.

The reasons for measuring L in feet and A in square inches will be readily seen from the following considerations. Work is composed of two factors, the *force* factor and the *distance* factor. The first must be measured in *pounds* and the second in *feet*. The product pA is the force factor, and since p is usually measured in pounds per *square inch*, A must be measured in square inches in order that pA may be the total mean load in *pounds*. The product $2LN$ is the distance factor, and hence $2L$ the distance traveled per revolution must be measured in feet, in order that $2LN$ may be the distance traveled per minute measured in feet. The product $(pA) \times (2LN)$ or $2pLAN$ will then give the work measured in foot-pounds as we have seen above.

We will now give by rule the operations necessary to find the indicated horse power, as expressed by the formula above.

Rule—Multiply together the mean effective pressure in pounds per square inch by the length of the stroke in feet, and this product by the area of the piston in square inches, and this product by the number of revolutions per minute, and this product by 2, and then divide the final product by 33,000. The quotient will give the indicated horse power.

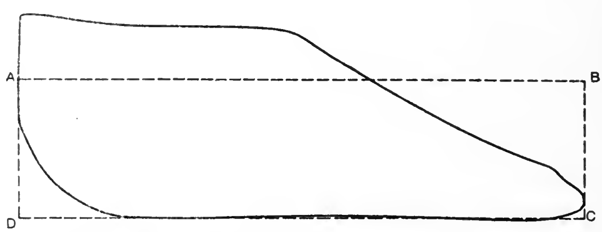


Fig. 262. Mean Effective Pressure from Indicator Card.

Now the various factors which enter into either the formula or rule for horse power, the length of stroke and area of the piston come from the dimensions of the engine, and the revolutions per minute from the counter, or by actually counting them, watch in hand. There remains the mean effective pressure p which must be found from the indicator cards, and to this part of the operation we now turn.

The mean pressure for a single card such as Fig. 262 gives simply the mean of the pressure in one end of the cylinder, say the top. To obtain this mean pressure we may proceed in a number of different ways. Fundamentally the mean of such a series of pressures as given by the indicator card, is found by dividing the area of the card by the length. This gives the side of a rectangle which would have the same area as the card. Thus in Fig. 262 if the rectangle ABCD has the same area as the card, then the side A D of the rectangle is the mean height of the card, and to the proper scale will give the mean pressure desired. Hence any method which will give the area of the card may be used for finding a mean height, and hence a mean pressure. In Part II., Sec. 9 [15] are given various rules and methods for finding the measure of an irregular area, illustrated by the example of an indicator card, and any of these may be used as there explained. The method most commonly used is to measure the ordinates on the dotted lines as in the figure there shown, take their sum, and divide by their number, 10. This multiplied by the scale of the indicator spring will give the mean pressure desired. The simplest method of locating the intervals for these dotted ordinates is that explained in Part II., Sec. 10 [4]. To carry this out we proceed as follows:

Let the card be represented in Fig. 263, then draw the lines at the ends as shown, perpendicular to the atmospheric line OA

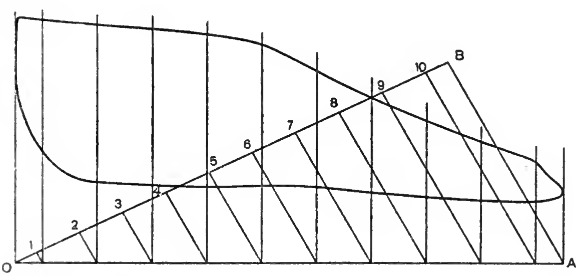


Fig. 263. Subdivision of Indicator Card for Obtaining Mean Ordinate.

and tangent to the card, thus fixing its length. Then lay off the line OB at an angle and on OB lay off first a half division O1, then nine whole divisions, and then a half division as shown. The divisions may be taken from ¼ to ½ inch in accordance with the length of the card. Then drawing a line from B to A and other parallel lines from the points of division on OB to OA, the locations for the ordinates are determined, and they may be drawn as shown. Where a large number of cards are to be worked up in this way, time will be saved by the use of a form of template or pattern for locating these points. Such an implement is shown in Fig. 264 and consists of a piece of hard wood with small steel points set in to the edge, spaced according to the lay out of points along OB Fig. 263. The distance between the extreme points is somewhat greater than the length of the longest card likely to be met with. Instead of steel points set in a block of wood, a thin plate of steel may be cut out and filed up so as to leave the points projecting at the desired intervals. In using this device it is simply necessary to draw lines tangent to the ends of the card as shown, and then to place one end of the template on one boundary line PR at any convenient point as P, and swing it to such an angle as will just bring the other end Q to the other line QS. The template is then pressed down so as to mark the paper with the points, and lines parallel to those at the ends are drawn through the points thus marked, as shown by the lines of the figure. In this way the ordinates spaced in the manner desired may be rapidly laid out and drawn in.

For summing the ordinates the method by the use of a strip of paper as explained in Part II., Sec. 9 [15] may be recom-

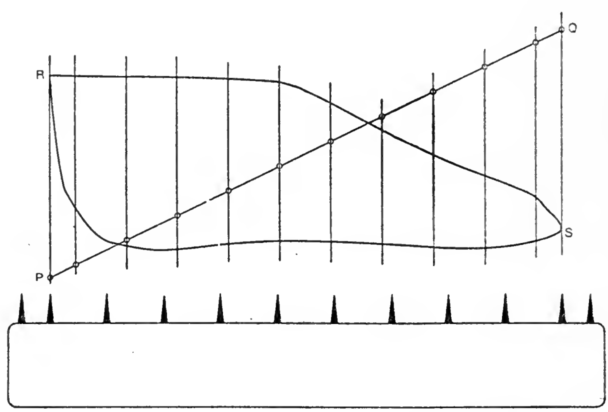


Fig. 264. Subdivision of Indicator Card for Obtaining Mean Ordinate.

mended as the simplest, quickest and most satisfactory available for the purpose.

Having thus in one way or another found the mean effective pressure for one card, the other one of the pair is taken in like manner, thus giving the mean effective for the other end of the cylinder or other stroke. These two values may then be averaged, and the result taken as the mean effective pressure for the revolution, thus furnishing the final factor p required in the formula or rule for horse power.

It must be noted that this operation is slightly in error by reason of the difference in area between the upper and lower sides of the piston. On the upper side the entire area is effective while on the lower side the piston rod takes out a small area in the center. To take account of this, we may compute the I.H.P. for each end of the cylinder separately. To this end we take each card by itself, say the head end first, and find the mean effective pressure which we may denote by p_1. Let the entire piston area be A_1. Then as before the mean load or average acting force is the product of the two, $p_1 A_1$. The distance acted through is L for each down stroke, and the number of down strokes per minute is equal to the number of revolutions N. Hence the distance per minute for the down strokes is LN and the I.H.P. for this end of the cylinder will be:

$$H_1 = \frac{p_1 A_1 L N}{33000} \text{ or } \frac{p_1 L A_1 N}{33000}$$

In a similar manner we then find the mean effective pressure for the bottom or crank end of the cylinder which we may call p_2. Then taking from A_1 the area of the piston rod, we have the effective area of the bottom of the piston which we may call A_2. Then similarly as in the head end we have for the I.H.P. in the crank end,

$$H_2 = \frac{p_2 A_2 L N}{33000} \text{ or } \frac{p_2 L A_2 N}{33000}$$

The total I.H.P. will then be the sum of these for the two strokes up and down, or:

$$\text{I H.P.} = H_1 + H_2 = \frac{p N}{33000} \quad (p_1 A_1 + p_2 A_2)$$

For illustration see example (7) below.

Mean Effective Pressure by the Aid of the Planimeter.

The planimeter, an instrument for measuring areas, is also frequently used for working up indicator cards, and where the number is large will be found of great service. Such instruments may be obtained of most makers of indicators or of dealers in mathematical instruments. General directions for their use will accompany them. The following hints may be given for their use with indicator cards.

Where the instrument has an adjustable bar it should be set so as to read the area in square inches. Where the bar is not adjustable the instrument is usually already set to read in terms of this unit. The order of procedure is then as follows:

(1) Draw lines at the ends of the card at right angles to the atmospheric line so as to be able to determine its length.

(2) Place the instrument and card in a suitable position, and read the record wheel, putting down the result, say 3.26 as below:

	Readings.	Differences.	Average.
First	3.26		
Second	7.08	3.82	
Third	10.92	3.84	3.83

(3) Then trace around the contour, usually in the direction *with* the hands of a watch for a second reading greater than the first, and come back carefully to the starting point. Then read again and set down the result, say 7.08 below the first as shown.

(4) Then repeat, tracing around as before, read and set down the result, say 10.92, below the others as shown. In making the last reading it will be noted that on the instrument itself we might be able to read only 0.92, but the increase upward from 3 to 7 shows that the wheel has passed the starting point and begun again, so that we must add the *ten* and write 10.92.

(5) We then take the difference of the readings, the first from the second and the second from the third and set down as shown, and then average these two numbers, thus finding in the present case 3.83 for the area in square inches. The reason for going around the area twice is to have two measurements, so that each will give a check on the other. If they differ widely an error somewhere is certain, and they must be repeated, while if nearly the same, as in the case given above, the error is no more than must be expected with such means, and the average may be taken as the value of the area desired.

(6) We next divide the area by the length of the card. Thus suppose in the case in hand that the length is 4.2 inches. Then $3.83 \div 4.2 = .912$ inches. This is the mean ordinate or mean height of the card in inches.

(7) We next multiply by the scale of the indicator spring and thus find the mean effective pressure desired. Thus suppose the spring to be 60 pounds to the inch. Then $60 \times .912 = 54.72$ pounds. This is then the mean effective pressure for the stroke as given by the card thus measured.

We then proceed similarly with the other card, and use the results for the determination of horse power in the manner already explained.

Illustrative Examples.

(1) The area of an indicator card is 2.87 sq. in. and its length is 3.8 in. What is the mean height?

Solution: $2.87 \div 3.8 = .755$ in.

(2) The scale of the indicator spring is 40 lbs. per inch. What is the *m.e.p.*?*

Solution: $.755 \times 40 = 30.2$ lbs.

(3) The ordinates measured in inches taken from an indicator card divided up as in Fig. 263 are as follows:

.91, 1.30, 1.44, 1.40, 1.35, 1.20, .95, .80, .70, .25, and the scale of the indicator spring is 60 lbs. per inch. Find the *m.e.p.*

Solution: Adding the lengths as given, we have for the sum 10.40. Hence dividing by 10 we have for the mean ordinate $10.40 \div 10 = 1.04$. Hence the *m.e.p.* is $60 \times 1.04 = 62.4$ lb.

(4) The total length between marks on a strip of paper used to measure the ordinates as described in Part II., Sec. 9 [15] is found to be 6.3 in. The scale of the spring is 20 lb. Find the *m.e.p.*

Solution: $6.3 \div 10 = .63$ in. = mean height, and

$.63 \times 20 = 13.6$ lb. = *m.e.p.*

(5) Given an indicator card with ordinates spaced as in Fig. 263. The pressures measured by a scale corresponding to the indicator spring are as follows:

18, 26, 28.4, 27.8, 27, 24.2, 19, 16, 14.3, 7.2. Find the *m.e.p.*

Solution: We add the pressures and find the sum 207.9. Divide this by 10 and we have 20.79 or 20.8 lb. as the value of the *m.e.p.*

(6) From the two cards of a pair the values of the *m.e.p.* are found to be 28.6 for one end and 32.2 for the other. The piston area is 1,213 sq. in., the stroke 39 in. and the revolutions 102. Find the I.H.P. neglecting the effect due to the area of piston rod.

Solution: The *m.e.p.* for the whole revolution is the mean of the values for the two ends or *m.e.p.* $= (28.6 \times 32.2) \div 2 = 30.4$.

Then stroke in feet $= 39 \div 12 = 3.25$.

$$\text{Then I.H.P.} = \frac{2 \times 30.4 \times 3.25 \times 1213 \times 102}{33000}$$

* This abbreviation is often used for the term *mean effective pressure.*

Multiplying out the factors of the numerator and dividing by the denominator we find I.H.P. = 741 Ans.

(7) Given the following:

Diam. of cylinder = 24 in.
Diam. of piston rod = 5 in.
m.e.p. from head end or $p_1 = 63.4$ lb.
m.e.p. from crank end or $p_2 = 58.8$ lb.
Stroke = 36 in.
Revolutions 110.

Find the I.H.P. both with and without the allowance for the area of piston rod.

Solution:

Area of 24 inch piston or $A_1 = 452.4$ sq. in.
Area of 5 inch piston rod or $a = 19.6$ sq. in.
Effective area of lower side of piston = difference, or $A_2 = 432.8$ sq. in.

Then neglecting the effect of the rod we should say:

$m.e.p. = (63.4 + 58.8) \div 2 = 61.1$

$$\text{and I.H.P} = \left(\frac{2 \times 61.1 \times 3 \times 452.4 \times 110}{33000}\right)$$

Working this out we find: I.H.P. = 552.8.

Taking account of the piston rod area we have for the head end:

$$H_1 = \frac{63.4 \times 3 \times 452.4 \times 110}{33000} = 286.8$$

For the crank end:

$$H_2 = \frac{58.8 \times 3 \times 432.8 \times 110}{33000} = 254.5$$

Adding we have:

$H = 541.3.$

There is thus seen to be in this case a difference of 11.5 horse power, constituting an error by the first method of some considerable amount. It is readily seen that this error will be relatively less the larger the cylinder, especially in the cylinders of a multiple expansion engine. Thus in the case given which was for the H.P. cylinder of a triple expansion engine the error is 11.5 horse power, or about 2 per cent. For the I.P. cylinder the error would be not far from 4.5 horse power or about .8 per cent., while for the L.P. cylinder it would be perhaps two horse power or about .3 per cent. This would give a resultant error of about 1 per cent. for the engine as a whole. While these figures would vary with particular circumstances, they will serve to illustrate the nature of the error, and the methods given show how to avoid it when so desired.

[4] Combined Indicator Cards.

The cards taken from the various cylinders of a multiple expansion engine, as for example those of Fig. 265, may be

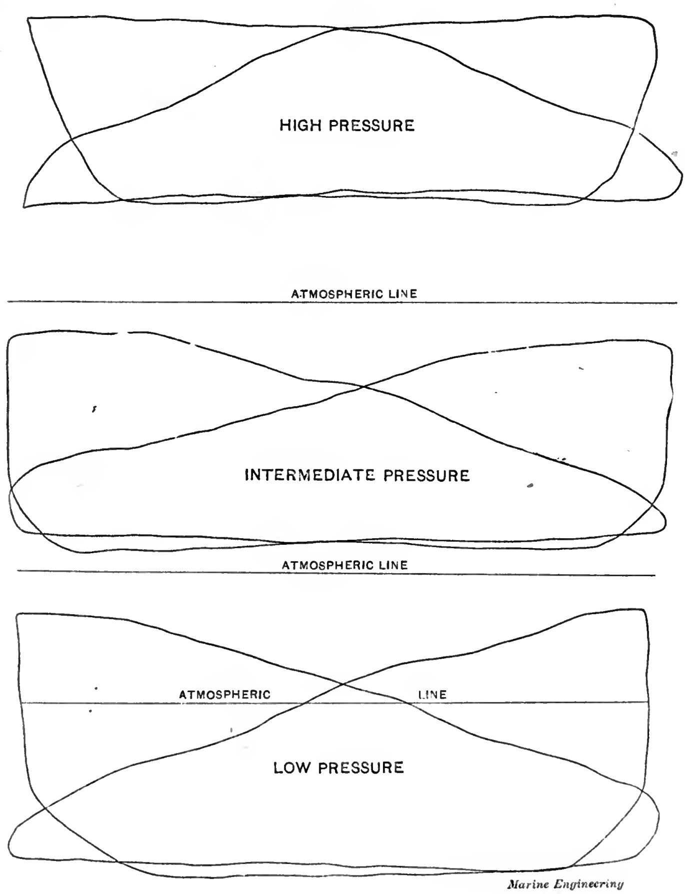


Fig. 265. Set of Indicator Cards from Triple Expansion Engine.

combined in such a manner as to show very instructively the continuous history of the expansion of the steam, that is the continuous relation between volume and pressure as the steam passes through the engine. To effect this combination it is necessary to lay down the various cards in one diagram and all to the same scale of volume and pressure. The details of the operation may be sketched out in the following steps:

(1) In Fig. 266 take the two lines at right angles, OX and OY, the former as an axis of volume and the latter as an axis of pressure.

(2) Determine in cubic feet for each of the cylinders the volume of the clearance (Sec. 67), and the volume swept by the piston.

(3) Lay off the lines AB, CD, EF at such distances from OY as to represent respectively the clearance volume in the

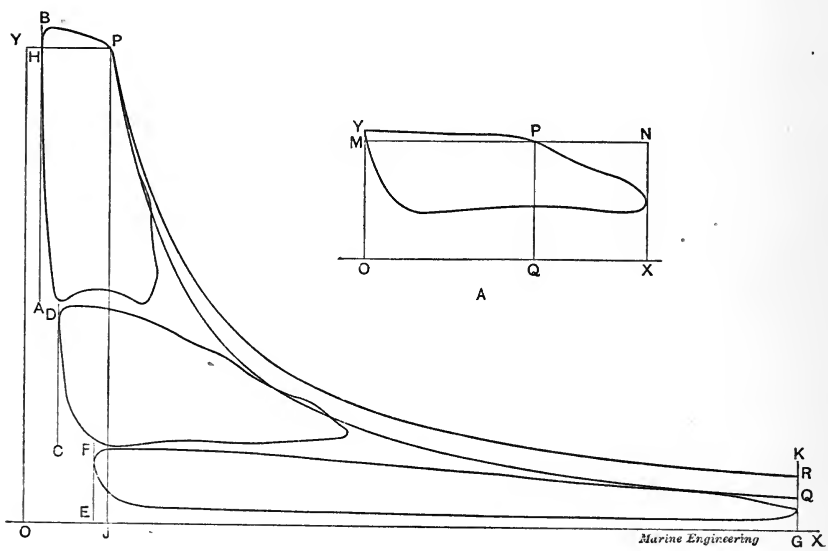


Fig. 266. Combined Cards from Triple Expansion Engine.

H.P., I.P., and L.P. cylinders, taking care to select the scale of volume such that the L.P. volume plus its clearance as measured between the lines OY and GK will come within the desired limits of the diagram.

(4) Lay off on each card the line of zero pressure or the perfect vacuum line, as shown by OX in the small diagram A.

(5) Take next the H.P. card as at A for example, and select any point such as P. Measure in any convenient units the distances MP and MN : multiply the volume of the cylinder by the former and divide by the latter. This will give the volume

swept in the H.P. cylinder from the beginning of the stroke to the point P.

(6) The corresponding point P of the combined diagram is then found by measuring from AB a distance HP representing this volume, and from OX a distance JP representing the pressure PQ on the card. This will give the point P, and other points are found in a similar manner, as many as may be needed to determine the form of the card as shown. It is to be especially noted that the H.P. card of the combined set is the same as that at A but drawn simply with different scales, and therefore more or less distorted in appearance.

(7) The points necessary to determine the other cards of the combination are found in a precisely similar manner, remembering that in each case volume is measured from the clearance line CD or EF, while the pressure must be measured from the line of zero pressure for the card and laid off from the corresponding line OX of the combined set.

This diagram shows the general manner in which the steam expands on its way through the engine. An expansion line PQ shows the general law of expansion as a continuous operation.

PR is an ideal expansion line laid down as a hyperbola, all points in the curve corresponding to the condition that the product of volume by pressure shall be constant, or in symbols, $pv =$ Constant. This shows the result of the so-called true hyperbolic expansion law, and as appears from the diagram, the actual expansion line is somewhat below this ideal line.

The equation to the actual expansion line may be expressed in the form $pv^n =$ Constant, where n is an exponent having values usually lying between 1.15 and 1.2. The equation $pv^{1.18}$ may be taken as very commonly representing this line in good average practice. The extent to which the area bounded by the line PR and the clearance lines on the left is well filled in. is an indication of the degree to which the performance of the actual engine approaches that of an engine having true hyperbolic expansion and with indicator cards as shown in Fig. 252. The relation between the actual engine and such an ideal case is usually expressed by a percentage factor known as the "card factor." For good practice with triple expansion engines, this factor will be found from .60 to .70. With quadruple expansion engines representative values are found from .55 to .60.

The diagrams of figures 265 are reproduced from an actual case and may be considered as representing good modern practice in general character and form.

At this point reference may be made to the effect on the distribution of power in a compound or multiple expansion engine, of linking up or cutting off earlier in the intermediate or low pressure cylinders. Taking first the case of a compound, linking up or shortening the cut-off on the L. P. cylinder will increase the power in this cylinder and decrease it in the high. This result at first sight seems contradictory to common experience, because in a single cylinder we are accustomed to associate an earlier cut-off with decrease of power. In the case of the compound, however, cutting off earlier in the L.P. cylinder gives a higher back pressure in the H.P. cylinder and a consequently higher initial pressure in the L.P. cylinder, and thus results in an actual addition to the L.P. indicator card area instead of a decrease as in the case of a single cylinder. At the same time the area of the H.P. card will be reduced and the power developed in this cylinder will be decreased correspondingly. Similarly for a multiple expansion engine and in general, cutting off earlier in any of the cylinders beyond the first or H.P. will result in an increased back pressure for the next preceding cylinder, and in a higher initial pressure for the cylinder itself, and thus in an actual addition to the area of the indicator card and a corresponding subtraction from the area of the card for the cylinder preceding.

Thus in Fig. 266, cutting off earlier in the L.P. cylinder will result in raising the upper line of the L.P. and lower line of the I.P. cards, and thus in increasing the area of the former and decreasing that of the latter. In like manner cutting off earlier in the intermediate cylinder will result in raising the upper line of the I.P. and lower line of the H.P. cards and thus in increasing the area of the former and decreasing that of the latter. In like manner cutting off later in any cylinder beyond the H.P. will result in similar changes but in the opposite direction. Thus a later cut-off in the I.P. cylinder will decrease the power developed in that cylinder, and increase the power developed in the H.P. cylinder. It thus results that a combination of changes such as a later cut-off in the I. P. cylinder and earlier cut-off in the L.P. will both tend to decrease the power developed in the I.P.; while an earlier cut-off in the L.P. cylinder and a later cut-off in the H.P. will both tend toward an increase of power developed in the I.P.

Sec. 56. STEAM ENGINE INDICATORS.

[1] Descriptive.

The indicator card has already been described in Sec. 55. It is the purpose of the indicator to draw this card. It must therefore provide for the proper combination of these movements. (1) A movement in step with the piston and proportional to it in amount so that all horizontal distances on the card shall bear a constant proportion to the corresponding parts of the stroke. (2) A movement at right angles to that in (1) and in direct proportion to the pressure per square inch on the piston in the end of the cylinder to which the indicator is connected.

The combination of these movements will then result in a diagram such as those shown in Sec. 55, and giving at each point of the stroke the pressure on the piston as desired, the upper line showing the pressure which urges the piston forward on one stroke and the lower line the pressure which resists its movement backward on the return stroke.

In Fig. 267 a modern indicator is shown. A is a drum to which the paper is attached by means of the clips as shown. This drum is given a motion back and forth about its axis by means of a connection with the crosshead through the so-called "reducing motion." By this means the drum is given a motion of some three to five inches in extent, just in step with the motion of the piston and proportional to it in amount. B is the indicator cylinder or barrel connecting with the end of the engine cylinder from which the card is to be taken. Within the cylinder, as shown, is a piston with a coiled steel spring above, resisting pressure from below the piston upward. To the piston rod is attached a linkage carrying at the end of the arm P the pencil point which is to trace the diagram upon the paper carried by the drum. The connection between the linkage and the piston rod is such that the former may be swung freely about the cylinder upon a ring to which it is attached. The pencil may thus be brought into contact with the paper on the drum or withdrawn from it as desired. An adjustable screw stop is provided, and so arranged as to arrest the movement of the pencil motion when swung around by the hand, and thus allow only light contact between the pencil point and the paper. In some cases a brass point is used instead of a pencil, the cards being of paper specially prepared so that the brass will leave a black mark upon it. Such points are strong and require no sharpening except at long intervals.

The object of the linkage which forms the pencil motion is to magnify the movement of the indicator piston, and thus

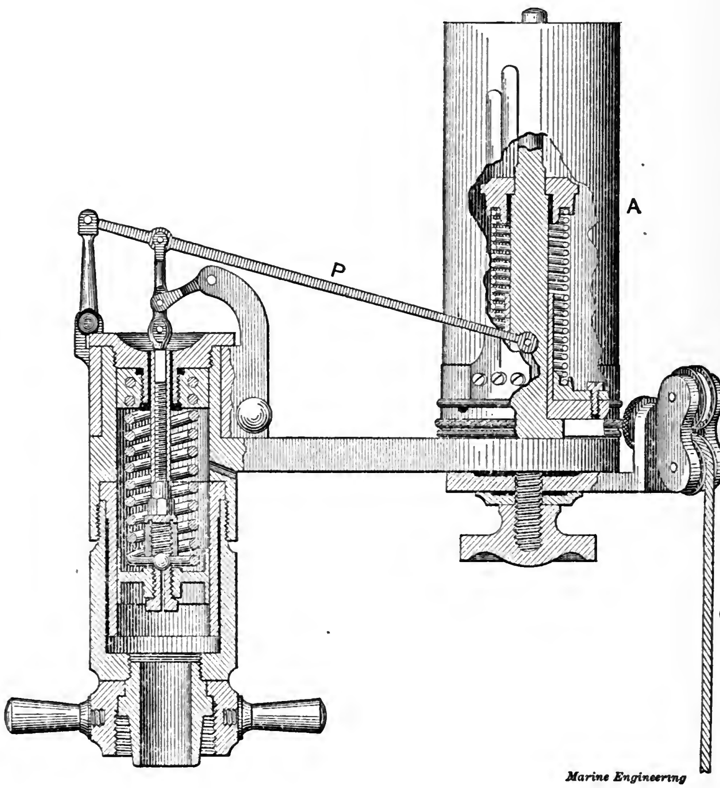


Fig. 267. Steam Engine Indicator.

to allow the use of stiff springs with a corresponding small movement of spring and piston. With high revolutions especially, this is found necessary in order to reduce as far as possible the disturbance in the diagram due to the inertia of the moving parts of the indicator. The linkage is thus a form of multiplying motion, or a reducing motion reversed, and it should give to the pencil a movement exactly proportional to that of the piston, but 3 to 5 times greater as may be desired.

The relation between the pressure per square inch and the actual movement at the pencil point fixes the so-called *scale* of the spring. This depends also on the actual area of the indicator piston, which is, however, usually about one-half square inch. Thus a 40 pound spring means a spring such that a pressure of 40 pounds *per square inch* on the indicator piston, or say an actual load of 20 pounds, will produce a movement of one inch at the pencil point.

By means then of the piston, spring and linkage, the second of the necessary movements as mentioned above is thus produced.

Returning to the drum the first of the motions above noted is obtained by some form of reducing motion as described below. The connection between the drum and the reducing motion is usually made by means of a cord C wrapped around a groove in the base as shown. The cord thus serves to pull the drum around in one direction while the return stroke is made by means of a coiled spring in the base. This spring opposes the motion given by the cord, and is therefore coiled up during the forward stroke. As soon, however, as the pull of the cord ceases the spring takes charge and uncoiling carries the drum in the reverse direction as fast as the cord will allow, thus keeping the latter taut and insuring the motion of the drum in step with the main piston in both directions as accurately as the form of reducing motion may determine.

A separate indicator may be provided for each end of the cylinder, or by suitable pipe connections and a three way cock, one indicator may be made to serve for both ends. In any case the cock which shuts off the indicator must be so arranged that when shut off from the cylinder the space below the piston will be connected to the outside air. The piston with equal air pressure on both sides will then come to a position of equilibrium, and the atmospheric line may be drawn.

[2] Reducing Motions.

The purpose of the reducing motion has already been stated. There are many different ways in which the desired movement may be given to the drum, some of them accurate in geometrical principle and some only approximate.

One of the most common is by means of links, levers and bell-cranks. The simplest of such forms is shown in Fig. 268. A is a pin attached to the crosshead. AB is a short link connecting the crosshead to a lever BD pivoted at C. The point D will then move in step and nearly in constant proportion to the piston, and from D the motion for the drum may be taken, either by a cord direct, or from the end E of a rod DE moving as shown. In such cases the cord should run in continuation of the line DE and not off at an angle as EF or DH. As a general rule in all such cases, the reducing motion should be so adjusted that the cord part should not undergo changes of angular direction, or at least such changes should be made as small as possible. Thus in Fig. 269 suppose the point from

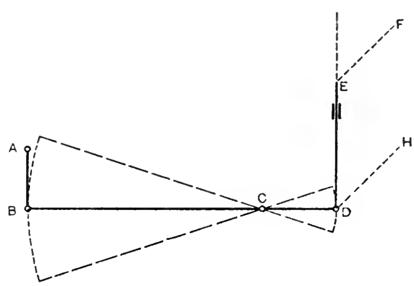


Fig. 268. Reducing Motion.

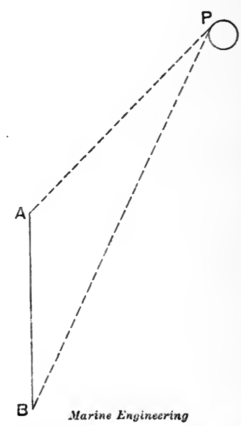


Fig. 269. Reducing Motion.

which the motion is taken to move through a path AB, and the indicator guide pulley to be at P. Then at one extreme the cord will be represented by PA, and at the other by PB. Such a change in the angular direction of the cord relative to the line of motion AB will result in error, and should be avoided by bringing P over AB or AB under P. It is not necessary that the motion of the point E Fig. 268 should be vertical so long as the gear is so arranged as to reduce to a minimum all angular changes in cords and connecting links. Thus the arrangement of Fig. 270, while containing a large number of joints and parts, may be as nearly correct as the simpler form of Fig. 268.

Instead of taking the motion direct from D a link DG connects this point with a bell-crank GHI pivoted at H. Then a second link IJ connects this to a second bell-crank JKL and a rod LE guided at M gives a point E from which the motion may be taken, or if more convenient the rod LE may be dispensed with and the motion taken from L direct. Such a complication of gear is of course not desirable, and the arrangement is shown simply as an illustration of a combination of links and bell-cranks which would still give the motion required.

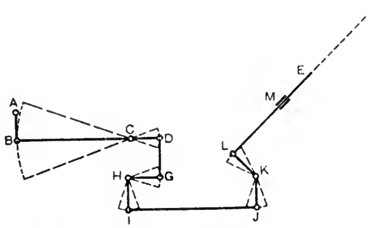


Fig. 270. Reducing Motion.

All such forms of reducing motion are approximate and not geometrically exact. The error is, however, in most cases small and is usually neglected, though if desired its nature and extent may be investigated by a suitable geometrical analysis of the gear.

Instead of taking the motion from the crosshead by means of a short link as AB Fig. 268, a lever BD Fig. 271 is sometimes provided, having a forked end and pivoted at C or D. A pin on the crosshead working in the slot or forked end gives the to and fro motion to the lever, while from D or C the desired motion is taken.

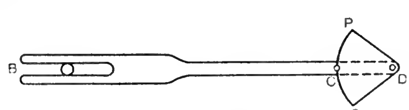


Fig. 271. Reducing Motion.

Instead of attaching the cord direct to C for example, a sector of wood PQD with center at D is attached to the arm, and the string is led off from the face of the sector. Such a sector may also be employed with the arrangements shown in Figs. 268 and 270. None of these motions is geometrically exact.

A form of pantograph consisting of jointed rods as shown in Fig. 272, may sometimes be used when there is room for it to work freely. A is attached to the crosshead and D or E is the fixed pivot. Then the other point E or D will provide a motion for the indicator drum which is geometrically exact. Here again however, the string should be so led that its angularity will not vary.

Instead of this arrangement of links the so-called lazy-tongs as shown in Fig. 273 is sometimes employed. This is also geometrically exact, and is in fact an equivalent to the pantograph in Fig. 272, without requiring quite as much room.

Various combinations of pulleys may also be used, as illustrated in Fig. 274. AB is an arm projecting from the crosshead and moving with it. To the end B of this arm is attached a cord wrapping around a light pulley P. Q is a smaller pulley on

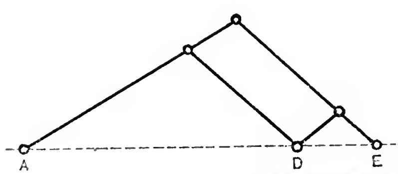


Fig. 272. Pantograph Reducing Motion.

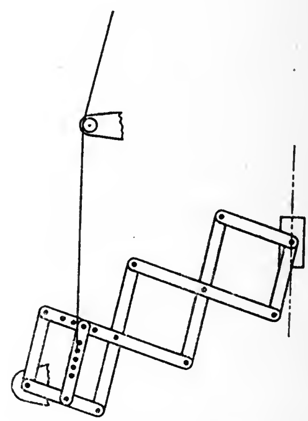


Fig. 273. Lazy Tongs Reducing Motion.

the same axis and moving with P. Wrapped on this is a cord CD, which may be led off in various directions to the indicator as shown by CD, CD_1, CD_2. This gear is geometrically exact.

Various other forms of reducing motion are also to be met with, but those described will be suffcient to show the forms most commonly available for marine practice.

[3] Taking an Indicator Card.

The instrument should first be examined and put into proper condition and adjustment. This should include the following points:

(1) The joints should all work freely, but without lost motion.

(2) The piston should not bind nor should it be so loosely fitted as to allow serious leakage. A slight leakage is, however, better than too snug a fit.

(3) The working surfaces of the barrel and piston should be carefully wiped and oiled. This should be repeated from time to time when a series of cards is being taken. The joints of the

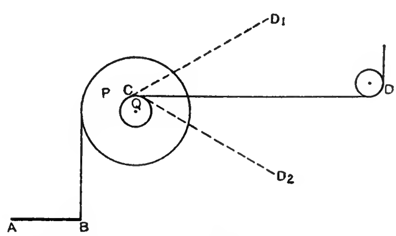


Fig. 274. Reducing Motion.

pencil motion should also be lubricated with clock oil as often as may be required.

(4) The pencil points should be sharpened and the screw stop so adjusted that the point can rest only lightly on the paper.

The operation of taking the card itself is briefly as follows:

The indicator is attached to the cock, a blank card is placed on the drum and the cord connection is adjusted so that the drum will have the proper stroke without coming against the

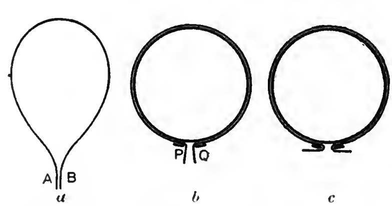


Fig. 275. Putting on an Indicator Card.

stop at either end. In attaching the blank card the most convenient way will be to bend the sheet of paper around and grasp both edges between the thumb and forefinger as at AB in Fig. 275a. Then slip over the drum and under the clips so that the latter will come outside the paper as shown at PQ, b. Then slip the paper down into place, pull and adjust so that it fits snug-ly, and bend the edges back as in *c*. The cord is then hooked on to the reducing motion and the drum takes up its movement with the main piston. The cock is then opened to the end of the cylinder from which the diagram is desired, and the pencil immediately takes up its motion corresponding to the varying pressures of the steam. The indicator piston should be allowed to work in this way for a few strokes, or until everything is warmed up into working condition.

When everything is in readiness the pencil motion is moved up against the stop so that the pencil resting lightly on the paper will trace its path for a complete revolution or longer if desired. Then remove and shut off the indicator from the cylinder. This will connect it with the air, the indicator piston will come to equilibrium under atmospheric pressure, and the atmospheric line may then be drawn. The drum connection is then unhooked, the paper removed, a fresh one replaced, and the next card taken when desired. If one indicator is used for both ends of the cylinder, both cards should be taken on the same paper with as small an interval between as possible. The cock is swung over for one end and the card taken, and then immediately swung over for the other end and the second card taken without loss of time. The cock is then closed off connecting the indicator with the air, and the atmospheric line is then drawn.

Each card as it is removed from the indicator should be marked with sufficient data to identify it, and make possible its use for the purpose intended. This should include at least the following items:

(1) Cylinder.
(2) End from which card is taken.
(3) Revolutions.
(4) Scale of spring.
(5) If a series of cards is being taken the time and serial number should also be set down.

The various other items usually printed on the back of the card may be filled in at a later time as may be convenient. When cards from both ends are taken on one paper, we must be able to assign each to its proper end of the cylinder. The most certain way of determining this is to shut off the connection to one end of the cylinder entirely, and then take the card from the other end. It will thus appear how the card from this end lies on the paper, whether with admission line to the right or left, and this will show how to mark the entire series of cards taken with the same arrangement of reducing gear, etc.

CHAPTER IX.

Special Topics and Problems.

Sec. 57. HEAT AND THE FORMATION OF STEAM.

[1] Constitution of Matter.

For the purpose of explaining or discussing the relations between matter and the forces of nature, all substances are supposed to be composed of enormously large numbers of indefinitely small parts called *molecules,* each one of which is supposed to be, in fact, the smallest portion of the substance which can exhibit its various properties. These molecules are furthermore not at rest, but are supposed to be in a state of more or less violent agitation or motion. If the motion of each molecule is about a fixed center so that they all retain their average positions fixed in the body, it is said to be a *solid* or in the *solid state.* If the motion of the molecules is about centers which themselves are free to move about in any direction, so that the average position of the molecules is not fixed and the body readily changes its form, it is said to be a *liquid* or in the *liquid state.* If the motion of the molecules is in straight lines hither and thither, bound to no center or location, but ever striving to fly as far apart as possible, the substance is said to be a *gas* or in the *gaseous state.*

In the solid and liquid states the molecules are bound together by forces of molecular attraction, so that they tend to maintain about the same average distance apart, and thus to fill the same volume. Any attempt to change this average distance between the molecules and thus to make the volume larger or smaller, must deal with these molecular forces. The only practicable way of doing this is through the agency of heat, as we shall see in the next section. In a gas the forces binding the molecules together have been overcome, and the molecules have been separated so much further apart that all traces of these attractive forces have disappeared, and instead we now have a repulsive force acting between the molecules and urging them as far apart as the limits of the volume which contains them will allow. Due to this property a gas will expand and fill any volume, no matter how large, the repulsive force or force of expansion, however, becoming weaker as the volume increases and the average distance between the molecules becomes greater and greater.

[2] Heat.

(1) *Heat and Its Relation to Matter.*—We know energy as the capacity for doing work. Also the energy of motion is called kinetic energy, while the energy of position or location relative to a given force, is called potential energy.

Heat is one of the many forms of energy. It is, in fact, the energy of the molecule, and the heat in a body means therefore simply the amount of such molecular energy which the body possesses. This energy of the molecule is partly kinetic or due to its motion, and partly potential due to its position relative to the molecular forces which act upon it. The addition of heat to a body or its subtraction from a body means therefore the addition or subtraction from the energy of its molecule, and hence the addition to or subtraction from its store of molecular energy. This addition or subtraction of heat is always accompanied by a series of changes in the state or condition of the body.

Thus if heat be added to a lump of ice at the melting point or 32° Fah. it first melts or changes from a solid to a liquid, remaining at the constant temperature of 32° and slightly contracting in volume meanwhile. If heat is still farther added the water grows warmer to the touch and as shown by the thermometer. At the same time it continues to contract slightly till it reaches a temperature of about 39° Fah. and then slowly expands. If under atmospheric pressure the increase of temperature and volume will continue with the addition of heat until the thermometer marks a temperature of 212° Fah. Then the further addition of heat occasions no further elevation of temperature, but instead, a new change of state. The water now passes into the state of vapor or steam, the temperature of both the water and the vapor formed from it remaining meanwhile constantly at the fixed temperature of 212°. After the water is completely vaporized, if the vapor be inclosed in a chamber of fixed volume and heat still further added, it will then be found that the pressure and temperature will continue to increase so long as additional heat is supplied. If instead the pressure is kept constant, the volume and temperature will increase as heat is added. If the temperature is kept constant, the pressure will fall as the volume is increased.

From the start then, as more and more heat has been added, the water has exhibited successively the three states of matter. As ice it is a solid; in its usual state or between 32° and 212° under atmospheric pressure, it is a liquid; and after it is completely vaporized and heat is still further added so as to carry the conditions considerably beyond those at which the vapor was formed, it becomes a gas.

We must here explain the difference which may be implied in the words *gas* and *vapor*. When a substance first changes from the liquid to the gaseous state, or while the pressure, volume and temperature are near those corresponding to such a change, the substance is more strictly called a *vapor*, or is said to be in the *vaporous condition*. If the substance is in the gaseous state but with pressure, volume and temperature conditions far removed from those corresponding to the change of state, the substance is more generally called a *gas*. There is no sharp line of difference between a vapor and a gas. The former means simply a substance in the gaseous state, but at or near the conditions corresponding to the process of change from one state to the other, while the latter means likewise a substance in the gaseous state, not far removed from the conditions corresponding to the process of change of state.

There are thus two chief kinds of change which the application of heat may produce.

(a) It may change the temperature of a substance accompanied by a change of pressure or volume or both, but without change of state.

(b) It may produce a change of state as from solid to liquid or liquid to vapor, accompanied usually by a change of volume, but without change of temperature. If, however, the pressure varies during the change of state, then the temperature at which the change occurs will also vary, but there will be no change of temperature directly accompanying the change of state. In consequence, during the change from solid to liquid or *vice versa*, the temperatures of both are the same, and similarly during the change from liquid to vapor or *vice versa*, the temperatures of both are the same.

It must be understood when changes are referred to as depending on the *addition* of heat, that the *subtraction* of heat will produce changes in exactly the opposite direction. Thus if the addition of heat causes a body to expand, the subtraction will cause it to contract: if the addition causes an increase of pressure the subtraction will cause a decrease: if the addition causes a change of the state from liquid to vapor, the subtraction will cause a change from vapor to liquid, etc.

(2) *Sensible and Latent Heat.*—Heat which causes a change of temperature in a body, as when water is heated and becomes hotter to the touch or to the thermometer, is called *sensible* heat. This corresponds to a change in the kinetic energy of the molecule, so that increase of velocity of the molecule and increase of its kinetic energy correspond within the substance to the growing hotter to the touch and to increase of temperature as observed on the outside.

Heat which is involved in a change of state but which produces no effect on the temperature of the substance (as in the melting of ice at 32° or the boiling of water at 212°) is called *latent heat*. This corresponds to a change in the potential energy of the molecule so that an increase in the average distance between the molecules (acquired in opposition to the molecular forces acting), and a consequent increase in their potential energy corresponds within the substance, to the change of state at constant temperature as observed on the outside.

It must be understood that there is really but one kind of heat, and that this division into sensible and latent is only a matter of convenience in order to signify the particular energy change which is effected within the body. The heat required to raise the temperature of a body or to increase its sensible heat is thus expended in increasing the velocity of the molecules of the body, and hence in increasing their kinetic energy. The heat required to effect a change of state, or to increase the potential energy is expended, on the other hand, in increasing the average distance between the molecules, and in increasing the volume of the body against whatever external forces may exist.

The gradual expansion of a body with increase of temperature is accounted for by assuming that as the velocity of the molecules is increased, their average path is increased also, and hence their average distance apart, and hence the volume of the body.

(3) *Temperature.*—We have seen above that temperature refers to the condition of a body as regards its sensible heat, or the kinetic energy of its molecules. Two bodies are said to be at the same temperature when there is no tendency for heat (that is, molecular energy) to flow from one to the other. This condition is measured by the thermometer, an instrument too well known to require particular description.

The Fahrenheit scale which is commonly used by engineers in the United States is graduated as follows: The temperature of melting ice is called 32° and that of boiling water 212°. The interval between the two is then evenly divided into 180 parts, and the same divisions are extended above and below as far as may be desired.

On the Centigrade scale the temperature of melting ice is called 0° and that of boiling water 100°. The interval is then evenly divided into 100 parts, and the divisions extended above and below as may be desired.

For transforming temperatures from one scale to the other we have the following equations:

$$F = 9/5\ C + 32^\circ$$
$$C = 5/9\ (F - 32^\circ)$$

where F and C denote respectively the temperatures on the Fahrenheit and Centigrade scales.

Examples: Transform 20° C into Fah.

Operation: $F = 9 \times 20 \div 5 + 32 = 36 + 32 = 68^\circ$.

Transform 20 Fah. to C.

Operation: $C = 5/9\ (20 - 32) = 5/9\ (-12) = -6\ 2/3^\circ$ or $6\ 2/3^\circ$ below zero.

Transform 77 Fah. into C.

Operation: $C = 5/9\ (77 - 32) = 5/9 \times 45 = 25^\circ$

Transform -22° Fah. into C.

Operation: $C = 5/9\ (-22 - 32) = 5/9\ (-54) = -30^\circ$

(4) *Heat Unit.*—Care must be taken to distinguish between *quantity* of heat and *temperature.* The first refers to the total amount of heat energy present in the substance, the second to the kinetic energy of a molecule. A large cup of warm water and

a small cup of hot water may both have the same quantity of heat, but not the same temperature. A cup of hot water and a barrel full of hot water may have the same temperature, but the quantities of heat will be very different.

For measuring quantities of heat, use is made of a *heat unit* defined as the amount or quantity of heat required to raise one pound of water one degree in temperature. Inasmuch, furthermore, as the amount thus required would vary slightly at different temperatures, it is necessary to fix the temperature at which the unit is to be defined. The temperature thus taken for the definition of the heat unit has sometimes been at the freezing point, or again at the point of maximum density of water which is about 39° Fah., or again at about 62°, or from 62° to 63°. It is very difficult to determine the amount of heat required to raise water one degree at or near the freezing point, while between 60° and 70°, or at about an average atmospheric temperature, the measurements are most readily made. For this reason a temperature within this range is to be preferred for definition of the heat unit.

The heat unit thus defined is often known as the *British Thermal Unit*, the name being usually abbreviated to *B. T. U.*

(5) *Joule's Equivalent.*—Since heat is but a form of energy it follows that it should be possible to transform heat into mechanical work, and *vice versa*. It is for the first purpose that the steam engine is used, while instances of the latter transformation are constantly before our eyes, as in the heat developed by the friction of a bearing, or in turning a chip from a bar of steel, etc.

It therefore becomes of importance to know the ratio of transformation, or how much mechanical work measured in foot-pounds corresponds to one heat unit as above defined. This has been made the subject of careful experiment extending over the past one hundred years, and the latest and most reliable results seem to give for this ratio the value 778. That is, one B. T. U. is equivalent to 778 foot pounds of mechanical work. This means that when in a steam or other heat engine heat is transformed into mechanical work, for every B. T. U. so transformed and disappearing as heat, 778 foot pounds of work will be obtained; or again when mechanical work is transformed into heat, for every 778 foot pounds so transformed and disappearing as work, one B. T. U. of heat will appear.

This number or ratio, 778, is known as *the mechanical equivalent* of heat, or frequently as Joule's *equivalent*, though its value as determined by Joule was somewhat smaller than the value given above.

(6) *Transfer of Heat.*—Heat may be transferred from one body or place to another in three different ways: by radiation, by conduction, and by convection.

By *radiation* we mean the transfer of heat through space in straight lines, as from the sun to the earth, or from a furnace fire to the face when the door is opened.

By *conduction* we mean the transfer of heat along a body from one molecule to the next, as when a slice bar becomes warm at one end if red-hot at the other.

By *convection* we mean the transfer of heat from one point of a liquid or gas to another by means of currents set up in the liquid or gas and carrying the heated molecules from one place to another, as in the circulation set up within a Scotch boiler.

Two other operations are also concerned in the transfer of heat from one substance to another. These are emission and absorption.

By *emission* we mean the giving off or transfer of heat from the molecules of one body to those of another. By *absorption* we mean the converse of this, the receiving of heat by the molecules of one body from those of another.

The heating of water in a boiler is due, at least in part, to all of these processes. The fire in the furnace radiates heat to the crown sheet; convection and draft currents convey the hot gas to the heating surface; there is emission from the hot gases and absorption by the metal of the heating surfaces; there is conduction through the metal from the fire to the water side; there is emission from the metal and absorption by the water; and finally there are convection currents developed in the water by means of which the temperature is more or less uniformly raised.

[3] Steam.

Steam or the vapor of water is the substance almost universally used as the medium through which the heat set free by the coal is transformed into the mechanical work of the engine or pump. Its properties are therefore of great importance for the engineer, and we may properly study briefly the more important at this point.

(1) *Formation of Steam.*—Let AB in Fig. 276 be a very tall vessel open at the top and having an inside cross-section of one square inch. Let us suppose at the start that there is in the bottom of this vessel one cubic inch of water at say 60° temperature. On top of the water let there be a piston as shown which we may suppose to move without friction, and to be without weight. In other words we wish simply, something to separate the water from the air. Then on the surface of the water there will be just the atmospheric pressure of say 14.7 pounds per square inch. Now let heat be applied at the bottom of the chamber. The first

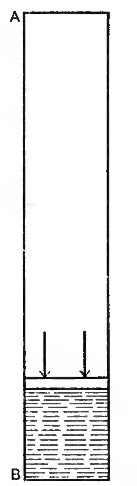


Fig. 276. The Formation of Steam.

result will be a transfer of heat through into the water, and a resultant rise in temperature of the water near the heating surface. In consequence of this the water will expand somewhat and thus become lighter than the other water above and farther from these surfaces. The heated water will thus tend to rise to the top and so displace the cooler water there, which will in consequence sink and thus in turn be brought into contact with the heating surfaces. There is set up in this way a general ascending current of warmer water and a corresponding descending current of cooler water by means of which the whole mass is gradually raised in temperature.

In this manner are formed the convection currents referred to in the preceding section, and to their formation in a steam boiler is due the circulation of the water, especially in those of the fire-tube or tank type.

In this way then the temperature of the water will gradually rise until it reaches 212°. The temperature then ceases to rise and steam begins to form, rapidly increasing the volume below the piston and thus forcing it upward. The steam formed is of the same temperature, 212°, as the water of which it is formed, the only changes being the increase of volume and the change of state. If we continue to supply heat at the bottom and prevent its escape from the sides of the tube or chamber, the water will thus gradually be all converted into steam, just balancing by its own pressure the atmospheric pressure on the top of the piston. The volume of steam thus formed would be 1,663 cubic inches, or in other words the tube would have to be 1,663 inches or 138.6 feet high to allow of the operation as we have supposed it to take place.

Suppose now that at the beginning the piston is loaded down with a weight so that the total load on the water is 20 pounds instead of 14.7. Then let heat be supplied as before. A like series of changes will follow, but the water instead of beginning to change state from liquid to vapor at 212° must be heated to 228° before the change begins, while the final volume will be only 1,244 cubic inches. If the initial pressure were 100 pounds then the temperature at which the change of state would begin would be 328°, and the final volume would be only 275 cubic inches. Similarly for 200 pounds pressure, the figures are 382° and 144 cubic inches.

On the other hand, if by means of an air pump the pressure on the water were decreased below that of the atmosphere the temperature of change would become lower, and the volume greater. Thus if the pressure is 10 pounds the figures are 193° and 2,385 cubic inches, and if 5 pounds they are 162° and 4,576 cubic inches. Similarly, to boil water at 32°, or the freezing point, it would be necessary to reduce the pressure to .089 pounds, while the volume of vapor formed would be about 21,170 cubic inches.

In general we thus find that as the pressure is increased or decreased there is a corresponding rise or fall in the temperature at which the formation of vapor begins, and that it remains fixed at this temperature during the process of change of state, and that the temperature of the vapor formed is also the same as that of the water from which it is formed.

It thus appears that steam is simply the vapor of water, or water in the vaporous state, as defined in [2]. The process of steam formation as above described is known as *boiling* or *ebullition*, or sometimes in the general sense as vaporization. The temperature at which the change of state occurs is known as the *boiling point* or *point of ebullition*.

At lower temperatures there is always a certain amount of slow vaporization going on, the pressure of the vapor formed being limited to that corresponding to the temperature of the water. It is by means of such slow vaporization that water is carried up for the formation of rain clouds from the surfaces of rivers, lakes and oceans, or that mud dries up in the roads, or clothes when hung out to dry. Thus if the temperature is 100° the maximum vapor pressure is about one pound. Considerable vapor will thus be formed, which will gradually find its way upward into the air, at least so long as the air is not saturated; i.e., already charged with vapor of the full pressure corresponding to the temperature. This is, however, a branch of the subject that we cannot here pursue further.

When, however, the temperature is such that the vapor pressure is sufficient to just balance the entire pressure on the surface of the liquid, then the vapor is formed with freedom, and more or less from the body of the liquid, producing the agitation and other conditions which constitute *boiling* or *ebullition* as referred to above.

We have thus examined in some detail the formation of steam at constant pressure. Let us next examine briefly its formation at constant volume, the case which corresponds to its generation in a steam boiler.

The boiler being first open to the air through the safety valve, the water is subject simply to atmospheric pressure. The heating of the water and first formation of steam proceed according to the process as described above. The air is thus displaced from the boiler and driven out through the safety valve or other escape. The safety valve or escape is then closed and in consequence the steam which is formed tends to increase the pressure, and as this rises the temperature of the boiling point will rise also. In this way as heat is added, a part of it is used in raising the temperature of the water and steam up to the ever-rising boiling point, while the remainder goes for the vaporization of a fresh portion of steam. In this way the volume of vapor remains practically constant, but the pressure and temperature rise together according to the regular law which relates the one to the other, while the increase of vapor is accommodated in the constant volume by the increase in density, or decrease in volume required per pound as the pressure rises.

It is thus seen that the temperature and volume of steam are closely dependent on the pressure to which it is subjected.

For engineering purposes pressure may be measured from two different starting points. The true or so-called *absolute* pressure is measured from the zero or no-pressure condition, and is the true or total pressure exerted by the gas or vapor in question. The ordinary steam gauge, however, as described in Sec. 17 [7] does not measure the absolute pressure, but simply the difference between this and the pressure of the atmosphere. This is due to the fact that the gauge is subjected to the pressure of the steam on one side of the tube and to that of the atmosphere on the other side. It thus measures simply the difference between the two. This is commonly called "gauge pressure." The absolute pressure is greater than the gauge pressure therefore, by the pressure of the atmosphere. This varies with the altitude and with other circumstances affecting the barometer. For engineering purposes it is very commonly taken at 15 pounds per square inch. A more correct average for the sea level is, however, 14.7, as noted above. It results therefore that having given the gauge pressure we may find the absolute pressure by adding 15 or 14.7, as we may choose, according to the degree of accuracy needed in the case in hand.

(2) *Saturated and Superheated Steam.*—When steam and water are present in the same vessel together and there is no tendency for the water to change into steam or the steam into water except as heat is added or taken away, the water and the steam are said to be in thermal equilibrium. When steam is thus in equilibrium in contact with water, it is said to be *saturated.* Thus during the entire process of steam formation illustrated in Fig. 276, the steam is saturated.

When there is no moisture or water in the liquid condition suspended in or mixed with the vapor, the steam is said to be *dry.*

When there is moisture or water suspended in or mixed with the vapor, the steam is said to be *moist* or *wet.* In such case the steam or vapor part of the mixture must itself be in the saturated condition as defined above, so that wet steam is simply a mixture of saturated water vapor and liquid water.

When steam is free from all suspended moisture, but is still in the saturated condition as determined by its pressure, temperature and volume, it is called *dry* and *saturated*. Thus the condition of the steam in the vessel of Fig. 276, just as the last bit of vapor is formed, is dry and saturated. During the operation the vapor would be dry and saturated provided it contained no suspended moisture. Practically this is a condition difficult to realize. The greater or less violence of the ebullition is apt to carry up a certain amount of water in the shape of fine mist into the steam space, from which it settles back only slowly, and so is liable to be carried over into the engine. The steam furnished by the average boiler under good conditions contains usually not less than from 1 to 2 per cent. of moisture, while under poor conditions the amount may rise to 5 per cent. and more.

Suppose now, referring to Fig. 276, that after the last bit of water has been vaporized, heat is still further applied, the pressure on the piston remaining the same. The temperature which during the process of vaporization has remained stationary, will now begin to rise, accompanied also by an increase of volume.

If we now recall the three stages which the water has passed through, all at constant pressure, it appears that during the first stage there was a rise in temperature of the water at nearly constant volume: during the second stage (that of steam formation) there was an increase of volume at constant temperature: during the third stage, as just described, there is increase of both volume and temperature.

If in this last operation instead of keeping the pressure constant the piston were held fast, thus keeping the volume constant, we should find with the addition of heat that both the temperature and the pressure would increase.

Steam in the condition resulting from these operations is said to be *superheated*. As compared with saturated steam its temperature and volume are greater for the same pressure, or its temperature and pressure are greater for the same volume, or its volume is greater and pressure less at the same temperature. It is clear that superheated steam cannot be in contact with water and remain superheated. It cannot therefore be moist.

If it were brought into contact with water it would lose its superheat and become saturated, forming by the heat given up, a little more vapor from the water present.

We may also put the relation between saturated and superheated steam into the following form:

Saturated steam contains only as much heat as absolutely necessary for its maintenance as steam at the given pressure. Superheated steam at the same pressure contains more heat than saturated.

The temperature and volume per pound for saturated steam correspond with the pressure as in the process of vaporization, and are respectively the lowest temperature and smallest volume at which steam of the given pressure can exist. The temperature and volume per pound for superheated steam are both larger for the same pressure than for saturated steam.

[4] Total Heat in a Substance.

(1) *Total Heat of Steam.*—The total heat of a body in a given condition is the total amount of heat required to produce this condition, reckoning from some starting point agreed upon. For steam this point is usually taken as 32° Fah., or the freezing point of water. The total heat per pound of steam at a given pressure and temperature, means then the total amount of heat, both sensible and latent, required to produce one pound of steam of the given pressure and temperature, from water at 32°. These quantities of heat are used in the solution of problems relating to the heat required for evaporation, boiler efficiency, gain by feed-water heating, etc. The value of the total heat in terms of the temperature is very closely given by the following approximate equation:

$$H = 1082 + .3t$$

while the latent heat is similarly given by the equation:

$$L = 1114 - .7t.$$

In these equations, H denotes the total heat, L the latent heat, and t the temperature.

Instead of using these or other similar equations, the values are more conveniently taken from tables prepared so as to give the various quantities for regularly varying values of the pressure. Thus from the Table it appears that at 14.7 pounds pressure absolute, or at the pressure of the atmosphere, it requires 180.9 B. T. U. to heat the water from 32° to the boiling point 212°, and then 965.7 B. T. U. to completely vaporize it at this point. The great excess of the latter or latent heat over the former or sensible heat may thus be noted. It is also seen that according to the table the heat required to raise the water from 32° to 212° is not exactly measured by the difference in degrees which is 180. The excess is due to the fact that the B. T. U. is defined for 1° rise from 62° to 63°, while the amount required to make 1° difference at other temperatures is slightly different, increasing on the whole for higher temperatures, so that between 32° and 212° the average is slightly greater than from 62° to 63°. It thus results that the sensible heat per pound of water involved in a temperature change is slightly greater than the number of degrees which measures such change. This difference is, however, so small that for most engineering purposes it may be neglected if more convenient, and the number of heat units per pound of water may be taken equal to the number of degrees difference in temperature.

(2) *Total Heat of a Mixture of Steam and Water.*—Steam as actually used is usually moist; that is, it contains a small fraction of water. To find how much heat is required to produce one pound of such a mixture of steam and water, a given fraction being steam and the remainder water, we have simply to remember that *all* of the water must be raised to the temperature of boiling, while only the given fraction is vaporized. If therefore S denotes the sensible heat and L the latent heat, while x is the fraction which is steam, or the *quality* of the steam as it is called, then the heat H required to produce one pound of the mixture will be:

$$H = S + xL.$$

The quality x is usually expressed on the percentage basis.

The following examples will illustrate the use of the steam tables:

(1) Find the sensible heat, the latent heat and the total heat in one pound of steam at 120 pounds gauge pressure.*

Ans. from the table, 322.1, 866.6, 1188.7.

(2) Find the heat in 1 pound of feed water at a temperature of 110°.

Ans. 110 — 32 = 78.

* For present purposes, and in the following problems, it will be sufficiently accurate to take the pressure of the atmosphere as 15 pounds per square inch. The absolute pressure is therefore found from the gauge pressure by simply adding 15.

(3) How much heat would be required to produce the steam in example (1) from the feed water in (2)?

Ans. The difference between the two or 1110.7.

Remark: These three examples show how we may find the heat necessary to produce a pound of steam of given pressure from feed water of any given temperature.

(4) How much heat is required to make 1 pound of steam at 150 pounds gauge pressure from feed water at 130°?

Ans. 1095.5.

(5) How much heat is required to produce 1 pound of moist steam of 92 per cent. quality at 150 pounds gauge pressure from feed water at 120°?

Solution:

The sensible heat per pound of steam is....	338.4
The sensible heat per pound of feed is......	88.0
The difference is.....................	250.4

The latent heat for one pound is 855.1. But since only 92 per cent. is vaporized, only .92 of this will be required. We have therefore:

$H = S + xL = 250.4 + .92 \times 855.1 = 1037.$ Ans.

(6) One engine requires per I.H.P. per hour, 16 pounds of steam of 96 per cent. quality at 150 pounds gauge pressure, the feed being at a temperature of 140°. Another engine requires 20 pounds of steam of 90 per cent. quality at 110 pounds gauge pressure, the feed being at a temperature of 110°.

Find the amounts of heat required per hour in the two engines, and hence their real comparison as heat engines.

By the methods illustrated above we find as follows:

For the first engine 16821 B. T. U. per hour.

For the second engine 20436 B. T. U. per hour.

The second engine requires therefore 3615 B. T. U. per hour more than the first, and in consequence is about 21.5 per cent. more expensive in terms of the heat required.

Further illustrations of these principles will be found in Sec. 58.

[5] Latent Heat in Passing from Ice to Water.

We have seen in [2] (1) that a certain amount of heat is absorbed in the melting of ice at the constant temperature of 32°. It is thus rendered latent or is taken up in effecting the change of state in the same general manner that heat is rendered latent in passing from liquid to vapor. It may be of value to note here the quantity of heat thus rendered latent.

This amounts to about 143 heat units, or B. T. U., per pound of ice melted from 32° into water at the same temperature.

Sec. 58. STEAM BOILER ECONOMY.

[1] General Principles.

There may be several different bases for boiler economy according to the particular feature held in especial prominence. The output of the boiler is estimated in terms of the steam produced, and we may have the following kinds of economy:

(1) Economy in coal consumption, increasing with the output of steam per pound of coal burned.

(2) Economy in weight of boiler, increasing with the output of steam per pound of boiler.

(3) Economy in first cost, increasing with the output of steam per dollar invested in the boilers.

(4) Economy of maintenance or total life, increasing as the life of the boiler is longer and the amount necessary for repairs is smaller.

It is never possible to fulfil in the highest degree the conditions for these various kinds of economy, and a compromise must always be made among them, though usually (1) or (2) will take the first place in the order of importance.

Without special note, however, the term economy is understood to refer to (1), though the considerations relating to the others should always be kept in mind. In some cases, as in torpedo boat and like practice, (2) may assume first place in the order of importance, and, perhaps, require some sacrifice relative to the others. We will now consider more especially the economy referred to under (1).

From the standpoint of coal economy or efficiency, the boiler is charged with all the coal that is thrown through the furnace doors, and is credited with the steam which it sends to the engine. Or to state the matter more definitely, it is charged with all the heat which could be gotten from this coal by perfect and complete combustion, and is credited with the heat which is transferred through and actually used in the formation of steam. If the efficiency were perfect, or if there were no loss, these two amounts of heat would be equal. Actually there are many losses, large and small, and, in consequence, the latter is considerably less than the former. The ratio of the two is known as the *boiler efficiency*. In practice its value varies from 50 to 75 or 80 per cent. Following are the more important sources of loss which occasion this drop in efficiency.

In the first place, a little of the fuel may fall unburned through the grate into the ash pit. Again, a little in the form of dust and small bits may be carried by a strong draft, either unburnt or only partially burnt, through into the tubes, uptakes or funnel. Still another small portion may escape as smoke, which consists almost entirely of very fine particles of unburnt carbon formed from the gases which are distilled away from the coal in the process of combustion. Still another portion of these gases may escape unchanged and unconsumed. Again, there may be an incomplete combustion of the carbon forming *carbon monoxide*, and giving only about 4,450 heat units per pound, instead of 14,500 which result from the complete combustion into *carbon dioxide*. Hence, whatever carbon escapes in the form of carbon monoxide is only partly burned, and may be considered as carrying away over two-thirds of the heat which would be liberated by complete combustion. These losses all occur in the furnace, and are due to poor firing and to imperfect combustion.

To reduce them to the lowest limit, the fireman must know his business, and be willing to attend to it with ceaseless care and diligence. In addition, there must be provided, by proper design, the necessary supply of air both above and below the grate, together with such arrangements as experience may show are needed for good combustion with the fuel in hand. At best this loss may be reduced to perhaps 2 or 3 per cent., while with carelessness or poor design, or both, it may easily reach values from 10 to 20 per cent.

The heat being thus more or less perfectly liberated in the furnace, is then passed on to the boiler heating surface, whose duty it is to transfer it through into the water on the other side. The energy is still to exist as heat, but it is to be transferred from the hot gas to the water, thus converting the latter into steam. This, however, cannot be perfectly accomplished, and thus arises a further loss. A part of the heat, instead of passing through the heating surface, goes up the funnel carried by the escaping gases, and so gets away into the outside air. Another and smaller part escapes by radiation into the fire room. These losses it is impossible wholly to avoid. It would be necessary to avoid all loss of heat by radiation, and to reduce the temperature of the products of combustion in the funnel to that of the outside air. The latter, especially, cannot be done for the best of reasons. In the first place, the temperature cannot be reduced below that of the steam and water in the boiler, because heat always flows naturally from a hot body to a cooler one, and it will, therefore, flow from the gas to the water only so long as the latter is the cooler of the two. The actual temperature of the escaping gases must be considerably higher than that of the steam, because in the first place sufficient heating surface to reduce them to nearly the same temperature could hardly be allowed; and, again, aside from blowers, the strength of draft is dependent on the temperature of the hot gas in the funnel, and for a satisfactory rate of combustion it is necessary to discharge the products of combustion at temperatures not less than 500 to 600 degrees. This loss is one therefore which exists in the nature of things, and cannot be reduced below some 20 or 30 per cent.

On the whole, then, the entire losses under the best conditions can hardly be reduced much below 25 per cent., while with poor conditions they may aggregate 40 to 50 per cent. The remaining fraction, or the 50 to 75 or 80 per cent., represents then the efficiency of the boiler as defined above.

Since a pound of average good coal has available some 13,000 to 14,000 heat units, it follows that the heat actually utilized per pound of coal is usually found between say 7,000 and 11,000 units.

In general the conditions favorable to high efficiency are the following:

(1) A free burning coal of good quality, with suitable furnaces and air supply for complete combustion.

(2) Moderate draft.

(3) Abundant heating surface.

Or, as a combination of (2) and (3), we may put:

(2) Moderate evaporation required per square foot of heating surface.

The opposite of these conditions will cause necessarily a loss in efficiency more or less pronounced according to circumstances.

[2] Evaporation Per Pound of Coal.

The efficiency of a boiler is often roughly estimated by the number of pounds of water evaporated into steam per pound of coal burned on the grates. This, according to conditions, may vary from 6 or 7 to perhaps 11. Remembering that it usually requires rather more than 1,000 heat units per pound of steam, the general agreement between these figures and those above for the heat utilized per pound of coal is readily seen. In fact, the figures for the heat utilized are derived really from a measurement of the pounds of steam evaporated per pound of coal, together with a knowledge of the heat required per pound of steam, the latter being derived, of course, from the conditions of the evaporation.

When we remember the great difference in the amount of heat required per pound of steam, depending on the temperature of the feed, the temperature of the steam, and whether the steam is moist or dry, it is clear that for any fair measure of boiler performance in terms of steam formed per pound of coal, these differences must be allowed for, especially in comparisons between boilers working under different conditions.

To this end it is customary to reduce the number of pounds evaporated to what it would be if the steam were dry and the temperature of both feed and steam were 212 degrees. In such case it would require to make one pound of steam simply the latent heat at 212 degrees, or 966 B. T. U. (British thermal units).

This is known as the reduced evaporation, or the equivalent evaporation from and at 212 degrees. It is really the number of pounds of steam which would be formed if each required 966 B. T. U., and is, therefore, simply a measure of the B. T. U. put into the steam per pound of coal.

The ratio between the number of B. T. U. actually required and the 966 is known as the *factor of evaporation*. These factors are often arranged in tabular form, assuming dry steam in each case, but with temperature of feed water and steam varying over the usual range.

The application of these various principles relating to boiler economy will be better understood by the solution of the following illustrative problems. In these problems we shall still for convenience consider the pressure of the atmosphere as 15 pounds per square inch, and the absolute pressure therefore as 15 pounds greater than the gauge pressure.

(1) Temperature of feed 110°, steam pressure 150 pounds, gauge. Thermal value of the coal 14,000 thermal units per

pound. Efficiency of boiler .64. Find the pounds of water evaporated into dry steam per pound of coal.

Operation:

From the steam tables:

Heat in the water at boiling point	338.4
Heat in the feed = 110 — 32 =	78
Difference	260.4
Latent heat	855.1
Heat required per pound of dry steam	1115.5

Heat available per pound of coal = .64 × 14,000 = 8,960.

Hence pounds of steam evaporated = 8960 ÷ 1115.5 = 8.03 Ans.

(2) Temperature of feed 140°. Steam pressure 200 pounds, gauge. Quality of steam 96 per cent. Thermal value of coal 14,400 thermal units per pound. Efficiency of boiler .70. Find the pounds of water evaporated per pound of coal into steam of the given quality.

Operation:

Heat in the water at boiling point	361.3
Heat in the feed = 140 — 32	108
Difference	253.3
Latent heat per pound = 838.9.	
Take .96 of this	805.3
Heat required per pound of moist steam	1058.6

Heat available per pound of coal = .70 × 14,400 = 10,080.

Hence pounds of steam evaporated = 10,080 ÷ 1058.6 = 9.52.

(3) Temperature of feed 100°. Steam pressure 120 pounds, gauge. Evaporation 8 pounds per pound coal. Assuming dry steam, what is the evaporation from and at 212°, and what is the factor of evaporation?

Operation:

Heat in the water at boiling point	322.1
Heat in the feed = 100 — 32 =	68.0
Difference	254.1

Latent heat per pound	866.6
Heat required per pound of dry steam	1120.7

Heat utilized per pound of coal $= 8 \times 1120.7 = 8965.6$.

Evaporation from and at 212° $= 8965.7 \div 966 = 9.28$.

Factor of evaporation $= 1120.7 \div 966 = 1.16$.

Evidently also

Equivalent evaporation $= 8 \times 1.16 = 9.28$.

(4) Same as in example (3) but assuming the steam of 95 per cent quality, what are the results?

Operation:

Heat in the water at boiling point	322.1
Heat in the feed = 100 — 32 =	68.0
Difference	254.1
Latent heat per pound = 866.6.	
Take .95 of this	823.3
Heat required per pound of moist steam	1077.4

Factor of evaporation $= 1077.4 \div 966 = 1.115$.

Equivalent evaporation $= 8 \times 1.115 = 8.92$.

(5) Temperature of feed 160°. Steam pressure 200 pounds, gauge. Quality of steam 97 per cent. Evaporation 8.8 pounds steam per pound coal. What is the equivalent evaporation from and at 212°, and what is the factor of evaporation?

Operation:

Heat in the water at boiling point	361.3
Heat in the feed = 160 — 32 =	128.0
Difference	233.3
Latent heat per pound = 838.9.	
Take .97 of this	813.7
Heat required per pound of moist steam	1047.0

Factor of evaporation $= 1047.0 \div 966 = 1.084$.

Equivalent evaporation $= 8.8 \times 1.084 = 9.54$.

(6) Compare the economy in (3) and (5).

Ans. in the ratio 9.28 : 9.54 or 1 : 1.028.

(7) Which is the more economical of the following cases:

(a) Coal at $4.00 per ton, 9.2 pounds of steam per pound of coal; quality of steam 98 per cent.; steam pressure 150 pounds, gauge; temperature of feed 110°.

(b) Coal at $3.20 per ton; 8.0 pounds of steam per pound of coal; quality of steam 96 per cent.; steam pressure 135 pounds, gauge; temperature of feed 120°.

By the methods of the preceding examples we find that the equivalent evaporations in the two cases are as follows:

(a) : 10.46.

(b) : 8.85.

The cost of steam in the two cases will therefore be in the compound ratio. (See Part II., Sec. 6 [2]):

$$(a) : (b) :: 4.00 : 3.20.$$
$$:: 8.85 : 10.46.$$

or $(a) : (b) :: 4.00 \times 8.85 : 10.46 \times 3.20.$

Whence $(a) : (b) :: 1.058 : 1$

or case (a) is nearly 6 per cent. more expensive than (b).

NOTE.—In solving the above examples the heat in the water at boiling point has been taken from the tables. Somewhat more quickly such problems may be solved by taking the heat in the water at boiling point as measured simply by the difference in temperatures, as explained in Sec. 57 [4] (1). The difference is small and is very commonly neglected. In the above examples, however, we have preferred to use the more exact values. The simpler form of solution will be illustrated by solving example (5) in this way, and the result will serve to show the nature of the difference between the two. We have thus:

Temperature of steam	387.7
Temperature of feed	160
To raise one pound feed to boiling point	227.7
Latent heat for one pound = 839.9.	
Take .97 of this	813.7
Heat required per pound moist steam	1041.4

Factor of evaporation $= 1041.4 \div 966 = 1.077$.

Equivalent evaporation $= 1.078 \times 8.8 = 9.49$.

[3] Evaporation Per Pound of Combustible.

In discussing further the details of steam boiler performance, it is often desirable to make the necessary allowance for the ash in the coal, or for the ash and moisture, so as to obtain the evaporation per pound of actual combustible matter. To this end it is only necessary to divide the evaporation per pound of coal determined as above by the fraction of the coal which is combustible, or what is the same thing, we may increase the result per pound of coal in the ratio in which the coal is greater than the combustible. The result will be the evaporation per pound of coal exclusive of ash or of ash and moisture. This result may relate, of course, either to the actual evaporation under the given conditions, or to the equivalent from and at 212°.

This will be illustrated by the following examples:

(8) In example (5) Suppose the ash to be 12 per cent., what are the actual and equivalent evaporations per pound of *moist* combustible?

Ans: Actual evaporation $= 8.8 \div .88 = 10$ lbs.
Equivalent evaporation $= 9.54 \div .88 = 10.84$ lbs.

(9) Under the same conditions suppose the ash and moisture to be 15 per cent., what are the evaporations per pound of *dry* combustible?

Ans: Actual evaporation $= 8.8 \div .85 = 10.35$.
Equivalent evaporation $= 9.54 \div .85 = 11.22$.

Sec. 59. STEAM ENGINE ECONOMY.

[1] General Principles.

In the following discussion it will be assumed that the reader has a general knowledge of the chief properties of steam and of its relation to the heat which it contains. We will then take up in an elementary way a discussion of the principles governing its economical use in a steam engine.

At the very outset it must be clearly understood that we derive the work of the engine from the heat which the steam contains, and not from the steam in itself. The steam is simply a carrier for the heat and the operation of the engine is simply a means for transforming into useful work a fraction of the heat which comes into the engine, and then rejecting the remainder with the steam which is its carrier. The larger the fraction of the heat which can be transformed into useful work the better the efficiency of the engine, and the constant aim is therefore to turn into useful work the largest possible fraction of the heat which enters with the steam.

It may be asked, why not turn all the heat into work, and so realize a perfect efficiency? Unfortunately a series of natural laws and limitations seems to prevent all hope of realizing such an ideal, and actually we must be content with turning into useful work a comparatively small fraction of the total heat supplied. First and foremost among the causes of this reduction in efficiency is a principle or law sometimes known as the second law of thermodynamics. This law fixes a limit on the fraction of heat which can be transformed into useful work, such limit depending on the extreme temperatures between which the substance is worked in the engine. Thus if t_1 is the temperature of the steam at admission and t_2 that at exhaust, so that t_1 and t_2 are the two temperatures between which the steam is worked, and $(t_1 - t_2)$ is the range, then the law asserts that no engine, no matter how perfect, can transform into useful work a fraction of the entering heat greater than $(t_1 - t_2) \div (t_1 + 461)$. As another way of stating this relation, the temperature may be supposed to be measured from a point 461, or more accurately 460.7 degrees, below the ordinary zero of the Fah. scale. This is called the *absolute zero*, and temperature measured from this zero is called *absolute temperature*. The difference of the temperatures would be the same no matter whether measured from the ordinary of absolute zero. The numerator of the above fraction is therefore the difference or range of temperature, while the denominator is the absolute temperature of the entering steam. The fraction of heat converted into useful work can therefore never exceed the *temperature range* divided by the *absolute temperature of the entering steam*. Thus to illustrate, suppose that $t_1 = 370$ and $t_2 = 140$. Then the fraction becomes $(370 - 140) \div (370 + 461$ or $230 \div 831 = .277$ or slightly over one-quarter.

These figures represent the limits for steam of about 160 pounds gauge pressure, and it therefore appears that for engines operating between these limits this law steps in, and at one stroke reduces the ideal efficiency from one to about one-quarter; or in other words we are forced, due to the operation of this law, to throw away about three-quarters of the total heat, and at the very best with the most ideally perfect engine could only transform into useful work the remaining one-quarter.

Such then is the very best that could be done by a so-called *ideal* engine. The working substance in the simplest form of such an engine must be carried through a series of changes or operations, four in number, and specified as follows:

(1) The first operation must consist of an expansion at con-

stant temperature, and all heat received from the source of supply must be received during this operation.

(2) The second operation must consist of an expansion with decrease of temperature during which, however, no heat as such is allowed to either enter or leave the substance.

(3) The third operation must consist of a compression during which the temperature remains constant and all heat removed from the body must be removed during this operation.

(4) The fourth operation must consist of a compression with increase of temperature, during which, however, no heat as such is allowed to either enter or leave the substance, and at the end of which the substance must find itself in the same condition as at the beginning of number (1).

Work is done by the substance during operations (1) and (2) and work must be done on the substance during (3) and (4). The difference between the work done *by* and *on* the substance will be the net work obtained from the heat in the substance, and the ratio of this to the total heat supplied during number (1) or the efficiency of the engine will be exactly measured by the difference between the temperatures of operations (1) and (3) divided by the larger increased by 461; or in symbols:

$$\text{efficiency} = \frac{t_1 - t_2}{t_1 + 461}$$

This is then the cycle and the efficiency of an ideal engine in the simplest form. There may be certain related variations in the operations (2) and (4) making a more complicated cycle, but with the same efficiency. This ideal marks, then, the highest possible limit of efficiency for any and all engines working between the given temperature limits t_1 and t_2.

In the table are shown the values of this limiting efficiency for engines with gauge pressure as indicated, all condensing and supposed to have a back pressure of 2.8 pounds absolute, or a lower temperature, t_2, of 140°.

An examination of this table shows that with the ideal conditions which correspond to the operation of this engine, the fraction of heat utilized with modern boiler pressures would range from 25 to 30 per cent. These conditions, however, are far from those which actually exist in practice. Every one of the conditions specified above is violated in greater or less degree, and the result is that with the operation of the engine under the best conditions obtainable in actual practice, the fraction realized will be only some 60 to 80 per cent. of the figures for the ideal case as given in the table below. These figures, 60 to 80 per cent. in the best practice, really measure the efficiency of the engine so far as the engineer is responsible. That is, nothing which he can do will serve to avoid the loss which reduces the limiting efficiency down to that for the ideal engine as given in the table above. His efforts are therefore limited to approaching as nearly as possible to the conditions of the ideal engine, and the figures 60 to 80 per cent. measure the degree of approach which modern engineering practice has made

TABLE.

Gauge Pressure at Engine.	Limiting Efficiency.
100	.248
110	.253
120	.259
130	.264
140	.269
150	.273
160	.278
170	.281
180	.285
190	.289
200	.292
210	.295
220	.299
230	.301
240	.304
250	.307

to this ideal. Thus, for illustration, if the ideal engine could transform 25 per cent. of the heat into useful work, a good actual engine working between the same temperature limits will be able to transform from 15 to 20 per cent., and similarly for other conditions.

To put the matter a little differently, any and all engines fail to transform into work all of the heat supplied to them. In the ideal engine as specified above, the part not transformed but rejected as heat is the least possible for all engines working between the same limits of temperature t_1 and t_2. In any actual engine the amount not transformed into work but rejected as heat is greater than in the ideal case. Such *additional* amounts of heat rejected and not transformed into work are called *wastes* or *losses*. That is, all differences between the performances of the ideal and actual engines are considered to be due to these so-called *wastes* or *losses*.

These various wastes may be classified as follows:

(a) *Radiation and Conduction Waste.*

This consists of heat which is radiated away from the hot surfaces of the cylinder, or conducted away through the columns and bed plate. The heat thus escaping avoids transformation into work and is therefore counted as a heat waste, or as an expense from which no corresponding return is received.

(b) *Initial Condensation.*

At the instant the steam valve opens, the steam rushes into the cylinder to find itself in contact with surfaces which have but recently been exposed to the influence of the condenser or external air. They are, therefore, at a temperature much lower than the steam, and in consequence a part of the heat is absorbed and a corresponding part of the steam is condensed. The heat thus absorbed by the surfaces of the cylinder and piston will be given up later during the exhaust period of the revolution, and thus communicated to the condenser. It thus appears that a thin skin of metal on the inside of the cylinder and on the faces of the cylinder head and piston may be considered in a sense as a place of hiding into which a portion of the heat slips on the entrance of the steam, and from which it escapes to the condenser or air without having taken part in the cycle of the engine, and hence without having contributed its part to the useful work done. The heat so escaping appears thus as an expense, but without any corresponding return in work, and therefore constitutes a heat waste.

(c) *Irregularities of the Cycle.*

We have specified above the four fundamental operations of the ideal engine cycle in its simplest form. In the actual engine none of these is realized, and the variations are all such as to count against the efficiency. Into the details of these points we cannot here enter, and the broad statement must suffice that with but few exceptions, the variations from the routine specified above for the ideal engine will count against the efficiency and occasion a heat waste greater or smaller as the circumstances may determine.

The Improvement of the Steam Engine.—From the preceding section it follows that there are two fundamental methods open for the improvement of the steam engine from the standpoint of economy.

(1) An increase in the temperature range and thus an increase in the ideal or limiting efficiency.

(2) The saving of some of the various wastes as noted above.

The first raises the ideal efficiency and hence with a given proportion of wastes will raise the actual efficiency as well. The second raises the actual efficiency by carrying it a little nearer to the ideal.

The temperature range may be increased in two ways,—the initial temperature can be raised, and the final temperature can be lowerd.

The continually advancing pressures in modern practice means a constant rise in the upper temperature, a constant increase in the ideal efficiency, and with the same proportion of losses, a corresponding rise in the actual efficiency. This is then the real significance of high pressures in modern practice so far as they are related to the question of economy.

Again by decreasing the back pressure from say 18 pounds for a non-condensing engine to say 3 pounds for a condensing engine, a very considerable decrease in the final temperature is obtained, a corresponding increase in the temperature range, and a resultant increase in actual efficiency. This is likewise the real significance of the influence of the condenser on the economy of the engine.

In general then, the proportion of heat wastes being the same, the economy will be better as the initial pressure is higher, and the back pressure is lower; or in general, as the range of pressure and temperature worked through is the greater.

We may turn next to the problem of reducing the wastes of the actual engine, as specified under the three heads above.

The waste due to radiation and conduction cannot be wholly avoided, but the former, which is by far the larger of the two, may be much reduced by suitable lagging or non-conducting covering. With such provision the loss under this head is usually very small compared with the other losses mentioned.

The waste due to the so-called initial condensation is one which may be reduced, but not wholly avoided. Before discussing the means suitable to this end some further explanation of the nature of the loss will be required.

As already pointed out, the action of the metal walls in producing this loss depends on their capacity when at a lower temperature, for absorbing heat from the steam (as during admission) and for giving it up when at a higher temperature (as during exhaust). The action depends, then, on the range of temperature between admission and exhaust, and on the particular readiness with which the walls absorb and reject heat according as they are cooler or hotter than their surroundings. There are therefore two distinct features to be considered—the range of temperature, and the readiness with which the iron absorbs and rejects heat under the conditions mentioned.

Now it is found that if the expansion through the entire temperature range is split up into a series of steps, each carried out in a cylinder by itself, the loss under consideration is less than if the entire expansion should take place in one cylinder. Carrying out this principle we have, of course, the multiple expansion engine with its total range of operation divided among several cylinders in series.

This, then, is the real significance of the multiple expansion (compound, triple, quadruple, etc.) engine, so far as its relation to economy is concerned—the splitting of the total expansion or of the total temperature range into a series of steps is found to reduce considerably one of the wastes, and so raise the actual efficiency of the engine.

Next turning to the other controlling feature of this loss,—the readiness of absorption and emission—it seems to be the case that once the internal surfaces become wetted or covered with a film of moisture, the absorption and emission of heat into and from the metal proceed with much greater readiness than when they are dry. In other words the passage of heat between metal with a moistened surface and moist steam is much more rapid than between the same metal with a dry surface and dry or superheated steam.

In the ordinary steam engine we have, therefore, an action of the walls due to the range of temperature employed, and to the natural capacity of cast-iron or steel to absorb and emit heat from and to the steam. This is further greatly augmented by the presence of a more or less complete film or layer of water over the surface, which arises from the condensation of the first entering saturated steam.

The use of superheaters, reheaters and jackets is found in a general way to decrease the readiness with which heat exchanges occur between the metal and the steam, and thus to decrease the amount of waste due to their actions. Thus in an engine using moderately superheated steam we should have the same general tendencies as noted above for the operation with saturated steam, but less augmented because of the smaller amount of moisture formed. In an engine using steam superheated to such an extent as to remain above the point of saturation during its entire passage through the cylinders, no moisture is formed and the action of the surfaces is limited to that which can take place between their dry surfaces and the dry superheated steam. The office of superheating is then simply to reduce the readiness with which the exchange of the heat between the metal surfaces and the steam is effected. The results show that in such case the reduction is real and productive of a considerable increase in economy.

Regarding the use of reheaters it seems likely that their beneficial action will be well marked in proportion as they are able to superheat the steam passing through them, and thus act as a superheater in stages, each for the cylinder next beyond.

The beneficial results gained by the use of steam jackets are in large measure due to action of the same character. The jacket containing steam at a temperature higher than that in the cylinder transfers heat into the inner surface of the cylinder walls and thus tends to keep it dry and to reduce the amount of heat exchange, and hence the corresponding waste. The steam jacket acts also to some extent to modify the character of the cycle as noted below, but most of its useful action may probably be put down to the hindering of heat exchanges between the walls and the steam in the cylinder.

It must not be forgotten, however, that whatever gain is thus effected within the cylinder is obtained at the expense of the heat drawn from the jacket, and the whole operation is therefore an attempt to reduce one loss by introducing another. If the latter is less than the saving in the cylinder, the net result will be a gain equal to their difference. If the latter is the greater of the two, the net result will be a loss, and if they are equal the net result will be no change in the economy of the engine. These relations account for the varying experience with jackets, but it now seems well assured that when properly fitted and operated, the result will show a gain of from 5 to 10 per cent. over similar conditions unjacketed.

We come now to the last principal division of the wastes of the actual steam engine,—those due to irregularities in the cycle, or in other words to variations from the routine of operations which would give the efficiency of the ideal engine as discussed above. In this respect but little can be done to improve matters. The use of jackets and reheaters may possibly affect the routine in such a way as to bring it somewhat closer to the ideal conditions, but this is by no means certain, and the benefit due to these appliances comes mostly from the decrease in cylinder condensation as explained above.

There are methods, however, of modifying the cycle of the engine by the use of a series feed water heater, in such way as to bring it somewhat nearer to the ideal cycle. Such a feed heater for a quadruple expansion engine may consist of say three chambers or heaters through which the feed passes in series. In the first it is heated by steam drawn from the L.P. receiver. It then passes on to the second chamber, where it is heated by steam drawn from the second I.P. receiver, and then goes on to the third chamber, where it meets with steam drawn from the first I.P. receiver. As the feed water thus becomes hotter and hotter it meets with steam of higher and higher temperature drawn from the successive higher receivers in the engine, and it is thus brought nearly to the temperature of the water in the boiler. The exhaust from pumps may also be turned into the first chamber, thus making it a means of taking heat from their exhaust and of returning it with the feed to the boiler. In some cases also live steam of full boiler pressure has been used in a last chamber to still further raise the temperature of the feed. Various modifications may be worked out in the details of the operation of such feed heaters, but in all cases their significance lies in the fact that the cycle of operations as a whole may in this way be brought a step nearer to the ideal cycle than would otherwise be the case. All such changes, if made in accordance with the proper principles, may therefore result in a saving of heat and in a gain in the economy of the engine, and in this fact lies the chief significance of the feed-water heater as a feature of modern engineering practice. See also Section 30 with reference to the same subject.

[2] Relation of Expansion to Economy.

The question of the *expansion of steam* and its influence on engine economy has long held an important place as perhaps the chief factor in engine economy, and it may therefore be well to refer to this feature in somewhat further detail. From the standpoint of the preceding discussion of the question we should say that the expansion of steam is favorable to economy because it brings the cycle of operations nearer to that for the ideal engine. These points may, however, be treated differently and in a more elementary manner, and we therefore proceed to discuss the question by the actual comparison of the two indicator diagrams given by an engine working with and without expansion.

Consider the two cards C D I H and C D E F H of Fig. 277. The first is the card that would be given by using the steam in an engine with stroke C D following to the extreme end, and then exhausting along D I H. The second card is such as would

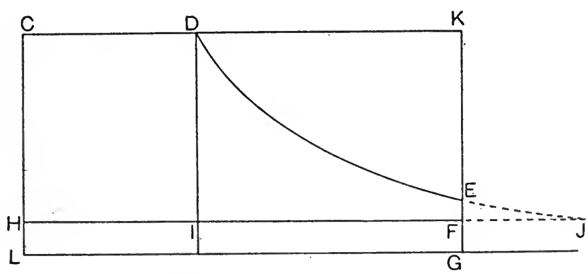


Fig. 277. The Expansion of Steam.

be given by the same steam used expansively cut-off taking place at D and the stroke continuing expansively to E. The back pressure in each case is represented by the line H F. The area C D I H represents the work in one case and the area C D E F H in the other, so that the difference or D E F I represents the gain by expansive working. In other words, if we use the steam full pressure up to D and then exhaust it, we throw away an amount of work measured by D E F I which might be saved by expansive working. It is likewise true that the exhaust opening at E causes a loss of work E J F which might be saved by continuing the expansion down until the forward pressure falls to the back pressure as at J. It is rare, however, that we can afford cylinders of sufficient size to carry the expansion to such a point, and, as may be seen, the amount thus lost is smaller and smaller as the final pressure E G is nearer the back pressure F G.

To illustrate the gain by expansive working let us suppose the initial pressure L C = 100 pounds, the back pressure L H = 3 pounds, and that the cut-off is at a series of points .1 .2 .3, etc. Then neglecting the effect due to clearance, the number of expansions will be as 10, 5, 3.3, etc., and the ratio of saving will be as given in the following table. The numbers in the column headed *e* give values of the ratio D E F I ÷ C D I H, or the ratio of the amount saved by expansion to the amount done before expansion begins.

Point of Cut-off.	Expansion Ratio.	*e*.
.1	10.	2.11
.2	5.	1.55
.3	3.30	1.18
.4	2.50	.91
.5	2.00	.69
.6	1.66	.51
.7	1.43	.36
.8	1.25	.23
.9	1.11	.11
.10	1.00	.00

It will be understood that the figures of the above table refer to indicator cards such as those of the diagram in which there is no allowance for clearance, compression, rounding off of corners, etc. These conditions are of course taken in order to simplify the necessary computations. The nature of the results would, however, be the same in the actual case, and these figures may therefore be taken as a sufficiently close indication for illustrative purposes.

It thus appears that the gain is proportionately greater the larger the number of expansions, and for the highest efficiency we should therefore carry the expansion to the highest limit. Practically there are two considerations which fix an early limit to this extension of the expansion range. The first is the limit of size. The greater the number of expansions the larger the engine and hence, especially for marine engines, we can seldom afford weight enough to give the number of expansions which other considerations might warrant. The second limitation comes from the increase of internal or cylinder condensation which increases with the number of expansions until finally the resulting waste would off-set the gain due to the increase in ideal efficiency.

This loss is decreased by the compounding of engines, so called, or by the splitting up of the total expansion into a series of steps in separate cylinders. Hence with multiple expansion engines we are able to employ higher rates of expansion without corresponding losses from cylinder condensation than with a single cylinder; and this, as we have seen in [1], is the real significance of the use of multiple expansion rather than simple engines.

[3] Economy of the Actual Engine.

We have already seen that the highest possible efficiency of an ideal engine under usual conditions will be found between 25 and 30 per cent., while the actual engine at the best will realize only some 60 or 70 per cent. of these figures, or an efficiency of say 15 to 20 per cent. in good practice. Now one horse power is 33,000 foot-pounds of work per minute, and from the value of the work equivalent of heat this is equal to $33{,}000 \div 778 = 42.42$ heat units per minute. Hence one horse power means the transformation of 42.42 heat units per minute into mechanical work. It follows that the heat which must be supplied to the engine in order to provide for one horse power will be given by dividing the number 42.42 by the efficiency at which the transformation is effected. But $42.42 \div 15 = 283. +$ and $42.42 \div 20 = 212. +$. Hence, placing the limits a little more broadly, it appears that in good practice we shall require from say 200 to 300 heat units per minute for each horse power developed in the engine. This corresponds to a range of 12,000 to 18,000 heat units per hour. Now remembering that each pound of steam brings to the engine roughly 1,000 heat units, it is clear that this will correspond to a range of steam consumption of say 12 to 18 pounds per I. H. P. per hour. These figures may be taken as covering the range of good practice from about the best at present attainable to a value only moderately fair for modern triple expansion engines, or good for the usual type of compounds.

Again each pound of coal burned may be expected to furnish under good conditions some 9,000 or 10,000 heat units to the water in the boiler, or to transform some 9 or 10 pounds of water into steam. Hence the coal required per I. H. P. per hour will be given by dividing the heat units or the pounds of steam required by the amount of either which may be expected from one pound of coal. This will give a coal consumption of from about 1.2 to 1.8 pounds per I. H. P. per hour, which may also be considered as representing the upper part of the range of good practice for triple and quadruple expansion engines under from moderately good to the best conditions at present attainable. In a few exceptional cases by the use of feed heaters, superheated steam and all means favorable to economy the consumption has been reduced to 1.0 pound coal per I. H. P. per hour.

For compound and simple condensing engines under good to moderate conditions the steam consumption will rise to from 20 to 30 pounds of steam, corresponding roughly to from 2 to 3 pounds of coal with good boiler economy, and to perhaps 2.5 to 3.5 pounds with poor boiler economy.

Farther along the line will come engines perhaps non-condensing, and of still lower efficiency, such for example as electric light, centrifugal pump, blower, winch, and steering engines. The steam required for them may rise to from 40 to 60 pounds or more per I. H. P. per hour, corresponding to a coal consumption of from perhaps 4 to 8 pounds, according to the boiler efficiency.

Still lower in the scale of economy we find the ordinary direct acting pump. Such pumps operate in the steam cylinder with almost no steam expansion, and the piston speed is very low, thus giving full time for the transfers of heat which cause cylinder condensation. Due to these and other less important causes the steam consumption may rise to 200 pounds and more per I. H. P. per hour, while rarely can it be brought as low as 100 pounds. This corresponds to a coal consumption from say 10 to 25 pounds, depending somewhat on the efficiency of the boiler. In terms of absolute efficiency these figures correspond to from about 1 to 3 per cent., the values thus ranging downward from the 15 to 20 per cent. given above as the highest values at present attainable.

Sec. 60. COAL CONSUMPTION AND RELATED PROBLEMS.

As we have already seen in Sec. 59 the coal required per I. H. P. per hour in good practice is usually found between say 1.5 and 2.0 pounds. Where especial attention is given to economy the figure may be reduced below the lower value down even to 1 pound per I. H. P. per hour, while by the neglect of due attention, or in cases where the conditions are such that economy must be sacrificed, the value may rise above the higher limit.

Let : c denote the pounds of coal per I. H. P. per hour.
H denote the I. H. P.

Then cH = pounds of coal per hour,

cH $\div$ 2,240 = tons of coal per hour,

and $\frac{24\,cH}{2240}$ or $\frac{3\,cH}{280}$ or $\frac{cH}{93.3}$ = tons of coal per day.

As a thumb rule for a quick estimate, we may remember that at a coal consumption of 1.86 pounds per I. H. P. per hour (a figure only moderately good), the coal required per day will be 20 tons per 1,000 I. H. P.

The use of the above formulae may be illustrated by the following examples:

(1) With a coal consumption of 1.78, how much coal will be required in the bunkers of a ship making a seven-day trip, the I. H. P. being 2,400 and a margin of 10 per cent. being allowed for emergencies?

Coal per day = $\frac{3 \times 1.78 \times 2400}{280}$ = 45.75 tons.

Coal for 7 days = 7 $\times$ 45.75 = 320.25, say..........320 tons.
Margin .... 32 tons.

Coal in bunkers = ...352 tons.

(2) Which will require the more coal per day, a ship with 9,800 I. H. P. at 1.82 lb. per I. H. P. per hour, or two ships each of 4,000 I. H. P. at 2.20 lb. per I. H. P. per hour?

For the first:

$$\frac{3 \times 1.82 \times 9800}{280} = 191 \text{ tons per day.}$$

For the second:

$$\frac{3 \times 2.20 \times 8000}{280} = 188.5 \text{ tons per day.}$$

Difference 2.5 tons.

(3) How long time can a vessel steam on 213 tons of coal and how far on a speed of 12 knots, the I. H. P. being 3,600 and the coal consumption being 1.68?

Coal per hour $= \frac{1.68 \times 3600}{2240} = 2.7$ tons.

Time $213 \div 2.7 = 78.9$ hours.

Distance $= 78.9 \times 12 = 947$ miles.

(4) A vessel's log shows 420 tons of coal used in a period of 9 days, 16 hours. The average I. H. P. was 2,120. What was the coal consumption per I. H. P. per hour?

Number of hours $= 9 \times 24 + 16 = 232$.

Amount used per hour $\frac{420 \times 2240}{232} = 4055$ lb.

Coal cousumption $= \frac{4055}{2120} = 1.91$ lb.

As a further development of the same problem we may often wish to find the coal burned per mile, or per ton-mile of displacement, or per ton-mile of cargo. These we may illustrate by the following examples:

(5) Given: Displacement... = 9,486 tons.
I. H. P........ = 12,000
Speed .... = 18 knots.
Coal consumption......... = 1.8 lb. per I. H. P. per hour.
Cargo .... = 2,000 tons.

Then:

Coal per hour $= 1.8 \times 12{,}000 = 21{,}600$ lb.

Coal per mile $= 21{,}600 \div 18 = 1{,}200$ lb.

Coal per ton-mile of disp. $= 1{,}200 \div 9{,}486 = .127$ lb.

Coal per ton-mile of cargo $= 1{,}200 \div 2{,}000 = .6$ lb.

(6) If the same ship were to be driven at but half the speed, only about one-eighth the I. H. P. would be required, or say 1,500 I. H. P., while the cargo might be increased to say 5,000 tons.

With the same engine economy we should then require:

Coal per hour $1.8 \times 1{,}500 = 2{,}700$ lb.

Coal per mile $= 2{,}700 \div 9 = 300$ lb.

Coal per ton-mile of displacement $= 300 \div 9{,}486 = .316$ lb.

Coal per ton-mile of cargo $= 300 \div 5{,}000 = .06$ lb.

(7) Again a case similar to one of the large modern ocean freighters:

Displacement = 27,000 tons.

Speed = 13 knots.

I. H. P. = 6,600.
Cargo = 15,000 tons.
Coal consumption = 1.3 lb. per I. H. P. per hour.
Then:
Coal per hour = $1.3 \times 6,600 = 8,580$ lb.
Coal per mile = $8,580 \div 13 = 660$ lb.
Coal per ton-mile of disp. = $660 \div 27,000 = .0244$ lb.
Coal per ton-mile of cargo = $660 \div 15,000 = .044$ lb.

At the other extreme take a torpedo-boat of the destroyer type as follows:

Displacement = 310 tons.
I. H. P. = 6,200.
Speed = 31 knots.
Coal consumption = 2.2 lb. per I. H. P. per hour.
Then:
Coal per hour = $2.2 \times 6,200 = 13,640$ lb.
Coal per mile = $13,640 \div 31 = 440$ lb.
Coal per ton-mile of disp. = $440 \div 310 = 1.42$ lb.

These examples illustrate the principle that per ton-mile, less coal is burned as the ship is larger and goes slower, while more is burned as she is smaller and goes faster. This is the result of the two facts.

(1) As the ship increases in size the power required per ton of displacement for a given speed decreases, and accordingly the larger the ship the less the coal required per ton at a given speed.

(2) For a given ship, as the speed increases, the power and hence the coal required increase nearly as the cube of the speed ratio, while the time for a mile or for a given voyage is reduced only in the simple ratio of the speeds. Thus to illustrate: If the speed is increased 10 per cent., or say from 10 knots to 11 knots, then the power will be increased nearly in the ratio $(11 \div 10)^3 = (1.1)^3 = 1.331$, while the time on the mile or on a given voyage will be decreased in the ratio $10 \div 11$. Hence the coal will be changed in the compound ratio $\frac{1.331}{1} \times \frac{10}{11}$ or 1.21 to 1. Hence it appears that in such case the increase of speed in the ratio 1.1 to 1 will increase the coal per mile or per voyage in the ratio 1.21 to 1. Or briefly an increase of 10 per cent. in the speed will mean an increase of about 20 per cent. in the total coal required. Similarly, of course, a decrease of 10 per cent in

the speed would mean a decrease of about 20 per cent. in the total coal required.

If there were no other principle involved, it would follow that the slower a given vessel goes the more cheaply could she make a given voyage. This would be true if we could consider only the power in the main engine, and assume for it a constant coal economy. This cannot be done, however, in the case of a single ship going at different speeds, because as the power in the main engines is decreased below its normal amount the coal required per I. H. P. increases continuously. Furthermore the power required for the various auxiliaries never decreases in the same ratio as the power of the main engines, and for certain auxiliaries the power is hardly affected by the change in the main engine. The coal required for the auxiliaries becomes therefore greater and greater relative to that required for the main engine. Due to these facts it follows that as the speed is decreased a point will be reached below which the saving in the total coal required per hour will be more than offset by the increase in the time required, so that for a given voyage the total coal expense will begin to increase rather than decrease. This point is known as the "most economical speed" and is the speed at which a given voyage can be made with the least expenditure of coal. Its value will depend largely on the amount and character of the auxiliary machinery in operation, but is often found at a speed somewhat above half the full power speed. In the mercantile marine it is rare that ships are operated at speeds other than those corresponding to normal full power conditions, so that the determination of a most economical speed is not of great importance in such cases. In the naval service, however, where economy may be of more importance than the reduction of time required for a voyage, ships are often operated at or about the most economical speed, and its determination and the principles fixing its location are of importance in such cases.

Sec. 61. THE LEVER SAFETY VALVE AND THE SAFETY VALVE PROBLEM.

The spring loaded safety valve as described in Sec. 17 [1] is used almost exclusively in modern practice. The ability to solve problems relating to the weighted arm safety valve (see also the same section), is, however, required of all candidates for U. S. Engineer's Licenses, so that it is of importance to thoroughly understand the method of solving the various problems which may arise in this connection. These problems are all special cases of the general problem in mechanics which has to deal with the equilibrium of a body under the action of a system of forces, and for a clear understanding of the matter the principles discussed and explained in Part II., Sec 12, must be kept well in mind.

The arrangement of a weighted arm safety valve is shown in skeleton in Fig. 278. The steam presses upward on the lower face of the valve V, and is opposed by three forces tending to keep the valve on its seat as follows:

(1) The weight of valve and spindle direct.

(2) The weight of the lever with center of gravity at some point N and pivoted at the fulcrum O.

(3) The weight proper at M acting with a leverage or arm equal to MO.

Now just as the valve is about to open, these two sets of

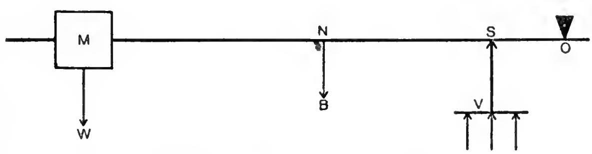


Fig. 278. The Safety Valve Problem.

forces, the one acting upward and the other acting downward, will balance. It is furthermore a principle of mechanics that when such forces are just on a balance, the product of the forces by their arms or leverages must make the same sum in each direction. Thus in the present case the up force at S may be considered as tending to cause motion of the lever about the fulcrum O, in the direction of the hands of a watch, while the down forces at B and W tend to cause motion about the same fulcrum in the opposite direction. We therefore measure the arms from the fulcrum point O.

Now if A is the area of the valve in square inches and p the steam pressure in pounds per square inch by the gauge, the total steam load on the valve will be the product pA. This acts directly upward, and, as noted above, is directly opposed, as far as it goes, by the weight of the valve and spindle. Denote this weight by V. Then the difference $(pA\text{-}V)$ is the actual or net force transmitted from the valve to the lever at S, and tending, as noted above, to turn the lever about O from left to right. Let

a denote the arm for this force, or the distance SO from the center of the spindle to the center of the fulcrum. Then $(pA-V)a$ is the product of force by arm for the upward forces. Let us save this and turn next to the remaining or downward forces. Let W denote the amount of weight at M, and l the arm or distance MO from the center of gravity of the weight to the fulcrum. Also let B denote the weight of the lever, and c the arm or distance NO, from the center of gravity of the lever to the fulcrum. Then $(Wl + Bc)$ is the sum of the products for the downward forces. By condition these are equal when the two sets balance and the valve may be considered as on the point of opening. Hence as an equation we shall have:

$$Wl + Bc = (pA-V)a \text{ or}$$

$$Wl + Bc = pAa-Va.$$

From this equation we can find the value of any one quantity which we may wish, provided we know all the others. Thus suppose we know all but W. Then we have:

$$W = \frac{pAa - Va - Bc}{l} \quad \ldots\ldots\ldots \quad (1)$$

Similarly if we know all but l we have:

$$l = \frac{pAa - Va - Bc}{W} \quad \ldots\ldots\ldots \quad (2)$$

and if we know all but p the pressure per square inch at which the valve will open with a given weight and location, we have:

$$p = \frac{Wl+Bc+Va}{Aa} \quad \ldots\ldots\ldots \quad (3)$$

We may readily express by rules the operations represented by these equations as follows:

Rule (1) To find the weight knowing the other quantities. Multiply together the pressure per square inch, the area of the valve in square inches, and the distance from the center of the valve spindle to the center of the fulcrum. From this subtract the product of the weight of the valve and spindle by the same arm SO, and also the product of the weight of the lever by its arm NO. Divide the remainder by the weight arm MO, and the quotient will be the weight desired.

Rule (2) To find the location of the weight or length of the arm MO knowing the other quantities.

Find the same difference as in rule (1) and divide by the weight W. The quotient will be the length of the arm MO.

Rule (3) To find the pressure at which a given valve and weight will lift.

Multiply the weight W by its arm MO; also the weight of the lever by its arm NO, and the weight of the valve and spindle by its arm SO. Add these three products and divide the sum by the product of the area of the valve times the arm SO. The quotient will be the pressure desired.

Example:
Let MO or $l = 28$ in.
Let NO or $c = 12$ in.
Let SO or $a = 4$ in.
Let diameter of valve $= 3\frac{1}{2}$ in.
Then area of valve or $A = 9.62$ sq. in.
Let weight of lever or $B = 5\frac{1}{2}$ lb.
Let weight of valve or $V = 4$ lb.
Let steam pressure or $p = 80$ per lb. per sq. in.
Then, $pAa = 80 \times 9.62 \times 4 = 3078.4$.
$Va = 4 \times 4 = 16$.
$Bc = 5\frac{1}{2} \times 12 = 66$.

Then $3078.4 - 16\text{-}66 = 2996.4$ and $W = 2996.4 \div 28 =$ 107 lb. in round numbers.

Or if W were known and l required, we should find the same numerator 2996.4, and then have:

$l = 2996.4 \div 107 = 28$ inches in round numbers.
Or if p is desired we should have:

$Wl = 107 \times 28$	$=$	2996
$Bc = 5\frac{1}{2} \times 12$	$=$	66
$Va = 4 \times 4$	$=$	16
Sum		3078

$Aa = 9.62 \times 4 = 38.48$.
We then have:
$p = 3078 \div 38.48 = 80$ lb. in round numbers.

General Remarks on the Problems.—The arm l is to be measured from the center of gravity of the weight W to the fulcrum or turning point O. Usually the weight is of regular form, circular or rectangular in elevation, so that its center of gravity is readily found. If the lever turns about a pin, then the arm l must be measured to the center of the pin. If it is provided with a link and knife edge bearing as in Fig. 78 then l is measured to the bearing edge. If the center of gravity of the weight W and the fulcrum O are not on the same horizontal line, then the arm l must be measured as the *horizontal* distance between verticals drawn through these points.

The center of gravity of the lever arm must be obtained practically either by measurement or by balancing on an edge in the familiar manner. If it is practically a uniform straight bar the method of measurement will be quite accurate; if it is tapering or irregular the method by balancing may be preferred. In any event, with usual proportions, as seen in the example above, the influence of the lever is relatively small, so that a slight error in the values relating to its weight or center of gravity would be of much less importance than a like proportional error in the weight W or its arm l.

The area of the valve, A, should be that, of course, of the lower face, or more accurately, of the opening at the lower edge of the seat.

The weights of the various parts are of course obtained by weighing. If this is not practicable, a fair approximation may be made by computation based on careful measurement. In such case the volumes are found by the most appropriate means according to the shape of the figure (see Sec. 77) and then by the use of the known weights of the substances per unit volume, (see p. 30) the weights may be found.

The general U. S. regulations relating to safety valves will be found among the extracts from the Rules of the U. S. Board of Supervising Inspectors given in Sec. 19.

Sec. 62. THE BOILER BRACE PROBLEM.

As we have seen in Sec. 16 all flat surfaces of any considerable size, in a boiler, require some support in addition to that which can be furnished by their own strength. In fact the whole idea of bracing is to subdivide by the braces a large flat surface into a sufficient number of smaller surfaces, each of which shall be self-supporting between the points where the braces are connected to the plate. The braces are then designed so as to be able to carry the entire load as a whole, and the parts of the plate between the braces are simply required to support, without undue change of form, the part of the load which comes upon them.

To illustrate by a diagram let Fig. 279 represent a part of a boiler head requiring bracing. Now imagine the plate entirely cut out around the dotted line, and then fitted in so exactly as to make a steam tight joint. The part thus cut out is therefore entirely separated from the remainder of the head, and without some especial support would be blown out immediately when steam was raised. Now suppose the braces to be so designed that they may be safely depended on to carry the entire load on the plate, and thus keep it securely in place in the head. It simply remains then for the parts of the plate between the braces to support themselves without losing their proper shape, and the support is thus made complete. In the actual boiler head, or in all cases where a flat surface has to resist pressure, the design of the braces is worked out exactly along these lines, and no

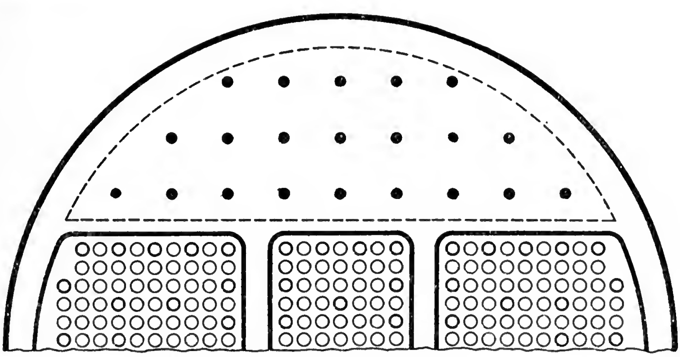


Fig. 279. Boiler Bracing.

account is taken of the strength of the plate for the general support of the load as a whole.

In designing boiler braces we have to consider two things:

(1) The total load to be supported and number of braces, or simply the load upon one brace as determined by their spacing and the steam pressure.

(2) The safe load per square inch of section of brace.

The total load depends on the area to be supported and on the gauge pressure. In figuring out the area, as in Fig. 279, it is customary to consider that a narrow strip of metal around the outside will be well supported by the shell or by the tubes. The width of such strip is usually taken as 2 or 3 inches, though there seems to be no good reason why it should not be taken as half the spacing or pitch of the braces. This amounts to considering the shell and tubes as effective bracing or support for that part of the plate near them, in the same manner and to the same extent as for the braces themselves.

If the area thus found is multiplied by the gauge pressure, the total load results. The spacing of the braces must then be taken in accordance with the principles and rules given in Sec. 19. The total number of braces is thus determined, and the average load per brace may be found by dividing the total load by the number. Or otherwise, after the spacing is decided upon, the load on each brace may be found by multiplying the area which it supports by the pressure per square inch. Thus in Fig. 280a the surface supported by the brace is considered as the dotted square, and the spacing being the same both ways, its area equals the square of the pitch. In some cases the spacing is not the same in both directions, as in Fig. 280b. In such case the

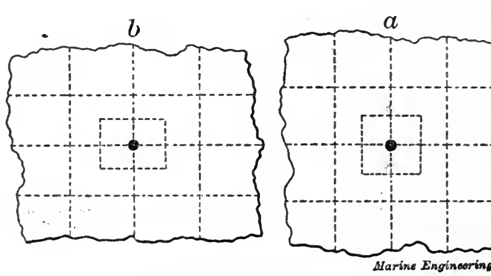


Fig. 280. Boiler Bracing.

supported area is found by multiplying together the two pitches. Sometimes, again, the braces are irregularly distributed, and the area supported by one brace may be roughly triangular or of other irregular shape. In such case the area which the brace may be fairly called upon to support must be taken by approximation, using the best judgment which can be brought to bear on the special circumstances.

When the braces are arranged in rows and columns as in Fig. 279, it will usually be found that due to the necessary spacing about the edges, the average load is slightly less than that which would correspond to an entire square or rectangle, and in consequence it is usually safer to take the load as determined directly by the area supported. Thus in Fig. 280a, let the side of the square be 7 inches: then the supported area is 49 square inches. Likewise in Fig. 280b, let the pitch be 14 inches in one direction and 16 inches in the other: then the supported area equals 14 × 16 or 224 square inches. Again in Fig. 67, suppose that three braces are to be used to support the approximately triangular area on the back tube sheet. In such case we make a fair allowance for the support about the edges, sketch in the area which the braces may be called on to support, sketch in the braces so as to divide the area as evenly as possible, and either compute the area of the whole triangle and divide by 3, or compute the smaller areas separately.

Turning now to the second chief question, the safe load to be allowed per square inch of section of brace, we find that the U. S. Rules provide that iron braces shall not be allowed more than 6,000 lb. per square inch of section; while for steel, if inspected according to regulation, the allowance may be as follows: From 1¼ in. diameter to 2½ in. diameter, not to exceed 8,000 lb.; and above 2½ in. diameter, not to exceed 9,000 lb., each per square inch of section. (See also Sec. 19).

It must be noted that in all cases the diameter is measured at the root of the thread or at the smallest section. For this reason the threads are usually *raised* so that the diameter at the bottom is not less than that of the body of the brace.

We have thus discussed the determination of two necessary items—the load which the brace is to support, and the safe load per square inch of section. It is clear then that if we divide the latter into the former, the quotient will be the necessary cross sectional area of brace. Having found the area, we find the diameter by means of a table of diameters and areas, or by the proper rule or formula of mensuration. See Part II., Sec. 9 [10].

These various operations may be expressed in the form of a rule as follows:

(1) Find the area to be supported by one brace, and multiply this by the gauge pressure per square inch. The product will be the load to be supported by the brace.

(2) Take the safe load per square inch of section in accordance with the rule above given.

(3) Divide the total load as found in (1) by the safe load as taken in (2) and the quotient will be the necessary area in square inches.

(4) Find the corresponding diameter either by the help of a suitable table or by means of the proper formula or rule of mensuration.

To illustrate the foregoing a few examples will be of aid.

(1) In Fig. 279, suppose the total area to be supported is found by measurement to be 3,784 square inches, the steam pressure being 160 pounds gauge. Find the total load.

Ans. Load $= 3784 \times 160 = 605{,}440$.

(2) In Fig. 280a, suppose the braces spaced 14 inches each way, the steam pressure being the same as in (1). Determine the load on one brace.

Ans. Load $= 14 \times 14 \times 160 = 31{,}360$.

(3) Suppose, instead, that we wished to space the braces 14 inches one way and 16 inches the other. Find the load on one brace.

Ans. Load $= 14 \times 16 \times 160 = 35{,}840$.

(4) Suppose that we space screw staybolts 6 inches x 6 inches. Find the load on one brace.

Ans. Load $= 6 \times 6 \times 160 = 5{,}760$.

(5) Suppose in (4) that the spacing were $7 \times 6\frac{1}{2}$. Find the load.

Ans. Load $= 7 \times 6\frac{1}{2} \times 160 = 7{,}280$.

(6) What would be the area and diameter of a steel brace in (2), allowing 8,000 lb. per square inch of section?

Ans. Area $= 31{,}360 \div 8{,}000 = 3.92$ square inches.

Corresponding diameter $= 2\frac{1}{4}$ inches nearly.

(7) What would be the area and diameter of a steel brace in (3), allowing 8,000 lb. per square inch of section?

Ans. Area $= 35{,}840 \div 8{,}000 = 4.48$ square inches.

Corresponding diameter $= 2$ 7-16 scant. Probably $2\frac{1}{2}$ inches would be employed.

(8) What would be the area and diameter of an iron stay in (4), allowing 6,000 lb. per square inch of section?

Ans. Area $= 5{,}760 \div 6{,}000 = .96$ square inches.

Corresponding diameter $= 1$ 1-8 inches nearly.

(9) What would be the area and diameter of an iron stay in (5), allowing 6,000 lb. per square inch of section?

Ans. Area $= 7{,}280 \div 6{,}000 = 1.213$.

Corresponding diameter $= 1\frac{1}{4}$ inches nearly.

(10) Suppose that screw stay bolts are spaced 7 x 7 inches, the steam pressure being 200 lb. gauge. Find the area and diameter of a steel stay, allowing 8,000 lb. per square inch of section.

Load $= 7 \times 7 \times 200 = 9{,}800$ lb.

Area $= 9{,}800 \div 8{,}000 = 1.225$ square inches.

Corresponding diameter = 1¼ inches nearly.

Load on Oblique Braces. In case the brace is not at right angles to the surface to be supported, proper allowance must be made for the increase of load on the brace due to the angle of obliquity. This problem is explained in Part II., Sec. 12 [14] (14).

Load on Forked Ends of a Brace. The load on the forked ends of a brace, as in Fig. 59, is greater than one-half the load on the brace, in a ratio depending on the angle of obliquity. This problem is explained in Part II., Section 12 [14] (13).

Sec. 63. STRENGTH OF BOILERS.

In order to examine the relation of the strength of a boiler shell to its diameter, thickness and the steam pressure, consider

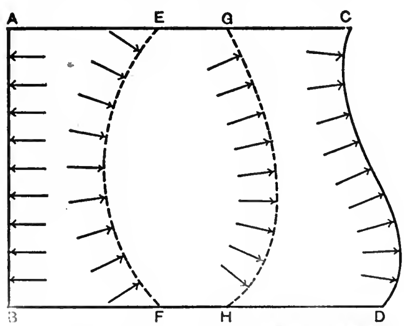


Fig. 281. The Strength of Boilers.

first a hollow chamber, as in Fig. 281, with parallel sides, AC and BD, a face AB perpendicular to these sides, and for the other end any other curved or irregular surface CD. Let this contain steam under pressure. Now it is a well-known fact of experience that under such circumstances the chamber will remain in equilibrium, and it will not move as a whole, and in particular will move neither to the right nor to the left. Hence the internal force acting to the right must equal that to the left. But the force acting to the right is the total resultant of all the forces acting on the curved surface CD, while the force acting to the left is the resultant of the parallel forces acting on the plane face AB. Hence numerically these two resultants must be equal, and this will be the same, no matter what the shape of the surface on the right, as, for example, EF or GH. Now AB is called the projected area of any curved surface, such as CD, EF or GH, the direction of projection being of course parallel to the sides AC and BD. Hence we may say that the total resultant force in any direction due to the pressure of steam or of any gas or vapor acting on the curved surface will equal the pressure of the projected area of such surface, the projection being taken in the direction of the resultant desired. This is a very general and very important principle in mechanics, and has many applications, one of which is to the problem of the strength of a boiler, as we will proceed to show.

In Fig. 282 let ABCD denote a cross section of a cylindrical boiler with steam pressure acting on the curved surface, as denoted by the arrows. Suppose a plane of division AB, and let us consider what it is which keeps the two halves from separating under the action of the steam pressure. The surface ACB is

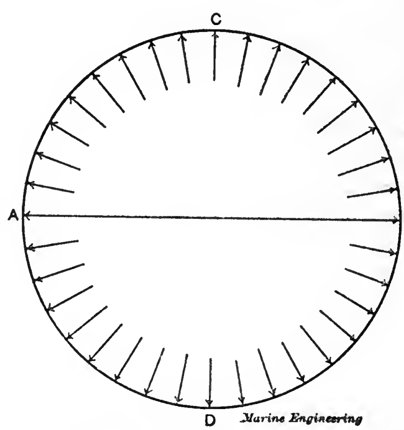


Fig. 282. The Strength of Boilers.

urged upward and the surface ADB is urged downward, while they are prevented from separating by the strength of the material at A and B.

Now the force tending to thus separate the two parts is evidently measured by the force urging ACB upward or ADB downward. As we have just seen, this equals the force on the projected area which is represented by AB. Suppose the axial length of the section which we are considering to be one unit, or one inch, and denote the diameter by d, the radius by r and the thickness of the shell by t. Then the area of AB equals d square inches, and if p is the pressure per unit area the total load on AB is pd. But as we have seen this is numerically the same as the force which urges ACB upward and ADB downward, and which is opposed simply by the strength of the material at A and B. The cross sectional area on each side will be $t \times 1$ or t; hence the total area of material will be $2t$. Let S denote the stress developed in the material per square inch of section. Then $2tS$ is the total stress developed, and this must equal the load pd. Hence we have the equation $2tS = pd = 2pr$. (1)

$$\text{or} \qquad t = \frac{pd}{2S} \tag{2}$$

$$\text{and} \qquad p = \frac{2tS}{d} = \frac{ts}{r} \tag{3}$$

In these equations (3) gives the value of the steam pressure p, which would produce the stress S in the metal of thickness t. If, however, a shell, as in Fig. 282, were formed with riveted joints the strength of the metal in the joint would be less than that of the plate itself in the ratio given by the efficiency of the joint as discussed in section 15. Hence, if S is to be the safe working stress in the metal of the joint, and e is the efficiency, the working pressure p must be reduced in the ratio of the efficiency, or from p to ep, in order to keep the stress in the joint down to the value S. Also if T is the ultimate strength of the metal, we do not wish S to rise above a certain fraction of T. The number by which we divide T to find the safe stress S is called the factor of safety. Denote this factor by f. Then in (3) substituting for S its value $T \div f$ and allowing for the efficiency of the joint we have:

$$p = \frac{2etT}{df} = \frac{etT}{rf} \tag{4}$$

$$\text{and} \qquad t = \frac{fpd}{2eT} \tag{5}$$

In a similar manner let us consider the strength of the boiler to withstand rupture around the shell. In this case the area of the head is $\pi d^2 \div 4$ and the load is $p\pi d^2 \div 4$. The section of metal carrying this load is measured by the circumference multiplied by the thickness, or by πdt. Hence if S is the stress developed per unit area, the total stress in the metal carrying the above load is πdts. Equating the load and the total developed stress we have:

$$\pi dtS = \frac{\pi pd^2}{4}$$

$$\text{or} \qquad p = \frac{4ts}{d} = \frac{2ts}{r} \tag{6}$$

Comparing this with (3) it is seen that in the case of the shell without seam or joint the pressure necessary to produce rupture around the circumference will be just twice that required for rupture along the length; or, in other words, the boiler is twice as strong for rupture around the circumference as for rupture along the length, and this is an important principle which should be borne in mind in dealing with questions relating to the strength of a cylinder against pressure from within. It also follows that the longitudinal seams must be made with the greatest care and of the highest efficiency, while joints of lower efficiency, so long as they insure tightness against leagage of steam, will be sufficient for the circumferential seams. To take account of the factor of safety and of the efficiency of the joint we must introduce in (6) the factors f and e, the same as in (3). This will give:—

$$p = \frac{4\,etT}{fd} = \frac{2\,etT}{fr} \qquad (7)$$

and
$$t = \frac{fpd}{4eT} \qquad (8)$$

For a bumped boiler head, as referred to in section 19, we consider that the head is a part of a sphere, and that all parts of such a surface are equally strong. Now for a sphere as a whole we have for the total load on a circumferential section the pressure p multiplied by the projected area of the hemisphere. But the latter is $\pi d^2 \div 4$, and therefore the load is $\pi pd^2 \div 4$. The total section of metal is πdt, snd if S is the stress developed per unit area, then the total stress is πdtS. Hence we have:

$$\frac{\pi\,pd^2}{4} = \pi\,dts$$

or
$$p = \frac{4ts}{d}$$

But, as seen above, this is the same as the value for a cylindrical shell for rupture around the circumference. Hence we have the principle that a sphere has the same strength in all directions as a cylinder of equal diameter for rupture around the circumference. It will be noted that this relates simply to the strength of a sphere or part of a sphere for pressure on the concave surface. For the strength of a head bumped inward or convex on the inside, there is no method of treatment by simple mechanics.

For the mechanics involved in the computations relating to plain boiler bracing reference may be made to section 62 and to Part II., section 12.

Sec. 64. LOSS BY BLOWING OFF.

In the days of the jet condenser, and when blowing off to reduce the density of the water in the boilers was the usual practice, the loss of heat occasioned by this operation was necessarily the subject of consideration, and it became necessary to be able to compute this loss in any given case. This is most easily done by the simple application of algebraic methods.

Let F denote the pounds of feed water in any given time, and f its density.

Let B denote the pounds of water blown out in the same time and b its density.

Then $(F-B)$ = pounds of water evaporated into steam in the same time. Likewise fF represents the amount of solid matter brought into the boiler during the given time, and bB represents similarly the amount blown out in the same time. Since the density of the water in the boiler remains constant at b, the amount of solid matter in the water must remain constant, and hence as much must be blown out as comes in by the feed, or:

$$fF = bB \qquad (1)$$

From this we readily derive the following relations:

$$\frac{F}{B} = \frac{b}{f} \qquad (2)$$

$$\frac{F-B}{B} = \frac{b-f}{f} \qquad (3)$$

Now let t_1 = temperature of feed,

t_2 = temperature of steam,

H = total heat in one pound of steam at given pressure.

Then $B\,(t_2 - t_1)$ = heat blown out, and $(F-B)\,[H-(t_1-32)]$ = heat put into the steam formed.

Then $(F-B)\,(H+32) + Bt_2 - Ft_1$ = total heat.

Hence ratio of loss e is given by:

$$e = \frac{B\,(t_2 - t_1)}{(F-B)\,(H+32) + Bt_2 - Ft_1}$$

By the aid of the ratios above, this expression is readily reduced to the following form:

$$e = \frac{f\,(t_2 - t_1)}{(b-f)\,(H+32) + ft_2 - bt_1}$$

These algebraic operations may be expressed by the following:

Rule—(1) Multiply the density of the feed water by the difference between the temperatures of the steam and of the feed.

(2) Subtract the density of the water blown out from the density of the feed.

(3) Add 32 to the total heat of 1 lb. steam.

(4) Multiply together the results in (2) and (3).

(5) Multiply the density of the feed by the temperature of the steam.

(6) Multiply the density of the water blown out by the temperature of the feed.

(7) Add the results in (4) and (5) and subtract from the sum the result in (6).

(8) Divide the result in (1) by that in (7) and the quotient expressed in per cent will give the percentge of loss.

Examples: (1) Density of feed, $f = 1$.
Density maintained in boiler, $b = 2$.
Pressure of steam = 100 pounds, gauge.
Temperature of feed $t_1 = 100°$.
Then from tables: $t_2 = 337.8$,
and $H = 1185$.

$$\text{Then loss ratio, } e = \frac{337.8 - 100}{(1185 + 32) + 337.8 - 2 \times 100}$$

$$\text{or} \qquad e = \frac{237.8}{1354.8} = 17.5 \text{ per cent.}$$

We may follow through the details somewhat differently, as follows:

The loss of heat per pound of water blown off equals $(t_2 - t_1)$. This equals 337.8—100 or 237.8.

The heat required per pound of water evaporated is $H-(t_1-32)$. This equals 1185—(100-32) or 1117. Now from (3) it appears that the amount evaporated is to the amount blown out as $(b-f)$ is to f or as 1 is to 1. That is, the amount evaporated equals the amount blown out. Hence for every 1117 heat units put into a pound of steam 237.8 are lost. Hence the percentage loss on the total heat employed is $237.8 \div (237.8 + 1117)$

$$\text{or} \qquad e = \frac{237.8}{1354.8} = 17.5 \text{ per cent. as before.}$$

(2) Density of feed, $f = \frac{7}{8}$.
Density maintained in boiler, $b = 1\frac{5}{8}$.

Steam pressure = 60 pounds gauge.
Temperature of feed $t_1 = 92°$.
Then from tables: $t_2 = 307.4$,
and $H = 1175.7$.

Then percentage of loss,

$$e = \frac{\frac{7}{8}(307.4 - 92)}{\frac{3}{4}(1175.7 + 32) + \frac{7}{8} \times 307.4 - 1\frac{5}{8} \times 92}$$

or $e = \dfrac{7 \times 215.4}{6 \times 1207.7 + 7 \times 307.4 - 13 \times 92} = \dfrac{1507.8}{8202} = 18.4$ per cent.

Or again by analysis: The loss of heat per pound of water blown off equals $(t_2 - t_1)$. This equals $307.4 - 92 = 215.4$. The heat required per pound of water evaporated is $H - (t_1 - 32)$. This equals $1175.7 - (92 - 32) = 1115.7$. Now from (3) it appears that the amount evaporated is to the amount blown out as ¾ is to ⅞ or as 6 : 7. Hence for every 6 lbs. evaporated there will be 7 lbs. blown out. The corresponding loss of heat is $7 \times 215.4 = 1507.8$. The corresponding amount of heat put into steam is $6 \times 1115.7 = 6694.2$. The total heat used is $1507.8 + 6694.2 = 8202$. The percentage of loss will be then

$$e = \frac{1507.8}{8202} = 18.4 \text{ per cent. as before.}$$

Sec. 65. GAIN BY FEED WATER HEATING.

As we have seen in Sec. 18 a certain fraction of the heat is lost by way of the funnel. In certain forms of feed-water heaters, a part of this loss is prevented by placing the heater at the base of the funnel to absorb the heat of the gases after they have left the tubes. In water tube boilers such arrangements are especially common, the heater consisting usually of a continuous coil of pipe jointed up with elbows or return bends, and through which the feed-water passes before going to the upper drum, or point of regular feed entrance.

It thus becomes a question of interest as to how much saving may be effected by the feed-water heater thus arranged to utilize a part of the heat in the waste funnel gases. This will be best illustrated by an example.

(1) Temperature of feed 110°. Pressure of steam 160 lb. gauge. Assuming dry steam, what will be the percentage gain by heating the feed-water to 170°?

From Table I for the first condition:

Total heat in 1 lb. steam = ...	1195
Heat in feed-water = 110—32 = ...	78
Heat required to form 1 lb. steam = ...	1117

For the second condition the heat units saved are measured by the difference in the feed-water temperatures, or in this case by 170 — 110 = 60.

Hence 60 heat units have been saved out of 1117, and in this case the heat required per pound of steam formed will be 1117 — 60 = 1057.

The percentage saving is measured by 60 ÷ 1117 = 5.7 per cent.

In case the feed-water is heated by exhaust steam which is not sent to the condenser, and of which the heat would be otherwise wasted, the gain is found in the same manner by dividing the rise in the temperature of the feed-water by the number of heat units needed to form one pound of steam without the heater.

In case the feed-water is heated by live steam from certain of the receivers, or by any steam which might otherwise have been used or the heat of which might have been saved, then the question of heater economy becomes much more complicated and cannot be determined by any process of simple computation. It becomes simply a question of where it is most advantageous to use the heat, whether in the heater or elsewhere, a question which in general can only be answered by the actual trial. See also Sec. 30.

Sec. 66. THE PROPORTIONS OF CYLINDERS FOR MULTIPLE EXPANSION ENGINES.

The total expansion of the steam in the multiple expansion engine is attained by expanding it in the high pressure cylinder from the point of cut-off to the end of the stroke, and then handing it over to a series of cylinders of continually increasing size until the steam which first filled the H. P. cylinder to the point of cut-off, finally fills the L. P. cylinder, and the expansion is complete. It would seem at first that the total number of expansions would be given by dividing the volume of the L. P. cylinder by that of the H. P. up to the point of cut-off. It is not quite true, however, that the volume of the entering steam is measured by the volume of the H. P. up to the point of cut-off, nor that its final volume is that of the L. P. cylinder. These simple relations are modified by the clearance in the manner described in Sec. 68. Due to this effect the actual number of expansions will usually be from .5 to 1 less than the apparent number given by dividing the H. P. volume up to the cut-off into the L. P. volume.

The number of expansions suitable in any given case will vary with the initial steam pressure and with the other conditions to be fulfilled.

With steam having an initial pressure of 150 to 180 pounds and used in triple expansion engines the number will usually vary from say 8 to 12; toward the lower values as the importance of the development of power per ton of machinery is greater, and the importance of coal economy is less, and toward the higher limit and perhaps even beyond in the reverse cases. With higher steam pressure, say from 180 to 220 pounds, and quadruple expansion engines, the number of expansions will be commonly found between 10 and 15, varying in one direction or the other according to the same general considerations as given above for the lower pressures.

Of this total expansion range not more than 1.4 to 1.6 is usually obtained in the H. P. cylinder with the usual cut-off between .55 and .75 of the stroke, and taking into account the effect of the clearance. This leaves the remainder to be obtained from the ratio between the volumes of the H. P. and L. P. cylinders, and assuming the same stroke this will equal the ratio between the areas of the cylinders. Hence with from 8 to 12 total expansions the ratio between the piston areas of the L. P. and H. P. will usually be found say from 5 to 7, while with a higher steam pressure and from 10 to 15 total expansions the ratio will be from say 7 to 10.

We shall not here enter into the details of the proportions of the cylinders of multiple expansion engines, and it will be sufficient to add to the foregoing the following simple rules by which suitable values for the diameters of intermediate cylinders may be found having given those of the high and low.

(a) For triple expansion engines.

(1) Take the square root of the H. P. diameter.

(2) Take the square root of the L. P. diameter.

(3) Multiply together the results of (1) and (2), and the result will give a value for the intermediate diameter.

Example: Diam. of L. P. = 50″.
Diam. of H. P. = 20″.25.

$$\sqrt{50} = 7.07.$$
$$\sqrt{20.25} = 4.5.$$
$$4.5 \times 7.07 = 31.8.$$

It is usually considered better to take the actual value slightly under rather than over the value given by the rules, and we may therefore take 31 or 31 ½ as a suitable diameter for the intermediate cylinder.

(b) For quadruple expansion engines.

(1) Take the cube root of the H. P. diameter.

(2) Take the cube root of the L. P. diameter.

(3) Square the result found in (1).

(4) Multiply together the results found in (2) and (3) and the product will give a value for the diameter of the first M. P.

(5) Square the result found in (2).

(6) Multiply together the results found in (1) and (5) and the product will give a value for the diameter of the second M. P.

Example: Diam. of H. P. = 27.
Diam. of L. P. = 80.

$$\sqrt[3]{27} = 3.$$
$$\sqrt[3]{80} = 4.31.$$
$$(3)^2 = 9.$$
$$9 \times 4.31 = 38.79.$$
$$(4.31)^2 = 18.58.$$
$$3 \times 18.58 = 55.74.$$

Here also it is usually considered better to take the actual diameters slightly under rather than over the values given by the rule. Hence in taking shop dimensions we may go under rather than over, and in the present case take say 38 and 55 as suitable values for the diameters of the two M. P. cylinders.

Sec. 67. CLEARANCE AND ITS DETERMINATION.

The term clearance is used in two senses. Clearance proper denotes the actual distance between the face of the piston and that of the cylinder head when the former is at the end of the stroke. That is, it is the least distance between the piston and the cylinder head. In amount it may vary from ¼ to ½ or ¾ inch, being naturally larger the larger the engine.

The clearance volume or the percentage clearance on the other hand is the actual volume contained between the face of the valve and the face of the piston when the latter is at the end of the stroke, or it is such volume expressed as a percentage of the volume swept by the piston. The clearance should be determined either by measurement and computation from the drawings, or by filling it with water and measuring the amount required. There are several methods of procedure. In the first place the valve must be disconnected and blocked in mid position, thus covering the ports. Care must also be taken to provide by the use of putty, if necessary, against leakage at either the valve or piston. Then place the engine on the center and by means of the indicator pipe fill the clearance volume with water by pailfuls, weighing each pailful before pouring in, and the amount left over in the last pailful. Then knowing the weight of the pail, the total weight of water poured in may be found. This reduced to volume by taking 62.5 lb. to the cubic foot will give the clearance volume in cubic feet, and this divided by the volume of the piston displacement will give the clearance percentage. If salt water were used, 64 instead of 62.5 would be used in reducing to volume.

Somewhat differently the mode of procedure may be as follows: Place the engine just one inch off the center as shown by measurements on the guides. Fill up the volume as before and note the weight required. Then move the engine up to the center slowly, catching the water as it is forced out and weighing as before. The amount forced out corresponds to 1 inch of piston displacement. Subtract this amount from the total, and the remainder represents the water in the clearance. Divide the latter by the amount representing one inch of piston travel, and the quotient is the number of inches corresponding to the clearance. This divided by the stroke will give the clearance percentage.

As an illustration of the first mode of procedure, suppose diameter $= 22$ inches, stroke $= 40$ inches, weight of water to fill clearance $= 85$ pounds. The volume of clearance $= 85 \div 62.5 = 1.36$ cubic feet. The volume of piston displacement $= 3.1416 \times 11 \times 11 \times 40 \div 1728 = 8.8$ cubic feet, nearly. Hence clearance percentage $= 1.36 \div 8.8 = 15.45$ per cent.

For the second mode of procedure let the figures be as follows: Total weight of water with engine 1 inch off center $= 99$ pounds. Weight of water forced out when engine is brought to center $= 13.5$ pounds. Difference $= 85.5$ pounds. Then 13.5 pounds represents 1 inch of piston travel, and 85.5 pounds the whole clearance. Hence 85.5 ÷ 13.5 = 6.33 inches = number of inches of piston travel giving a volume equal to that in the clearance. Hence 6.33 ÷ 40 inches = 15.8 per cent. = clearance percentage.

Sec. 68. THE EFFECT OF CLEARANCE IN MODIFYING THE APPARENT EXPANSION RATIO AS GIVEN BY THE POINT OF CUT-OFF.

As we have seen in Sec. 67, the clearance volume is defined as the volume or space between the piston when at the end of the stroke and the face of the valve. It comprises the "clearance proper" or space between the piston when at the end of the stroke and the cylinder head, together with the volume of the ports or passages leading from the valve face to the cylinder.

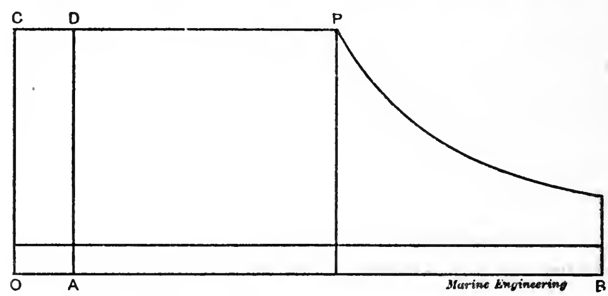


Fig. 283. The Effect of Clearance on the Expansion Ratio.

The volume of the clearance expressed as a fraction of the volume swept by the piston is usually known as the clearance ratio or per cent., and is usually found in marine practice from .10 to .15, though in some cases it may rise as high as .20. The steam within this volume takes part, of course, in all expansions and compressions to which the steam in the cylinder as a whole is subjected, and its influence on the apparent expansion ratio must therefore be considered.

If there were no clearance volume, then the expansion ratio would be given by dividing the total volume swept by the piston, by the volume up to the point of cut-off. But this would be the same as taking the reciprocal of the cut-off ratio. Thus, for example, if the cut-off were at 1/2 stroke the expansion ratio would be 2; if at 1/3 stroke, 3; if at 2/3 stroke, 3/2 or 1.5, etc. With a clearance volume, however, this is modified as shown by Fig. 283. Let AB denote the volume swept by the piston and OA the clearance volume to the same scale, or otherwise let AB denote the length of the stroke and OA the clearance volume reduced to stroke by dividing the volume by the piston area. Then if cut-off is at some point X the actual volume of steam within the cylinder and ready to expand is denoted by OX rather than by AX. Again at the end of the stroke when the piston reaches B, the final volume of the steam is OB. Hence the real expansion ratio is OB/OX and denoting its value by *r* we have:

$$r = \frac{AB + OA}{AX + OA}$$

Now dividing both numerator and denominator of this fraction by AB we have:

$$r = \frac{1 + \frac{OA}{AB}}{\frac{AX}{AB} + \frac{OA}{AB}}$$

Now AX ÷ AB is the cut-off ratio, and OA ÷ AB is the clearance ratio or per cent. Denote the first of these by *a* and the second by *c*. Then we have:

$$r = \frac{1 + c}{a + c}$$

Examples:

(1) Cut-off at ½ stroke, clearance 10 per cent.

Find the true expansion ratio.

Operation: $a = \frac{1}{2} = .50.$

$c = 10$ per cent. $= .10.$

Hence $r = \frac{1.00 + .10}{.50 + .10} = \frac{1.10}{.60} = 1.83$ Ans.

(2) Cut-off at 60 per cent, clearance 15 per cent. Find the true expansion ratio.

Operation:— $r = \frac{1.00 + .15}{.60 + .15} = \frac{1.15}{.75} = 1\ 53$ Ans.

Sec. 69. ENGINE CONSTANT.

As seen in Sec. 55 [3] we have for the horse power formula:

$$H = \frac{2pLAN}{33{,}000} = (pN)\left(\frac{2\,LA}{33{,}000}\right)$$

Now the factors $2LA \div 33{,}000$ are always the same for any one cylinder, while the other two (pN) will vary according to the conditions. We may therefore compute in advance the value of the factor ($2LA \div 33{,}000$) and then to find the H.P. we shall have simply to multiply these by the other two, of which one, p, is found from the cards, and the other, N, from the counter. This factor $2LA \div 33{,}000$ is called the "engine constant," and is often thus computed as a matter of convenience, especially when large numbers of cards are to be worked up.

For the power in one end of the cylinder only we have simply to take the factors $LA \div 33{,}000$ with N and the value of p found from the corresponding card. To allow for the area of the piston-rod on the lower side of the piston in the formula for the full power, we may use the average area top and bottom with the average mean effective pressure. When there is a difference in the values of the mean effective pressure top and bottom, this will not give quite the same result as if the two ends were taken separately. The difference, however, is in all ordinary cases quite unimportant. See also Sec. 55 [3].

Example: Given a cylinder of diam. = 36″, stroke = 42″, diam. of piston-rod = 5″. Find the constant neglecting the piston-rod, and also allowing for it as above explained.

Area of cylinder = 1017.9 sq. in.
Stroke = 3.5 feet.
Constant = $(2 \times 3.5 \times 1017.9) \div 33{,}000 = .2159$.

Next to allow for the piston-rod we have for its area 19.6 square inches. Taking this from 1017.9 we have 998.3 square inches as the area of the lower side of the piston. The mean of the upper and lower sides is then 1008.1. It may be noted that in all such cases the mean may be most easily obtained by taking from the upper area one-half the piston-rod area, or in this case, by taking 9.8 from 1017.9, giving 1008.1 as above. We then have:

Constant = $(2 \times 3.5 \times 1008.1) \div 33{,}000 = .2138$.

Sec. 70. INDICATED THRUST.

The indicated thrust in pounds may be defined as the indicated power in foot pounds divided by the product of the pitch of the propeller multiplied by the revolutions per minute.

Let:
H = I. H. P.
p = pitch of propeller.
N = revolutions per minute.
T = indicated thrust.

Then by formula:

$$T \text{ in pounds} = \frac{33,000\ H}{pN}$$

This may be reduced to tons by dividing by 2,240, and we have T in tons $= \frac{33,000\ H}{2.240\ pN} = \frac{14.73\ H}{pN}$

Dividing the above value of T in pounds by the value of the reduced mean effective pressure as given by equation (1) in Sec. 71, and we have:

$$\frac{T}{p_m} = \frac{33,000\ H}{pN} \times \frac{2LNA}{33,000\ H} = \frac{2\ LA}{p}$$

It thus appears that the ratio of the indicated thrust to the reduced mean effective pressure is measured by twice the stroke times the L.P. area divided by the pitch of the propeller. All of these are constants for any given engine, and it thus follows that the ratio between the indicated thrust and the reduced mean effective pressure is a constant, or in other words that the former is in a constant ratio to the latter. The indicated thrust which is often considered as a rather vague quantity is thus related to the reduced mean effective pressure, a much better known quantity.

The indicated thrust may also be considered as the actual thrust which would be exerted if the propeller worked without slip, and if all the power developed in the cylinders were used in driving the ship forward at the speed thus produced. Actually a part of the power is lost in the friction of the engine and in the water due to the operation of the propeller, while the latter does not operate without slip. In consequence the actual thrust exerted on the thrust block is usually found somewhere about two-thirds the indicated thrust, computed as above.

Example:

The I. H. P. is 1,640, the pitch 13 feet, and revolutions 148. Find the indicated thrust in pounds and in tons:

$$T \text{ in pounds} = \frac{33,000 \times 1,640}{13 \times 148} = 28,130$$

T in tons $= 28,130 \div 2,240 = 12.56.$

Sec. 71. REDUCED MEAN EFFECTIVE PRESSURE.

In the multiple expansion engine the power, as we know, is developed in the various cylinders, as equally as the designer is able to bring about. The *reduced mean effective pressure* may be defined as the mean effective pressure which, acting in the low pressure cylinder alone with the same piston speed, would produce the same power as the actual engine with its series of cylinders.

Taking the usual formula for power, as in Sec. 55 [3] we have:

$$H = \frac{2pLAN}{33,000}$$

and solving for p we have

$$p = \frac{33,000\ H}{2\ LAN} = \frac{33,000\ H}{(2\ LN)A} \qquad (1)$$

Hence if the entire power were to be developed in the L. P. cylinder the necessary mean effective pressure would be found by the operations indicated by this equation, and such would be the reduced mean effective pressure, or the mean effective pressure reduced to the L.P. cylinder. The operations indicated by the above equation may be expressed by a rule as follows:

Rule:

(1) Multiply the indicated horse power by 33,000.

(2) Multiply twice the length of the stroke in feet by the revolutions (giving piston speed) and this by the area of the low pressure piston in square inches.

(3) Divide the result found in (1) by that found in (2) and the quotient is the reduced mean effective pressure desired.

To obtain a somewhat different expression for the reduced mean effective pressure denote the areas of the three pistons H.P., I.P. and L.P. of a triple expansion engine, for example, by A_1, A_2, A_3, and the corresponding values of the mean effective pressure in these cylinders by p_1, p_2, p_3. Then the total power H of the formula (1) above may be expressed as follows:

$$H = \frac{(2LN)p_1A_1 + (2LN)p_2A_2 + (2LN)p_3A_3}{33,000}$$

According to (1) this value of H is to be multiplied by 33,000 and divided by 2 LN times A_3 the L.P. piston area. This will give the following as the value of the reduced mean effective pressure:

$$p = p_1\frac{A_1}{A_3} + p_2\frac{A_2}{A_3} + p_3 \qquad (2)$$

According to this formula the procedure for finding the reduced mean effective would be as follows:

(1) Divide the mean effective for the H.P. cylinder by the ratio between the L.P. and H.P. piston areas. This reduces the

H.P. mean effective to the L. P. piston.

(2) Divide the mean effective for the I.P. cylinder by the ratio between the L.P. and I.P. piston areas. This reduces the I.P. mean effective to the L.P. piston.

(3) Add together the results found in (1) and (2) and the mean effective for the L. P. The sum will be the total mean effective reduced to the L.P. piston.

Example:

Given for a triple expansion engine the following:

Diam. H.P. cylinder	= 24″
" I.P. "	= 38″
" L.P. "	= 60″
Length of stroke	= 42″
Revolutions	= 106

From sets of indicator cards suppose the mean effective pressures found as follows:

For the H.P. $p_1 = 61.9$
" " I.P. $p_2 = 30.2$
" " L.P. $p_3 = 13.8$

Find the total I. H. P. and the reduced mean effective pressure.

For the piston areas we have from a table of areas of circles:

$A_1 = 452.4$
$A_2 = 1134.1$
$A_3 = 2827.4$

Then finding the I H. P. in each cylinder we have as follows:

I. H. P. in H. P. cylinder = ...	629.6
I. H. P. in I. P. cylinder = ...	770.2
I. H. P. in L. P. cylinder = ...	877.2
Total I. H. P. = ...	2277.0

Then according to rule (1) for the reduced mean effective:

$$p = \frac{33,000 \times 2277}{7 \times 106 \times 2827.4} = 35.82$$

According to rule (2) for the same we should have as follows:

$$p = 61.9 \times \frac{452.4}{2827.4} + 30.2 \times \frac{1134.1}{2827.4} + 13.8$$

$$\text{or } p = 9.9 + 12.12 + 13.8 = 35.82.$$

The results are of course the same, since the two operations are simply two methods of computing the same quantity. If the I.H.P., revolutions, length of stroke and L.P. piston area are given, then the first method would be used. If the mean effectives in the various cylinders are given, together with the revolutions, length of stroke, and piston areas, then the second method may be used without necessarily finding the I. H. P. at all.

Problems:

(1) Given I. H. P. = 5,120.
L. P. area = 5,612.
Stroke = 48″.
Revolutions = 112.

Find the reduced mean effective pressure. Ans. 33.6.

(2) From a pair of H. P. indicator cards the mean effective pressure is found to be 72.6 lb. The diameters of the H. P. and L. P. are respectively 18 and 48 inches. Find the high pressure mean effective reduced to the L. P. piston. Ans. 10.2.

(3) In the same engine as in (2) the mean effective pressure for the I. P. cylinder is found to be 33.2 lb., and the L. P. diam. is 29 inches. Find the I. P. mean effective reduced to the L. P. piston. Ans. 12.1.

(4) In the same engine as in (2) the L. P. mean effective pressure is 14.1 lb. Find the entire reduced mean effective pressure. Ans. 36.4.

Sec. 72. PRESSURE ON MAIN GUIDES.

The load on the crosshead guides comes from the load on the connecting rod and the obliquity of its line of action. The mechanics of this problem is considered in Part II., Sec. 12 [14] (15) and the maximum value of the load, which is found when the crank is at right angles to the center line, is readily computed in the manner there shown. It thus appears that the maximum load on the guide will bear the same relation to the load on the piston that the length of crank does to the connecting rod. This method of computing the load will be illustrated by an example.

(1) At about mid stroke given the pressure on the top of the piston 180 lb. per square inch and on the bottom 88 lb. per square inch. The ratio of connecting rod to crank is 4.5 to 1. The area of the piston is 404 square inches. Required the maximum load on the guide.

Net pressure on the piston = 180 — 88 = 92 lb. per square inch.

Net load on the piston = 92 × 404 = 37,168 lbs.

Maximum load on guide = 37,168 ÷ 4.5 = 8,260 lbs.

The safe load on guides is usually taken at from 50 to 70 lbs. per square inch. In this case therefore taking 60 lbs. as a safe load per square inch we should have:

Area needed = 8,260 ÷ 60 = 138 square inches.

Sec. 73. FORCE REQUIRED TO MOVE A SLIDE VALVE.

The net load on a slide valve is the difference between the steam loads on the two sides. On the back we have a load due to the full steam pressure in the steam chest. On the inside we have a more variable load due partly to the pressure in the steam chest or cylinder, and partly to the exhaust pressure. For the low pressure cylinder exhausting into the condenser the exhaust pressure is small and is usually neglected. The area of the face subjected to pressure from the cylinder is also relatively small, and for the purposes we have now in view is usually omitted. The load on such a valve is therefore taken simply as the load on the back, the pressure per square inch, multiplied by the area in square inches. Denote the pressure by p and the area by A. Then the total load will be pA. The resistance to the motion of the valve which must be overcome by the valve rod will be the load pA multiplied by the coefficient of the friction between the valve and its seat. Let f denote this coefficient, and F the force necessary to move the valve. Then we have:

$F = fpA$.

The values of f will depend on the condition of the surfaces and on the lubrication. With well fitted and lubricated surfaces its value should not exceed .01 to .02. With dry surfaces, especially if they should begin to abrade, its value may rise to .10 and more.

Example:

Given a low pressure slide valve with dimensions 50 inches by 60 inches : average excess of pressure in valve chest over condenser, 26 lbs. Coefficient of friction .02. Find load on valve stem.

Area $= A = 50 \times 60 = 3{,}000$ square inches.

Load $= pA = 26 \times 3{,}000 = 78{,}000$.

Load on valve stem $= fpA = .02 \times 78{,}000 = 1{,}560$ lb.

In designing a valve stem relative to such a load it must be given a large factor of safety in order to provide for starting the valve from rest or where partly stuck to the seat, and also for extra stresses due to the effects of inertia.

For a flat slide valve on a high or intermediate cylinder, an estimate must be made of the load on each side and the difference taken. Without serious error the net pressure may be taken as the difference between the average pressure in the valve chest and in the next following receiver. If then p_1 and p_2 are these pressures, $(p_1 - p_2)$ will be the difference, and $(p_1 - p_2) A$ the average load on the valve. The remainder of the operation is, of course, the same as explained above for the low pressure valve.

Sec. 74. AMOUNT OF CONDENSING WATER REQUIRED.

In Sec. 57 we have seen how to find the amount of heat required to form one pound of steam of given temperature and pressure from a pound of feed-water of given temperature. To condense the steam and reduce it back to the condition of the feed-water will require the subtraction of the same amount of heat. Hence we may find the heat to be taken from each pound of steam in the condenser in exactly the same manner as in Sec. 57. Now suppose the condensing water as it comes in to have a temperature of t_1, and as it is discharged, a temperature of t_2. Then the temperature of each pound will be raised $(t_2 - t_1)$ degrees. This means that it will absorb $(t_2 - t_1)$ units of heat. Then if we divide this into the number of heat units which must be taken from each pound of steam, it will give the number of pounds of condensing water which must be provided to condense one pound of steam. Then if we know the amount of steam to be condensed, the total amount of condensing water is readily found.

This may be illustrated by the following example:

Pressure of steam at exhaust = 3.5 pounds, absolute.

Corresponding temperature = 148°.

Temperature of condensed water = 130°.

Temperature of condensing water at entrance or $t_1 = 62°$.

Temperature of condensing water at discharge or $t_2 = 98°$.

Then from the steam tables we find that 1,029 heat units per pound must be subtracted in order to condense the steam and reduce it to the condition of the water in the condenser.

We have also $t_2 - t_1 = 98 - 62 = 36$ = number of heat units absorbed per pound of condensing water. Then $1,029 \div 36 = 28.6$ = number of pounds of condensing water per pound of steam.

Let us suppose an engine of 2,000 I. H. P. requiring 16 lbs. of steam per I. H. P. per hour to be condensed under these conditions. Then for the total weight of water W we have:

$W = 2,000 \times 16 \times 28.6 = 915,200$ pounds per hour.

And $915,200 \div 60$ = lbs. per mt. = 15,253.

Then $15,253 \div 64$ = cu. ft. sea water per mt. = 238.

In all ordinary cases the number of heat units to be subtracted will not differ much from 1,000, and for a rough estimate this number is often taken without detailed computation from the steam tables. Then, varying with the season of the year and the locality, we may expect that each pound of condensing water will absorb from say 25 to 50 heat units, and hence that the condensing water required per pound of steam will vary from 40 to 20.

Sec. 75. WORK DONE BY PUMPS.

It is sometimes desired to find the net work done by a pump in handling a certain amount of water. This may be computed closely if we know the conditions under which the pump operates. It is shown in mechanics that work may be divided into a volume factor and a pressure per unit area factor, and this form of the expression for work is usually most convenient for use in such cases.

Let us take first the case of a boiler feed pump feeding against a gauge pressure of 160 pounds and supplying 16 pounds water per I. H. P. per hour for 2,100 I. H. P. Then the amount of water supplied will be $2,100 \times 16 = 33,600$ pounds per hour. This equals $33,600 \div 60 = 560$ pounds per minute. Taking 62.5 pounds per cu. ft. this will occupy a volume of $560 \div 62.5 = 8.96$ cu. ft. This volume of water is pushed into the boiler against a total pressure of $160 + 14.7$ or say 175 pounds per square inch or $175 \times 144 = 25,200$ pounds per square foot. Hence we have: Work per minute $= 25,200 \times 8.96 = 225,792$ ft. lbs. Reducing this to horse power we have: H.P. $= 225,792 \div 33,000 = 6.84$. This is the net work, and assuming that there is no leakage or loss of steam. Actually there will be such a loss, raising the amount of water which the pump must deliver by from 5 to 10 per cent or more.

Now between the steam cylinder where the total work is developed and the net delivered work as above determined, there is a series of losses. These may be classified as follows:

(1) Loss due to the friction of the water in the pipes and to the inertia or resistance of the valves. These items form an extra resistance which must be overcome in addition to the regular pressure in the boiler.

(2) Loss due to the friction of the pump itself. This likewise forms an extra resistance as in (1).

(3) Loss due to the slip of the pump. The pump plunger and valves are rarely tight and a certain amount of "slippage" to the water is sure to occur. It results that the volume displaced by the pump plunger will be greater than the volume delivered to the feed pipe, and the work to be done in the water cylinder will be increased in about the same ratio. The slip is quite a variable feature, being quite small with good workmanship and careful attention, and large under contrary conditions. With the usual run of boiler feed pumps, however, it will rarely be less than 5 per cent., and with lack of care may readily rise to from 10 to 20 per cent.

The sum of losses (1) and (2) will be found usually between 15 and 20 per cent., and hence the sum of the total losses may be expected to vary between perhaps 20 or 25 and 40 per cent.

In the present case, for illustration, assume the loss by leakage of steam at joints, etc., as 6 per cent. Then the water actually delivered to the boiler will require a net work of $6.84 \div .94 = 7.28$. Assume the total losses between steam cylinder and feed pipes to be 33 per cent. Then the I. H. P. in the steam end will be $7.28 \div .67 = 10.87$, and we should therefore expect that under moderate to fair conditions such a pump would require from 10 to 11 I. H. P. With the pump plunger and valves leaking badly, stiff working parts and generally poor conditions, the amount will, of course, rise far above these figures. The full capacity of the feed pump would also be, of course, considerably above these values. We are here simply concerned with an estimate of the power actually required and the net power delivered under a given set of conditions.

Again consider the case of a centrifugal pump for the same engine handling we will say 30 pounds condensing water per pound of steam condensed. Then the total amount of water handled per hour will be $2,100 \times 16 \times 30 = 1,008,000$ pounds.

In this case the resistance to be overcome is due chiefly to forcing the water through the condenser tubes. In some cases also the discharge outlet is slightly above the surface of the water and this additional lift increases the work to be done.

The most convenient method of computation in this case is to estimate the total head equivalent to the resistance occasioned by the condenser tubes, and the lift of the water above the surface. This will be the total height of water which would produce the same pressure as must be overcome by the pump at the tips of the vanes. In usual cases we may assume the head corresponding to the resistance in the condenser tubes at from 4 to 5 feet. If the water is discharged at or below the water level this will then be the total head against which the pump works. In case, however, the pump draws from the bilge as when used for freeing the ship of water, then the total head will be the total lift plus the head due to the condenser tubes, and its value may rise in such cases to 20 feet and more.

In such cases the work done is the product of the weight of water handled as the force or resistance factor, multiplied by the head as the distance factor.

In the present case, assuming a total head of 6 feet we have:

Work per mt. $= 16{,}800 \times 6 = 100{,}800$ ft. lbs.

or H. P. $= 100{,}800 \div 33{,}000 = 3.05$.

The efficiency of such pumps is usually found between .30 and .50. That is, between 50 and 70 per cent of the power developed in the steam engine operating the pumps is lost, chiefly in the slip of the pump. Hence under fair conditions we may assume that this 3.05 H. P. will be about 40 per cent. of the I. H. P. of the engine. Hence the latter will be greater than 3.05 in the ratio of 100 to 40 or $2\frac{1}{2}$ times. Hence we have:

I. H. P. required $=$ about $2\frac{1}{2} \times 3.05 =$ about 7.5.

These examples will serve to show the methods to be used in working such problems, and if the principles involved are kept clearly in view they may be similarly applied to the solution of many problems likely to present themselves in engineering work.

Sec. 76. DISCHARGE OF STEAM THROUGH AN ORIFICE.

It may be sometimes convenient to be able to compute approximately the amount of steam which will escape into the atmosphere from a chamber under a given pressure through an aperture of given area.

Let p be the pressure, supposed to be not less than 25 lbs., absolute. Let A = area of aperture in square inches, and W = weight discharged in pounds per second. Then Napier's rule for the approximate value of W is as follows:

$$W = \frac{pA}{70}$$

Thus given $A = .5$ sq. in. and $p = 140$ we have:

$$W = \frac{140 \times 1}{70 \times 2} = 1 \text{ lb. per second.}$$

Take the following problem: What weight of steam would be discharged per hour through a small hole or crack of area .005 sq. in. under a pressure of 200 pounds per sq. in.?

Using the formula we have:

$$W = \frac{200 \times .005 \times 3600}{70} = 51.4 \text{ pounds per hour.}$$

We may thus realize the importance of small leaks.

Sec. 77. COMPUTING WEIGHTS OF PARTS OF MACHINERY.

The determination of the weights of various parts of marine engines and boilers is often necessary as a part of an estimate of costs for repairs or for other purposes. Such determinations are usually made by numerical computation, and consist in finding first the volume of the piece in question, and then its weight by the use of factors such as those given in the Table on p. 30. The chief part of the computation is therefore mensuration, the principles of which are given in Part II., Sec. 9. We will here add some general suggestions regarding the application of these rules, with some additional methods which may be used in special cases.

[1] Units to be Used.

The dimensions will usually be taken either from a drawing or directly from the piece in question. It may be recommended as a general rule to reduce all dimensions to inches as the unit rather than feet or feet and inches, the latter requiring the use of duo-decimal notation and methods as explained in Part II., Sec. 4 [5]. Where the pieces are small and fractions of an inch are to be dealt with, it will usually be most convenient to reduce them to decimal form rather than to express them as common fractions. In brief the inch as the unit and numbers expressed decimally are recommended as a general rule in such computations. The factors for reducing cubic inches to pounds are then used as given in the Table on p. 30, and the weight is readily found.

[2] Approximations and Short Cuts.

In computations of this character absolute accuracy does not exist. In fact with no physical measurement can absolute accuracy be attained. In practical life we wish simply an approximation, a value sufficiently near for the purposes in view; a value so near that the error is not of commercial or financial importance. All engineering measurements and computations recognize this principle, and of the acquirements which may come to the engineer with experience, none is of greater value than that which enables him to know where to stop his computations, how far to carry his measurements and approximations, what error will be of importance, and what insignificant. Thus if we are to measure the dimensions of a coal bunker in order to compute its volume, it is evidently absurd to note the figures to the fraction of an inch. We can be by no means sure that the length, for example, is uniform within any such limit, and the difference due to a variation of $\frac{1}{4}$ or $\frac{1}{2}$ inch either way will be insignificant for the purpose in view. On the other hand, if we are to measure a journal in order to make a new one which shall fit in the same bearing, the admissible error is only a few thousandths of an inch, and the utmost attainable accuracy will be in order. So likewise if we are finding the weight of a sheet of boiler plate. an error of $\frac{1}{8}$ inch in the length or breadth will introduce no significant error in the final result, while such an error in the thickness would cause a most serious error in the result. The former would make a difference of perhaps one part in 1,000 or so, while the latter might cause an error of one part in 8 or 10.

Another point which may also be remembered is that a large relative or percentage error is more permissible in some small part of the whole than in a large part. Thus in a boiler an error of 10 per cent in the weight of the tubes may be of less importance than one of 1 per cent in the weight of the shell and heads, while an error of 50 per cent in the high pressure cylinder cover, for example, might make less difference than one of 10 per cent in the low pressure cover. Of course there should be no excuse for making any 50 or 10 per cent errors, but the principle may be borne in mind as a legitimate means of saving time when a roughly approximate value must be determined.

The most common approximations are those which make the computation of volume simpler by substituting for the actual body some other of simpler form, and with such dimensions as

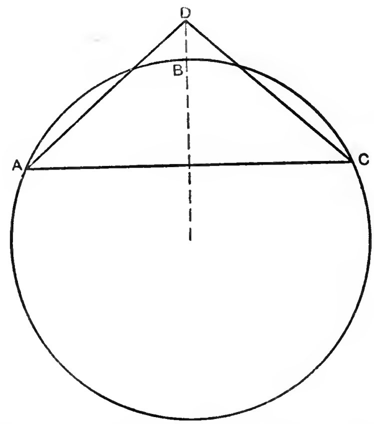


Fig. 284. Approximate Area of Segment of Circle.

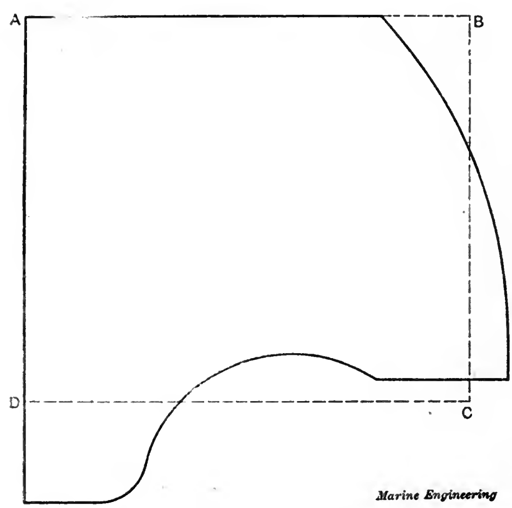


Fig. 285. Approximate Area of Boiler Plate.

to be of equal volume so far as judgment may be able to determine. Such substitutions are often employed, but they must be used with judgment and care in order that the possible error introduced may not be larger than permissible. No general rules can be given for such approximations, but the most common consist in substituting a rectangle or triangle or sometimes a circle, for a more irregular area; or a cylinder or regular prism or plate for some irregular volume.

Thus in Fig. 284, if we wish to find quickly an approximate value of the segment of the circle ABC, we may sketch in a triangle ADC, so taking the sides that the area left out shall be judged equal to that taken in, and hence the area of the triangle may be taken as an approximation to that of the segment. This area is then readily found by the usual rule for a triangle.

Again, in computing the area of a front boiler tube sheet, as shown in Fig. 285, we may for a first approximation substitute by judgment for the actual contour a rectangle ABCD, and thus quickly obtain a value which may be sufficiently close for the purpose in hand.

Again we may often add by judgment something to one of the dimensions of a piece in order to provide for additional or irregular parts which would not be included in the regular geometrical figures dealt with. Thus in finding quickly the approximate weight of a cylinder casting, provision may be made for the flanges by adding by judgment an appropriate amount to the length of the casting; or similarly for a piece of shafting with flanged couplings at the ends.

It is often necessary to divide a more or less complicated piece into several parts, each of which may be of some relatively simple form. In some cases the volume of one simple form may be subtracted from that of another, thus giving as the remainder the volume of a more or less irregular form. Thus, to find the volume of a pair of brasses with square backs and sides, we may find the volume from the outside dimensions as though the block were solid, and then the volume of the cylindrical hole, and take the one from the other.

Many such little devices will suggest themselves in connection with the details of the work, but it will be unnecessary to here enter further into the subject.

In connection with the rule of Pappus, Part II., Sec. 9 [30], we may note the following method of applying it to the determination of the weight of such forms as a piston, cylinder head, etc. The operations are as follows:

(1) The cross sectional drawing is supposed to be at hand.

(2) A copy of the half cross section, as shown for a piston

in Fig. 286, is prepared on thick, uniform paper, and then cut carefully out with a sharp-pointed penknife.

(3) This is weighed on delicate scales, and also balanced on the knife edge, the line AB containing the center of gravity being thus found.

(4) A square of the paper containing any convenient number, say 100 square inches of area, is also cut out and weighed. This divided by the area will give the weight of the paper per square inch.

(5) The weight of the paper half section is divided by that

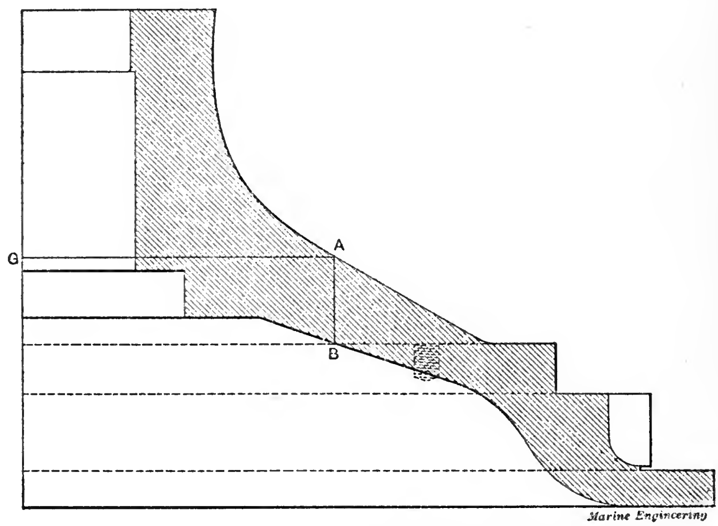


Fig. 286. Volume of Piston by Rule of Pappus.

of the square inch. The quotient will be the area of the paper half section in square inches.

(6) This area is multiplied by the square of the scale ratio of the drawing. Thus if the drawing is to a scale 1 inch = 1 foot, it is in the ratio 1 : 12 with the original, and we multiply by 12 × 12 or 144. If the scale is 1½ inches = 1 foot it is 1 : 8, and we multiply by 64. If to a scale of 3 inches = 1 foot it is 1 : 4 and we multiply by 16. The result thus found will be the area of the actual full-sized half section in square inches.

(7) We then multiply the distance AG scaled off according to the scale of the drawing and expressed in inches, by 6.2832, and the product by the area as found in (6). The result will be the volume in cubic inches.

Instead of the preceding, we may less accurately find the area by taking it in parts and using substituted simpler forms, as above explained. We may then by judgment assume the location of G and then proceed as above in (7).

Thus, for example, suppose we find as follows:

Scale of drawing $1\frac{1}{2}$ inches $= 1$ foot.

Weight of paper section 240 grains.

Weight of paper per square inch 36 grains.

Arm $AG = 1.7$ inches on the paper or 13.6, as scaled from the drawing.

Then area $= 240 \div 36 = 6.67$ inches.

This multiplied by 64 gives 426.7 square inches as the area of the actual half section.

Then volume $= 13.6 \times 6.2832 \times 426.7 = 36462$ cubic inches.

CHAPTER X.

Propulsion and Powering.

Sec. 78. MEASURE OF SPEED.

For measuring the speed of steamships the customary unit is the *knot*. While this term is often used as a distance, it is really a speed or velocity. As adopted by the United States Navy Department, it is a speed of 6080.27 ft. per hr. The British Admiralty knot is a speed of 6080 ft. per hr. For all ordinary purposes the United States and British knots may be considered the same. It is often necessary to reduce knots to feet per minute or *vice versa*. To this end we divide 6080 by 60 and find 101.33 ft. per minute as the equivalent of one knot. Hence the following rules:

To reduce knots to feet per minute multiply by 101.33.

To reduce feet per minute to knots, divide by 101.33.

In the inland waters of the United States, and to some extent on the coast for tugs, yachts, launches, etc., the *mile per hour* is used as the unit, instead of the knot. A statute mile consists of 5,280 feet. Hence one mile per hour equals 5,280 ÷ 60, or 88 feet per minute. Hence:

To reduce miles per hour to feet per minute, multiply by 88.

To reduce feet per minute to miles per hour, divide by 88.

Also:

To reduce miles per hour to knots, divide the former by 1.1515.

To reduce knots to miles per hour, multiply the former by 1.1515.

Sec. 79. PROPULSION.

To propel a ship through the water some kind of a propulsive thrust must be obtained. This is the fundamental problem of propulsion. Thus when a boat is poled along a shallow creek the thrust is obtained as a reaction from the bed of the creek against the end of the pole and thence to the man who is pushing it, and thence to the boat.

In the usual case, however, there is no bottom to be reached, and the only thing outside the ship which can be gotten hold of for the purpose of gaining a reaction is the air or the water. For all cases with which the engineer is concerned it comes to the latter, and so the problem is to get from the water a reaction or force directed forward by means of which the ship may be pushed through the water. To understand how this is possible we must remember the property of *inertia*, one of the fundamental properties of matter. It is this property which enables all matter to resist any effort made to change its condition of rest or relative motion, and to react back on the means by which such a change is effected. Thus a push of the hand may serve to set in motion a weight hanging by a rope, but while the condition of rest is being overcome the weight will react back on the hand with a force equal and opposite to that which the hand exerts upon the weight. Similarly, when a shot is fired from a gun, the inertia of the shot causes it to react back against the gas and so to the gun, causing the well known recoil.

From the fundamental principles of mechanics it follows that to obtain a thrust or reaction forward it is only necessary to produce in a certain mass of water an increase in velocity, such increase being directed from forward aft, or at least having a component in that direction. There will then be a reaction directed from aft forward, or having a component in that direction, and such reaction exerted on the means used to produce the change of velocity may be utilized as a propulsive thrust.

This is carried out in practice by either a screw propeller or paddle wheel, and remembering the principles above stated, it appears that the immediate purpose of the propeller or paddle wheel is simply to produce an increase in the velocity of the water directed from forward aft. In consequence of this the water will exert a forward reaction on the propeller or paddle wheel, and thus produce the thrust required to propel the ship through the water. It may be added that this increase of velocity from forward aft, referred to above, may be obtained either by taking hold of water at rest and giving it a motion sternward, by taking hold of water already moving sternward and giving it a still higher velocity in the same direction, or by taking hold of

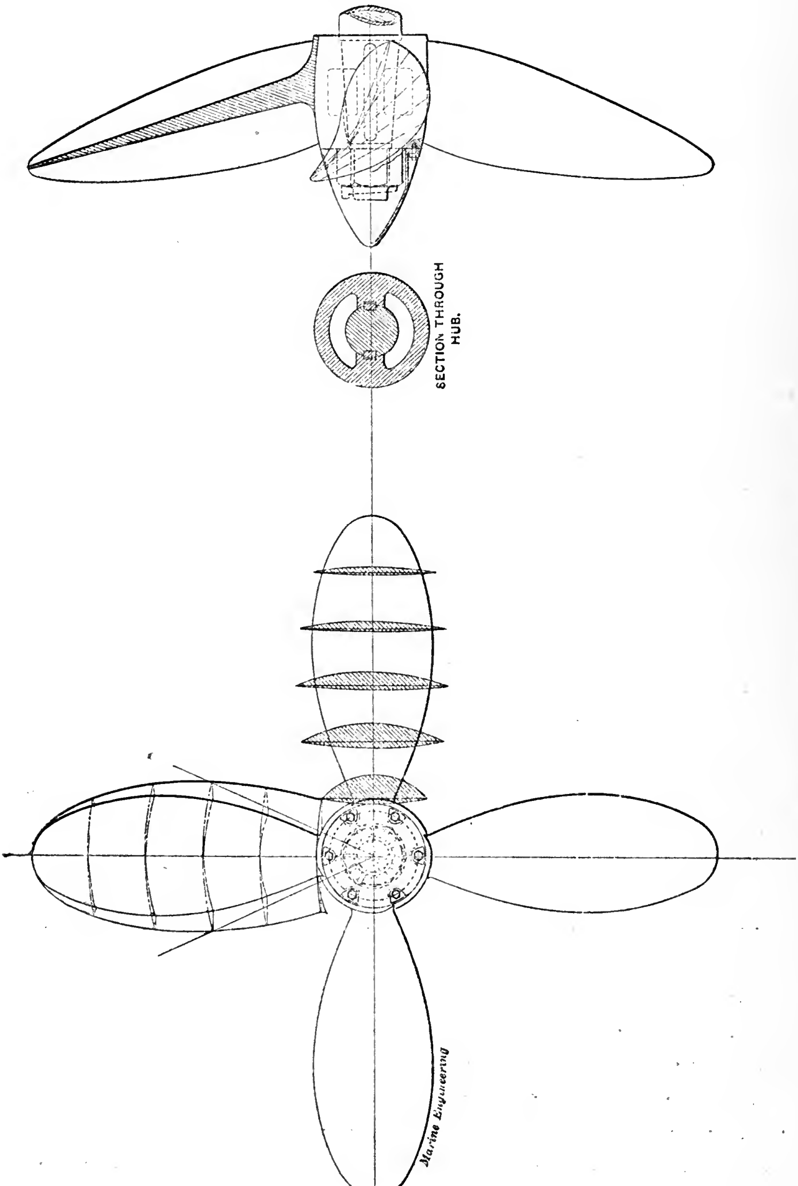


Fig. 287. Screw Propeller, Blades Cast with Hub.

water moving forward and decreasing such forward motion, stopping it and leaving it at rest, or reversing it to a sternward motion. In all cases it is the *change* of velocity which is of importance. Thus a change from rest to 5 feet per second aft, or from 3 feet per second aft to 8 feet per second aft, or from 5 feet per second forward to rest, or from 2 feet per second forward to 3 feet per second aft, will each give the same forward thrust.

We shall not go further into the theory of propulsion, but in the next section will give certain definitions relating to screw propellers and the solution of a few simple problems.

Sec. 80. SCREW PROPELLER.

[1] Definitions.

A screw propeller as shown in the frontispiece and in Figs. 287, 288, consists of a hub and a certain number of blades, usually two, three or four. The blades have on their rear or driving side an approximately helical surface—that is, a surface similar to that which forms the faces of an ordinary screw thread. In this view a two-bladed screw propeller may be considered as a small part of a double-threaded bolt, the threads being cut very deep, and all portions being cut away down to the hub except the parts retained for blades. Similarly a three or four-bladed propeller may be considered as a small part of a triple or quadruple-threaded bolt similarly cut away except for the parts retained for blades.

The *hub* or *boss* is the central portion to which the blades are attached, and through which they receive their motion of rotation in a transverse plane relative to the ship.

A propeller is said to be *right hand* or *left hand*, according as it turns with or against the hands of a watch when looked at from aft and driving the ship ahead.

The *face* or *driving face* of a blade is to the rear. It is that face which acts on the water and so receives the forward thrust.

The *back* of a blade is therefore on the forward side. Care must be taken not to confuse these terms.

The *leading* and *following edges* of a blade are respectively the forward and after edges.

The *diameter* of a propeller is the diameter of the circle swept by the tips of the blades.

The *pitch* of a propeller is the same as the pitch of the screw thread, of which it may be considered as forming a small part—

that is, it is the longitudinal distance between the successive turns of the helical surface. This definition will hold, however, only when the pitch is the same over the entire face of the blade. In many cases the pitch varies from one point to another, and we must therefore understand the term as relating, in such cases, to a small element of the driving face only. From this view the pitch may be defined as the longitudinal distance which the ship would be driven for one revolution were this element to work

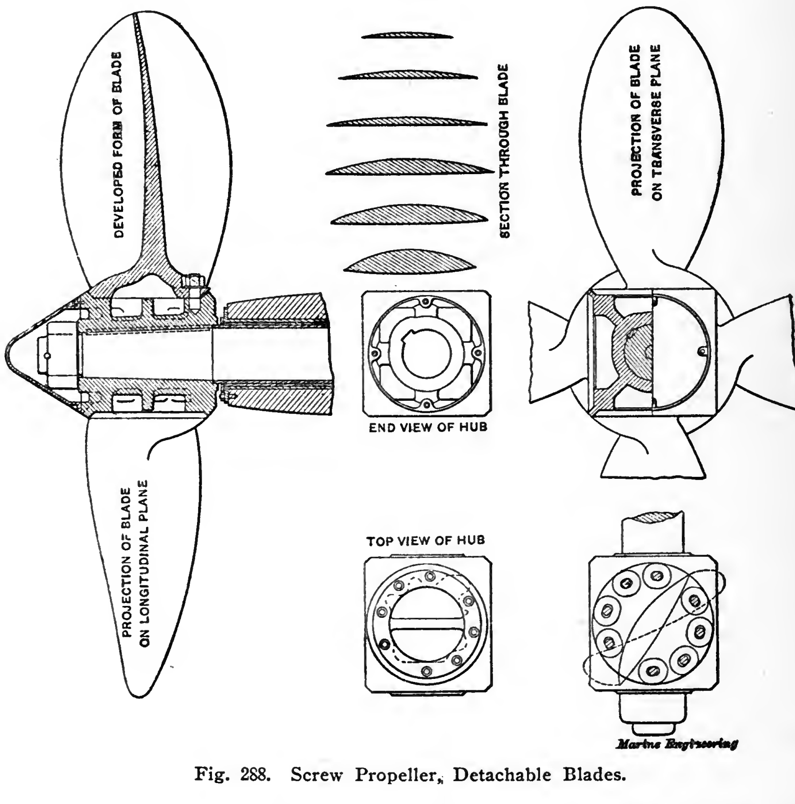


Fig. 288. Screw Propeller, Detachable Blades.

on a smooth, unyielding surface, as, for example, the corresponding surface of a fixed nut. The pitch thus defined will depend on the location and inclination of the surface at the point or element considered, and its value may thus vary from one point to another over the entire face of the blade. The pitch is thus said to be uniform or variable, as its value remains the same or changes from point to point over the driving face. If it increases as we go from the hub to the tip of the blade it is said to increase radially. If it is greater on the following than on the leading edge, it is said to increase axially. The latter is usually implied by the simple term *increasing* or *expanding* pitch.

The *pitch ratio* is the pitch divided by the diameter, or the ratio of pitch to diameter.

The *area, developed area* or *helicoïdal area* of a blade is the actual surface of the driving face. For the propeller as a whole it is the sum of the areas of all the blades.

The *projected area* is likewise the area of the projection on a transverse plane, of one blade or of all the blades collectively.

The *disk area* is the area of the circle swept by the tips of the blades.

The pitch has been defined as the distance which the propeller in one revolution would drive the ship if it worked on a smooth, unyielding surface. Instead of working on such a surface, however, the propeller works on the water, a yielding medium, and in consequence the water recedes somewhat under the action of the propeller and the ship moves forward per revolution a distance less than the pitch. The difference between the pitch and the distance the ship actually moves per revolution is called the *slip*, or, more concisely, the slip per revolution. The ratio of this slip to the pitch is called the slip *ratio*, or simply the *slip* stated in per cent as a slip of 20 per cent, 30 per cent, etc.

Let p denote the pitch of the propeller in feet.
N " " revolutions per minute.
u " " velocity of the ship in knots.
s " " slip ratio.

Then $101.3\,u \div N$ is the distance traveled by the ship per revolution, and $p - (101.3\,u \div N)$ is the slip per revolution. Hence for the slip ratio we have:

$$s = \frac{p - \dfrac{101.3\,u}{N}}{p}$$

Multiplying both terms of the fraction by N we have:

$$s = \frac{pN - 101.3\,u}{pN} \qquad (1)$$

The term pN is the distance the ship would go per minute if there were no slip, while $101.3\,u$ is the distance which is actually made good. The difference or $pN - 101.3\,u$ may therefore be called the slip per minute, and the quotient of this by pN is the slip ratio. This latter equation for s is the one by means of which its value is usually computed.

It may be well to note at this point that while slip implies a certain loss of effectiveness in the propeller, it is a loss which is necessary in the very nature of the case. We have already seen that to obtain a propulsive thrust we must give to a certain body of water an increased velocity sternward. This means that the water must yield under the action of the propeller, and it is this yielding or falling sternward which thus gives rise at the same time to both the slip and the propulsive thrust. We cannot therefore have the thrust without the slip: we must accept the latter to obtain the former.

We must now introduce a further consideration. We have defined slip as the difference between the pitch on the driving face and the advance per revolution. The latter admits of being defined in two ways according as we take for our point of reference a point in the outlying still water, or a point in the water about the stern of the ship and in which the propeller works. So far as we are concerned with the movement of the ship through the water as a whole, the former is the natural point of reference. For various considerations connected with the operation of the propeller itself, however, the latter is the more important. Let us note briefly the condition of the water close about the stern of the ship and in which the propeller works.

The ship in moving through the water will throw into forward motion a skin of water extending from the surface of the ship for several inches outward. Very near the surface of the ship this will move with nearly the velocity of the ship, while as the distance from the surface is increased the velocity will rapidly decrease and soon become insensible. The water thus given forward motion by the skin of the ship will finally be found at the stern, where, still further influenced by wave and stream line motion, it forms the so-called "wake." The forward velocity in the wake at different points in a transverse plane at the stern is quite irregular, rising as high as 50 to 75 per cent of that of the ship at points near the surface and near the stern post, and decreasing irregularly and gradually to nothing at the outlying still water. For single screw ships the average value in that part of the wake directly influenced by the screw is usually from 10 to 20 per cent of the speed of the ship. For twin screws located somewhat aside from the strongest part of the wake the values are usually found between 6 and 12 per cent of the speed of the ship.

Now with reference to the propeller, it is evident that so far as it is concerned individually and as an appliance for developing thrust, it should be judged relative to the water immediately about it and in which it works rather than relative to an outlying body of undisturbed water upon which it has no direct influence. The slip which is given by taking the speed relative to the wake is therefore called the *true slip*, while that given by taking the speed relative to the outlying still water (the speed as usually considered) is called the apparent *slip*.

To show the relation between the true and apparent slips let v denote the forward velocity of the wake and u that of the ship as before, both measured in knots and relative to the outlying still water. Then $(u - v)$ is the speed of the advance of the propeller through or relative to the wake. Also using the same notation as above,
$pN - 101.3\,(u - v)$ is the true slip per minute, while, as before, $pN - 101.3\,u$ is the apparent slip per minute. Denoting the latter by S_1, and the former S_2, we have:

$$S_1 = pN - 101.3\,u.$$
$$S_2 = pN - 101.3\,(u - v) = S_1 + 101.3\,v.$$

It thus appears that the difference between the two slips in feet per minute is simply the wake velocity, as we should expect.

To reduce to slip ratio we use pN as the divisor in each case, and denoting the resulting ratios by s_1 and s_2 we have:

$$s_1 = \frac{pN - 101.3\,u}{pN} \text{ as before, and:}$$

$$s_2 = \frac{pN - 101.3\,(u - v)}{pN} = s_1 + \frac{101.3\,v}{pN}$$

Where the term *slip* is used without special definition, the *apparent* slip is usually intended.

In the preceding discussion of slip we have used the term pitch as though it were of constant value over the entire surface of the blade. If such is not the case, then the term must be understood as referring to a mean or average value. Such an average value of the pitch of a propeller might be defined in a variety of ways, but engineers are not as yet agreed upon the method most suitable. This point, however, is one which cannot be further developed in the present work.

Problems.

(1) *To find the apparent slip.*

From the preceding value of the apparent slip ratio s we may derive the following:

Rule:—(1) Multiply the pitch by the revolutions per minute.

(2) Multiply the speed in knots by 101.3.

(3) Subtract the result in (2) from that in (1) and divide the difference by the result in (1). The quotient will be the slip ratio.

Example: Given speed of ship, 11 knots; pitch, 20 feet; revolutions, 72. Find the apparent slip.

Operation: $20 \times 72 = 1440$.

$11 \times 101.3 = 1114.3$.

$1440 - 1114.3 \div 1440 = 325.7 \div 1440 = 22.6$ per cent, ans.

(2) To find the speed, having given the other items.

From the preceding equation we derive the following value for u:

$$u = \frac{pN\,(1 - s)}{101.3} \qquad (2)$$

Whence the following:

Rule:—(1) Multiply the pitch by the revolutions.

(2) Subtract the slip per cent from 1.00.

(3) Multiply the result in (1) by that in (2).

(4) Divide the result in (3) by 101.3. The quotient will be the speed in knots.

Example: Given pitch, 18 feet; revolutions, 110; apparent slip, 18 per cent. Find the speed.

Operation: $18 \times 110 = 1980$.

$1.00 - .18 = .82$.

$.82 \times 1980 = 1624$.

$1624 \div 101.3 = 16.03$ knots, ans.

(3) To find the revolutions, having given the other items.

From the preceding equation we derive the following value for N:

$$N = \frac{101.3\,u}{p\,(1 - s)} \qquad (3)$$

The use of this formula will be illustrated by the following example:

Given speed, 22 knots; pitch, 26 feet; apparent slip, 16 per cent. Find revolutions.

Operation: $101.3 \times 22 = 2228.6$

$(1.00 - .16) \times 28 = 23.52$

$2228.6 \div 23.52 = 94.8$ revolutions per minute, ans.

(4) To find the pitch, having given the other items.

From the preceding equation we derive the following value for p.

$$p = \frac{101.3\, u}{N(1 - s)} \qquad (4)$$

The use of this formula will be illustrated by the following example:

Given speed, 28 knots; apparent slip, 21 per cent; revolutions, 380. Find the corresponding pitch.

Operation: $101.3 \times 28 = 2836.4$

$380 \times (1.00 - .21) = 300.2$

$2836.4 \div 300.2 = 9.45$ feet, ans.

[2] Varieties of Propellers.

Screw propellers are found in the greatest variety according to the number, shape, style and arrangement of blades. In modern practice the number of blades is usually either three or four, the former being perhaps more commonly met with in twin screws and the latter in single screws.

The shape of the blades may be oval or elliptical, as in Figs. 287, 288, or broadening somewhat toward the tip with rounded corners, or of any intermediate or similar form which may be desired. The oval or generally rounded form of blade is most commonly met with in modern practice. The blades may also be bent or curved in various ways. Thus in propellers for small boats the blades are often bent back, as in Fig. 287, so as to throw them somewhat farther from the stern post. They are also sometimes curved in the plane of rotation so that the entering edge is well rounded as it enters the water in going ahead. Combined with these there may be various modifications of pitch, as above referred to. The normal or standard propeller in modern practice may, however, be considered as one having plain blades of uniform pitch, of oval or elliptical form, and standing at right angles to the axis, as in Fig. 288. Most of the variations from this type are based on fancy rather than on definite engineering reasons. So far as is known at present, the simple normal type, as specified above, is the equal of any of those of variable pitch or of special form or shape of blade, and while it may be that some special combination of pitch, shape and form of blade may give a higher efficiency than can be obtained from the normal type, yet up to the present time such results have not been proven.

Small propellers are usually made complete or in one casting, as in Fig. 287. Large propellers are made either in one casting or with separate or sectional blades, as in Fig. 288. In the latter case the root of the blade carries a circular flange fitting into a corresponding recess in the hub. This serves to secure the blade to the hub by stud-bolts passing through the flange and fitted with nuts countersunk below its outer face. The general details of this arrangement are shown in the figure. The holes in the flanges are usually made slightly oblong, thus providing for a slight change in the pitch by turning the flange back and forth, and thus changing the average obliquity of the blade to the axis of the propeller. Once adjusted as desired the holes are filled by packing pieces so that no further change can result from the accidental slipping of the flange under the nuts. In this connection it may be noted that the change of pitch resulting from such a twisting of the blade is not the same for all parts of the blade, but varies from root to tip. It follows, if the blade is made of uniform pitch, that it will remain so only so long as it is set at the corresponding angle, and that if it is twisted to and fro the pitch will be increased and decreased, but not uniformly; so that in all positions but this one the pitch will no longer be uniform, but variable. For a moderate angle of twist, however, the change from uniformity is but slight, and changes of average pitch up to perhaps 10 to 15 per cent may be made without serious departure from the average.

The chief advantages of the separate blades lie in the possibility of varying or adjusting the pitch as just described, and in the readiness with which repairs may be executed. A separate blade broken or defective may be readily removed and replaced with a new one, this operation in small vessels being sometimes accomplished without placing the vessel in dry dock. One or two blades may also be carried as spare parts or shipped by rail, or otherwise, much more readily than an entire propeller.

The attachment of the propeller to the shaft is shown by the figures. The taper is usually about 1 inch in the diameter per foot. The propeller is prevented from turning on the shaft by one or more keys fitted, as shown. The after end of the shaft is fitted with a nut which serves to hold the propeller against any tendency to slip off when backing, and this is often covered with a conical tail piece, as shown, in order to reduce the eddy formation just aft of the boss. It may be noted that for a right-hand propeller the nut is usually left hand, and *vice versa,* it being considered that this arrangement reduces the liability of loosening or backing off.

If water is allowed to come into contact with the taper of the shaft on which the propeller boss is secured it may give rise to considerable corrosion, and this may set the boss so firmly on the shaft that great difficulty will be experienced in its removal. In order to prevent the contact of water with the taper, various means may be employed. In one method the brass liner on the shaft is carried along nearly to the forward end of the taper, and a rubber ring is placed just forward of the boss or in a counter-bore. As the boss is forced on, the rubber ring is compressed between the boss and the liner, and thus a water-tight joint is made. See Fig. 288. In other cases the liner is carried into a counterbore in the boss and a red lead joint is made between the two.

For the examination or repair of the stern bearing it may become necessary to remove the propeller and withdraw the tail shaft forward into the ship. This is of course an operation requiring the docking of the ship. For the removal of the propeller the first attempt may be made with steel wedges between the forward face of the boss and the stern post, or after end of the stern bearing. The space between the two is made up with the metal blocking necessary, and the wedges are then inserted and driven one from either side. In this way a tremendous strain can be exerted and the boss will be started unless seriously corroded or jammed unduly tight. In the latter cases recourse must be had to hydraulic jacks, or to a heavy ram or to heating the boss in order to expand it in size and break the connection with the shaft. Once the boss is started the weight of the propeller may be taken by chain hoists suspended from the counter, and the shaft may then be drawn forward into the tunnel.

In the case of twin screw ships with the common form of strut, the same general means may be employed, except that it should be remembered that the strain set up is carried on the strut, and the means taken should not go so far as to endanger its rupture or the undue straining of its fastenings.

[3] Materials.

Cast iron, cast steel, brass, gun metal and the various bronzes are the materials used for screw propellers. Cast iron is the cheapest, but is relatively weak and brittle, and the blades must necessarily be thicker and less efficient than if made of steel or bronze. Cast steel is stronger than cast iron, and the sections may be accordingly decreased with a resultant gain in efficiency. The surface of cast steel is naturally not as smooth as with cast iron, but with improved methods of production the difference is not important. Brass and the various bronzes have naturally a smoother surface, and seem furthermore to have a lower coefficient of skin resistance. This, added to their strength and good casting qualities, makes possible a smooth and relatively thin blade with sharp edges, all of which are features favorable to good efficiency. With the best bronzes the ultimate strength may vary from 50,000 to 60,000 pounds per square inch of section. With cast steel the ultimate strength will reach still higher, or, say, to 65,000 pounds per square inch. With gun metal an ultimate strength of 25,000 to 35,000 pounds may be expected, while with common brass and cast iron not more than 20,000 to 25,000 pounds can be depended on.

Of the various materials available, manganese bronze may perhaps be considered as possessing the best combination of desirable qualities, such as strength and stiffness, good casting qualities, resistance to corrosion, etc. Care is needed in the manipulation of the various bronzes in melting, pouring and cooling, in order to insure uniformity and the full realization of the valuable properties of the alloy. The greater cost of such bronzes restricts their use, however, to warships, yachts and launches, ocean liners and other cases where the importance of a saving in propulsive efficiency may be considered worth the increased cost of the propeller.

The durability of propeller blades is usually in the order: bronze, cast iron, cast steel. The two latter usually suffer by general corrosion and local pitting, the average life being usually from five to ten years. The life of bronze blades is practically indefinite.

[4] Measurement of Pitch.

To determine the pitch of a given propeller three measurements are necessary. See Fig. 289. These are:

(1) The radius OA at which the pitch is desired.

(2) The angle or part of the complete circumference corresponding to the distance on the blade between A and B, the two points between which the pitch is to be found.

(3) The advance BC parallel to the line of the shaft, corresponding to this part of a complete revolution.

In the figure, A and B are points on the face of the blade, and are at a constant distance OA from the shaft center line OO. AC is an arc of a circle which lies in a plane through A and perpendicular to the shaft. The angle AOC is therefore the one referred to in (2), and the distance BC is the corresponding advance. Then BC is the same fraction of the entire pitch that AOC is of a complete circle, or the same fraction that the length

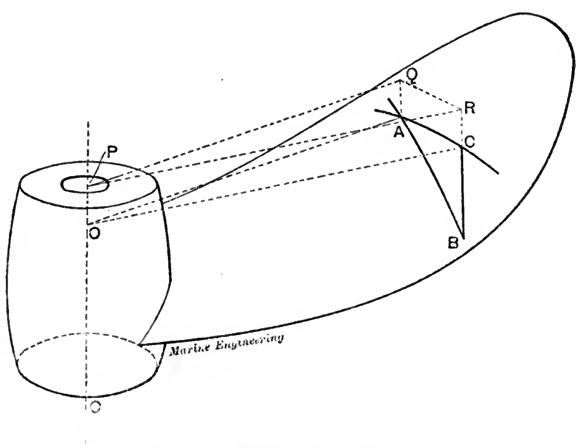


Fig. 289. Measurement of Pitch.

AC is of a complete circumference with OA as radius. This complete circumference will be $6.2832 \times OA$. Hence the proportion:

$$AC : 6.2832 \times OA :: BC : \text{pitch}.$$

$$\text{or pitch} = \frac{6.2832 \times OA \times BC}{AC}$$

It is not, however, as easy to measure AC as AB, so that we may put for AC its equal $\sqrt{\overline{AB}^2 - \overline{BC}^2}$, and we then have

$$\text{pitch} = \frac{6.2832 \times OA \times BC}{\sqrt{\overline{AB}^2 - \overline{BC}^2}}$$

A brief outline of the operations is as follows:

(1) Select the points A and B at and between which the pitch is desired, making sure that they are at equal distances from the shaft center line. This can be done by squaring down from a straight edge or other reference line PQ, PR, placed across the hub and at right angles with the shaft. Then measure the length AB.

(2) The propeller being leveled up, measure the distance BC from a level through A vertically down to B. Or if the propeller cannot be leveled, measure from B in a direction parallel to the shaft out to a line through A in a plane at right angles to the shaft. Or measure from Q down to A and from R down to B and take their difference BC.

(3) Multiply the distance OA or its equal PQ by 6.2832, and by the length BC all in the same units of measure.

(4) Square the lengths AB and BC, subtract the square of the latter from that of the former and extract the square root of the difference.

(5) Divide the result found in (3) by that found in (4) and the quotient will be the pitch desired.

Thus, for example, suppose

AB $=$ 20 inches, BC $=$ 13 inches and OA $=$ 48 inches.

Then $6.2832 \times 48 \times 13 = 3920.7$

Also $\sqrt{400 - 169} = \sqrt{231} = 15.2$.

Then $3920.7 \div 15.2 = 238$ inches $=$ 21 ft. 6 in. $=$ pitch.

If the pitch is variable instead of uniform, the operation is precisely the same, but the result found must be considered merely as the mean or average value of the pitch between the points A and B. For other parts of the blade a similar process will give the pitch at those points.

When the propeller is in place on the ship it is sometimes more convenient to carry out the principles involved in this method of measuring pitch somewhat differently, as follows: Let the propeller be turned so as to bring one of the blades horizontal. Then select the place at which the pitch is desired, and hang over the blade at this point a cord with two weights, as shown in Fig. 290. Care must be taken that the two points A and B at which the cord touches the edges of the blade are at the same distance from the center. It is then readily seen that the points A and B of Fig. 290 correspond to the similar points of Fig. 289, except that in Fig. 290 they are of necessity taken on the extreme edges of the blade. We then level up a bar PQ and measure the distances, AB and BC, as noted above, using them in the same way for finding the pitch. Or we may measure AC directly and use this with BC in the proportion above.

As a rough and ready rule it may be remembered that the pitch of a propeller will equal the length of a circumference at the place on the blade where the slope of the face is 45°, or where it is equally inclined to the shaft and to the transverse direction. Starting near the shaft, the inclination to the longitudinal is small, but increases toward the tip, passing at some point through the value 45°. At this point let the radius be r. Then

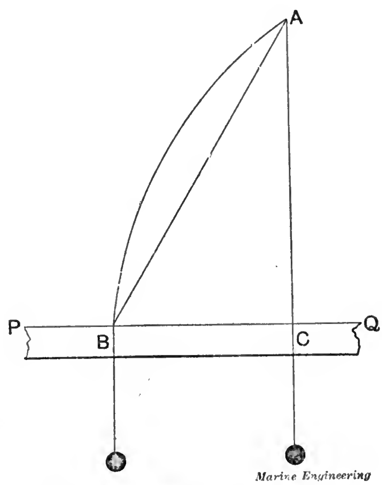


Fig. 290. Measurement of Pitch.

pitch $= 2\,\pi r = 6.2832\,r$. In this way an approximate idea may often be quickly obtained of the pitch of a wheel by estimate without special measurement, except for the radius or diameter at which the blade has the slope of 45°.

The details of the above methods for finding pitch may vary considerably, but the description given will serve to show the principles involved, and with reasonable mechanical skill no trouble will be found in carrying out the measurements required.

Sec. 81. PADDLE WHEELS.

In addition to the screw propeller the *paddle wheel* is the other appliance used for ship propulsion. In Fig. 291 is shown in skeleton a common radial paddle wheel. In this type of wheel the paddles or floats are rigidly fixed to the arms, the latter being connected at their inner ends to a hub, which is carried on the shaft. In this manner the motion of the shaft is transmitted to the floats, and these, acting on the water, drive it sternward and thus receive the forward thrust which is required for the propulsion of the vessel.

In Fig. 292 is shown a feathering paddle wheel. In this arrangement the floats are hung on axes and are swung in such way that they enter and leave the water nearly in an edgewise direction. In this way there is less disturbance of the water and a smoother action of the wheel is obtained. Such arrangement is especially suitable for ships operating under widely varying conditions of draft, for the floats of a deeply immersed radial

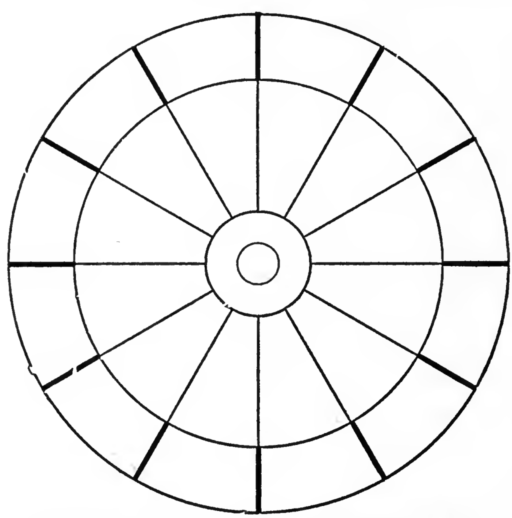


Fig. 291. Radial Paddle Wheel, Skeleton Diagram.

wheel enter and leave the water at a great obliquity and there would be considerable loss by oblique action.

There are two chief methods by which the proper motion may be given to feathering floats, depending on whether the paddle shaft has an outer or spring bearing on the outside of the paddle box or is overhung; that is, provided simply with a bearing on the rail, the paddle wheel itself being then mounted on the overhung end of the shaft. In the former case the arrangement will be understood from the skeleton drawing of Fig. 293. The stationary excentric A has its center forward of the wheel center, as shown. To the excentric strap is attached a drive link HB, connected by pin joint to an arm BC, carrying a float DE.

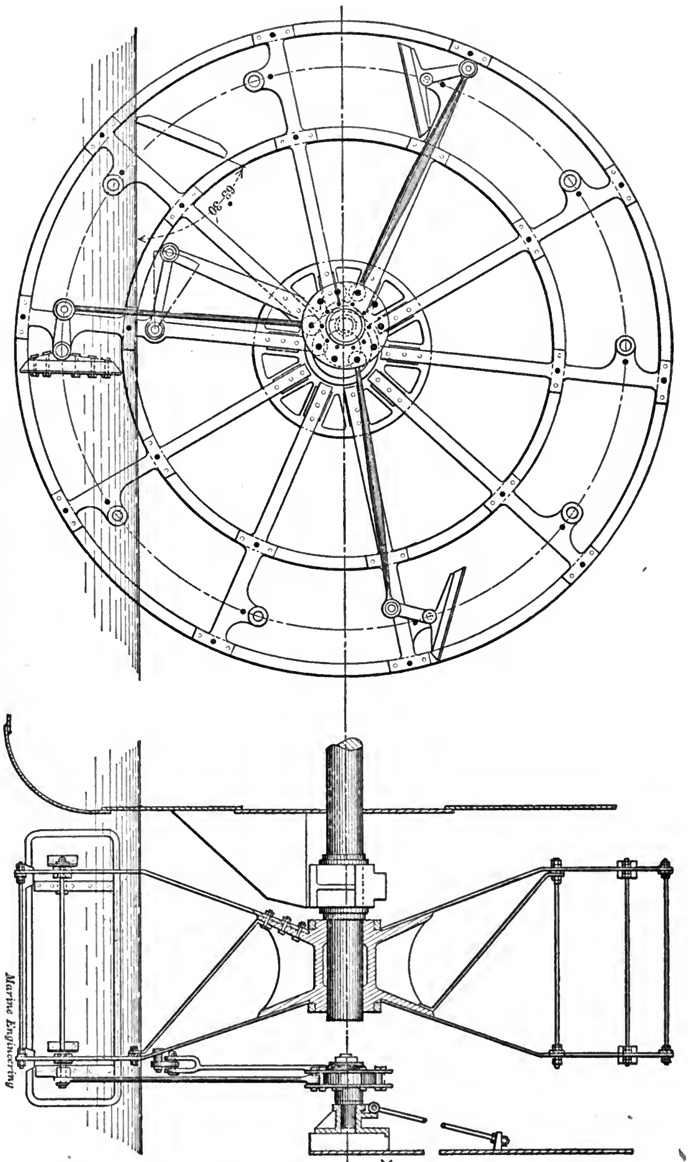


Fig. 292. Paddle Wheel With Feathering Floats.

The other floats, mounted in a similar manner, are connected by pin joint links to the excentric strap, as shown. As the wheel turns the drive link HB carries the strap around the excentric sheave, and with it the series of connected links. This gives a see-saw motion to the ends of the arms BC and thus swings the floats in the manner desired.

When the paddle shaft has no outer bearing, as in the arrangement shown in Fig. 292, the disc carrying the links may be mounted on a supporting pin carried on the outer side of the guard. It may then be given motion through a drive link and connections, as shown, giving a similar see-saw motion to the floats, as in the former case.

In modern practice the arms of paddle wheels are made of

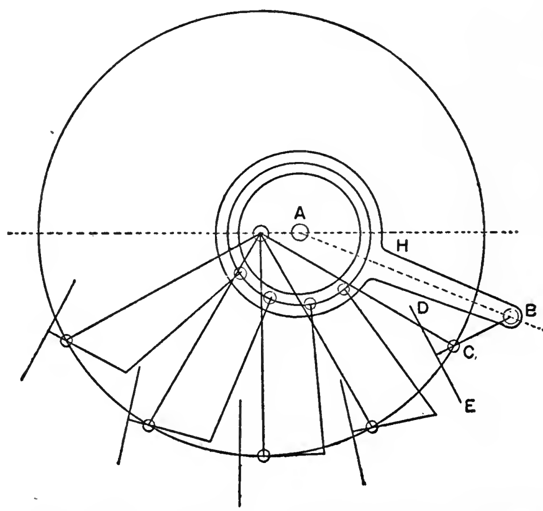


Fig. 293. Paddle Wheel, Skeleton of Arrangement for Feathering Floats.

steel, the hubs of cast iron or cast steel, and the floats of wood or boiler plate; in the latter case often curved in cross section.

In estimating the pitch of the paddle wheel or what corresponds to pitch in the screw propeller, we must consider it as the circumference of the circle traveled by the floats. Since, however, a float as a whole is made up of a series of strips or elements at varying distances from the center, each such element will have its own circumference and therefore its own pitch, and will try to drive the ship at a speed corresponding to such pitch. The paddle wheel as a whole has therefore a varying pitch, increasing from the outer to the inner edge of the float. The resultant mean pitch is considered as the circumference traveled by a point called the center of effort. The proper basis for the determination of this point, and hence of the true mean pitch of a paddle wheel is, however, not definitely known, and can only be determined by the aid of extended experimental investigation. In the absence of such definite basis it is sufficient for all practical purposes to take it at the center of the float radially, though its true location would lie somewhat outside this point. Counting the circumference through this point as the pitch, the actual distance traveled by the boat per revolution is less by the amount of the slip, which is usually found from 15 to 25 or 30 per cent.

The circle whose circumference is equal to the distance traveled per revolution is sometimes known as the rolling circle. It is so called from the fact that the speed of the boat is the same as though it were carried on wheels of this diameter, which rolled on a supporting surface as wagon wheels along a smooth, level road.

The solutions of problems relating to the revolutions, diameter and slip of paddle wheels are found in the same general manner as for the screw propeller, and the resulting equations are similar to those found in Sec. 80 [1], with the substitution for p of D, as defined below, and 88 for 101.3.

Let D = diameter of rolling or pitch circle.
N = revolutions per minute.
u = speed in miles per hour.
s = slip ratio.

Then, as with the screw propeller, we have:

$$s = \frac{DN - 88\,u}{DN} \qquad (1)$$

$$u = \frac{DN\,(1 - s)}{88} \qquad (2)$$

$$N = \frac{88\,u}{D\,(1 - s)} \qquad (3)$$

$$D = \frac{88\,u}{N\,(1 - s)} \qquad (4)$$

These may be illustrated by the following examples:

(1) Given diameter of rolling circle, 36 feet; revolutions, 45 per minute; speed, 14 miles per hour. Find the slip ratio. Operation:

$D\,N = 36 \times 45 = 1620$

$88\,u = 88 \times 14 = 1232$

$1620 - 1232 \div 1620 = 388 \div 1620 = 23.3$ per cent.

(2) Given diameter of rolling circle, 30 feet; revolutions, 50 per minute, what speed can be made, allowing a slip of 30 per cent?

Operation:

$D N = 30 \times 50 = 1500$

$1 - s = 1 - .30 = .70$

$.70 \times 1500 = 1050$

$1050 \div 88 = 11.9$ miles per hour $= 10.35$ knots.

(3) Given a speed of 18 miles per hour, a slip of 24 per cent, and a wheel whose rolling circle has a diameter of 40 feet. Required the number of revolutions per minute.

Operation:

$88 \times u = 88 \times 18 = 1604$

$1 - s = 1 - .24 = .76$

$D (1 - s) = 40 \times .76 = 30.4$

$1604 \div 30.4 = 52.8$ revolutions per minute.

(4) Given a speed of 14 miles per hour; revolutions, 40 per minute; slip, 28 per cent. Find the corresponding diameter of rolling circle.

Operation:

$88 \times u = 88 \times 14 = 1232$

$1 - s = 1 - .28 = .72$

$N (1 - s) = 40 \times .72 = 28.8$

$1232 \div 28.8 = 42.8$ feet.

Sec. 82. POWERING SHIPS.

The subject of the powering of ships is one which can be here only referred to in a brief and elementary way. The usual problems are to find the power required to drive a given ship at a proposed speed, or the probable speed for a given ship with a given power. Such problems require a knowledge of the relation between power, speed and the ship. In the present state of our information on this subject, such relation cannot be accurately expressed by any ordinary formula or equation. Several approximate formulae have, however, been employed for the solution of such problems, and among them none has perhaps been of wider general usefulness than the so-called Admiralty coefficient formula.

Let $H =$ I.H.P.

$D =$ displacement in tons.

$v =$ speed in knots.

$K =$ a coefficient.

Then according to this formula we have:

$$H = \frac{D^{\frac{2}{3}} v^3}{K}$$

and solving for speed:

$$v = \sqrt[3]{\frac{HK}{D^{\frac{2}{3}}}}$$

and solving for the coefficient:

$$K = \frac{D^{\frac{2}{3}} v^3}{H}.$$

The whole point in the use of the formula is to properly select the values of the coefficient K in accordance with the special features of the case, including the form and size of the ship, proposed speed, probable efficiency of propulsion, etc. The safest plan is to find values of K from the trial data of actual ships of about the same size, character of form and speed, as the proposed case, and to be guided by such values in the selection of the coefficient for the proposed case. There are other special methods for obtaining from the trial data of ships of similar form, by the so-called law of comparison, the suitable values for a proposed case, even when the sizes and speeds differ considerably from those of the proposed case. Into the details of these points, however, we cannot here enter. Some general suggestions regarding the value of K with a few illustrative examples must suffice.

We find then by experience that in general the value of K is greater (and hence the I.H.P. *relatively* less) as the ship is larger, but more especially as she is longer, also as she is narrower in proportion of length to beam, and as she is finer in form, especially in the water lines.

In the reverse cases the values of K will be smaller, and the I.H.P. *relatively* larger. The values of K are also smaller and the I.H.P. relatively larger as the speed is higher in proportion to the length, or, more exactly, as the speed is higher in proportion to the square root of the length. For small launches and such craft driven at speeds in miles or knots greater than $\sqrt{L}$ in feet, the values of K will be quite small, ranging perhaps from 100 to 150. At lower speeds equal to or less than $\sqrt{L}$ the values will rise to perhaps 200 and more with fine form and small proportion of beam to length. For yachts and craft of similar form, moderately fine and at fairly high speeds, values of 200 above and below will be found. For torpedo boats, with their narrow proportions and fine form, their excessive speeds carry them into a set of conditions where the coefficients are larger and the power required relatively less than we might expect. Varying with size and other conditions, values of 200 above and below are found for boats of this character. For mercantile vessels of moderate size, rather full form and moderate speed, the values will be usually found from perhaps 220 to 250. For larger mercantile vessels at moderate speeds or for those of moderate size under exceptionally good conditions the values may rise from 250 to 300. For fast passenger boats varying with size, and other conditions, values from 220 to 280 may be expected. For naval vessels, cruisers and battleships, from 200 to 250 is the usual range.

These various values are not intended as marking definite limits, nor can they enable a person without individual judgment to properly select a suitable value for a given case. They are intended simply as general suggestions of the range of values commonly met with.

We will now solve a few examples to illustrate the use of this formula.

We may first note that $D^{\frac{2}{3}}$ means the cube root of the square, or the square of the cube root of the displacement in tons, and hence with a table of squares and cubes or square and cube roots, the value desired may be readily found. In some hand books values of $D^{\frac{2}{3}}$ are given directly.

(1) Given $D = 3200, v = 12$ and take $K = 230$. Required the power.

From a table of squares we have $(32)^2 = 1024$, and hence $(3200)^2 = 10{,}240{,}000$. Looking in the column of cubes of numbers of three figures we find that the nearest cube is 10,218,313, and that the number corresponding is 217. This will be sufficiently near for all practical purposes, and is therefore taken as the value of $D^{\frac{2}{3}}$. We have also $(12)^3 = 1728$. Hence we have:

$$H = \frac{217 \times 1728}{230} = 1630 \text{ Ans.}$$

(2) Given a yacht of displacement 366 tons, to be driven at a speed of 18 knots. Assume $k = 200$ and find the necessary power. In this case $(366)^2 = 133{,}956$ and the number corresponding to the nearest cube is 51.2. Also $(18)^3 = 5832$. Hence

$$H = \frac{51.2 \times 5832}{200} = 1493.$$

(3) Given $D = 7243$, $v = 16$ and take $K = 240$. Find the power. Without important error we may drop the last 3 tons so as to bring the number within the range of the usual tables of squares and cubes. We have then $(724)^2 = 524{,}176$, and hence $(7240)^2 = 52{,}417{,}600$. The number corresponding to the nearest cube is 374. Also $(16)^3 = 4096$. Hence:

$$H = \frac{374 \times 4096}{240} = 6283.$$

(4) What speed may be expected from a liner of 15,400 tons displacement and 26,000 I.H.P.? Take, in this case, $K = 250$. Then $(154)^2 = 23716$ and hence $(15400)^2 = 237{,}160{,}000$. The number corresponding to the nearest cube is 619. Then we have

$$v = \sqrt[3]{\frac{26{,}000 \times 250}{619}} = \sqrt[3]{10{,}500} = 21.9.$$

(5) Given $D = 8320$, $v = 13$ and $H = 3400$. Required the value of K.

We have $(832)^2 = 692{,}224$ and hence $(8320)^2 = 69{,}222{,}400$. The number corresponding to the nearest cube is 411. Also $(13)^3 = 2197$.

Hence we have:

$$K = \frac{411 \times 2197}{3400} = 265.$$

Sec. 83. REDUCTION OF POWER WHEN TOWING OR WHEN VESSEL IS FAST TO A DOCK.

It is well known that the power developed by the engine when the ship is towing is less than when she is running free, the steam pressure and cut-off being the same; also that at a dock trial (engines running, but ship fast to the dock) the power is considerably less than may be developed in free route under the same steam pressure and point of cut-off.

To explain these results let us first assume that the ship, boilers, engine and propeller are all properly designed for a given speed. This means that with a given boiler pressure (say 180 lbs.) the boilers will be able to supply steam enough to drive the engines at the designed revolutions (say 100) and thus develop the designed power, while the propeller with a certain slip (say 20 per cent) will drive the ship at the designed speed (say 16 knots). Now it must first be noted that all of these conditions go together, and if any one of them is disturbed it will react on all the others. The next point is that a constant set of pressures throughout the engine means a constant reduced mean effective pressure, a constant turning moment on the shaft and a nearly constant thrust, and hence a nearly constant resistance overcome. Now at the regular speed, if the resistance is increased, as by taking up a tow, what will be the immediate result? Evidently the speed will decrease until at some reduced speed the nearly constant thrust will balance the resistance, and the motion will become uniform again. The greater the increase in resistance at the regular speed—that is, the larger the tow—the lower the speed at which the nearly constant thrust will be able to balance the resistance and thus produce steady conditions.

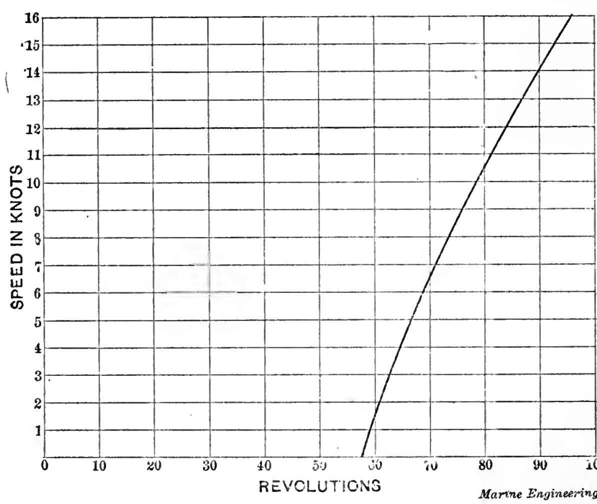


Fig. 294. Diagram Showing Relation Between Revolutions and Speed for Constant Turning Moment.

The whole question as regards power developed now turns on the revolutions at this reduced speed. In other words, how do revolutions and speed vary for a constant turning moment, and a nearly constant thrust developed or resistance overcome? This is best answered by experimental data, which give us a relation similar to that shown in Fig. 294. Revolutions are laid off horizontally and speed vertically. If we suppose that 90 revolutions and 14 knots are the designed conditions, as indicated on the curve, then the diagram shows how the revolutions and speed vary for a constant thrust. In particular the curve shows, as the tow is increased in amount and the speed for a given thrust decreases more and more, so likewise do the revolutions decrease, though at a slower rate. Hence with the decrease of speed the slip constantly increases. The relations as shown by this curve furnish the key for the solution of all questions regarding the variation of the power. The work done by the engine is a product of the revolutions into other factors, and since these other factors include merely dimensions of the engine and mean effective pressure, and since by assumption all of these remain constant, it is evident that under the conditions assumed the power developed will vary directly with the revolutions. Hence the diagram shows the *relative* decrease of power with decreasing speed, as well as the actual decrease in revolutions.

If we should continue to add to the tow indefinitely we should at length reach a condition similar to that of a vessel tied to a dock; that is, a condition where the speed has become reduced to nothing and the revolutions reduced in quite marked degree. The exact relation between revolutions in free route and when fast to a dock will of course depend on the special circumstances. The mean effective pressure remains the same, however, and the work done and power developed will therefore vary simply with the revolutions according to a law similar to that shown in the figure.

To sum up the matter, therefore, the power falls off because the revolutions fall off, and the revolutions fall off because the speed falls off, and because at the reduced speed and increased slip the constant turning moment of the engine can no longer turn the propeller at the original number of revolutions.

The reason why of the facts expressed by the curve in Fig. 294, on which the whole matter turns, is to be found in the fundamental relation between revolutions, slip, thrust, etc., a complete discussion of which is of course beyond our present purpose. Once these relations accepted as experimental truth, however, the desired explanation is seen to flow from them as a necessary consequence.

Sec. 84. TRIAL TRIPS.

The general purpose of a trial is to determine the power or speed which may be maintained for a certain distance or time. In addition to these fundamental purposes, information relating to the general problem of resistance and propulsion may also be gained, as well as that bearing on other points which may be the object of special inquiry. We shall not here refer especially to the determination of power, as that has been already sufficiently treated in Sec. 55 [3].

For the determination of speed alone it is sufficient to obtain observations of *distance* and *time*. The revolutions should, however, be also taken, in order that the slip of the propeller may be found. For speed trials we may use a long course, as, for example, from 20 to 100 miles or more, over which but one run, or, more commonly, one run in each direction is made; or, on the other hand, a short course of 1 or 2 miles, over which as many runs may be made as desired.

For marking off the course, buoys or ships at anchor are often used for the long course, while range marks on shore for the limits and buoys for the direction and location are commonly employed for the short course. At each end of the short course there should be plenty of room for making turns and gathering headway before entering the course for the return run. The free space available for this purpose should be not less than from one-half to one mile.

To eliminate the error due to the tide on the long course run, tidal observations should be made from vessels anchored along the course by means of a patent log or equivalent device, and from the results the average tidal influence may be determined. To eliminate the tidal error on short course trials the runs are made in both directions and an average is taken. This may be either a simple average or the result of a "continued average," as illustrated below.

Suppose four runs made, two in each direction, and let the resulting speeds in order be those entered in the column on the left.

North	17.2			
		17.00		
South	16.8		17.075	
		17.15		17.10
North	17.5		17.125	
		17.10		
South	16.7			

An average is first made of Nos. 1 and 2, then of 2 and 3, and then of 3 and 4, and these are put in the second column. Then these are averaged in like manner and put in the third column, and these are again averaged for the final result, which, in the above case, is 17.10. With six runs the operation is carried out in the same way. While this is often considered as the only correct way of averaging such a series of runs, it may be shown that such is by no means the case, and, as a matter of fact, that under ordinary conditions the simple average will give quite as probable a result as the more complex method. In the above case the simple average would give 17.05, as against 17.10, a difference of .05 knot, and the former value is quite as likely to be correct as the latter.

In some cases it is desirable to make a complete speed trial

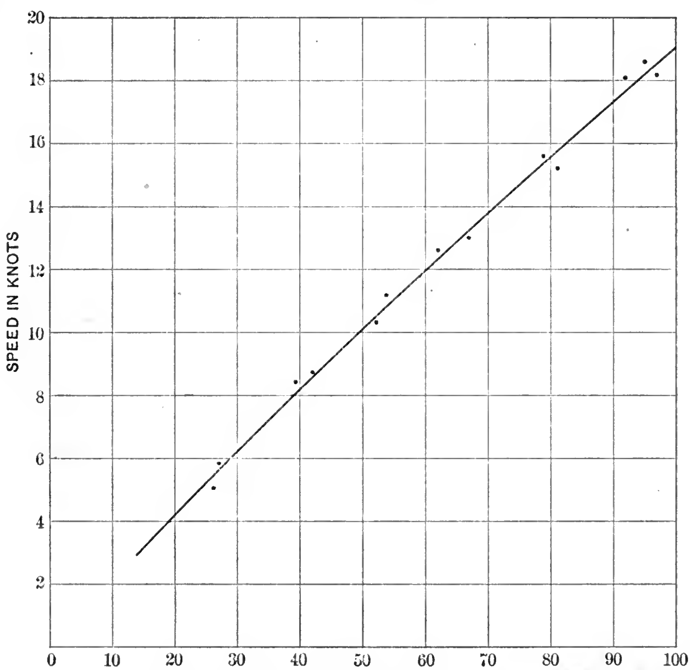


Fig. 295. Diagram of Revolutions and Speed.

and thus obtain a series of values of the power, revolutions and speed from full power conditions down. This may be done on the measured mile or short course by making runs in pairs, the conditions for each pair remaining as nearly constant as possible, while from one pair to another the conditions change over the complete range to be included in the trial. The average results in speed, power and revolutions are then used for plotting the curves showing the various relations desired. Such a curve showing the relation between revolutions and power is shown in

Fig. 296, together with the spots which may represent the actual single observations. The curve may then be drawn through and among the spots as a method of getting a graphical average; or otherwise the values may be averaged numerically and plotted as a series of averages, and the curve then drawn through them.

As a somewhat shorter method, a series of runs may be taken in each direction, beginning at the highest, and at constantly decreasing revolutions. The results for speed and revolutions are then plotted, as shown in Fig. 295, and a fair curve drawn through and among the spots. This is then taken as the

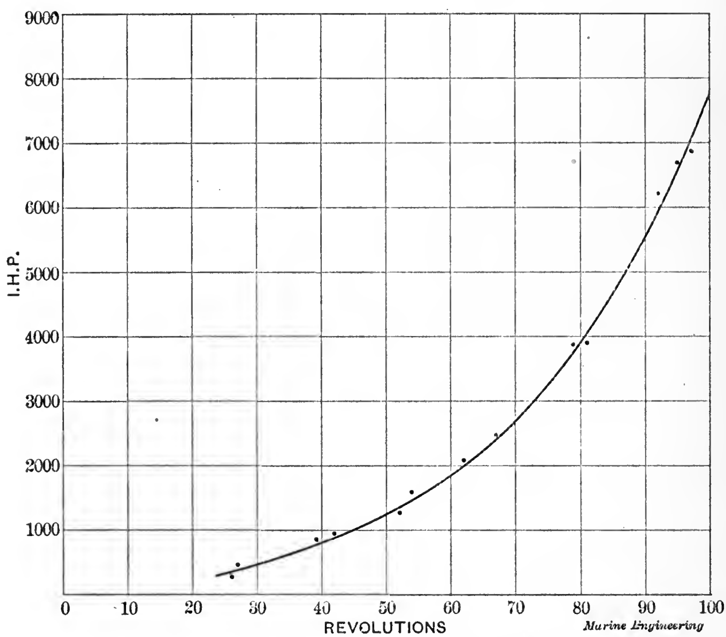


Fig. 296. Diagram of Revolutions and Horse Power.

relation between revolutions and speed. The relation between revolutions and power is also plotted, as in Fig. 296. Then, by the aid of these two, the speed-power curve may be plotted, as shown in Fig. 297. We will now note briefly the computations arising in connection with such trials. The observations with which we are here concerned are simply *time* and *revolutions*.

Let the course be a measured mile (marine or statute, as the case may be), and let the time on the course be t, expressed in minutes and decimals. This is usually determined by a stop-watch reading the half or quarter second. The value cannot, however, be depended on as accurate to much within 1 second. Then if v denotes the speed we have:

$$v = 60 \div t$$

The revolutions will be obtained from the counter by subtracting the readings at the entrance and end of the course. This will give the number of revolutions for the course. Let this be denoted by R. Then

$$\text{Revolutions per minute} = \frac{R}{t}$$

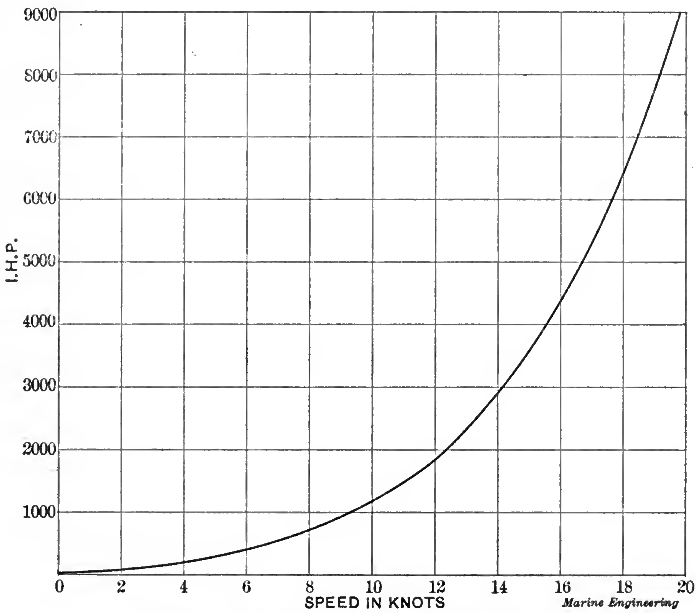


Fig. 297. Diagram of Speed and Horse Power.

Also let $p =$ pitch of propeller.

Then assuming the course to be a nautical mile, we have:

$$(pR - 6080) = \text{slip in feet, and } \frac{pR - 6080}{pR} = \text{slip ratio.}$$

These computations may be illustrated by the following example:

Let the course be a mile of 6,080 feet, and let the following data be taken:

	Counter at entrance.	Counter at end.	Time.
Run north	106,248	106,654	4 m. 26 sec.
Run south	107,112	107,542	4 m. 33 sec.

Pitch of propeller 18 feet.

Required the mean speed and slip for the two runs.

For the run north the revolutions are 106,654 — 106,248 = 406.

For the run south the revolutions similarly found are 430. The average revolutions are then 418. The speed north is $60 \div 4.433 = 13.53$. The speed south is $60 \div 4.55 = 13.19$. The mean speed is then 13.36.

The slip in feet $= 418 \times 18 - 6080 = 7524 - 6080 = 1444$.

The slip ratio $= \frac{1444}{7524} = 19.2$ per cent.

Sec. 85. SPECIAL CONDITIONS FOR SPEED TRIALS.

In speed or power trials the purpose is often the development of the maximum speed or power, and in the present section we may note the more important points connected with the fulfilment of these purposes.

Boilers. Where there is a record performance to be made there must be no loss of evaporative efficiency due to accumulations of soot and ashes on the fire side, or of oil, scale or mud on the water side. Hence especial care must be taken to see that the boilers are thoroughly clean on both fire and water sides.

Fuel. If possible, the fuel should be of the highest grade and carefully selected with reference to clean, free burning qualities.

Engines. The engines should be adjusted with the various joints sufficiently loose to avoid danger from heating, and at the same time not sufficiently loose to hammer seriously. Special attention must be paid to oiling gear, and also to the provision for supplying water in case the bearings tend to become hot.

Ship. The ship should be lightened as much as possible—that is, the displacement should be made as small as possible. This point must not be carried so far, however, as to decrease the draft or change the trim sufficiently to bring the tips of the propeller blades too near the surface of the water. In the latter case the propeller will fail to develop the necessary thrust and the highest speed cannot be attained. The bottom of the ship should also be thoroughly clean and fresh painted, or, if coppered, clean and polished, if not too large. The propeller should also be looked after, and if there is opportunity it should be cleaned and the edges sharpened.

CHAPTER XI.

Refrigeration.

Sec. 86. GENERAL PRINCIPLES.

In connection with the general principles of refrigeration reference should be made to Section 57, where the general nature of heat and its relation to matter is discussed. The fundamental problem of refrigeration is the abstraction of heat from some body or substance A, or the maintenance of such substance A at a temperature lower than that of the surrounding air. This is most conveniently brought about by bringing the substance A into relation with another cooling substance B at a lower temperature, such that heat may readily pass by conduction from one to the other. The heat then flows from A to B, and thus the end desired is brought about. It is clear, however, that unless there is a continuous renewal of the substance B, or some way of removing the heat which flows into it, then in the end the two substances A and B will come to the same temperature, and if this is lower than that of the surroundings, it will gradually rise until both A and B are in equilibrium with their general surroundings. Again it is seen that if the substance A is kept below the temperature of its surroundings there will be a constant flow of heat from these surroundings (structural material, earth, air, water, etc.) into it, and this heat must be as constantly removed by conduction into the substance B at still lower temperature, which again must be renewed in order to maintain its capacity for absorbing heat from A. Again we may have two different cases, according as the substance A is cooled by the conduction of its heat directly into B, or first into some intermediate substance such as air and then into B. In other words, the substance B may affect A directly, or through the intermediate action of the air surrounding both of them.

Thus in making artificial ice the water to be frozen is brought as directly as possible under the influence of the cooling substance, whatever it may be, while in certain systems for the refrigeration of meat, etc., in a refrigeration or cold storage room, the meat is kept cool by the action of the cold air of the room, which, in turn, is cooled by the action of the cooling substance, usually conducted through coils of pipe about the walls of the room. In other systems of refrigeration or cold storage, air itself is made the cooling substance, and the cooled air is passed through the storage room, thus acting directly upon the substances stored therein.

With the foregoing by way of a statement of general principles we come now to the production of the cooling substance. In other words, how shall we find or produce a substance whose temperature is far below the usual temperature of the air, and which may be made available for the purposes outlined above? To this end we have two general methods; one involving a change of state, either physical or chemical or both, and the other the compression, cooling and expansion of an elastic gas such as air. Illustrations of these methods will be found in the systems of refrigeration described in the following sections.

Sec. 87. REFRIGERATION BY FREEZING MIXTURES.

If salt and ice are mixed there is a tendency for the mixture to pass into the liquid state. As to just why this is so we are not concerned, but simply with the fact. In answer to this tendency the two substances pass rapidly from the solid into the liquid state, thus forming a brine, or, more exactly, the ice passes from the solid to the liquid state and the salt dissolves in the liquid. Now ice cannot pass from a solid to a liquid state without absorbing heat. This point has been referred to in Sec. 57 [1], where it was also noted that the term latent heat is applied to the heat which is thus involved in a change of physical state. Heat must therefore be supplied from somewhere in order that the change from ice to liquid may take place. Furthermore, due to the physical and chemical forces at work, this change takes place faster than heat can be supplied from the general surroundings, and since it must come from somewhere it is drawn from the substances themselves, brine, salt and ice, which are thus reduced to a much lower temperature than they would have if separate. Thus a mixture of ice and salt forms a brine having a temperature of about — 5 Fah., or about 37° below the freezing point of ice. Such solutions or mixtures are known as freezing mixtures, and the brine thus formed may then be used as a cooling or freezing agent in whatever way may be most convenient.

Thus in the ordinary manner of freezing ice-cream and ices the ice-salt mixture is commonly used, inclosed in a casing which surrounds an inner vessel containing the substance to be frozen. In the manner above described then the ice will be melted, a brine will be formed, and heat will be withdrawn first from the freezing mixture itself, and then from the substance in the inner vessel, which thus becomes cooled down to its freezing point. Next its own latent heat will be drawn upon, and thus finally the substance becomes frozen as desired.

There are many such mixtures composed of various chemicals, and by means of which temperatures from 0 to — 50 Fah. may be obtained. For the general purpose of refrigeration, however, the use of such mixtures has not been found as efficient as other means, and it will not be necessary therefore to further refer to them in the present connection.

Sec. 88. REFRIGERATION BY VAPORIZATION AND EXPANSION.

We will next consider the application of substances like ammonia or carbonic acid, which are gaseous at the usual temperatures, but may be liquified at low temperatures and under suitable pressures. Anhydrous ammonia (ammonia without water) has a boiling point under atmospheric pressure of 37° below zero. With higher pressures the boiling point rises. As we have already seen in Sec. 57, a liquid passing into the state of vapor absorbs a certain amount of heat, and this latent heat, as it is termed, will be drawn either from the surroundings or from the liquid itself, or from both, thus lowering their temperatures. In order to utilize these properties of ammonia the refrigerating apparatus consists of the following:

(1) A series of expansion evaporating coils which are placed in the refrigerator room or space to be cooled, or which in other systems are surrounded by a liquid which is used as the immediate cooling agent in the coils of the refrigerator room.

(2) A reservoir containing liquid ammonia which is allowed to flow as may be required into the expansion coils.

(3) A pump which withdraws the vapor from the expansion coils and then compresses it back into the coils of a condenser, which are surrounded by cool water. Here, under the influence of the pressure and moderate temperature, the vapor becomes condensed to liquid and then flows to the reservoir from which it started. The two fundamental parts of the process are therefore (1) evaporation and expansion, (2) compression and condensation. In this manner the continuous operation of the pump insures the formation and continuous flow of ammonia vapor at low temperature through the cooling coils. The vapor is thus in condition to absorb heat through the metal of the coils from the surroundings, either air or liquid, and thus the refrigeration is effected. Where a liquid is employed as the immediate cooling agent it is usually a brine made with common salt, which, after having been cooled down by giving up its heat to enable the ammonia to evaporate, is then circulated by a brine pump through the coils of the refrigeration room.

It has been well said that the action of the ammonia in this round of operations is like a sponge. It vaporizes and expands in the expansion coils and absorbs or soaks in the heat from its surroundings. Then it is compressed and the heat is forced or "squeezed" out, and it is ready for a new round, thus acting as a carrier of heat away from the substance to be cooled.

It may aid further in understanding the action of the liquid ammonia in the refrigerating coils to note that relative to the air or brine surrounding these coils the ammonia is situated somewhat like the water in a water-tube boiler with hot gas on the fire side of the tubes. The furnace gases are hot relative to the water, and so heat tends to flow through the tubes into the water, thus forming steam. So is the temperature of the air or brine about the coils far above that at which ammonia would naturally exist in the liquid form under the pressure in the coils. In fact, the liquid ammonia when first admitted is itself far above this temperature, so that the first result is a tendency for the liquid to fly into vapor immediately, drawing on its own sensible heat to supply the necessary latent heat, and thus cooling the remaining liquid and the vapor formed. Then the heat from the surrounding air or brine flows in and thus the vaporization is completed. We may thus say that the liquid ammonia is boiled into vapor chiefly by the heat which it draws from the surrounding air or brine. So likewise if we should set a jar of liquid ammonia into a snow bank, the latter would be warm relative to the boiling point of the ammonia under atmospheric pressure, and in consequence the snow would play the part of the fire in the usual case with water and supply the heat which would serve to vaporize the ammonia, the snow becoming thereby cooled by reason of its loss of heat.

Having thus sketched the general outline of the process, the following additional points may be mentioned:

The liquid ammonia is usually under a pressure of 125-175 lbs. per square inch, corresponding to temperatures from about 70 to 90 degrees. By the action of the ammonia pump, which draws the vapor from the refrigerating coils, the pressure in the latter is maintained at from 30 to 60 lbs., corresponding to temperatures from 0 to 30 Fah. The ammonia condensing coils are in some cases immersed in water, which is renewed in order to maintain the temperature as low as convenient, while in other cases the condensation is brought about by allowing a spray of water to fall over them from top to bottom.

Except for details of the apparatus employed, the principles outlined above, as well as the leading features of the equipment, are the same for a variety of substances which may be used as refrigerating agents.

Thus, sulphur dioxide, carbon dioxide, sulphuric ether, methylic ether and still more recently liquid air may be and have been used in the general manner above described.

Sec. 89. PRINCIPAL FEATURES OF AMMONIA REFRIGERATING APPARATUS.

In Fig. 298 is shown diagramatically the arrangement of apparatus in the De La Vergne system of ammonia refrigeration. In all such systems involving the compression of a gas it is most important that the compressed gas be as completely discharged as possible at the end of the stroke, else it will re-expand on the return stroke, thus preventing the inflow of a full charge of fresh gas and thus reducing the effective capacity of the machine. In the system illustrated in these figures this is accomplished by injecting into the compressor at each stroke a certain quantity of oil which fills all clearances and thus insures the delivery of practically the entire charge of gas. This oil likewise acts to lubricate the moving parts and to seal the stuffing-box, piston and valves, and thus to prevent leakage. It also acts to some extent to absorb the heat resulting from compression, thus to reduce the expenditure of work required. In Fig. 299 is shown the double acting compression cylinder used in this system. The two passages marked "Suction" and "Discharge," respectively, connect the compressor with the pipe system.

On the up stroke gas flows through the lower suction valve into the space behind the moving piston, while the gas above the piston, after being compressed to the condenser pressure, is discharged through the upper valves (in the loose head) into the discharge passage.

On the down stroke gas flows into the cylinder through the

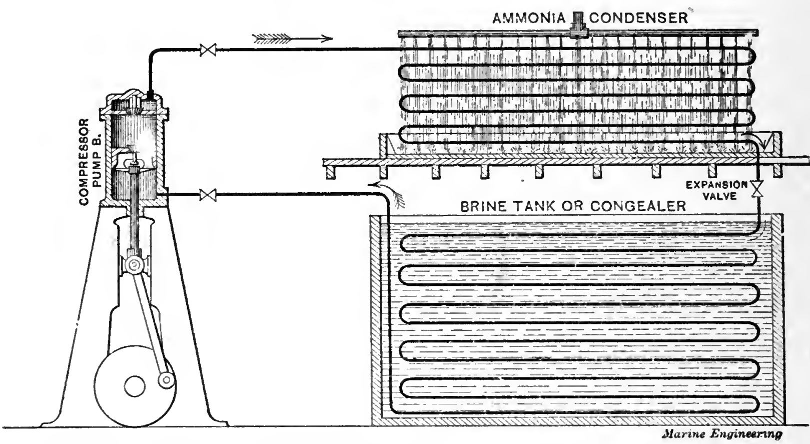


Fig. 298. Diagram Showing Arrangement of Ammonia Refrigerating Machinery.

upper suction valve, and the gas below the piston is compressed and passes through the lower discharge valves into the discharge passage. The piston in its downward course closes successively the openings of these two discharge valves. When the lower is closed, however, the upper one communicates with the chamber in the piston, and the gas and oil still remaining below the piston are discharged through its valves into the chamber and out by the upper discharge valve.

The oil is injected directly into the compressor after the compression of the full cylinder of gas has commenced, and thus does not reduce the capacity of the machine.

The compressed gas and oil thus delivered from the compressor cylinder pass on to the oil cooler. The cooled oil drops into the bottom of the tank, while the gas continues into the condenser, where it is liquified and collected in a second tank. Two forms of condenser may be employed. In one form the condensing coils are immersed in a tank of cold water, which, by suitable pumping arrangements, is continuously withdrawn and renewed in order to maintain as low a temperature as convenient. In the other form of condenser water is sprayed over the coils, falling from top to bottom, while the gas enters preferably at the

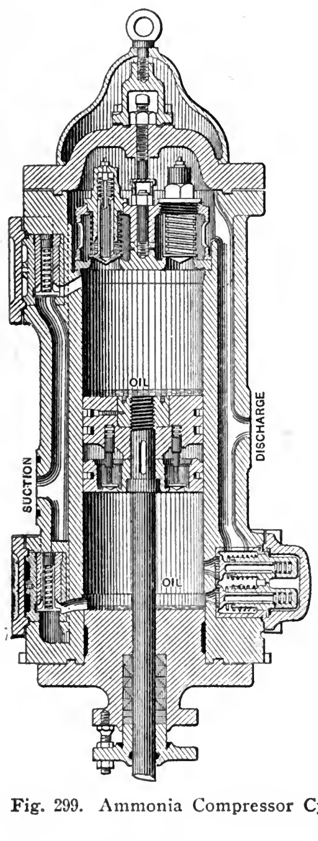


Fig. 299. Ammonia **Compressor Cylinder.**

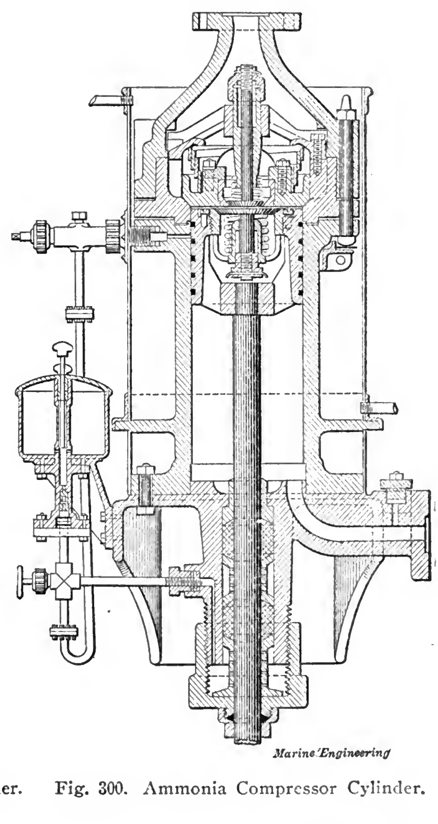


Fig. 300. Ammonia Compressor Cylinder.

bottom and passes upward in a direction opposite to that of the water. The latter type of condenser is the more efficient of the two for the same amount of water, while it has the further advantage that ammonia leaks are readily detected by the odor, while with the submerged condenser the escaping gas is absorbed by the water, and its escape is not so readily noted.

From the collecting tank the liquid ammonia passes through the expansion cock into the expansion coils. The latter may be located directly within the space to be refrigerated, or they may be surrounded by a brine, which in turn is pumped through the refrigerating coils proper. The former method is the more efficient inasmuch as a loss always attends a multiplication of such processes. The chief objection to this method lies in the somewhat greater liability of ammonia leaks and the resulting presence of ammonia in places where it may be objectionable. The expansion cock must be capable of nice adjustment in order to make possible the proper control of the flow of liquid ammonia into the coils. In the system here shown this is accomplished by making the orifice on the delivery side of the cock in the shape of a very narrow wedge, the point of which is the first to open. Movement is then imparted to the plug by a worm and wheel, thus insuring adjustment of the most delicate character.

The point of chief importance in connection with the expansion coils, and, in fact, in connection with the entire piping system, is that of the joints. Screwed connections are far more liable to leak under ammonia than under steam, and the utmost care is needed in regard to this feature. For the best results special joints are required. In a representative joint of this character the thread into which the pipe screws does not reach entirely to the outside of the fitting, but instead a smooth annular space is provided around the pipe beyond the termination of the thread. This recess is filled with solder, the pipe and fitting being well tinned, and thus a screwed and soldered joint is made which is found tight against ammonia under all pressures employed.

The cooling efficiency of the refrigerating coils is also much increased by clamping to the pipes thin disks of cast iron. These disks are made in halves, and are placed at intervals of 6 to 10 inches on the pipes. They effect an increase in the surface in contact with the air, and thus an increase in the heat, which can be withdrawn from the air and conducted to the cooling substance within the pipe.

In the Eclipse system of refrigerating machinery the same general principles are involved, but with some differences in the apparatus employed. The compressor, as shown in Fig. 300, is single acting, the gas being compressed on the upper side of the piston only. No oil is used to fill the clearance spaces, and the clearance is reduced to a negligible quantity by working the piston almost metal and metal against the head. This is made practicable by making the pump head movable so that it may operate as a large valve the full size of the bore of the cylinder, and through the seat of which, if need be, the piston might pass without injury. Under normal conditions this entire head does not work as a valve, the discharge being through a small steel valve in the center of the head, as shown in the figure. The compressor cylinder is surrounded with the water jacket, which absorbs a part of the heat generated by the compression of the gas. For controling the expansion in this system a special form of valve is used which provides for close adjustment of the opening and is fitted with an index and pointer so that the amount of opening may be ascertained and such adjustments made as have been by trial found to best suit each case.

Sec. 90. REFRIGERATION BY THE EXPANSION OF A COMPRESSED GAS.

We will next examine the principles involved in the use of air as a refrigerating agent through compression, cooling and expansion.

If compressed air is allowed to expand against a resistance, thus doing work, while at the same time no heat is allowed to enter, the air will lose a part of its heat, the equivalent in amount of the work done. That is, the work is done at the expense of the heat in the air, which gives it up as called for and becomes correspondingly cooled in consequence. Thus, for example, if air at 100 pounds absolute pressure and at a temperature of 70 degrees were allowed to expand to 15 pounds pressure, and no heat be permitted to enter, the temperature would become reduced by about 220 degrees, or to about 150 degrees below zero.

Actually the absorption of some heat could not be avoided, and hence the actual temperature reached would be higher than thus indicated. The equipment for utilizing these properties of air is substantially as follows:

(1) A cylinder in which air is drawn from the atmosphere or from the refrigerating coils and compressed. The air thus becomes heated and its temperature is raised. It is therefore sent next to (2), a cooling coil working after the manner of a surface condenser for steam, or for ammonia vapor, as above described. The difference here is, however, that no effort is made to condense the air, but simply to cool it down to somewhere about at-

mospheric temperature. The air is next sent to (3), a cylinder in which the air expands and does work and becomes thereby cooled. From this cylinder the cooled air on the return stroke is forced out through the refrigerating coils, and thus the cooling action is brought about. The compression and expansion cylinders are connected to the same crank shaft in such manner that the work done by the expanding air will aid in effecting the compression of the incoming fresh charge. The difference in the work of compression and that furnished by expansion plus the friction of the machine must be made up by the motor operating the machine.

Sec. 91. PRINCIPAL FEATURES OF COMPRESSED AIR REFRIGERATING APPARATUS.

We will now briefly describe the leading features of the Allen dense-air refrigerating machine as illustrating this type of apparatus.

Experiment has shown the advantage of using relatively high pressures throughout the apparatus. The air in the refrigerating coils is therefore kept at about 60 lbs. gauge pressure.

From these coils the compressor cylinder draws its air, which is then compressed to 200 lbs. or over, and this, after cooling, is again expanded back to 60 lbs. and sent to the coils again, so that the same air is used over and over. To make up for leakage a small air compressor is added, capable of delivering air at the lower pressure into the inflow pipe leading to the main compressor. Again the cold air on its way back from the coils at a temperature usually below freezing is used in a special cooler to further cool the compressed air at high pressure before it is sent to the expansion cylinder.

Referring to Fig. 301, the following are features of the cylinder with their uses:

(A) the steam cylinder which furnishes the power required.

(B) The air compressor cylinder. This is usually surrounded by a water jacket to assist in cooling the air as it is compressed.

(C) The cooling coil surrounded by water. Through this the air passes and thus becomes cooled nearly to the temperature of the external air.

(D) The return air cooler for still further cooling the compressed air by allowing it to give a part of its heat to the air returning from the refrigerating coils to the compressor inflow, as noted above.

(E) The expansion cylinder. The cooled air is admitted to about one-third stroke and is then cut off. The charge is thus expanded to about three times its original volume, and is thus brought to about the same pressure as it had when entering the compressor cylinder, but at a much lower temperature. The air is then discharged through a pipe which leads it to the refrigerating coils or to the point where its capacity for absorbing heat is to be utilized.

(F) A trap through which the air passes after leaving the

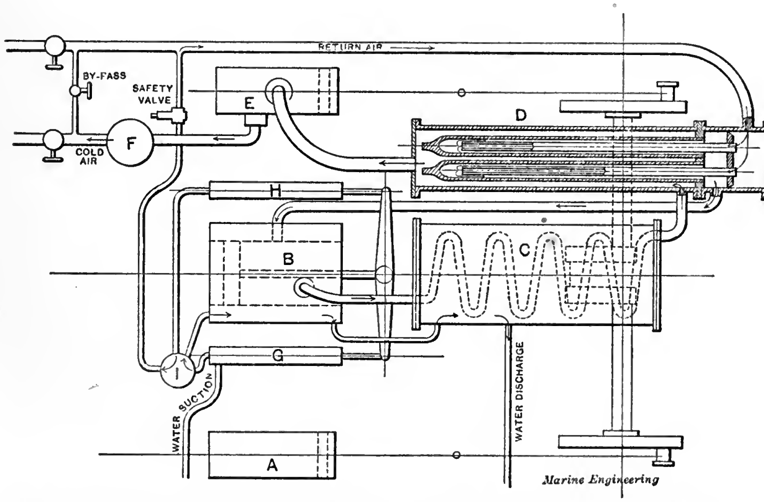


Fig. 301. Diagram Showing Arrangement of Compressed Air Refrigerating Machinery.

expansion cylinder and in which are gathered the lubricating oil carried by the air from the compressor cylinder as well as the frost which results from the freezing of the moisture in the air. This leaves the air pure and dry and in the best possible condition for carrying out the refrigeration in the coils beyond.

(G) A pump for supplying water for the water jacket around the compressor cylinder, for the bath around the cooling coils (C), and for the trap (I).

(H) A small air compressor for supplying the loss due to leakage.

(I) A trap for taking the moisture from this supplementary air supply so as to have it enter the machine as dry as possible.

Sec. 92. OPERATION AND CARE OF REFRIGERATING MACHINERY.

Before starting refrigerating machinery, whether newly installed or after any considerable period of disuse, all piping and joints should be tested for leaks. This may be done, no matter what the system be, using the compression pump to compress air into the piping up to whatever pressure may be considered suitable. The seriousness of the leakage may then be estimated by the rapidity with which the pressure is lost after allowing the pump to stop. The larger leaks may be determined by the noise made by the escaping air. For the smaller ones the joints are sometimes covered with soap-suds so that the escaping air may show itself by blowing a cluster of bubbles. After the points which may show leaks have received proper attention, the system should, for a considerable time, hold the pressure without sensible loss. In this connection, however, it must be remembered that the air as it leaves the compressor will be heated by the work of compression, and as it loses this excess heat in the coils there will be a corresponding loss of pressure. After equality of temperature with the outside air has been reached, however, the further loss of pressure should not be appreciable. With an air refrigerating plant the presence of leaks is of course of less importance than with ammonia. In the former a leak may result in a slight decrease in the capacity of the machine, while in the latter the capacity of the machine is not only affected, but ammonia will be lost as well. With the air machine no further preliminaries are needed beyond the examination necessary to insure the proper mechanical condition of the compressor and steam cylinders. With the ammonia machine, however, it is necessary next to exhaust the air from the entire system by working the pumps and discharging through valves provided for this purpose. When the gauges show the highest vacuum which can be maintained, the valves are closed and the system is ready for charging.

The ammonia is usually provided in steel flasks containing a known weight of the liquid. To introduce the charge the flask is connected to the charging valve according to the directions which usually accompany them. In the meantime the machine is run at slow speed with the suction and discharge valves open and the condenser ready for operation. The expansion valve is then closed and the valve of the flask opened, thus allowing the ammonia to be exhausted into the system. In this manner one flask after another is exhausted into the system until the liquid shows to a suitable height in the glass gauge of the receiver. The charging valve is then closed and the expansion valve opened and regulated until with the machine running at a normal speed the pressure gauges steady down to the pressures usually maintained, as mentioned in Sec. 88, and a frosting of all parts of the expansion coils in contact with the air shows that the refrigerating action is in vigorous operation. The weight of ammonia in the flask being known, its complete discharge may be determined by weighing before and after connection with the machine.

For the routine care of refrigerating machinery the same principles apply as for steam engines and pumping machinery in general, and they need not be here repeated in detail. A few special points may, however, be mentioned.

In ammonia machinery, through leaky stuffing-boxes and in other ways, air may occasionaly find its way into the system. Its presence will decrease the efficiency of the machine, and it must therefore be removed as soon as possible. The most noticeable symptoms of such trouble will probably be a rise in the condenser pressure. Purging valves are usually fitted on the condenser or elsewhere through which such air may be drawn off. To this end it is desirable to stop the machine for a time to allow the air to collect. It may be then drawn off through a rubber hose or other suitable means and discharged under water. The rise of bubbles will show that air is escaping, while on the other hand the presence of crackling or snapping sounds will indicate that the air is all exhausted and that ammonia is escaping and being absorbed by the water.

Reference has been made to the importance of joints and piping. This part of the system must receive especial care both in the original installation and in the routine attention. For ammonia in particular, as already noted, special joints are usually required, and both ammonia and air will find smaller leaks than steam. It must also be remembered that the chemical action between ammonia and copper renders impossible the use of copper, brass or bronze through all parts of the installation with which the ammonia may come in contact.

With compressed air refrigerating machinery it sometimes happens that the ports or passages through which the air first passes in its expansion become clogged with a deposit of snow or ice, due to the freezing out of the moisture contained in the air. In the Allen dense-air machinery, where the same air is used over and over again, additional moisture can only come from the small make up air supply, and most of this is removed by the trap provided for this purpose. In operation this trap should be watched in order to make sure that its action is efficient and that there is no danger of the passage of water over into the expansion system. In routine operation it is usually desirable to clean the machine by heating it up and blowing out all the oil and ice deposits. To this end the valves in the main pipes leading the air to and from the coils are closed, thus shutting off the machine from the remainder of the system. A by-pass is then opened, connecting the main expansion pipe beyond the oil and snow trap with the main return from the coils. Connections are then opened in the so-called hot air pipe leading from the compressor cylinder to the expander cylinder, and the expander inlet valve is partly closed. Live steam is then let slowly into the jacket of the oil trap in order to thaw out all ice and hardened oil, and the machine is run moderately for a time, during which the blow off valves of the trap and expander cylinder are frequently opened until everything appears clean. Then the machine is readjusted to its normal condition and run as before.

If it should be suspected that any considerable quantity of oil and water have gotten into the pipe system and are clogging the surfaces, the pipes may be cleaned by running hot air through them and drawing off the oil and water at the bottom of the manifolds of the refrigerating coils.

CHAPTER XII.

Electricity on Shipboard.

Sec. 93. INTRODUCTORY.

In the present chapter we shall discuss briefly and from the practical standpoint the application of electricity on board ship for lighting, and as a source of auxiliary power. The limitations of space prevent, of course, the development of the subject in detail, and we shall therefore give by way of introduction a few

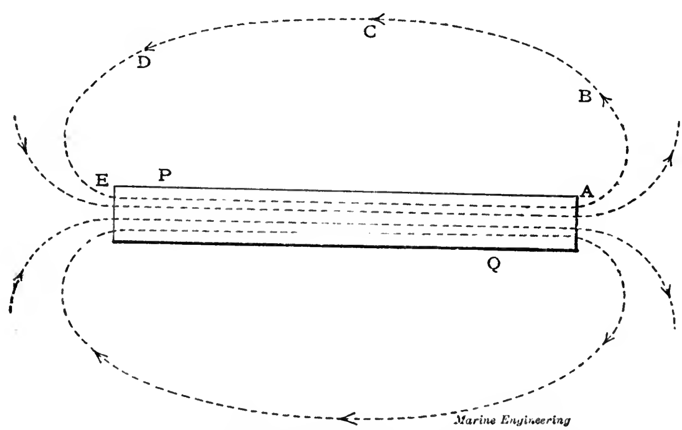


Fig. 302. Simple Bar Magnet, Showing Lines of Force.

statements and definitions which must be taken for granted by those not already familiar with the elements of electrical theory.

(1) We first suppose that the reader is familiar with a *common magnet* and its more well-known properties.

(2) The name *magnetic field* is applied to the space around a magnet and through which the magnetic forces act. Let PQ, Fig. 302, be a magnet, with one pole at E and one at A, and sup-

pose the latter to correspond to the north end of a compass needle.

It must be understood that the magnetic forces act really in closed paths, as indicated by the dotted lines in the figure. That is, if we should map out the direction in which the north end of a long, thin magnet would be urged, beginning with A, we should trace out a path ABCDE. That is, a north magnet pole, if free to move by itself, would tend to move along the path from A around to E, in the direction of the arrow, and would so move unless prevented by some external force. Hence the magnetic force acts all along this line from one end to the other, thus marking out what is called a line of force. Now, to complete the circuit it is considered that the same force acts on through from E

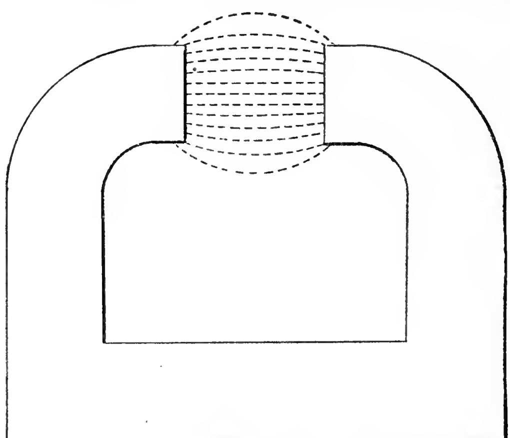


Fig. 303. Horseshoe or Bent Magnet, Showing Lines of Force.

to A inside the iron or steel, although we are not able to measure the actual force there in the ordinary way. The entire space around and within a magnet is thus occupied with these lines of force, and in its widest sense therefore the magnetic field, or field of force of a magnet, would include all space. As we move away from the immediate vicinity of the poles, however, the force becomes weaker and weaker, and finally at no great distance becomes very small. Practically, then, the field of force includes only that part of space within which the magnetic forces are measurably large in amount. By changing the form of the magnet, as in Fig. 303, the sensible part of the field becomes limited to the space between the two poles, as shown by the dotted lines, and it also becomes quite uniform in strength.

(3) All the phenomena connected with what we call electric currents in wires or other conductors take place as though a current of something was flowing around the closed circuit. Scientists do not assume that there is in reality anything actually flowing within the wire. In fact, the nature of electricity and of the electric current are not satisfactorily known, and so in the absence of more definite information we speak of the *electric current* simply as an aid in the discussion.

(4) The fundamental principle which seems to connect magnetism with electric currents is found in the following fact:

A wire or conductor in which an electric current is flowing is surrounded by a field of magnetic force, as shown in Fig. 304. If the current is flowing away from the observer, or along the direction in which he looks, then the direction of the force is such that a free north magnetic pole would tend to go round and round the wire in circular paths in the direction shown by the dotted lines.

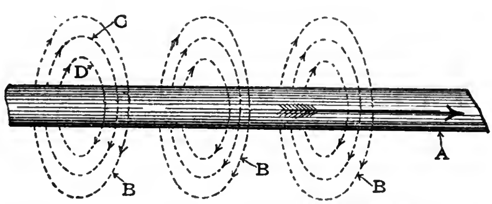


Fig. 304. Lines of Force About a Wire Carrying an Electric Current.

(5) If we bend the wire in which the current is flowing into the form of a circle, as in Fig. 305, then these separate effects combine and we have a field of force just the same as though the wire circuit were a very short, flat magnet, the two poles being very close together. If we have many turns of wire on a spool then all these effects are added, and we have a magnetic field of still greater force and distributed almost exactly as though the core of the spool were a bar magnet. If, in fact, we put in a piece of soft iron for the core of the spool, then we shall find that the iron itself becomes magnetised, and adds its force to that of the current, thus producing a still stronger field of force. A rather more exact way of stating this is to say that the current which flows tends to produce a magnetic field, but that there is a certain resistance to the setting up of this force, and that this resistance depends on the substance through which it is to be set up. If there is no metal core, then the magnetic force must pass through

air around the entire circuit. It so happens that the resistance to the setting up of magnetic forces is very much less in iron than in air, so that if an iron core is put in, the total resistance in the circuit of the magnetic forces is very much reduced, and the same electric current can then set up a much stronger field than through air all the way.

(6) We can now state the fundamental principle upon which the operation of the electric generator depends.

If a wire is so moved as to cut across the lines of magnetic force, then there will be generated a force tending to set up a current of electricity in the wire. This is known as the electromotive force, and is usually abbreviated into E.M.F. If, then, the ends of the wire are connected so as to form a closed circuit,

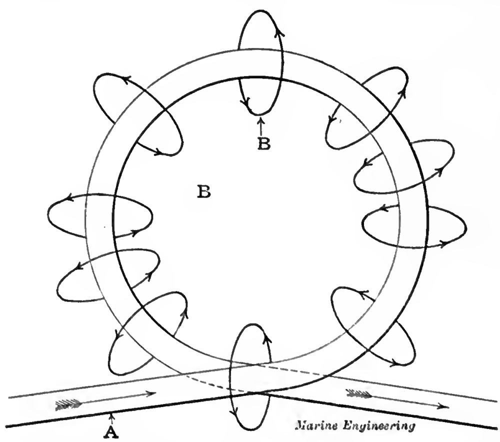


Fig. 305. Lines of Force About a Coil Carrying an Electric Current.

and the movement is such that the amount of magnetic force which passes through the circuit of the wire undergoes a change, either increase or decrease, then a current of electricity will be set up in the wire, and will continue as long as such change is in progress. Thus, in Fig. 306, on the right, if the loop of wire should be moved from a strong to a weak field, as shown, then the amount of force passing through the circuit would decrease and a current would be set up, as shown, and lasting as long as the change was in progress. If, further, the coil should move on from the weak to a strong field, there being no change in the direction of the lines of force, then a current would be developed in the opposite direction to that developed by the movement from strong to weak. If, however, at the same time the direction of the lines of force should change, then the direction of the current would remain unchanged, as shown in the figure. If also, as shown on the left, the loop were to be turned sideways, so that a smaller amount of force could pass through, then also a current would be set up and would last as long as the change was in progress.

In all such cases the direction in which the current will flow may be determined by the following rule: If we look at the loop along the lines of force in the direction in which a free north magnetic pole would tend to move, and the change in the amount of force passing through the loop is a decrease, then the current will flow in the right-hand direction, or with the hands of a watch. If the change of force is an increase, the current will of course flow in the opposite direction.

The generation of electricity in all forms of electric gener-

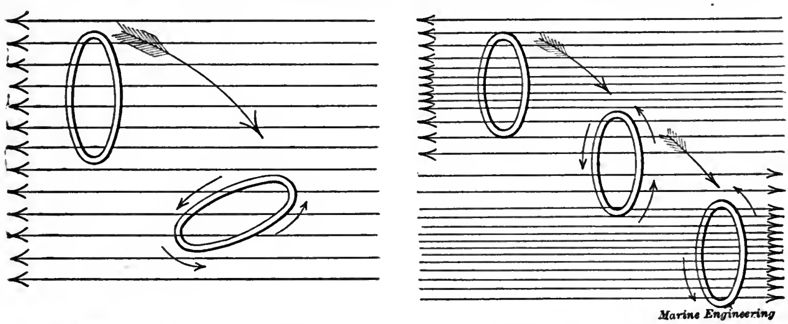


Fig. 306. Development of Electro-Motive Force by Moving a Coil of Wire in a Magnetic Field.

ators, and its use in all forms of motors, depends fundamentally upon the few principles explained above.

Electro-Motive Force.—The force in answer to which the electric current is set up and maintained is known as the *electro-motive force.* Relative to the electric current this force plays a part quite similar to that played by steam pressure, or a difference of steam pressures, in causing a flow of steam from one place to another. In fact, the term electric pressure, or difference of electric pressures, is now quite commonly used by engineers instead of electro-motive force. The phrase electro-motive force is commonly abbreviated to E.M.F. The unit of E.M.F. is known as the *volt.*

Resistance.—The electric current, in flowing around the circuit, meets with a certain resistance, and it is in fact this resistance which the E.M.F. must constantly overcome. This resistance to the flow of the electric current may be likened to the resistance of a pipe to the flow of water within it.

Electric resistance depends on the material of the circuit, on its length, and on its cross-sectional area. The greater the length the greater the resistance, and the greater the cross-sectional area the less the resistance, both in direct proportion. Substances are usually divided into conductors and non-conductors. There is no substance so good a conductor, however, but that it opposes some resistance to the passage of a current, and there is no non-conductor so perfect that it will not allow the passage of minute currents, especially under very high pressures.

The unit of resistance is called the *Ohm*, and is defined as the resistance which is opposed to the passage of an electric current by a tube of pure mercury of a certain size and length.* The result, then, of the action of electro-motive-force against electric resistance is to produce an electric current, and the unit of current thus produced is called the *Ampere*. This is defined as the current which will be produced by an E.M.F. of one Volt acting through a resistance of one Ohm. It signifies a certain rate of flow, just as if we should give a special name to a flow of say 20 gallons of water per minute through a pipe.

Electric Power is represented by the product of E.M.F. and rate of flow, just as in the case of the flow of water in a pipe it is represented by the product of the pressure per square foot and the rate of flow expressed in cubic feet.

The unit of electric power is called the *Watt*, and is the power required to maintain for a minute a flow of one *ampere* under a pressure of one *volt*. Similarly a current of 8 ampères under a pressure of 50 volts will require 400 watts.

To connect electric with mechanical power it is found that in round numbers 746 watts equals one horse power. This must not be taken to mean that one indicated horse power in a steam engine will give 746 watts in a generator. There are losses between the two due to the fact that the electric generator cannot transform all the mechanical energy it receives into electric energy, and thus wastes a certain amount, which appears chiefly as heat. The number 746 gives the ratio which would exist if there were no losses of any description whatever.

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* Cross section of one millimeter square or about 1/25 inch, and length of 1063. millimeters or about 42. inches.

The ordinary commercial unit of electric power is the kilowatt, or 1,000 watts. It is thus equivalent to about 1 1-3 horse power of mechanical work.

Ohms Law.—We will now state the fundamental law which gives the relation between the current, the E.M.F. and the resistance. It is simply that the current in ampères equals the E.M.F in volts divided by the resistance in Ohms, or in symbols

$$C = E \div R \text{ or } E = CR.$$

Likewise, if we denote the electric power by P we have:

$$P = CE = C^2R$$

This measures the power which is required to maintain the current in the circuit. If no other effects are produced it is wholly expended in heating the circuit, and the heat thus developed will be measured by the power multiplied by the time or in symbols

$$H = C^2Rt.$$

Now, without further discussion of the principles of electrical engineering, we will proceed to a brief description of the apparatus more commonly met with on shipboard.

Sec. 94. THE DYNAMO.

We may consider the dynamo simply as an apparatus designed for the development of electric current, or for the transformation of mechanical into electrical energy. It consists essentially of two electro magnets, which we may call A and B. A is stationary, and between its poles B rotates, carrying its coils of wire through the field produced by A. The rotating magnet and its coils is known as the *armature*, and the fixed magnet and coils as the *poles* and *field coils*. In Fig. 307 is shown one of the rotating coils A of the armature, while its successive positions are shown by 1, 2, 3, 4. The field of force due to the field coils and poles is also indicated by the arrows. In accordance with the principles above stated an E.M.F. is thus generated in the armature coils. It may be likewise seen that the E.M.F. will act alternately in opposite directions in these coils, changing as their plane is about midway between the poles, or in the position shown at A. All these elementary forces thus generated in the separate coils of an actual armature are gathered together and produce the full pressure at the terminals.

Electric generators are of two chief varieties, *direct* and *alternating* current. In the latter the E.M.F. generated is allowed to produce the current in the circuit back and forth, alternating in direction as the coils pass between the poles. The current may thus be likened to a series of surges to and fro, but without continuous flow in either direction. In the former, or direct current machine, these impulses are so taken up by the commutator that they are all adjusted or turned in one direction, and we have, therefore, a continuous flow rather than a series of surges back and forth.

It is found by experience that either form may be used for the development of light by either the incandescent or arc system, while also either, by the use of appropriate motors, may be used for the development of mechanical power.

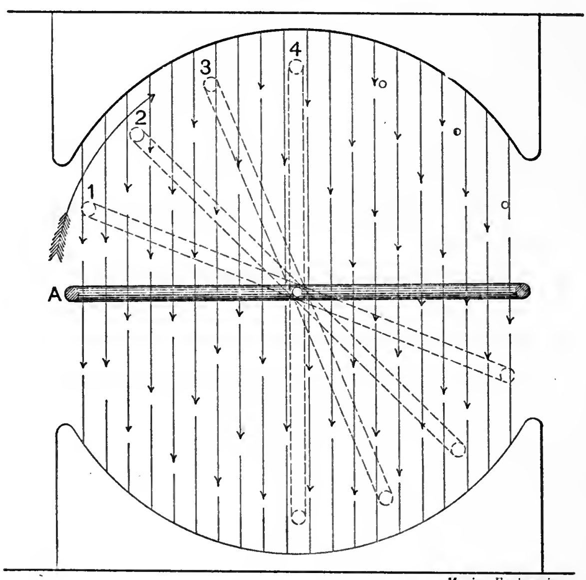

Fig. 307. Action of Single Coil of an Electric Generator.

For use on board ship the direct current form is usually employed. A discussion of the relative advantages and disadvantages of the two forms is beyond our present limits, but it may be said in a word that for lighting the limited space of a single ship the alternate current system does not possess the advantages which it may in the case of distribution over a wider area, and for the operation of motors its use would entail some disadvantages and complexity from which the direct current system is free.

Restricting our attention therefore to direct current machinery, the simplest is found in the so-called series dynamo. In this type the circuit as a whole leads continuously around both the armature and the field coils, thence into the external circuit at one terminal, and around back to the machine at the other terminal. If instead of this arrangement the wires are lead as in Fig. 308, the current leaving the armature is divided, and one part flows around the field coils while the other goes through the external circuit. This constitutes the so-called shunt-wound dynamo. In such case the field coils consist of many turns of fine wire, while in a series machine they consist of a few turns of larger wire.

A combination of the series and shunt-windings constitutes the so-called compound wound machine. For the purpose of

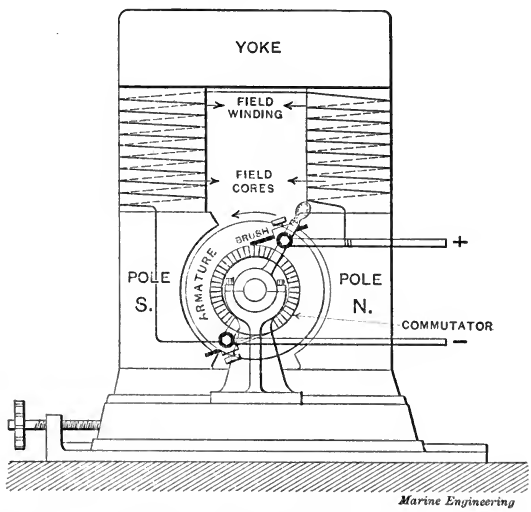

Fig. 308. Shunt-Wound Electric Generator, Outline Diagram.

delivering a constant potential or electric pressure the shunt-wound machine is usually employed, and it is this type which is commonly met with on shipboard.

We will now pass immediately to a brief description of a modern electric generator applicable to marine purposes.

According to size the typical marine generator is made with four or six poles, distributed uniformly around the circumference of the armature, which is of the ring type, as shown in Fig. 309. The machine is shunt wound according to the diagram of Fig. 308. The commutator, as shown in Fig. 310, consists of a series of bars of special bronze or copper formed into a cylinder or drum, and with mica or other insulating material between them. Each bar of the commutator is joined to the winding on the armature, and the connections are such that the E.M.F. generated in the

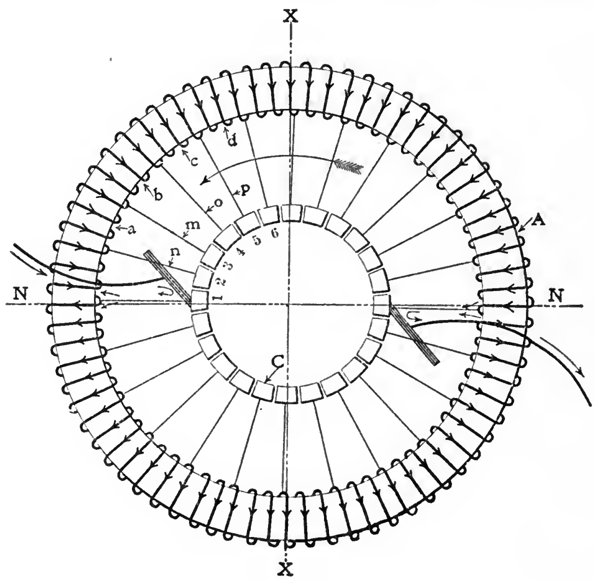

Fig. 309. Ring Armature.

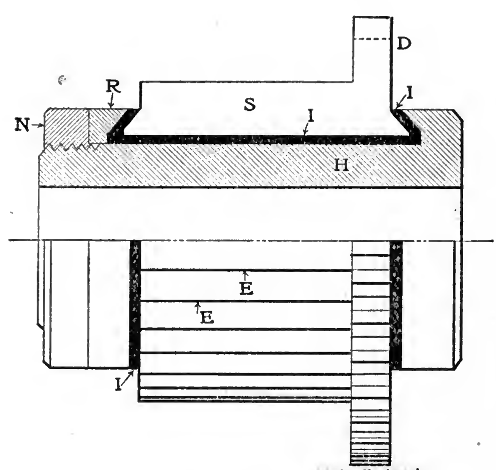

Fig. 310. Commutator.

entire series of coils is so directed as always to urge the current out through one of the brushes and in through the other. The brushes are usually in the form of a block of carbon, and are carried in metal holders, as shown in Fig. 311, and provided with means for holding them by adjustable spring pressure up to their contact with the commutator, while at the same time the frame which carries them may be rotated as a whole about the axis of the machine, thus bringing them to bear on different parts of the commutator for purposes of adjustment with varying load. With a four-pole machine the brushes are usually placed one nearly on top and one at about right angles. This makes them more accessible than if placed at opposite sides of the commutator, as is necessary in two-pole machines.

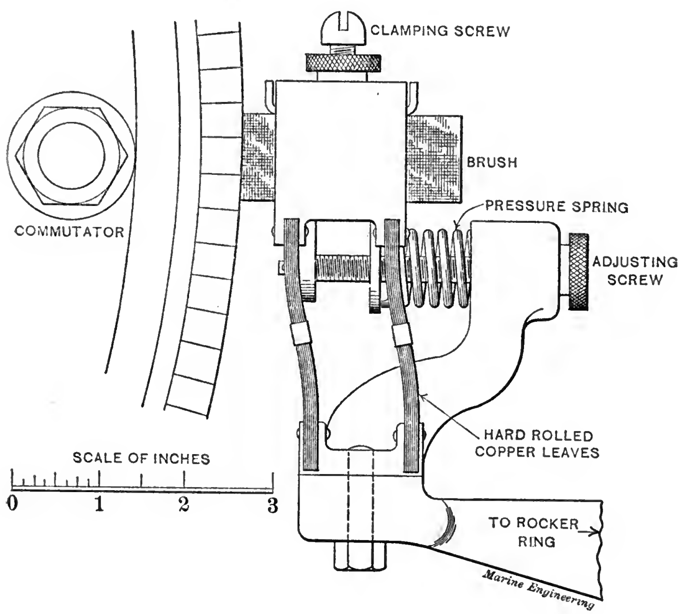

Fig. 311. Brush Holder.

In the selection of such a generator special attention should be paid to points of mechanical excellence, especially in connection with the bearings and balancing of the armature, the construction of the brush carriers, the binding posts, and other like points entering into the construction of the generator as a machine. The capacity of the generator should be such that it will be able at its normal rating to supply the expected demand for light and power. Generators are expected to be able to stand a certain amount of over load, and if of proper design and in a cool and well ventilated room will run safely at far above their rated power. It must be remembered, however, that the dynamo room of a ship is rarely cool or well ventilated, and that the overheating of armature and field coils is the chief limitation on the capacity of an electric generator. Hence in the unfavorable situation in which such machines are usually installed, they should not be expected to carry any considerable over load.

The typical marine generating set consists of a generator as above, together with engine set on one base and coupled direct together. The usual number of revolutions at which such sets are operated ranges from perhaps 300 to 500, according to size. The engine may be either a simple or compound, or even a triple-expansion, though the latter are but rarely employed in marine practice. In the design and construction of such an engine the chief points to be held in view are solidity and strength, especially in the running parts, but without undue excess of weight, generous bearing surfaces and provision for continuous and sure supply of lubricant.

With such small high-speed engines the presence of water in the cylinder is likely to produce serious damage, and especial care should be taken in regard to the relief and drainage valves, both hand and automatic systems being preferably fitted.

The governor provided with such engines is of the shaft automatic type, and should control the revolutions within about 2 per cent. for change from full load to no load, or *vice versa*.

In the care and operation of such engines no points are involved which have not already been discussed in connection with other engines, and it will not therefore be necessary to further consider these topics.

MOTORS.

Electric motors are coming into use on shipboard for a variety of purposes connected with the application of auxiliary power, but chiefly for running hoists and ventilating fans. It will be sufficient for our present purposes to note that a motor is essentially the same as a generator, and only differs in the manner of its use. With the latter we supply mechanical energy and withdraw electric energy. With the former we supply electric energy and withdraw mechanical energy. The motor contains the same parts as a generator, and similarly related, and differs only in being thus operated in the inverse manner to the generator.

Sec. 95. WIRING AND THE DISTRIBUTION OF LIGHT AND POWER.

The conductors used for the distribution of electricity may be either of single copper wire, varying in size according to the current to be carried, or of several small wires made up into a cable and of corresponding capacity. The insulation must be of extra good quality in order to stand the severe conditions to which it may be subjected on shipboard.

The materials available for the insulation of wire are rubber, either gum or vulcanized, gutta percha, and various special compounds, together with braided or wrapped coverings of cotton, linen or silk. Gutta percha is rarely used except for submarine cables and for other special purposes. It must be understood that none of these substances singly and no combination of them is a perfect non-conductor. There is always a small leakage through the insulation and air, and the purpose of the covering is to reduce this leakage to a safely negligible amount. For marine work it is especially necessary to protect the insulation against the effects of dampness and corrosion, as under these influences deterioration may set in, causing a breaking down of the resistance to the passage of the current and an increase in the leakage.

The usual form of marine lighting wire is insulated with vulcanized rubber or other special compound, covered with braided cotton, the whole covered with a coating of waterproof and preservative material. For switchboards and mountings for fixtures and all attachments, slate, marble and porcelain are used.

The conductors are run either in iron pipe or in double wooden mouldings. The former or the conduit system is to be preferred, as the conductors are thereby given the protection of the iron pipe. The conduit system is, however, much more expensive than the moulding system, and the latter is therefore more commonly employed, except in warship work and where high first cost is not an objection.

The chief point in either system is to give to the wires and to the various junctions and connections the greatest possible protection from mechanical injury and from the action of dampness and water, and all parts of the distributing system exposed to the weather or to the sea should be made as nearly water-tight as the limitations of first cost will permit.

The method to be adopted for the distribution of electricity depends somewhat on the purpose for which it is to be used.

The most common use on board ship is for operating incandescent lamps, where as nearly as may be the same pressure is required at each lamp. For this purpose the so-called distribution in parallel is required. . This is illustrated by the diagram of Fig. 312, where G is the generator and *ll* the lamps. It is seen that each lamp has its own bridge across from the two mains LL, and may thus be turned on or off independent of the operation of the others.

In the early days of electric installations on shipboard the iron or steel hull was frequently made a part of the circuit, or, as it was sometimes stated, the return was made through the hull. To use the hull in this way it was simply necessary to connect one

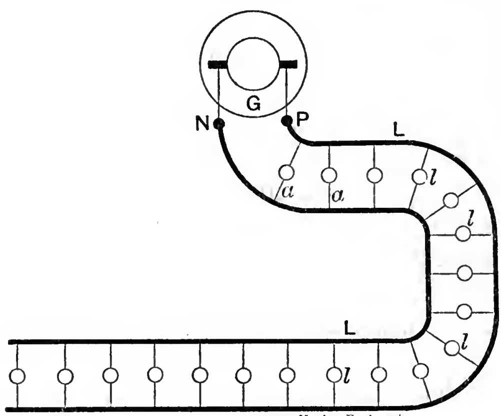

Fig. 312. Distribution in Parallel.

pole of the generator to the ship, and then connect one lead from each lamp to the hull likewise.

Great difficulty, however, was experienced in making and keeping good contact between the conductors and the ship. Corrosion was especially active at these points, and they were found to be the source of constant trouble. The presence of stray current returning through the ship was also believed to be a factor in the corrosion and deterioration of parts of the ship's machinery.

This system was therefore open to grave objections, and is now rarely met with. The modern practice is to employ a complete wire conductor for the external circuit, and to keep the current insulated entirely from the hull, as indicated in Fig. 312.

Arc lamps are very commonly operated in series, as shown in Fig. 313, each lamp thus receiving the same current which flows around the entire circuit. On board ship the chief or only use of arc lamps is for the searchlight, and the number therefore will usually not be greater than one or two, except for warships, where a larger number may be employed.

We may also need current supplied at various special points for the operation of motors for running ventilators, hoists, etc.

The entire distribution may therefore comprise various circuits according to the use to be made of the current and the point at which it is to be delivered. It is furthermore found desirable to split up the incandescent lighting distribution into a series of circuits, each of which is led from a main distributing point. The entire distribution will thus comprise several main circuits, all of which, however, will be led from the central point.

This distribution point is actually furnished by the *switch-*

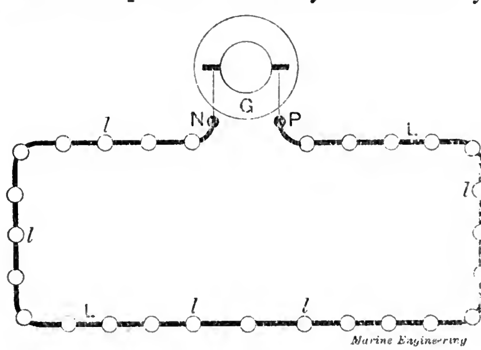

Fig. 313. Distribution in Series.

board, a slate or marble slab, on which are mounted the various instruments, switches, circuit breakers and connections needful for safely carrying out the distribution in the various ways in which it may be required.

We will now refer briefly to the more important items usually grouped on the switchboard:

Switch or *Cut Out*.—The object of a switch or cut out is to break the circuit and thus stop the flow of current. With a single pole switch the circuit is broken at one point only. This is the usual form fitted in the sockets of electric lamps, etc. With a double pole switch the circuit is broken at two points. The switches mounted on the switchboard are usually of this type, and are to be preferred, since by this means the line is entirely cut off from connection with the generator.

Fuses.—A fuse is a short piece of fusible alloy, so adapted to the proper current in the circuit that any marked increase in current strength will cause the fuse to melt, due to the increase in the heating effect. The result of this will be a break in the circuit and an interruption of the current.

This is therefore an automatic device for preventing the rise of the current strength beyond a certain value. In the same manner as with switches, we may have single pole or double pole fuses. The latter are to be preferred, as they give a double chance of breaking the circuit, and hence a greater relative margin of safety.

Circuit Breakers.—It is found that the ordinary fuse does not satisfactorily provide protection against momentary variations of current strength due to excessively rapid fluctuations in the generator or in the outer circuit. To prevent possible damage from such cause a magnetic circuit breaker is often employed. This consists of an electro-magnet, through the coil of which the main current flows. Any sudden increase of current will correspondingly increase the magnet strength, and the armature is so adjusted that in answer to this excess of pull it will move toward the magnet poles, thus breaking the circuit by the motion produced, and interrupting the current as desired.

Ammeter.—This is an instrument through which the current is passed in order to measure or indicate its strength. The indication is shown by a finger moving over a dial, graduated usually so as to read the current in ampères. The Ammeter must be connected in series with the machine and main circuit.

Volt Meter.—This is an instrument which measures or indicates the E.M.F. or difference of electrical pressure between the two points to which its wires are attached. It serves a similar purpose as the steam gauge for the boiler. The indication is shown by a finger moving over a dial graduated usually so as to read the pressure in volts. It is usually connected with one wire to each pole of the generator so as to give the entire pressure in the external circuit.

Rheostat.—This is simply an arrangement for varying the total resistance of the external circuit, and thus of varying the pressure available for that part beyond the instrument. It consists essentially of a series of coils of wire, more or less of which can be brought in and made a part of the circuit by moving a handle across a series of contact points.

Rheostats are often required in connection with the use of arc lights, or with the governing and control of shunt-wound generators and motors.

The complete installation of instruments, switches, connections, etc., thus brought together on the switchboard has as its object the possibility of shifting the load from one generator to another, of connecting the various circuits singly or in different combinations with any one or any combination of generators, and at the same time providing for safety and for the proper measurement of the pressure and current.

Sec. 96. LAMPS.

Electricity is used on shipboard for operating incandescent or glow lamps for general lighting, for operating arc lamps for search-lights, and for operating motors.

Incandescent Lamps.—The usual style of incandescent lamp consists of a filament of carbon in a glass bulb exhausted of its air and sealed air-tight. The carbon filament is connected to metal terminals, through which connection with the external circuit is made. The resistance of the carbon filament is very high and the passage of the current gives therefore a heating effect, which is sufficient to raise it to the luminous point. If the air were not exhausted from the bulb the carbon would be immediately burned up or converted into carbon dioxide by union with the oxygen. Since, however, the oxygen is almost entirely removed such combustion is limited to the smallest fraction of the filament when the current is first turned on, and it then remains nearly unchanged for a period of time ranging from 600 to 1,000 hours. Gradually the filament seems to disintegrate or lose its strength, and finally fails by snapping in two, or the bulb becomes blackened by a deposit of carbon on its inner surface. The usual lamp for ship fitting is one of 16 candle power, requiring about 110 volts pressure, and taking about ½ ampere of current. It thus follows that such a lamp will require about 55 watts of electrical energy. To supply this will require the equivalent of not far from 75 at the steam engine, or about 1-10 H.P. It follows, then, that we may expect to operate roughly about ten such lamps per I. H. P. at the engine. More powerful lamps, such as 32 or 50 candle power will require more current, and hence more energy and more horse power in proportion.

Arc Lamps.—The simplest form of arc lamp is shown in

Fig. 314. It consists of a pair of carbon rods, separated by a slight gap as indicated. These carbons form part of an electric circuit, which is completed by the leads to the generator, as shown. To start the lamp the rods must be brought together, thus completing the circuit and permitting the current to flow. The instant the current is set up, however, the rods are separated a slight distance. The space between the two is then filled with hot air and carbon dioxide, and across this the current is able to pass. The resistance is so great, however, that intense heat is developed, and this brings the gases and the particles of

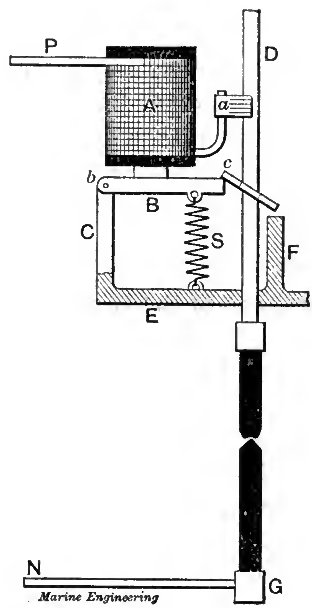

Fig. 314. Arc Lamp, Simple Form.

carbon which are torn off and hurled across the gap all to a state of incandescence. As the lamp burns, the carbons are slowly consumed and the gap thus widens. There is always some width of gap for which the lamp operates best, and a constant adjustment must therefore be made. It is found further that the two carbons do not wear equally or similarly.

Referring to Fig. 314, the current is supposed to flow through the lamp downward, and the upper or positive carbon will then wear in a crater-like form, while the lower or negative carbon will take a more rounded or pointed form, as shown. The positive end will also wear away about twice as fast as the negative end, and this will require a special form of adjustment in order to keep the gap at the same place in reference to a lense or mirror. It is also found that most of the light comes from the crater-like cavity, so that the carbons must be so mounted as to allow the maximum amount of light to escape from this source.

A lamp of this character, mounted and provided with suitable lenses and adjustments for controlling the carbons and for manipulating and turning the beam of light in any direction as desired, constitutes a search-light, as installed on board ship.

We shall not attempt here the description of the modes of automatic adjustment for the carbons or of the optical parts of the lamp, as the present purpose is to give only a general idea of the engineering side of the problem.

Such lamps require a pressure of from 50 to 60 volts and a current of from 50 to 150 amperes. This corresponds to from 4 to 10 horse power at the lamp, or say 5 to 12 at the engine.

Sec. 97. OPERATION AND CARE OF ELECTRICAL MACHINERY.

[1] Routine Care.

In looking over a generator for the first time the whole machine should be carefully examined, both as to its electrical and mechanical features. All the leads of the wires should be followed through and the binding posts and contacts examined to insure that they are in proper condition. The commutator and brushes should be carefully looked over to see if the segments of the former are in good condition, and if the surface is smooth and free from scores and uneven spots.

Carbon brushes are very commonly employed in modern marine practice, but whatever the form of brush the fit of its end on the commutator should be noted, and if necessary it should be refitted so as to bed suitably on the surface. The final adjustment of the brushes cannot be made until the machine is running, but the adjusting and holding devices will be carefully examined and the brushes may be placed in approximate adjustment, according to judgment. The bearings and journals and provision for oiling should then be examined, and carefully cleansed if the condition requires it. The armature should be turned by hand to make sure that there is the necessary clearance between it and the pole pieces, and that it has the proper freedom of motion.

These various points having been attended to and everything being in a satisfactory condition, the machine may be started. A shunt machine usually excites or builds up best when the external circuit is cut out, so that it operates simply through its own armature and field coil. As the voltage rises, as indicated on the voltmeter or pilot lamp, the external circuit may be switched in and the generator will then settle down to its work. The proper lead or adjustment of the brushes is the next thing to be attended to. This must be ascertained by trial until a position is finally found where the sparking is reduced to the smallest limits. If in modern machines the sparking at the brushes is in any degree pronounced, it may be taken as a safe indication that the brush adjustment is not quite as it should be. The exact adjustment will, however, vary as the load changes, and hence in case the load is changing rapidly there will be more sparking than with a steady load.

If excessive sparking occurs and cannot be controlled by the brush adjustment, the machine should be shut down at once, as this is an indication that it is out of electrical balance, and to continue running it would mean the possibility of burning out the armature or seriously injuring the commutator and brushes. In some cases such sparking may be due to a dirty commutator, and may be corrected by simply cleaning it. The pressure of the brushes on the commutator should also receive attention, as if too heavy undue wear will occur, while if too light they may jump and run irregularly, thus causing sparking.

A very small amount of lubricant on the commutator is usually found to aid in smooth running and in saving the surface from scoring. In amount it should be very small, a drop or two of hydrocarbon oil or a little vaseline or a rub of one of the standard preparations sold for the purpose is sufficient.

[2] Faults.

One of the most troublesome features connected with the operation of electric machinery is the possibility of the occurrence of more or less sudden failure at some point, resulting in the sudden extinction of a certain group of lights or the stoppage of a motor, or even the interruption of the entire plant in case the fault lies at the generator itself.

In all such cases the immediate cause of the trouble is the interruption of the current, either in whole or in part. This may be due to a variety of causes, as follows: (1) The resistance in the circuit may be enormously increased, thus cutting down the current strength proportionately. (2) The current may be diverted in some other direction, finding an easier path, and thus avoiding the circuits in which it belongs. (3) The generator itself may fail to develop the necessary E.M.F., and hence while the circuits may be in perfect condition the current will be weakened in corresponding degree.

The increase of resistance in a circuit may be due to poor contact at binding posts or junctions, arising from improper mechanical construction or fitting, or from the presence of oxide due to corrosion, or from the reduction in the cross section of a conductor due to corrosion, or to any like cause which decreases the cross section of the conductor or changes its electric conductivity. In case the conductor is broken or the separation is complete, the resistance across the air gap becomes practically infinite, and the current is completely interrupted.

The current may be more or less shunted or diverted from its proper path by the accidental establishment of other paths, and the consequent re-arrangement of the current distribution.

The generator may fail entirely to develop the E.M.F., if, for example, the field coil circuit should become broken and the cores thus lose their magnetism.

In locating a fault the circuit in which it occurs must first be determined, and then the various points at which it might exist must be examined, one after another. The circuit in which the fault is located can usually be inferred from the extent of the disturbance. If only a single light goes out, it is evidently limited to the circuit belonging that light alone, and may be looked for in the lamp itself or in the fuse block, if it has one. If all the lights on a single circuit go out, but no others, the trouble is of course limited to that circuit, and may probably be found at the junctions with the main feeders. If, on the contrary, the lights in all of the circuits go out or fall off in candle power, the trouble is in the main circuit and must be sought for at the switchboard or generator itself. At the switchboard the cut outs and fuses must be examined, and if the trouble is here it will be soon located. At the generator the brushes should be examined to make sure that they run with proper contact on the commutator. The contact between the brush and holder and all binding posts and contacts about the machine should also be examined, in order to make sure that the trouble is not in a poor contact at these points. If nothing is found here, it is probable that the fault lies in the armature or field coils, and that possibly they have become burned out or unsoldered or otherwise disconnected at some point.

Leakage faults may occur where the insulation is partially or wholly destroyed, and the result of such leakage will be a loss of effect in the lighting and power circuits. Thus, for example, the insulation between the brush carriers and the base of the machine may become faulty by wear or by the accumulation of copper dust and oil and thus form a more or less ready path from one brush to the other through the base of the machine. In such case the current traversing both the field coils and external circuit will be cut down, and due to the former the strength of the magnetic poles will be diminished, and the E.M.F. developed will fall off correspondingly. In case these leakage contacts are good, the entire machine will shut down, due to failure of current in the magnet coils and consequent failure of the magnetic field. In order that such trouble may exist, both brushes must be thus connected to the base. If one only is connected, the machine is said to be grounded. No immediate trouble may occur, but if any accidental connection is set up between the external circuit and the structure of the ship the leakage circuit will be complete and trouble will develop immediately, its character and extent depending on what point of the external circuit is thus grounded with the machine.

The repair of a fault in the external circuit is often a simple matter of cleaning and readjustment. In the generator itself, however, the exact location and repair of a fault may require a complete dismantling, and perhaps a partial re-building of the machine.

Practical Marine Engineering

Part II

Practical Marine Engineering

PART II.

COMPUTATIONS FOR ENGINEERS.

Sec. 1. COMMON FRACTIONS.*

[1] Units of Measurement and Definitions.

One of the principal duties of the engineer is to measure things. Thus he may be called on to find the length of a section of shafting, the diameter of a piston-rod, the weight of a screw-propeller, the volume of an oil-tank, or the capacity of a coal-bunker. Measuring consists in nothing more or less than comparing the quantity to be measured with another quantity of the same kind, and so finding how many times the latter is contained in the former. Thus in measuring a length of shaft with a foot rule we really compare the length of the shaft with the length of the rule, and so find, for example, that the shaft is 14 feet in length; that is, that it is 14 times as long as the rule, or that its length contains the length of the rule 14 times. All these are simply different ways of saying the same thing. Now the foot rule or the foot in such an operation is called the *unit*.

Again, in measuring the weight, say, of a screw propeller, the operation really amounts to making a comparison between the weight of the propeller and the standard weight called the *pound*. Here likewise the pound is the *unit*.

It is easily seen that the unit must be the same kind of quantity as the thing to be measured, else no direct comparison can be made between them: thus a unit length to measure length, a unit area to measure surface, a unit volume to measure volume, a unit weight to measure weight, etc.

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*The reader is supposed to be somewhat familiar with the general subject of fractions as presented in the elementary text books of arithmetic. The present section is not intended as a complete discussion of the subject, but rather as a short compendium of the more important operations, based on a point of view somewhat different from that given in the usual text books.

Now if we wish to measure the length of our section of shafting with some degree of accuracy we shall probably find that it is not an exact number of feet. We may perhaps find its length between 14 and 15 feet. We then proceed to measure the inches and find, let us say, 14 feet 5 inches, with an additional length less than one inch. With still greater accuracy we might go on and find perhaps 14 feet $5\frac{7}{16}$ inches as the length correct to a sixteenth. Now let us set down this length as follows: 14 feet, 5 inches, 7 sixteenths inch. It is easily seen that if the *foot* is a unit of length so is the *inch*, and so is the *sixteenth-inch*. Here then in measuring a piece of shaft we have used three kinds of units, the foot for most of the length, the inch for most of the remainder, and the sixteenth-inch for the little that finally remained. We know very well that the sixteenth-inch is obtained by simply dividing the inch up into sixteen equal parts and taking one of them, while the inch is gotten similarly by dividing the foot into twelve equal parts and taking one of them. It thus appears that these units are all related together; the sixteenth comes from the inch, and the inch from the foot.

A unit which is thus obtained by taking one of the equal parts into which a larger or principal unit may be divided is often called a *fractional unit*. Thus the inch is a fractional unit relative to the foot; while the half-inch, quarter-inch, eighth-inch, etc., are all fractional units relative to the inch. In general we can always suppose any given quantity considered as a principal unit, to be divided up into any number of equal parts, as 3, 8, 16, 100, 144, 2196, etc., and we can then take one of these parts as a fractional unit and with it proceed to measure any given amount of the same kind of quantity.

Now when we measure quantities with fractional units the result is known as a *fraction*. Thus the fraction five-twelfths ($\frac{5}{12}$) is the measure of a quantity in terms of the unit *one-twelfth*, just as five inches is the measure of a quantity in terms of the unit *one inch*, and in each case the measure is simply five such units. Or to put the same thing the other way around, if we measure a quantity in terms of a fractional unit such as one-twelfth ($\frac{1}{12}$), for example, and find that the measure is five such units, then we write the result $\frac{5}{12}$. and such an expression is known as a *fraction*. For the engineer it is always most natural to bear in mind this idea of a fraction, and to consider that it is simply the measure of a certain quantity in terms of a fractional unit.

When we are handling fractions simply for exercise in arithmetic we do not always stop to ask what kind of a unit it is, or what kind of a quantity we are dealing with. For an exercise in arithmetic it makes no difference, but the engineer in actual problems always knows what he is dealing with, and what kind of a unit is meant.

In the usual way of writing fractions, as $\frac{5}{12}$, $\frac{7}{16}$, $\frac{3}{10}$, $\frac{23}{144}$, etc., the number below the line is called the *denominator* and shows into how many equal parts the larger or principal unit is divided in order to furnish the smaller or fractional unit. The denominator thus shows the relation of the fractional unit to the principal unit. The number above the line is called the *numerator* and shows how many of these fractional units are used to measure the quantity in question.

PROPER FRACTION. In a proper fraction such as $\frac{3}{4}$ the numerator is less than the denominator, showing that the quantity measured is less than the principal unit.

IMPROPER FRACTION. In an improper fraction as $\frac{17}{12}$, the numerator is greater than the denominator, showing that the quantity measured is greater than the principal unit.

MIXED NUMBER, OR WHOLE NUMBER AND FRACTION. Such an expression means that the quantity is measured in terms of two units. Thus $7\frac{5}{12}$ means seven principal units and five fractional units, the latter unit being one-twelfth the former. This is exactly similar to the measurement of length in feet and inches, or weight in pounds and ounces. Thus if the foot is the principal unit, $7\frac{5}{12}$ means simply 7 feet and 5 inches, or if the pound is the principal unit, $8\frac{9}{16}$ means 8 pounds and 9 ounces. We may also recall the illustration used near the beginning of this section where three units were used to find the length of a piece of shafting.

[2] Reduction of a Mixed Number to an Improper Fraction.

This means simply the reduction of the measure all to terms of the smaller or fractional unit, just as we may reduce a measure in feet and inches all to inches. Thus to reduce $7\frac{5}{12}$ to an improper fraction we see that in each of the seven principal units there are 12 fractional units, and hence 7×12 or 84 such units in the whole number. In addition, there are 5 more fractional units, and therefore $84 + 5$ or 89 in all. The reduced value is therefore $\frac{89}{12}$. In a similar way if the principal unit were the foot we should reduce 7 feet and 5 inches to inches by multiplying the 7 by 12 and adding in the 5, giving 89 inches similar to the 89 twelfths ($\frac{89}{12}$) above. We have therefore the following:

Rule.—Multiply the whole number by the denominator and add in the numerator. The result is the numerator of the improper fraction, and the denominator is the same as before.

Problems—Reduce to improper fractions the following:

$$2\tfrac{5}{6},\ 8\tfrac{4}{7},\ 12\tfrac{7}{16},\ 8\tfrac{11}{12},\ 12\tfrac{13}{16},\ 17\tfrac{23}{45},\ 264\tfrac{123}{144}.$$

[3] Reduction of an Improper Fraction to a Mixed Number.

To reduce an improper fraction such as $\frac{31}{12}$ to a mixed number we must evidently find first how many principal units there are. Since 12 fractional units make 1 principal unit, it is evident that the number of principal units will be found by dividing 31 by 12. The quotient will then give the number of principal units and the remainder will give the remaining number of fractional units. Thus $31 \div 12 = 2$ principal units and 7 remainder, or 7 fractional units over; or $\frac{31}{12} = 2\frac{7}{12}$. Hence the following:

Rule—To reduce an improper fraction to a mixed number, divide the denominator into the numerator and the quotient will be the whole number, while the remainder will be the numerator of the fraction, and the denominator will be as before.

Problems—Reduce to mixed numbers the following:

$$\tfrac{76}{13},\ \tfrac{198}{144},\ \tfrac{289}{101},\ \tfrac{56}{21},\ \tfrac{95}{76},\ \tfrac{143}{117},\ \tfrac{264}{84},\ \tfrac{764}{19}.$$

[4] Reduction of Fractions Without Change of Value.

If we change the size of a unit of measure we shall change the measure of the quantity in like proportion. Thus, for example, the number measuring the diameter of a bolt in sixteenths as a unit will be twice as great as if measured in eighths as a unit. Thus $\frac{6}{16}$ and $\frac{3}{8}$ represent the same quantity, one measure being in sixteenths and the other in eighths. It follows that we can multiply or divide both terms of a fraction (the numerator and denominator) by the same number without changing its value. Thus $\frac{2}{3}$, $\frac{4}{6}$, $\frac{6}{9}$, $\frac{10}{15}$, $\frac{14}{21}$, $\frac{16}{24}$, $\frac{20}{30}$, $\frac{42}{63}$, etc., all represent the same quantity measured in terms of different units, and it is seen that the $\frac{2}{3}$ may be changed into any of the other forms by multiplying both numerator and denominator by the same number, and similarly any of these latter forms may be reduced back to the $\frac{2}{3}$ by dividing both numerator and denominator by the same number.

$$\text{Thus } \frac{2 \times 7}{3 \times 7} = \tfrac{14}{21} \text{ and } \frac{16 \div 8}{24 \div 8} = \tfrac{2}{3}.$$

It may often be convenient to reduce a fraction to another of equal value but having some particular or specified denominator. To this end we divide the denominator desired by the denominator of the fraction, and multiply both terms of the fraction by the quotient. That is, we must multiply both numerator and denominator by some number which will produce the desired denominator. Thus to reduce $\frac{2}{3}$ to a fraction whose denominator is 42 we divide 42 by 3 and find 14. We then multiply both terms of $\frac{2}{3}$ by 14 and thus find $\frac{28}{42}$ as the fraction desired.

Lowest Terms. A fraction is said to be reduced to its lowest terms where there is no whole number which will divide both numerator and denominator without a remainder. Thus in the foregoing string of fractions $\frac{2}{3}$ is in its lowest terms while none of the others is. To reduce a fraction to its lowest terms we seek a factor which will divide both numerator and denominator and divide, continuing the operation until no further reduction can be made.

Example—Reduce $\frac{18}{24}$ to its lowest terms. We may first see that 2 will divide both terms without remainder. Dividing we have $\frac{9}{12}$ as a reduced value. We then see that 3 will again divide both terms, and thus find $\frac{3}{4}$ as the lowest reduction. We may also note at first that 6 will evenly divide both terms, and thus find $\frac{3}{4}$ by a single operation.

Problems—Reduce the following to their lowest terms:

$$\tfrac{72}{216}, \tfrac{350}{770}, \tfrac{116}{48}, \tfrac{864}{1728}, \tfrac{42}{81}, \tfrac{231}{1331}, \tfrac{56}{44}, \tfrac{72}{24}, \tfrac{19}{57}.$$

ADDITION, SUBTRACTION, AND MULTIPLICATION OF COMMON FRACTIONS.

Considering fractions as representing the measures of various quantities, we may be called upon to perform upon them the four fundamental operations of mathematics—addition, subtraction, multiplication, and division. These we will briefly consider in order.

[5] Addition of Common Fractions.

We cannot combine directly $\frac{2}{3}$ and $\frac{3}{4}$ into a single quantity any more than we can 2 feeet and 3 inches, or 6 miles and 8 feet. The reason is that the units are not the same in the two quantities which we seek to combine, and before the combination can be effected we must reduce the measures to the same unit in each. To this end we take advantage of the operations explained in [4] and reduce the fractions to a common denominator or common unit of measure. We naturally seek for this denominator as small a number as possible, and hence proceed according to the common rule for finding the L. C. M. (*least common multiple*) of the denominators. We then proceed to express the various fractions all with this L. C. M. as the common denominator by the method explained in [4]. We may then add the numerators and reduce the result as may be possible. This is the foundation for the usual rule, which may be expressed as follows:

Rule. (1) Find the L. C. M.* of all the denominators for a new denominator.

(2) Divide each denominator into this L. C. M. and multiply the corresponding numerator by the quotient for a new numerator.

(3) Add the new numerators thus found, and the result is the numerator of the sum desired.

(4) Write this numerator over the L. C. M. or common denominator, and reduce to the lowest terms.

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* For convenience in connection with these operations, we give as follows the rule for finding the least common multiple of a series of numbers—*i. e.*, the smallest number which will contain each without a remainder.

Rule.—Write the numbers in a line (as (1) below), and select any number (as 4 in this case) which will divide at least two of them without remainder. Divide and set down the quotients underneath, except where the division would not be exact, in which case bring down again the number itself (as shown in line (2) below). Proceed with this line the same as with the first, and so continue until no two numbers have a common divisor. Then multiply together all the numbers remaining on the last line, together with all the divisors, and the product will give the least common multiple desired.

Example.—Find the L. C. M. of 8, 36, 20, 6.

Operation:—

4)	8	36	20	6	....Line (1)
3)	2	9	5	6	....Line (2)
2)	2	3	5	2	....Line (3)
	1	3	5	1	....Line (4)

L. C. M. $=4\times3\times2\times3\times5=360$.—Ans.

Examples. Add $\frac{5}{18}$ and $\frac{1}{4}$.

The L. C. M. of 18 and 4 is 36. We then have:

$$\frac{5}{18} + \frac{1}{4} = \frac{10 + 9}{36} = \frac{19}{36}.$$

Add $\frac{3}{16}$, $\frac{2}{3}$ and $\frac{5}{6}$.

The L. C. M. of 16, 3 and 6 is 48. We then have:

$$\frac{3}{16} + \frac{2}{3} + \frac{5}{6} = \frac{9 + 32 + 40}{48} = \frac{81}{48} = \text{by reduction } \frac{27}{16}.$$

If there are but two fractions to be added and both have 1 for a numerator, a short rule for their addition is as follows: Write the sum of the denominators over their product and the fraction thus formed is the sum desired.

Thus $\frac{1}{8} + \frac{1}{6} = \frac{14}{48} = \frac{7}{24}$.

[6] Subtraction of Fractions.

This operation requires reduction to the same unit of measure in the same way and for the same reason as in the case of addition. We have hence the usual rule, which may be expressed as follows:

(1) Find the L. C. M. of the two denominators for a new denominator.

(2) Divide each denominator into this L. C. M. and multiply the corresponding numerator by the quotient for a new numerator.

(3) Subtract the new numerators, and the result will be the numerator of the difference desired.

(4) Write this numerator over the L. C. M. or common denominator, and reduce to lowest terms. Thus to subtract $\frac{1}{4}$ from $\frac{7}{18}$ we have as follows:

$$\frac{7}{18} - \frac{1}{4} = \frac{14 - 9}{36} = \frac{5}{36}.$$

If both numerators are 1 a short rule for the subtraction of the fractions is as follows: Write the difference of the denominators over their product and the fraction thus formed is the difference desired.

Thus $\frac{1}{2} - \frac{1}{5} = \frac{3}{10}$.

Problems in Addition and Subtraction of Fractions. Perform the following additions:

$(\frac{1}{2} + \frac{1}{3} + \frac{2}{7})$, $(\frac{3}{8} + \frac{2}{3} + \frac{1}{12})$, $(\frac{16}{9} + \frac{2}{3})$, $(\frac{3}{5} + \frac{17}{10})$, $(\frac{243}{16} + \frac{7}{24})$.

Perform the following subtractions:

$(\frac{3}{8} - \frac{1}{12})$, $(\frac{17}{16} - \frac{7}{24})$, $(\frac{814}{144} - \frac{13}{16})$, $(\frac{430}{864} - \frac{104}{288})$, $(\frac{17}{64} - \frac{1}{4}.)$

Note.—In the operation of multiplication and division we should always distinguish between the *operator* and the *subject* or *thing operated on.* Thus in 6 times 5, the number 6 is the operator and 5 is the subject. The latter is usually the measure of some quantity. The former is the sign of an operation to be performed, and this distinction, which is most important, must not be forgotten.

[7] Multiplication of Fractions.

We shall first consider the operator as a whole number and the subject as a fraction. Thus suppose that we wish to multiply $\frac{5}{12}$ by 6. The operation is exactly the same as if we wished to multiply 5 inches by 6. The result in the latter case is 30 inches and in the former it is 30 twelfths, or as we may write it: $\frac{30}{12}$, or by reduction $\frac{5}{2}$, or $2\frac{1}{2}$. This illustrates the familiar principle that to multiply a quantity we must multiply its measure, and since in a fraction the numerator is the measure, we multiply the numerator to multiply the fraction.

Furthermore, it is plain if we *multiply* the denominator of a fraction that we *decrease* the size of the fractional unit, and hence with the same numerator or same number of such units we *decrease* the value of the fraction in like proportion. Similarly if we *divide* the denominator we *increase* the size of the fractional unit, and hence with the same numerator or same number of such units we *increase* the value of the fraction in like proportion. Hence if we can divide the denominator of the fraction by the given multiplier and leave the numerator the same, it will have the same effect as multiplying the numerator and leaving the denominator the same. Thus $\frac{7}{12} \times 3 = \frac{7}{4}$.

This may also be seen by first multiplying the numerator and then reducing to lowest terms. Thus $\frac{7}{12} \times 3 = \frac{21}{12} =$ by reduction $\frac{7}{4}$. Hence we have the following rule for multiplying by a whole number:

Rule—Multiply the numerator of the fraction or divide the denominator of the fraction by the given number.

Problems—Perform the following multiplications:

$\frac{2}{3} \times 2$, $\frac{2}{3} \times 3$, $\frac{6}{11} \times 4$, $\frac{7}{12} \times 6$, $\frac{19}{14} \times 7$, $\frac{21}{36} \times 12$, $\frac{27}{432} \times 36$.

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[8] Divisions of Fractions.

As before, we first consider the divisor or *operator* as a whole number, and the dividend or subject as a fraction. Then we may remember that division is simply the inverse of multiplication, and that by inverting the procedure for the latter we shall effect the former. Thus to divide $\frac{18}{23}$ by 2 we divide 18 by 2 and have $\frac{9}{23}$ as the result. Or again we may multiply the denominator, thus dividing the value of the fractional unit and thus dividing the value of the fraction as explained in [7]. Thus $\frac{18}{23} \div 2 = \frac{18}{46}$. This being reduced to its lowest terms gives $\frac{9}{23}$ as before.

The operations on fractions involved in multiplication and division by whole numbers may be summarized as follows:

Multiplying the $\left\{\begin{matrix}\text{numerator}\\ \text{denominator}\end{matrix}\right\}$ $\left\{\begin{matrix}\text{multiplies}\\ \text{divides}\end{matrix}\right\}$ its value, and dividing the $\left\{\begin{matrix}\text{numerator}\\ \text{denominator}\end{matrix}\right\}$ $\left\{\begin{matrix}\text{divides}\\ \text{multiplies}\end{matrix}\right\}$ its value.

Problems—Perform the following divisions:

$\frac{6}{7} \div 4$, $\frac{2}{9} \div 3$, $\frac{6}{17} \div 2$, $\frac{21}{35} \div 7$, $\frac{16}{23} \div 4$, $\frac{15}{6} \div 3$, $\frac{231}{144} \div 11$.

[9] Multiplication and Division by Fractions.

In these operations the *operator* is expressed as a fraction, and the latter in this case is therefore not the measure of a quantity, but the sign of something to be performed. When the fraction is used as a multiplier it is simply a short-hand way of expressing two operations (1) a multiplication by the numerator and (2) a division by the denominator. Thus if $\frac{2}{3}$ is used as a multiplier it is simply a short-hand way of expressing a multiplication by 2 and a division by 3. Thus $8 \times \frac{2}{3}$ is another way of expressing $8 \times 2 \div 3$ or $8 \div 3 \times 2$.

Similarly and since division is exactly the inverse of multiplication, when a fraction is used as a divisor it is simply a short-hand way of expressing two operations: (1) a division by the numerator, and (2) a multiplication by the denominator. Thus if $\frac{2}{3}$ is used as a divisor it is simply a short-hand way of expressing a division by 2 and a multiplication by 3. Thus $8 \div \frac{2}{3}$ is another way of expressing $8 \div 2 \times 3$ or $8 \times 3 \div 2$.

When the thing operated on is also a fraction these principles work out as follows: $\frac{3}{4} \times \frac{2}{5}$ means that $\frac{3}{4}$ is to be multiplied by 2 and divided by 5. But we can multiply by multiplying the numerator, and we can divide by multiplying the denominator. Hence

$\frac{3}{4} \times \frac{2}{5} = \frac{3 \times 2}{4 \times 5} = \frac{6}{20} = \frac{3}{10}$. Hence for the multiplication of one fraction by another we have the usual rule as follows:

Rule—Multiply together the two numerators for a new numerator and the two denominators for a new denominator, and the fraction thus formed is the product desired.

Similarly for division, $\frac{3}{4} \div \frac{2}{5}$ means that $\frac{3}{4}$ is to be divided by 2 and multiplied by 5. But this is the same as the pair of operations expressed by using $\frac{5}{2}$ as a multiplier. Hence $\frac{3}{4} \div \frac{2}{5} = \frac{3}{4} \times \frac{5}{2} = \frac{15}{8}$.

Hence for the division of one fraction by another we have the usual rule as follows:

Rule—Invert the terms of the divisor and proceed as in multiplication.

This might naturally be expected by remembering the relation between multiplication and division, and that one is the exact inverse of the other.

For the multiplication of a series of fractions into each other these principles work out as follows: $\frac{1}{2} \times \frac{3}{5} \times \frac{2}{7} \times \frac{5}{9}$ means that $\frac{1}{2}$ is first considered as a subject and $\frac{3}{5}$ as an operator. Then the result of this is the subject and $\frac{2}{7}$ is the operator and so on. The final result will be therefore $\frac{1 \times 3 \times 2 \times 5}{2 \times 5 \times 7 \times 9}$ $= \frac{30}{630}$. In this result we have a numerator 30 whose factors are the numerators of the individual fractions, while similarly the denominator 630 has for its factors the individual denominators. Applying here the principles of [4] we may cross out from numerator and denominator any pair of common factors. This will shorten the operation and give the result in its lowest terms. Thus shortened the above case becomes:

$$\frac{1 \times \cancel{3} \times \cancel{2} \times \cancel{5}}{\cancel{2} \times \cancel{5} \times 7 \times \underset{3}{\cancel{9}}} = \frac{1}{21}$$

The propriety of striking out a common factor in both numerator and denominator may also be seen by remembering that such a pair of factors denote, one a multiplication and the other a division by the same number. These operations will offset each other and may therefore be omitted entirely.

Cancellation. This striking out of common factors from both terms of a fraction is known as *cancellation*, and is often of great value in simplifying an operation before proceeding with the actual multiplication and division. The following are additional illustrations.

$$\frac{14}{33} \times \frac{16}{42} \times \frac{21}{8} \times \frac{22}{18} = \frac{\cancel{14} \times \overset{2}{\cancel{16}} \times \overset{7}{\cancel{21}} \times \overset{2}{\cancel{22}}}{\underset{3}{\cancel{33}} \times \underset{3}{\cancel{42}} \times \cancel{8} \times \underset{9}{\cancel{18}}} = \frac{14}{27}$$

$$\frac{4}{81} \times \frac{15}{16} \times \frac{16}{60} \times \frac{27}{2} = \frac{\overset{1}{\cancel{4}} \times \cancel{15} \times \cancel{16} \times \cancel{27}}{\underset{3}{\cancel{81}} \times \cancel{16} \times \underset{4}{\cancel{60}} \times 2} = \frac{1}{6}$$

We often meet with expressions like $\frac{2 \times 3 \times 16}{9 \times 8}$ in which the number of factors in numerator and denominator is not the same. All such expressions represent a series of multiplications of whole numbers and fractions as $\frac{2}{9} \times \frac{3}{8} \times 16$, or they may be considered as denoting a series of operations of multiplication and division, and in either case it follows that their reduction may be effected by cancellation as just described. Still otherwise we may consider such expressions as consisting of a numerator and denominator each resolved into factors, and hence in ready condition for reduction by cancellation. Thus in the expression above we have:

$$\frac{2 \times \cancel{3} \times \overset{2}{\cancel{16}}}{\underset{3}{\cancel{9}} \times \cancel{8}} = \frac{4}{3}$$

Problems—Perform the following operations:

$(\frac{2}{3} \times \frac{6}{7})$, $(\frac{2}{5} \times \frac{7}{8})$, $(\frac{3}{7} \times \frac{12}{11})$, $(\frac{8}{27} \times \frac{3}{8} \times \frac{2}{3})$, $(\frac{6}{17} \times \frac{15}{8} \times \frac{1}{3})$, $(\frac{2}{3} \times \frac{4}{5} \times \frac{25}{6})$.

$(\frac{16}{3} \div \frac{4}{7})$, $(\frac{28}{9} \div \frac{7}{3})$, $(\frac{17}{144} \div \frac{13}{24})$, $(\frac{42}{65} \div \frac{7}{13})$ $(\frac{8}{23} \div \frac{7}{16})$, $(\frac{2}{3} \div \frac{17}{6})$.

[10] Complex Fractions.

In complex fractions either or both numerator and denominator may consist of fractions or mixed numbers.

Thus for example: $\frac{2\frac{1}{2}}{\frac{3}{5}}$, $\frac{\frac{1}{3}}{\frac{2}{7}}$, $\frac{\frac{2}{5}}{3\frac{1}{5}}$.

All such expressions may most conveniently be considered as ways of indicating operations, the numerator being the subject of the operation and the denominator being the operator expressed as a divisor.

Thus

$$\frac{2\frac{1}{2}}{\frac{3}{5}} = 2\tfrac{1}{2} \div \tfrac{3}{5} = \tfrac{5}{2} \div \tfrac{3}{5} = \tfrac{5}{2} \times \tfrac{5}{3} = \tfrac{25}{6}.$$

$$\frac{\frac{1}{3}}{\frac{2}{7}} = \tfrac{1}{3} \div \tfrac{2}{7} = \tfrac{1}{3} \times \tfrac{7}{2} = \tfrac{7}{6}.$$

$$\frac{\frac{2}{5}}{3\frac{1}{2}} = \tfrac{2}{5} \div 3\tfrac{1}{5} = \tfrac{2}{5} \div \tfrac{16}{5} = \tfrac{2}{5} \times \tfrac{5}{16} = \tfrac{1}{8}.$$

The reduction of such expressions becomes therefore simply a matter of the application of the rules and methods already given.

Problems—Reduce the following:

$$\frac{3\frac{2}{3}}{8\frac{4}{5}},\ \frac{\frac{1}{2} + 2\frac{1}{3}}{\frac{2}{3} - \frac{6}{5} \times \frac{1}{3}},\ \frac{\frac{4}{5} - \frac{3}{2} \div \frac{5}{2} + \frac{3}{10}}{2\frac{2}{5} \times \frac{5}{6} - 1\frac{3}{8}}.$$

Sec. 2. DECIMAL FRACTIONS.

[1] In decimal fractions the denominator is always 10, 100, 1000, etc., or some power of 10. Instead of writing them in the usual way, however, a device is made use of to indicate the denominator without actually writing it. This is simply to write the numerator and to place a *dot* at such a point that there shall be as many figures on the right as there are ciphers in the denominator, using ciphers to the left of the significant figures if necessary to make up the needed number of places. In writing a mixed number the whole number part stands on the left of the *dot or decimal point*, which thus becomes a point of separation between the whole number and the fraction.

Thus			
	$\frac{7}{10}$ is written		.7
	$\frac{7}{100}$ "	"	.07
	$\frac{7}{1000}$ "	"	.007
	$\frac{23}{10000}$ "	"	.0023
	$2\frac{7}{10}$ "	"	2.7
	$162\frac{43}{1000}$ "	"	162.043

In each case it is seen in the decimal that there are as many places on the right of the dot as there are ciphers in the denominator, and in this way the value of the denominator or unit of measure is indicated. There are no principles involved in decimal fractions different from those already discussed in common fractions, and the only difficulties in using them arise from the peculiar manner in which they are written. We will therefore give without further discussion the rules for the handling of decimals, all of which may be seen to follow from the principles already laid down.

[2] To Reduce Decimals to Lower Terms.

Evidently all ciphers standing on the right of the decimal may be struck off as they disappear from both numerator and denominator. Thus .3500 = .35.

[3] To Raise Decimals to Higher Terms.

Add ciphers to the right of the numerator as may be desired. Thus .35 = .350 = .3500.

[4] To Reduce a Decimal Fraction to a Common Fraction.

Write the numerator and denominator as in common fractions and reduce to lower terms if possible.

Example—Reduce .35 to a common fraction.

Operation: $.35 = \frac{35}{100} = \frac{7}{20}$.

[5] To Reduce a Common Fraction, Proper or Improper to a Decimal.

Take the numerator for the dividend and the denominator for the divisor and proceed as in division of whole numbers, adding ciphers to the right of the dividend as may be necessary. Then point off according to the following rule:

If the numerator or dividend is less than the denominator or divisor, the first figure of the quotient must stand on the right of the decimal point as many places as the number of ciphers added to the right of the dividend in order to enable it to contain the divisor. If the dividend is greater than the divisor place the point after the figure of the quotient given by bringing down the last figure of the dividend in the operation of division.

It should be noted that this gives the general method for dividing one whole number by another and expressing the result decimally.

Examples—Reduce $\frac{16}{250}$ to a decimal.

Operation:

```
250) 1600 (.064
     1500
     ----
     1000
     1000
     ----
```

Two ciphers are annexed in order to obtain the first figure 6 in the quotient. Hence this figure must stand in the second place to the right of the decimal point, and a cipher is added on the left of the 6 to bring it to this position.

Reduce $\frac{1647}{25}$ to a decimal.

Operation:

```
25) 1647 (65 88         220
    150                 200
    ---                 ---
     147                 200
     125                 200
     ---                 ---
```

The figure 5 of the quotient is given by bringing down the last figure 7 of the dividend, and hence the decimal point must come between this and the next figure of the quotient.

Problems—Reduce the following to decimals:

$\frac{2}{5}$, $\frac{12}{5}$, $\frac{122}{5}$, $\frac{18}{250}$, $\frac{17}{130}$, $\frac{18}{4}$, $\frac{143}{55}$, $\frac{1721}{1440}$, $\frac{18}{9600}$, $\frac{14}{1900}$, $\frac{23}{7}$.

[6] To Add Decimals.

Set down the decimals in a column so that the points shall all stand under each other. Then add as in whole numbers and bring down the decimal point in the sum under those standing above.

Example—Add .025, .64, .231, .4685, .003.

Operation:*

```
 .025
 .64
 .231
 .4685
 .003
 ------
1.3675
```

[7] To Subtract Decimals.

Set down the subtrahend under the minuend, the two decimal points one under the other, adding ciphers to the right if necessary to fill out to the same number of places. Then subtract as in whole numbers and bring down the decimal point under those above.

* If desired, the numbers may be filled out all to the same number of places, by adding ciphers on the right, such operation being, in fact, the reduction of the decimals all to the same denominator or unit of measure. In the summing, however, such ciphers play no part, and therefore the filling out is unnecessary in the actual operation.

Example—Subtract .263 from .83.
Operation:

$$\begin{array}{r} .830 \\ .263 \\ \hline .567 \end{array}$$

[8] To Multiply Together Two Numbers Expressed Decimally.

Multiply as in whole numbers and point off for the decimal portion as many places as there are in the two factors taken together.

Examples—

$$\begin{array}{rcrcr} .16 & \times & 24. & = & 3.84 \\ 72. & \times & .0004 & = & .0288 \\ .162 & \times & .041 & = & .006642 \\ 21.14 & \times & 13. & = & 274.82 \\ 21.14 & \times & .13 & = & 2.7482 \\ 21.14 & \times & 1.3 & = & 27.482 \end{array}$$

Problems—Perform the following multiplications:

$$\begin{array}{rcr} 2.72 & \times & 1.4 \\ 143.26 & \times & 24.2 \\ 12.16 & \times & .018 \\ 4214.3 & \times & 22.3 \\ .14 & \times & .21 \\ .06 & \times & .0084 \end{array}$$

[9] To Find the Quotient of Two Quantities Expressed Decimally.

Clear both dividend and divisor of decimals by moving the point to the right an equal number of places in each, adding ciphers as may be necessary. The number of places moved will be that necessary to clear the term of higher order. Thus modified, consider the two terms as whole numbers and proceed with the division, pointing off according to the rule given above for the reduction of a common fraction to a decimal.

Examples—

$$.16 \div 2.5 \text{ or } \frac{.16}{2.5} = \frac{16}{250} = .064$$

$$1.6 \div 2.5 \text{ or } \frac{1.6}{2.5} = \frac{16}{25} = .64$$

$$1.6 \div .25 \text{ or } \frac{1.6}{.25} = \frac{160}{25} = 6.4$$

$$16. \div .025 \text{ or } \frac{16}{.025} = \frac{16000}{25} = 640.$$

$$.016 \div 250 \text{ or } \frac{.016}{250} = \frac{16}{250000} = .000064$$

$$1.6 \div .243 \text{ or } \frac{1.6}{.243} = \frac{1600}{243} = 6.5761 +$$

Problems—Perform the following divisions:

$7.25 \div 16$
$72.54 \div 6.4$
$.00864 \div .14400$
$8.64 \div .0144$
$.96 \div .024$
$1.25 \div .0025$

Sec. 3. PERCENTAGE.

[1] In percentage, fractional reations are expressed decimally, but it is understood that the denominator shall always be 100. The fractional unit is therefore always one one-hundredth of the principal unit, and therefore occupies to it the same relation as that of the cent to the dollar. The words *per cent* are commonly represented by the symbol %, so that 16% is the same as 16 per cent or .16, while 5% equals similarly .05, etc. Care must be had not to fall into confusion with the use of both the decimal point and % mark. Thus .4% is not .4, but .4 of one per cent or .004. A number written in percentage is usually an operator, and not a quantity or measure by itself. Thus 16% is not a quantity by itself, but rather expresses the relation between two quantities, or represents an operation to be performed on one quantity in order to obtain another. The handling of percentages is the same as that of decimals, remembering that the term *per cent* is simply a special name for a fractional unit one one-hundredth of the principal unit, whatever the latter may be, and that it is with this fractional unit that all quantities are measured in percentage operations. Remembering the rules for decimals we readily see the following:

Examples—

$$16\% \text{ of } 80 = 80 \times .16 = 12.8$$
$$3\% \text{ of } 2.1 = 2.1 \times .03 = .063$$
$$23\% \text{ of } \$145.24 = \$145.24 \times .23 = \$33.4052, \text{ or } \$33.41.$$

From the above it follows that:

To reduce any number expressed in per cent to terms of decimals, we divide by 100 and express the result decimally, or shift the decimal point two places to the left; while to reduce any decimal to terms of per cent we multiply by 100 or shift the decimal point two places to the right.

Thus:

$$.16 = 16.\%$$
$$1.60 = 160.\%$$
$$.016 = 1.6\%$$
$$.0016 = .16\%$$

To Find the Percentage Ratio Between Two Numbers.

Rule—Multiply the dividend or numerator by 100 and then divide and express the result as a decimal according to the rules of Section 2 [9].

Thus: $7 \div 25$ or $\frac{7}{25} = \frac{700}{25}\% = 28\%$.

Similarly:

$$3 \div 50 \text{ or } \frac{3}{50} = 6\%$$

$$7 \div 16 \text{ or } \frac{7}{16} = 43.75\%$$

$$6.4 \div 160 \text{ or } \frac{6.4}{160} = 4\%$$

$$6.4 \div 16 \text{ or } \frac{6.4}{16} = 40\%$$

$$6.4 \div 1.6 \text{ or } \frac{6.4}{1.6} = 400\%$$

$$6.4 \div .16 \text{ or } \frac{6.4}{.16} = 4000\%$$

In percentage problems it is usually required to find certain percentages of various quantities, or to find the percentage relations between various quantities. The only difficulty likely to arise is not with the operations themselves, but with a correct interpretation of the problem and a clear understanding as to the relations desired.

Examples—

(1) A broker buys a ship for \$160,000 and sells her for \$172,000. What per cent does he gain?

Solution: The amount of gain is \$12,000. To find what per cent this is of \$160,000 we proceed as above and find 12,000 ÷ 160,000 = 7.5%.

(2) A broker buys a ship for \$160,000 and sells her so as to gain 6%. What was the selling price?

Solution: Six per cent of \$160,000 = .06 × \$160,000 = \$9,600. Hence selling price = \$160,000 + \$9,600 = \$169,600.

(3) A broker sells a ship for \$171,200 and thereby gains 7% on her cost. What was the cost?

Solution: Since he gains 7% or 7 cents on every dollar of cost, there will be \$1.07 in the selling price for every \$1.00 in the cost price. Hence the cost price will be as many dollars as 1.07 is contained in 171,200 or 160,000.

Problems—

(4) Thirty-six pounds of gun metal contain the following:

Copper	32 pounds
Tin	1 pound
Zinc	3 pounds

Find the percentage composition.

Ans.	Copper	$\frac{32}{36}$ or	88.89%
	Tin	$\frac{1}{36}$ or	2.78%
	Zinc	$\frac{3}{36}$ or	8.33%

(5) A ship starts on a voyage of 2,200 miles. After going 800 miles what per cent of the voyage remains to be covered?

Ans. 63.6%.

(6) A ship starts on a voyage of 1,800 miles. After three days she has made 42% of the distance. With 12% increase of speed for the rest of the time how long will it take her to finish the voyage?

Ans. 6.3754 days.

(7) A marine engine requires 2.1 lbs. of coal per I. H. P. per hour. After certain changes are made the figure is reduced to 1.89 lbs. What is the percentage gain?

Ans. 10%.

In one ton of coal (2,240 lbs.) there was found to be 250 lbs. of ashes. What per cent of the coal was combustible?

Ans. 88.8%.

Sec. 4. COMPOUND NUMBERS.

WEIGHTS AND MEASURES.

[1] Long or Linear Measure.

Inches.	Feet.	Yards.	Rods.	Furlongs.	Miles.	Meters.
1	.0833	.0278	.00505	.000126	.000016	.0254
12	=1	.333	.0606	.00152	.000189	.305
36	3	=1	.182	.00455	.000568	.914
198	16½	5½	=1	.025	.00313	5.029
7920	660	220	40	=1	.125	201.166
63360	5280	1760	320	8	=1	1609.3
39.371	3.281	1.094	.199	.00497	.000621	= 1

SPECIAL MEASURES.

6	feet	= 1 fathom.
120	fathoms	= 1 cable's length.
6080.27	feet	= 1 nautical mile (United States).
6080	feet	= 1 nautical mile (British).
3	nautical miles	= 1 marine league.

[2] Avoirdupois Weight or Measure.

Drams.	Ounces.	Pounds.	Tons.		Kilos.
			Long or British.	Short or Legal.	
1	.0625	.0039	....	....	....
16	= 1	.0625	....	....	.02835
256	16	= 1	.000446	.0005	.4536
573440	35840	2240	=1	1.12	1016
512000	32000	2000	.89	= 1	907
564.38	35.27	2.2046	.000984	.001102	= 1

[3] Square Measure.

Square Inches.	Square Feet.	Square Yards.	Square Meters.
1	.00694	.000772	.000645
144	=1	.1111	.0929
1296	9	= 1	.8360
1550	10.765	1.196	= 1

[4] Cubic or Volume Measure.

Cubic Inches.	Cubic Feet.	Cubic Yards.	Cubic Meters.
1	.0005788	....	....
1728	= 1	.037	.0283
46656	27	= 1	.7645
61033	35.32	1.308	= 1

[5] Liquid Measure.

Cubic Inches.	Gills.	Pints.	Quarts.	Gallons.		Barrels.
				U. S.	British.	
1	.1154	.02885	.0144	.00433	.00361	......
8.665	=1	.25	.125	.03125	.3125	......
34.659	4	=1	.50	.125	.125	......
69.318	8	2	=1	.25	.25	......
231	32	8	4	=1	.833	.003175
277.274	32	8	4	1.2	=1	.003175
....	1008	252	126	31½	31½	=1

[6] Dry Measure.

Cubic Inches.	Pints.	Quarts.	Pecks.	Bushels.
1	.02976	.01488	.00186	.0004641
33.6	=1	.5	.0625	.015625
67.2	2	=1	.125	.03125
537.6	16	8	=1	.25
2150.42	64	32	4	=1

[7] Shipping Measure.

1 register ton = 100 cubic feet.

1 United States shipping ton = { 40 cubic feet or / 32.14 United States bushels.

1 British shipping ton = { 42 cubic feet or / 32.72 imperial bushels.

[8] The Metric System of Weights and Measures.

MEASURES OF LENGTH.

10 millimeters (mm)........... = 1 centimeter........... cm.
10 centimeters................. = 1 decimeter............dm.
10 decimeters.................. = 1 meter................. m.
10 meters....................... = 1 dekameter.......... Dm.
10 dekameters................ = 1 hektometer..........Hm.
10 hektometers............... = 1 kilometer........... Km.

Measures of Surface (Not Land).

100 square millimeters (mm^2)..	=	1 square centimeter ...cm^2.
100 square centimeters........	=	1 square decimeter....dm^2.
100 square decimeters..........	=	1 square meter.........m^2.

Measures of Volume.

1000 cubic millimeters (mm^3)...	=	1 cubic centimeter.....cm^3.
1000 cubic centimeters.........	=	1 cubic decimeter.....dm^3.
1000 cubic decimeters..........	=	1 cubic meter..........m^3.

Measures of Capacity.

10 milliliters (ml)...............	=	1 centiliter..............cl.
10 centiliters	=	1 deciliter...............dl.
10 deciliters....................	=	1 liter....................l.
10 liters.......................	=	1 dekaliter.............Dl.
10 dekaliters	=	1 hektoliter ....Hl.
10 hektoliters.................	=	1 kiloliter...............Kl.

Note.—The liter is equal to the volume occupied by 1 cubic decimeter of pure distilled water at its temperature of maximum density, or 39.2° F.

Measures of Weight.

10 milligrams (mg)...........	=	1 centigram ....cg.
10 centigrams...............	=	1 decigram..............dg.
10 decigrams...............	=	1 gram..................g.
10 grams....................	=	1 dekagram............Dg.
10 dekagrams	=	1 hektogram...........Hg.
10 hektograms...............	=	1 kilogram.............Kg.
1000 kilograms................	=	1 ton ....T.

Note.—The gram is the weight of 1 cubic centimeter of pure distilled water at a temperature of 39.2° F.; the kilogram is the weight of 1 liter of water; the ton is the weight of 1 cubic meter of water.

[9] Conversion Tables.

English Measures into Metric.

English.	Inches to Millimeters.	Feet to Meters.	Pounds to Kilos.	Gallons to Liters.
1	25.4001	.304801	.45359	3.78544
2	50.8001	.609601	.90719	7.57088
3	76.2002	.914402	1.36078	11.35632
4	101.6002	1.219202	1.81437	15.14176
5	127.0003	1.524003	2.26796	18.92720
6	152.4003	1.828804	2.72156	22.71264
7	177.8004	2 133604	3.17515	26.49808
8	203.2004	2.438405	3.62874	30.28352
9	228.6005	2.743205	4.08233	34.06896

METRIC MEASURES INTO ENGLISH.

Metric.	Meters to Inches.	Meters to Feet.	Kilos to Pounds.	Liters to Gallons.
1	39.3700	3.28083	2.20462	.26417
2	78.7400	6.56167	4.40924	.52834
3	118.1100	9.84250	6.61386	.79251
4	157.4800	13.12333	8.81849	1.05668
5	196.8500	16.40417	11.02311	1.32085
6	236.2200	19.68500	13.22773	1.58502
7	275.5900	22.96583	15.43235	1.84919
8	314.9600	26.24667	17.63697	2.11336
9	354.3300	29.52750	19.84159	2.37753

USE OF THE CONVERSION TABLES.

Example: Change 243.6 feet into meters.

$$243.6 = 200 + 40 + 3 + .6.$$

These parts separately may all be directly converted from the table by an appropriate use of the decimal point. Thus, keeping simply three places of decimals:

	200 feet	=	60.960 meters.
	40 feet	=	12.192 meters.
	3 feet	=	.914 meters.
	.6 feet	=	.183 meters.
Hence, adding:	243.6 feet	=	74.249 meters.

Other examples may be solved in an entirely similar manner.

[10] Reduction of Compound Numbers.

Example—(1) Reduce 7 miles, 12 rods, 10 feet to feet. We may proceed in two ways: (a) We may reduce the rods to feet and add in the 10 thus: $12 \times 16\frac{1}{2} + 10 = 198 + 10 = 208$. Then reduce the miles to feet and add in the 208 thus: $7 \times 5{,}280 + 208 = 36{,}960 + 208 = 37{,}168$ feet; or (b) we may reduce the miles to rods and add in the 12 thus: $7 \times 320 + 12 = 2{,}240 + 12 = 2{,}252$. Then reduce the rods to feet and add in the 10 thus: $2{,}252 \times 16\frac{1}{2} + 10 = 37{,}158 + 10 = 37{,}168$ feet. The result will, of course, be the same in either case. The factors for making the reductions may be either taken from the tables of Sec. 4 or readily found by separate computation.

Example—(2) Reduce 37,168 feet to miles, rods and feet.

This is the inverse of the above and may likewise be solved in two ways: (a) We may reduce the feet to miles with feet as a remainder, thus: $37,168 \div 5,280 = 7$ miles and 208 feet remainder. Then reduce the feet to rods with feet again as a remainder, thus: $208 \div 16\frac{1}{2} = 12$ rods and 10 feet remainder. Hence, 37,168 feet = 7 miles, 12 rods, 10 feet; or (b) we may reduce the feet to rods with feet as a remainder, thus: $37,168 \div 16\frac{1}{2} = 2,252$ rods and 10 feet remainder. Then reduce the rods to miles with rods as a remainder, thus: $2,252 \div 320 = 7$ miles and 12 rods remainder. We have, therefore, as before, 37,168 feet = 7 miles, 12 rods, 10 feet.

These examples will sufficiently illustrate the mode of procedure so that all similar problems may be readily solved.

Addition, Subtraction, Multiplication and Division of Compound Numbers.

[11] Addition of Compound Numbers.

Example—Add the following:

16 miles	43 rods	12 feet
21 miles	308 rods	9 feet
7 miles	318 rods	$7\frac{1}{2}$ feet
44 miles	669 rods	$28\frac{1}{2}$ feet
or 46 miles	30 rods	12 feet

The columns are added as in whole numbers and the sums are brought down. This gives a correct result, but it is not in its simplest form, since $28\frac{1}{2}$ feet are more than 1 rod, and 669 rods are more than 1 mile. We therefore reduce the feet to rods, giving 1 rod and 12 feet, and then carry the 1 to the column of rods, giving 670. We then reduce the rods to miles, giving 2 miles and 30 rods, and carry the 2 to the column of miles, thus giving the final result as stated in the second line of the answer.

[12] Subtraction of Compound Numbers.

Example—In the following subtract the lower from the upper:

36 miles	18 rods	14 feet
21 miles	43 rods	6 feet

In the column of rods, the 43 being greater than the 18, we cannot subtract directly, and therefore borrow 1 mile or 320 rods from the 36 miles, thus putting the minuend into the form as follows:

35 miles	338 rods	14 feet
21 miles	43 rods	6 feet
14 miles	295 rods	8 feet

We may then subtract each column as in whole numbers, thus giving the result as shown. In the actual operation the borrowing may be done mentally, thus making it unnecessary to rewrite the minuend as shown.

[13] Multiplication of Compound Numbers.

Example—Multiply 14 miles, 276 rods, 12 feet by 5.

Operation:

14 miles	276 rods	12 feet
		5 feet
70 miles	1,380 rods	60 feet
74 miles	103 rods	$10\frac{1}{2}$ feet

We first multiply each term by the multiplier 5 and obtain the result entered in the first line below. This is correct, but not in its simplest form, since 60 feet are more than 1 rod and 1,380 rods more than 1 mile. We therefore reduce the feet to rods, giving 3 rods and $10\frac{1}{2}$ feet, and carry the 3 to the column of rods, giving 1,383. We then reduce the rods to miles, giving 4 miles and 103 rods and carry the 4 to the column of miles, giving the final result as stated in the second line of the answer.

[14] Division of Compound Numbers.

Example—Divide 142 miles, 296 rods, 15 feet by 6.

Operation:

6)142 miles	296 rods	15 feet
$23\frac{2}{3}$ miles	$49\frac{1}{3}$ rods	$2\frac{1}{2}$ feet

We first divide each term separately by the divisor 6 and obtain the result as written below the line. This is correct, but not in its simplest form, since $\frac{2}{3}$ mile may be reduced to rods and feet, and $\frac{1}{3}$ rod may be reduced to feet. We therefore simplify as follows:

23	49	$2\frac{1}{2}$
	213	11
23	262	$13\frac{1}{2}$

Since there are 320 rods in 1 mile, we have $\frac{2}{3}$ miles $= \frac{2}{3} \times$ 320 rods $= 213\frac{1}{3}$ rods. We set down the 213 in the column of rods, and, adding the $\frac{1}{3}$ to the other $\frac{1}{3}$ belonging with the 49, we have $\frac{2}{3}$ rod in addition. Since there are $16\frac{1}{2}$ feet in 1 rod, we have $\frac{2}{3}$ rod $= \frac{2}{3} \times 16\frac{1}{2} = 11$ feet. This we enter in the column of feet and add, giving the final result as shown.

These examples will sufficiently illustrate the principles involved in the addition, subtraction, multiplication and division of compound numbers so that all similar problems may be readily solved.

Sec. 5. DUODECIMALS.

It is often necessary to determine areas or volumes, the dimensions of which are given in feet and inches. To this end we may either reduce the dimensions to feet and decimals, or treat them directly by the method of *duodecimals*. In this system of expressing numbers, use is made of a series of numerical units, each one-twelfth of the unit of next higher order. The fundamental unit will be either the linear foot, the square foot, or the cubic foot, according to the geometrical nature of the quantity to be expressed. Lengths are thus expressed in feet, inches and twelfths; or, as they are termed feet, primes and seconds. Similarly areas are expressed in square feet, primes and seconds, and volumes in cubic feet, primes and seconds, the prime in each case being one-twelfth the foot, and the second one-twelfth the prime.

Duo-decimals should be written with a series of accents thus: $6^\circ - 7' - 10''$. The $^\circ$ stands for the fundamental unit, the foot; which may be a length, an area or a volume, according to the problem; the single accent $'$ stands for the prime, and the double accent $''$ for the second.* Thus, if the unit were a foot of length, the above expression would mean 6 feet, 7 primes or inches, and 10 seconds or $\frac{10}{12}$ inch. If the unit were a square foot it would mean 6 square feet, $\frac{7}{12}$ of a square foot and $\frac{10}{12}$ of $\frac{1}{12}$, or $\frac{10}{144}$ of a square foot. Hence, in all, 6 square feet and 94 square inches. If the unit were a cubic foot it would mean 6 cubic feet, $\frac{7}{12}$ of a cubic foot and $\frac{10}{12}$ of $\frac{1}{12}$ or $\frac{10}{144}$ of a cubic foot. Hence, in all, 6 cubic feet and 1,128 cubic inches. The importance of noting the character of the fundamental unit is thus clearly indicated.

ADDITION OF DUODECIMALS. The addition of duodecimals is carried on exactly as in compound numbers, quantities of the same order being written under each other and their sums, where necessary, reduced to units of higher order by division by 12.

* Care must be taken to avoid confusion between this method of writing duo-decimals and the common method of expressing feet and inches by one and two accents.

Example—Add

$$\begin{array}{r} 17^\circ - 8' - 10'' \\ 14^\circ - 2' - 9'' \\ 7^\circ - 1' - 8'' \\ \hline \text{Sum: } 39^\circ - 1' - 3'' \end{array}$$

SUBTRACTION OF DUODECIMALS. The subtraction of duodecimals is carried on exactly as in compound numbers, quantities of the same order being written under each other and units of a higher order being borrowed where necessary.

MULTIPLICATION OF DUODECIMALS. This operation, which is the one of greatest importance in the treatment of duodecimals, is carried on according to the following:

Rule—Set down the two quantities, with terms of the same order under each other. Multiply each term of the multiplicand by each term of the multiplier. The order of any such product will be determined by adding the indices of the two terms used. If the product is greater than 12, reduce to the next higher order by dividing by 12, and set down the quotient and remainder in their proper columns. Proceed in this way, taking care in the product to set down terms of the same order under each other. Then add and reduce where necessary to the higher order by dividing by 12.

Example—Multiply together:

$$\begin{array}{llll} 6^\circ - & 10' - & 4'' & ''' \\ 4^\circ - & 8' & & \\ \hline 4 & 6 & 2 & 8 \\ 3 & 1 & 8 & \\ 24 & 4 & 4 & \\ \hline 31^\circ & 11' & 2'' & 8''' \end{array}$$

We have first $8' \times 4'' = 32''' = 2'' - 8'''$, which is set down. We have next $8' \times 10' = 80' = 6' - 8''$, which is set down. We have next $8' \times 6^\circ = 48' = 4^\circ$, which is set down. In the same way we use the other term of the multiplier, 4°, and then add and reduce the columns as shown.

For purposes with which the marine engineer is concerned, feet and inches are alone involved, and operations with duodecimals are correspondingly simplified.

Examples—(1) Find the area of a boiler plate $12^\circ - 5'$ long by $6^\circ - 7'$ wide.

Multiplying as before, we find for the product $81^\circ - 8' - 11''$, or taking the result to the nearest prime, $81^\circ - 9'$ or $81\frac{9}{12}$ square feet or 81 square feet — 108 square inches.

(2) Find the volume of a bunker $26^\circ — 4'$ by $9^\circ — 7'$ by $7^\circ — 10'$.

Taking first the product of $26^\circ — 4'$ by $9^\circ — 7'$ we have $252^\circ — 4' — 4''$. Then multiplying this by $7^\circ — 10'$ we have $1{,}976^\circ — 10' — 11'' — 4'''$, or taking the result to the nearest prime, $1{,}976^\circ — 11'$, or $1{,}976\frac{11}{12}$ cubic feet or 1,976 cubic feet — 1,584 cubic inches.

Sec. 6. RATIO AND PROPORTION.

[1] Simple Proportion.

The ratio between two numbers is simply their numerical relationship expressed as the quotient of the first divided by the second. Thus the ratio of 6 to 3 is 2; of 1.2 to 3 is .4; of 4 to 5 is .8, etc. Ratio is often expressed by the sign : which is simply an abbreviation of the sign of division ÷. Thus $6 : 3 = 2$; $1.2 : 3 = .4$, etc. Ratio is also expressed by the sign of division or in the form of a fraction. Thus

$$6 : 3 \text{ or } 6 \div 3 \text{ or } \tfrac{6}{3} = 2$$
$$1.2 : 3 \text{ or } 1.2 \div 3 \text{ or } \tfrac{1.2}{3} = .4, \text{ etc.}$$

[1] A *proportion* is a statement of the equality of two ratios. The equality is commonly expressed by the symbol : :

Thus: $6 : 3 :: 4 : 2$ ....(1)

A proportion may also be written in other ways, as follows:

$6 : 3 = 4 . 2$ ....(2)

$\frac{6}{3} = \frac{4}{2}$ ....(3)

A proportion always contains four terms, two for each ratio, and when written as in (1) and (2) the first and last terms are called *extremes* and the second and third are called *means*. To solve a proportion three terms must always be known and then the remaining term can always be found. From the nature of a proportion it follows that the product of the extremes must equal the product of the means, and this gives the usual rule for solution as follows:

Rule—If the unknown term is an extreme, multiply together the two means and divide by the other extreme. If the unknown term is a mean, multiply together the two extremes and divide by the other mean. In either case the quotient will give the remaining term desired.

In a proportion two of the terms always relate to one kind of quantity, while the other two relate to another. Likewise two of the terms always relate to one set of conditions, while the other two relate to another set. In problems to be solved by proportion, therefore, where three terms must be given, two terms will be of one kind and the remaining term will be of another and of the same kind as the answer. Also two terms will relate to the given set of conditions, and the remaining term to the set involving the unknown term or answer.

Again, proportions are of two kinds, *direct* and *inverse.* In a direct proportion the relation between the two quantities different in kind is such that both increase or decrease together. In an inverse proportion the relation between these two quantities is such that one increases as the other decreases. Thus if the question involves the relation between distance and speed, the time being the same, the proportion is direct, because, for a given time, the more speed the more distance, or the more distance the more speed. On the other hand, if the question involves the relation between speed and time for a given distance, the proportion is inverse, because for a given distance the more speed the less time, or the more time the less speed, and *vice versa.* Our general knowledge of the relation between the quantities involved will thus enable us to determine whether the proportion will be direct or inverse.

Example of a direct proportion:

If a ship steams 1,400 miles in 5 days, how far will she steam in 8 days at the same rate?

Here two of the given terms are days and one is miles, being of the same character as the answer desired. Also two relate to one performance or set of conditions (1,400 in 5 days), while the other relates to another set (the desired performance in 8 days). Our right to solve this example by direct proportion lies in the fact that the relation between the 5 days and 1,400 miles must be the same as that between the 8 days and the distance desired, and hence there must be an equality in the ratios showing these relations, and similarly for problems of like character.

A simple *direct* proportion, like that involved in the example above, may be stated in various ways, of which two will serve our present purpose.

$$\text{(a) } \overset{(2)}{\text{Ans. (miles)}} : \overset{(2)}{8\text{ (days)}} :: \overset{(1)}{1400\text{ (miles)}} : \overset{(1)}{5\text{ (days)}}$$

$$\text{(b) } \overset{(2)}{\text{Ans. (miles)}} : \overset{(1)}{1400\text{ (miles)}} :: \overset{(2)}{8\text{ (days)}} : \overset{(1)}{5\text{ (days)}}$$

The numbers in ( ) relate to the two sets of conditions, the first being the set given, and the second being the set for which the distance is desired. It is thus seen that the proportion may be stated as in (a), with each set of conditions forming a ratio, the items of proportion alternating: miles, days, miles days; or it may be stated as in (b), with the set of conditions alternating 2d, 1st, 2d, 1st, the two items of each ratio being the same.

Example of an inverse proportion:

If a ship at 10 knots does a certain trip in 5 days, how many days will be required at 12 knots?

The relation between time and speed, as we know, is such that, for a given distance, the more speed the less time, or the less speed the more time. The time is, therefore, said to be in such case *inversely* proportional to the speed. That is, for example, if the speed is *doubled* the time will be *halved*, etc.

An inverse proportion may be stated as follows:

$$\text{(c)}\ \overset{(2)}{\text{Ans.}}\ \text{(days)} : \overset{(1)}{10}\ \text{(knots)} :: \overset{(1)}{5}\ \text{(days)} : \overset{(2)}{12}\ \text{(knots)}$$

$$\text{(d)}\ \overset{(2)}{\text{Ans.}}\ \text{(days)} : \overset{(1)}{5}\ \text{(days)} :: \overset{(1)}{10}\ \text{(knots)} : \overset{(2)}{12}\ \text{(knots)}$$

The difference between these methods of statement and those above for direct proportion will be readily seen by comparison.

As another general way of stating a proportion whether direct or inverse we may proceed as folows:

Put the *answer*, or letter representing it, for the first term, and the other quantity of the same kind for the second term. Then, according as the first term or answer should be greater or less than the second term, write the third and fourth terms in the same order. That is if the relation of the quantities is such that the *answer* should be greater than the second term, write the third and fourth terms in the order *greater — lesser:* or if the *answer* should be less than the second term, write the third and fourth terms in the order *lesser — greater.*

After having become familiar with the methods of stating a proportion, the names of the quantities and the numbers denoting the sets of conditions may, of course, be omitted, and we should have instead of (a) and (b)

$$\text{Ans.} : 8 :: 1400 : 5$$
$$\text{Ans.} : 1400 :: 8 : 5$$

Solving either of these according to the rule given, we have:

$$\text{Distance} = 8 \times 1400 \div 5 = 2240 \text{ ans.}$$

Likewise instead of (c) and (d) we should have:

$$\text{Ans.} : 10 :: 5 : 12$$
$$\text{Ans.} : 5 :: 10 : 12$$

Solving either of these we have:

$$\text{Days} = 5 \times 10 \div 12 = 50 \div 12 = 4\tfrac{1}{6} \text{ Ans.}$$

[2] Compound Proportion.

In many cases the result depends on more than one changing condition. In such case the problem is treated by the method of compound proportion as illustrated in the following example.

When coal is $4.20 per ton the fuel bill for a ship requiring 34 tons per day on a voyage of 2,100 miles is $1,020. At the same speed what would be the bill on a 2,800-mile voyage, with coal at $3.60 per ton, and a consumption of 40 tons a day?

The resulting cost evidently depends on three conditions, and the proportion is stated as follows:

$$\text{Ans.} : 1020 :: \begin{matrix} 3.60 & : & 4.20 \\ 40 & : & 34. \\ 2800 & : & 2100 \end{matrix}$$

This consists really of three simple direct proportions, each stated exactly according to the rules already given, as may be readily seen. For the solution of such a proportion the same rules may be used, but the "product of the means" or the "product of the extremes" must here be understood to mean the product of all the separate numbers forming these terms.

Thus for the foreging proportion we multiply together the means and divide by the extremes, giving as follows:

$$\text{Ans.} = \frac{3.60 \times 40 \times 2800 \times 1020}{4.20 \times 34 \times 2100} = \$1371.43.$$

Cancellation can usually be made to assist in the reduction of such expressiŏns, and we thus readily find the answer as stated.

Where the relations are such that the answer is inversely proportional to certain of the varying conditions, the ratio involving these conditions must, of course, be so stated as to make this part of the proportion inverse instead of direct. In fact, each of the ratios in the second part of the proportion should be carefully examined in order to make sure whether it should be stated *direct* or *inverse*.

Example involving both direct and inverse proportion:

When coal is $4.00 per ton, the fuel bill for steaming a certain distance at a speed of 12 knots, with a ship requiring 48 tons per day, is $1,800. For the same distance, what would be the bill if the ship steams 10 knots on 40 tons per day, with coal at $3.20 per ton?

The statement would be as follows:

$$\text{Ans.} : 1800 :: \begin{cases} 3.20 : 4.00 \text{ (direct)} \\ 12 : 10 \text{ (inverse)} \\ 40 : 48 \text{ (direct)} \end{cases}$$

In this case we have:

$$\text{Ans.} = \frac{1800 \times 3.20 \times 12 \times 40}{4.00 \times 10 \times 48} = \$1440$$

Problems in Proportion.

(1) An engine with 34 lb. mean effective pressure gives 1,400 I.H.P. All other conditions remaining the same, what would the engine give with 39 lb. mean effective pressure?

Ans. 1,606 I.H.P.

(2) An engine making 98 revolutions per mt. gives 1,800 I.H.P. All other conditions remaining the same, what would the engine give if the revolutions were 90?

Ans. 1,653 I.H.P.

(3) An engine with stroke of 3 ft. gives 900 I.H.P. All other conditions remaining the same, what would be the power with stroke of 42 in.?

Ans. 1,050 I.H.P.

(4) An engine cylinder whose area is 1,200 sq. in. gives 800 I.H.P. All other conditions remaining the same, what would be the power with an area of 1,000 sq. in.?

Ans. 666.67 I.H.P.

(5) A given engine has mean effective pressure 33, revolutions 120, stroke 42 in., and gives 1,800 I.H.P. Other conditions remaining the same, what would be the power with a mean effective pressure of 38, revolutions of 100 and stroke of 4 ft.?

Ans. 1,974 I.H.P.

(6) A boiler with tubes 7 ft. long, and 2¾ in. diam., has 2,168 sq. ft. of tube-heating surface. What would be the surface, with the same number of tubes, 7 ft. 6 in. long and 2½ in. diam.?

Ans. 2,112 sq. ft.

(7) An engine with 34 lbs. mean effective pressure and 98 revolutions develops a certain power. If the mean effective pressure were 38 lbs., what revolutions would give the same power?

Ans. 87.7 rev.

(8) A given engine has a mean effective pressure 33, revolutions 120, stroke 42 in., and develops a certain power. With mean effective pressure of 38 and stroke of 3 ft., what would be the revolutions for the same power?

Ans. 121.6 rev.

(9) A pump making 40 double strokes per mt. can empty a tank in 1½ hrs. In what time could the same pump, making 30 double strokes per mt., empty a second tank, 20 per cent larger than the first?

Ans. 2.4 hours.

(10) A propeller at 100 revolutions and 20 per cent slip gives a speed of 18 knots. What will be the speed at 120 revolutions and 25 per cent slip?

Ans. 20.25 knots.

Sec. 7. EVOLUTION AND INVOLUTION.*

[1] EVOLUTION is the operation of raising a number to successive powers, or of multiplying it into itself as a factor a certain number of times. The number of times the number is used as a factor is called the *index* of the power, and is indicated by a small figure written to the right and above as follows:

$$4^2 = 4 \times 4 = 16$$
$$5^3 = 5 \times 5 \times 5 = 125$$
$$3^5 = 3 \times 3 \times 3 \times 3 \times 3 = 243$$

Evolution involves, therefore, simply continued multiplication. The powers most commonly used by the engineer are

---

* Hand books with convenient tables of powers and roots are so common at the present day that the engineer has small use for the actual operations of raising to powers or of extracting roots. In practice the use of such tables is always counseled as tending toward accuracy and speed. The importance of these operations, however, merits a brief outline of the process as given herewith.

the second and third, or *square* and *cube,* as they are commonly called.

INVOLUTION is the inverse of evolution, and consists in finding a number which, used on itself as a factor a certain number of times, will produce a *given* number. The former is then called the root of the latter. The number of times the root is used as a factor is called the *index,* and is represented either by writing this number at the upper left hand angle of the sign of involution, $\sqrt{}$, or by the use of a fractional index, written as in evolution. The roots most commonly used by the engineer are the *square* and *cube* roots, corresponding to the second and third or square and cube powers. When square root is indicated by the symbol $\sqrt{}$, it is customary to omit the 2 from the upper angle. Thus we have:

$$\sqrt[3]{27} \text{ or } (27)^{\frac{1}{3}} = 3 \text{ because } 3 \times 3 \times 3 = 27.$$

$$\sqrt{49} \text{ or } (49)^{\frac{1}{2}} = 7 \text{ because } 7 \times 7 = 49.$$

Occasionally the engineer has to deal with the index $\frac{2}{3}$. which is simply a short way of indicating two operations. (1) Raising to the square. (2) Extracting the cube root. Thus $64^{\frac{2}{3}}$ means the 3d root of the 2d power of 64, or the 2d power of the 3d root of 64. Either order of operation will give the same result. Thus:

$$(64)^{\frac{2}{3}} = \sqrt[3]{64 \times 64} = \sqrt[3]{4096} = 16$$

$$\text{or } (64)^{\frac{2}{3}} = \left(\sqrt[3]{64}\right)^2 = 4^2 = 16$$

[2] To Extract the Square Root.

This is best illustrated by an *example.* Find the square root of 746.2.

*Rule**—(1) Point off the given number into periods of two figures each, beginning at the right or at the decimal point, if there is one, and in the latter case point both ways, adding ciphers on the right of the decimal as may be necessary to complete the periods. Thus:

* The brackets [ ] contain the numbers in the example which correspond to the special indication of the rule, and thus make its application to the given case more easily followed.

Number	*Pointed off.*
2643	26·43
867	8·67
424.362	4·24.36·20
.024	.02·40
.6	.60

(2) Write a 0 on the left, thus heading two columns (1) and (2), as shown. Find by trial the greatest number [2] which when squared is equal to or less than the left hand period. Put this on the right as the first figure of the root.

(1)	(2)
0	7·46.20(27.316 +
2	4
2	346
2	329
40	1720
7	1629
47	9120
7	5461
540	363900
3	
543	
3	
5460	
1	
5461	
1	
54620	

(3) Place this figure [2] under the 0 in col. (1) and add. Multiply the sum [2] by the root figure [2], and place the product [4] under the left hand period in col. (2). This product will be, of course, the square of the root figure. Subtract and bring down the next period for a partial dividend [346].

(4) Again place the root figure [2] on the left and add [4]. Annex a 0 and the result [40] will be a trial divisor for the next root figure.

(5) Divide and take on trial the resulting whole number for the next root figure [7].

(6) Bring down this figure [7] in col. (1) under the 0. Add and multiply the sum [47] by the same root figure [7] and place the product [329] in col. (2) under the partial dividend. Subtract and bring down as before for a new partial dividend [1720].

(7) Place the root figure [7] again in col. (1), add and annex a cipher for a new trial divisor [540].

(8) Find another root figure as before and proceed in this manner till as many figures are obtained as are desired.

If the product found, as in (6), is greater than the partial dividend, it indicates that the trial figure was too great, and the next lower must be taken. If at any time the trial divisor is greater than the partial dividend, enter a 0 in the root, bring down the next period in col. (2) for a new partial dividend, annex a 0 on the right in col. (1) for a new trial divisor and proceed as before.

[3] To Extract the Cube Root.

This is best illustrated by an example. Find the cube root of 12.593.

(1)		(2)	(3)
0		0	12.593 (23.26
2		4	8
2		4	4593
2		8	4167
4	66	1200	426000
2	3	189	320168
60	690	1389	105832000
3	2	198	
63	692	158700	
3	2	1384	
	694	160084	
	2	1388	
	6960	16147200	

Rule—Point off the given number into periods of three figures each, beginning at the right or at the decimal point, if there is one, and in the latter case point both ways, adding ciphers on the right of the decimal as may be necessary to complete the periods. Thus:

Number	*Pointed Off.*
1724	1˙724
17243	17˙243
17.24	17.240
.64	.640
.0032	.003˙200

(2) Write ciphers on the left thus, heading the columns (1), (2) and (3), as shown. Find by trial the greatest number [2] which when cubed is equal to or less than the left hand period. Put this on the right as the first figure of the root.

(3) Place this figure under the 0 in col. (1), add, multiply sum [2] by root figure [2], place product [4] in col. (2), add, multiply sum [4] by root figure [2] again, and place product [8] in col. (3) under left hand period. This product will be, of course, the cube of the root figure. Subtract and bring down the next period for a partial dividend [4593].

(4) Again place the root figure [2] in col. (1), add, multiply sum [4] by root figure, put the product [8] in col. (2), and add [12].

(5) Again place the root figure [2] in col. (1), and add as before [6].

(6) Annex one 0 to the result in col. (1) [60] and two 0s to that in col (2) [1200]. The latter is then a trial divisor for the next root figure, which is thus seen to be probably 3.

(7) The same process of bringing down, adding and multiplying in cols. (1), (2) and (3) is then repeated as just described, and as shown in the example. Continue in this way until as many figures of the root are found as are desired.

If the final product to be entered in col. (3) is greater than the partial dividend, it indicates that the trial figure was too great, and the next lower must be taken. If at any time the trial divisor is greater than the partial dividend, enter a 0 in the root, bring down the next period in col. (3) for a new partial dividend, annex one 0 in col. (1) and two 0s in col. (2) for a new trial divisor, and proceed as before.

It will be noted that in these methods for square and cube root the former requires for each root figure two operations in col. (1) and one in col. (2), while the latter similarly requires three operations in col. (1), two in col. (2), and one in col. (3). Care should be taken that none of these are omitted, and that the entire process is carried through with regularity and order.

Sec. 8. MATHEMATICAL SIGNS, SYMBOLS AND OPERATIONS.

Mathematical signs are simply shorthand methods of indicating mathematical language. Those most commonly met with are the following:—

$+$ The sign of addition called *plus*. This means that the two numbers or quantities between which it is placed are to be added. Thus $12 + 3$ is read 12 plus 3 and means that 12 and 3 are to be added, the result being 15.

— The sign of subtraction called *minus*. This means that the number or quantity which follows the sign is to be subtracted from that which precedes it. Thus 12 — 3 is read 12 minus 3 and means that 3 is to be taken from 12, the result being 9.

× The sign of multiplication. This means that the two numbers or quantities between which it is placed are to be multiplied together. Thus 12 × 3 is read 12 times 3 or 12 multiplied by 3, the result being 36.

÷ The sign of division. This means that the number or quantity which precedes the sign is to be divided by that which follows. Thus 12 ÷ 3 is read 12 divided by 3, the result being 4.

/ A sign of division. A fraction is really a mode of expressing division, and a common way of writing a fraction all in one line is to make use of the oblique line. Thus 12/3 means the same as $\frac{12}{3}$ or 12 ÷ 3 or 4. Frequently the horizontal line — as shown is used, thus: $\frac{1}{2}$, ½, 1-2 all indicate one-half, or 1 divided by 2. The horizontal line used in this connection must not be confused with the *minus* sign; usually the sense is plainly indicated by the connection in which it is used.

. Placed before and in line with the bottom of a number is a decimal point, showing that the number is the numerator of a fraction which has some power of 10 for its denominator; as .1 $= \frac{1}{10}$, .25 $= \frac{25}{100}$, which reduced to its lowest terms is 1-4. See section 2.

: The sign of ratio. This signifies the ratio or numerical relationship of the two quantities between which it is placed, and is equivalent to a sign of division, since the quotient of the first quantity divided by the second is the measure of the ratio between them. Thus 12 : 3 is read the ratio of 12 to 3 or 12 is to 3, and the real measure of this ratio is 12 ÷ 3 or 4.

:: The sign of proportion or equality of ratios. This sign is placed between two ratios and signifies that they are equal. Thus 12 : 3 :: 20 : 5 is read 12 is to 3 as 20 is to 5, or the ratio of 12 to 3 equals the ratio of 20 to 5. This is seen to be the case since 4 is the measure of each ratio.

= The sign of equality. This signifies that the two quantities separated by the sign are equal in value. Thus 12 ÷ 3 = 4 is read 12 divided by 3 equals 4, and thus states the equality between the two sides of the relationship.

Equation. Two quantities or expressions related by the sign of equality, =, form an equation. Thus

$$4 + 2 = 6$$
$$a = 3$$
$$a + b = c$$

( ), [ ], { }. *Parentheses or brackets.* These symbols mean that all numbers or quantities within a parenthesis or pair of brackets are to be considered as one quantity and thus treated in all numerical operations. Thus 2 (3 + 4) means that 3 + 4 is to be taken as the single quantity 7, and then multiplied by 2. Similarly

$$2\ (3+4-2)=10$$
$$3\ [16-2\ (3+4-2)]=18$$

NOTE: The sign multiplication, ×, is often omitted between a number and a parenthesis or bracket containing a quantity into which it is to be multiplied, as in the expressions here shown.

In some cases a bar or *vinculum* is drawn over numbers thus to be taken together. Thus $2 \times \overline{3 + 6}$ means 2 multiplied by the quantity 3 + 6, or 2 × 9, or 18. The difference between $2 \times \overline{3 + 6}$ and $2 \times 3 + 6$ or 12 will be noted. In reducing quantities thus affected by the signs +, —, ×, ÷ and connected by brackets, the difference in significance between the sign +, — and × ÷ must be carefully noted. Thus 2 × 3 + 6 means 2 times 3 added to 6 or 12 and not 2 times (3+6) or 18. As a general principle, it may be remembered that the signs + and — effect a separation of the expression into separate terms, while the signs × and ÷ bind together the two quantities between which they stand into a single term.

Examples—These principles are further illustrated by the following:

$$3 \times 16 - 2 \times 3 + 4 - 2 + 4 \times 3 - 1 = 55$$
$$3[16 - 2 \times 3 + 4 - 2 + 4 \times 3 - 1] = 69$$
$$3[16 - 2\ (3 + 4) - 2 + 4\ (3 - 1)] = 26$$
$$3[16 - 2\ (3 + 4 - 2)] + 4\ (3 - 1) = 24$$

$\sqrt{}$ This sign placed over a quantity denotes the extraction of a root. If no number is placed at the upper left angle, it denotes the square root; otherwise a root corresponding to the index thus indicated.

Thus $\sqrt{49}$ denotes the square root of 49 or 7, $\sqrt[3]{27}$

denotes the cube root of 27 or 3, $\sqrt[4]{16}$ denotes the fourth root of 16 or 2, etc.

3^2 or 4^3 or a^4. A small number written to the right and above another number or quantity is called an *index* or *exponent*, and signifies that the lower number or quantity is to be used as a factor on itself a number of times equal to the index. Thus:

$$3^2 = 3 \times 3 = 9$$
$$4^3 = 4 \times 4 \times 4 = 64$$
$$a^4 = a \times a \times a \times a$$

′ ″ These signs set to the right and above any figure or figures (superior) signify feet and inches. These signs are much used in dimensioned drawings.

$\perp$ Signifies *perpendicular to.*

$\angle$ Signifies *angle.*

∟ Signifies *right angle.*

$\therefore$ Signifies *hence* or *therefore.*

$\because$ Signifies *because.*

π Denotes the ratio between the circumference and diameter of a circle. Its value is usually taken as 3.1416.

□ Square sometimes used in denoting pressures; as 220 lbs. per □″ or per square inch.

% Signifies *per centum* or *per hundred.*

FORMULAE.

A formula is simply a brief way of denoting a series of mathematical operations. Once understood, the directions given by a formula are much more readily followed than when given in the form of a rule. In fact a formula may be considered as simply a brief or short hand way of expressing the same directions as are given by the rule in ordinary words. In formulae, quantities are usually represented by letters, as in the well-known horse power formula:

$$I.H.P. = \frac{2\,p\,L\,A\,N}{33{,}000}$$

In this formula p denotes the mean effective pressure per square inch of piston area, L the length of stroke in feet, A the piston area in square inches, and N the revolutions per minute.

In thus writing letters to represent quantities the sign of multiplication, $\times$, is usually omitted. Thus in the foregoing the numerator $2\ p\ L\ A\ N$ means the same as $2 \times p \times L \times A \times N$, or that the continued product is to be taken of these five factors. It must be noted, however, that where both factors are numbers the sign for multiplication cannot be omitted. Thus, 23 does not mean 2×3, but $20 + 3$ or 23.

Division may be expressed by the usual sign, but it is more commonly indicated by writing the dividend as the numerator and the divisor as the denominator of a fraction. Or in general we multiply by putting a factor in the numerator, and divide by putting a factor in the denominator. Thus in the horse power formula the product $2\,p\,L\,A\,N$ is to be divided by 33,000.

As a further illustration, take the formula

$$p = \frac{T\,t}{6\,R}$$

In this formula p is the pressure per square inch in a boiler, T is the tensile strength of the material of the shell, t is the thickness of the plate, and R is the half diameter or radius of the shell. The whole gives the pressure per square inch allowed by U. S. rules on marine boilers. The formula directs us, in order to find the desired pressure, to multiply together T and t and to divide the product by 6 times R; or, in the words of the U. S. rule: "Multiply one-sixth of the lowest tensile strength . . . by the thickness . . . and divide by the radius or half diameter." In this formula all dimensions are in inches and the result is the pressure per square inch allowed on the boiler. Thus let $T = 60{,}000$, $t = 1\frac{1}{4}$ in. and $R = 6$ feet or 72 in. Then:

$$p = \frac{60{,}000 \times 1.25}{6 \times 72} = 174 \textit{ pounds per square inch.}$$

Take again the formula

$$p = \frac{112\ t^2}{L^2}$$

In this formula p is the pressure per square inch allowed on a flat surface of a boiler supported by staybolts, t is the thickness of the plate expressed in sixteenths, and L is the pitch of the bolts, or distance from center to center. The formula thus directs us to multiply 112 by the square of the thickness of the plate in sixteenths, and then to divide the product by the square of the pitch of the bolts. Thus let the thickness be 9-16 in. and pitch be 7 in. Then we have:

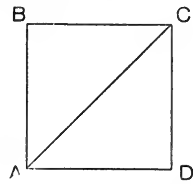
$$p = \frac{112 \times 9 \times 9}{7 \times 7} = 185 \textit{ pounds per square inch.}$$

Sec. 9. GEOMETRY AND MENSURATION.*

[1] Square.

A *square* is a figure, such as A B C D, having four sides all equal, and four angles all equal, each being a right angle.

DIAGONAL, A C. To find the length of a diagonal, A C, having given a side of the square, as A B:



Rule—Square the side, multiply by 2, and take the square root;

$$\text{or } A\,C = \sqrt{\overline{A\,B}^2};$$

or *Rule*—Multiply the side by 1.4142.

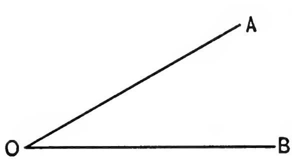
Example: $A\,B = 16$. Then $A\,C = \sqrt{2 \times 256} = \sqrt{512} = 22.627$,

$$\text{or } A\,C = 16 \times 1.4142 = 22.627.$$

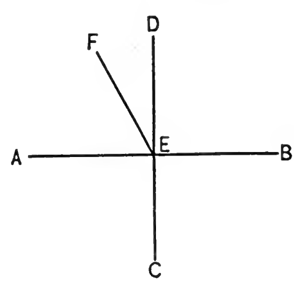
AREA, A B C D. To find the area of a square, having given the length of a side, as A B:

* The following definitions are here given as introductory to this section. Other definitions will be given as the terms are introduced.

An *angle* is formed when two lines, $O\,A$ and $O\,B$, having different directions meet in a point. as O. The angle refers, then, to the difference in direction of the two lines, and its measure is a measure of such difference in direction. An angle is usually denoted by three letters, the one at the apex being placed between the other two. Thus, in *Fig. a* the angle would be called $A\,O\,B$ or $B\,O\,A$.



a



b

When a line, $C\,D$, meets another line, $A\,B$, in such a way that the four angles at E are all equal, the two lines are said to be *perpendicular* to each other, or, in more common terms, one line is square with the other. An angle such as those formed at E is called a *right angle*.

An angle less than a right angle, as $A\,O\,B$, *Fig. a*, is called an *acute* angle. An angle greater than a right angle, as $F\,E\,B$, *Fig. b*, is called an *obtuse* angle.

Rule—Square the side, or multiply it by itself;

$$\text{or Area} = \overline{A\,B}^2 = A\,B \times A\,B.$$

Example: $A\,B = 6$. Then area $= 6 \times 6 = 36$.

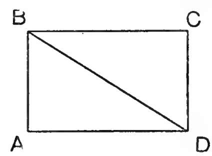
[2] Rectangle.

A *rectangle* is a figure, such as A B C D, having four sides, the opposite sides being equal and parallel (A B = D C and B C = A D), and four angles all equal, each being a right angle.

DIAGONAL, B D. To find the length of a diagonal, B D having given the two sides, as B C and C D:

Rule—Square the two adjacent sides, add, and take the square root;

$$\text{or } B\,D = \sqrt{\overline{B\,C}^2 + \overline{C\,D}^2}.$$



Example: $B\,C = 6$, $C\,D = 8$. Then $B\,D = \sqrt{36 + 64} = \sqrt{100} = 10$.

AREA, A B C D. To find the area of a rectangle, having given the two sides, as A B and A D:

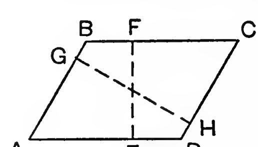
Rule—Find the product of the two adjacent sides;

$$\text{or Area} = A\,B \times A\,D.$$

Example: A B = 6, A D = 8. Then area = $6 \times 8 = 48$.

[3] Parallelogram.

A *parallelogram* is any figure, such as A B C D, having four sides and four angles, the opposite sides being equal and parallel, and the oposite angles being equal.



AREA, A B C D. To find the area of a parallelogram, having given a side and the perpendicular distance between this and the side opposite:

Rule—Multiply one side by the perpendicular distance between it and the side opposite:

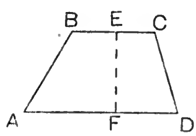
or Area = A D × E F;
= also A B × G H.

Example: A D = 16, E F = 9. Then area = 9 × 16 = 144.

[4] Trapezoid.

A *trapezoid* is any figure, such as A B C D, having four sides and four angles, two of the sides, as B C and A D, being parallel.

AREA, A B C D. To find the area of a trapezoid, having given the parallel sides and the perpendicular distance between them:



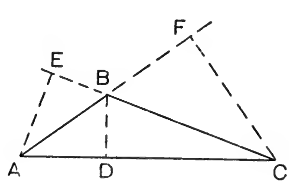
Rule—Multiply the half sum of the parallel sides by the perpendicular distance between them,

$$\text{or Area} = \left(\frac{B\,C + A\,D}{2}\right) \times E\,F.$$

Example: $B\,C = 10$, $A\,D = 16$, $E\,F = 8$. Then area $= \left(\frac{10 + 16}{2}\right) \times 8 = 104$.

[5] Triangle.

A *triangle* is any figure, such as A B C, having three sides and three angles. In a triangle placed as in the figure, A C is



called the *base*, and B D—the perpendicular distance from B to A C—is called the *altitude*.

ANY SIDE, A B. To find the length of any side, having given the triangle complete:

Rule—Square the other two sides and add, and according as the angle between them is greater or less than 90 degrees, add or subtract twice the product of one of these sides by the projection* of the other upon it. Then take the square root of the result thus found;

$$\text{or } A\,B = \sqrt{\overline{A\,C}^2 + \overline{B\,C}^2 - 2\,A\,C \times D\,C}.$$

$$\text{Similarly } B\,C = \sqrt{\overline{A\,B}^2 + \overline{A\,C}^2 - 2\,A\,C \times A\,D},$$

$$\text{and } A\,C = \sqrt{\overline{A\,B}^2 + \overline{B\,C}^2 + 2\,B\,C \times B\,E}.$$

Example: $A\,C = 12$, $B\,C = 9$, $D\,C = 8$. Then

$$A\,B = \sqrt{144 + 81 - 2 \times 12 \times 8} = \sqrt{33} = 5.745.$$

Area. To find the area of a triangle, having given the triangle complete, or any side and its perpendicular distance from the opposite vertex:

Rule—Multiply any side by the perpendicular distance from the opposite vertex to such side (produced, if necessary, to meet the perpendicular), and take half the product thus found; or take half the product of the base by the altitude;

$$\text{or Area} = \tfrac{1}{2}\,(A\,C \times B\,D),$$
$$= \tfrac{1}{2}\,(B\,C \times A\,E),$$
$$= \tfrac{1}{2}\,(A\,B \times C\,F).$$

Example: A C = 120, B D = 32. Then area = ½ (120 × 32) = 1,920.

[6] A Right-Angled Triangle.

In a *right-angled triangle* one of the angles, as C, is a right angle. The side opposite is called the *hypothenuse*.

Hypothenuse, A B. To find the length of the hypothenuse, having given the other two sides:

Rule—Square the other two sides and add, and take the square root of the sum;

$$\text{or } A\,B = \sqrt{\overline{A\,C}^2 + \overline{B\,C}^2}.$$

Example: $A\,C = 9$, $B\,C = 12$. Then $A\,B = \sqrt{81 + 144} = \sqrt{225} = 15$.

Side A C or B C. To find the length of one of the sides about the right angle, having given the hypothenuse and the other side:

---

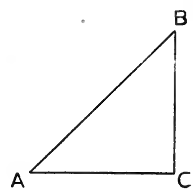
* Let $B\,D$ be drawn perpendicular to $A\,C$. Then $D\,C$ is called the *projection* of $B\,C$ upon $A\,C$. Similarly $A\,D$ is the projection of $A\,B$ upon $A\,C$, $A\,F$ the projection of $A\,C$ upon $A\,B$ produced, and $E\,C$ the projection of $A\,C$ upon $B\,C$ produced.

Rule—Square the hypothenuse and the given side. Subtract the squares, and take the square root of the difference;

$$\text{or } A\,C = \sqrt{\overline{A\,B}^2 - \overline{B\,C}^2}.$$

Example: $A\,B = 15$, $B\,C = 12$. Then $A\,C = \sqrt{225 - 144} = \sqrt{81} = 9$.

AREA, A B C. To find the area of a right-angled triangle, having given the two sides about the right angle:



Rule—Multiply together the two sides about the right angle, and take half their product;

or Area = ½ (A C × B C).

Example: A C = 9, B C = 12. Then area = ½ (9 × 12) = 54.

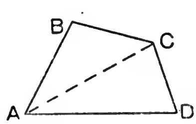
These rules are special cases of those for the general triangle, as in [5].

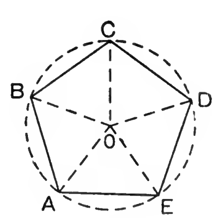
[7] Trapezium.

A *trapezium* is a figure, such as A B C D, having four angles and four sides, no two of the latter being parallel.

AREA, A B C D. To find the area of a trapezium, having given the figure complete:

Rule—Divide the trapezium into two triangles, and proceed with each separately, and then add.





[8] Regular Polygons.

A *regular polygon* is a figure, such as A B C D E, having any number of equal sides and a like number of equal angles. They are named as follows:

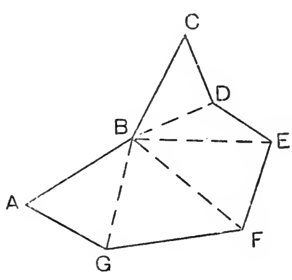
Number of Sides.	Names.
3	Triangle
4	Square
5	Pentagon
6	Hexagon
7	Heptagon
8	Octagon
9	Nonagon
10	Decagon

AREA. To find the area of any regular polygon, as A B C D E, having given the figure complete:

Rule—Divide the polygon into as many triangles as there are sides, the apexes all being at the center. Find the area of one of these, and multiply by the number of sides.

[9] Irregular Figures.

AREA. To find the area of a figure, such as A B C D E F G having given the figure complete:



Rule—Divide into triangles in any convenient way; proceed with each separately, and add the results.

[10] Circle.

A *circle* is a figure bounded by a curved line, every point of which is equally distant from a point within, called the center. The distance across from one side to the other through the center is called the *diameter* (see A B or F G). The diameter is usually represented by *D*. The distance from the center to the curved boundary line is called the *radius,* and is plainly one-half the diameter (see A C, F C, etc.). The radius is usually represented by *r*. The curved boundary line A F H B G K A, is called the *circumference*. Any part of the circumference, as A F, F H, etc., is called an *arc*.

CIRCUMFERENCE. To find the circumference, having given the diameter:

Rule—Multiply the diameter by 3.1416, or more exactly by 3.1415927, or less exactly by $\frac{22}{7}$. This ratio is frequently denoted by the symbol π.

or Circumference = 3.1416 × Diameter = π D.

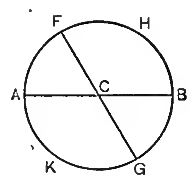
Example: Diameter = 11. Then circumference = 11 × 3.1416 = 34.5576.

DIAMETER. To find the diameter, having given the circumference:

Rule—Divide the circumference by 3.1416, or more exactly by 3.1415927, or less exactly by $\frac{22}{7}$.

or Diameter = Circumference ÷ 3.1416,
or Diameter = Circumference × .31831.

Example: Circumference = 48.7. Then diameter = 48.7 ÷ 3.1416 = 15.50 +



AREA, A H B K. To find the area of a circle, having given the diameter or the radius:

Rule—Multiply the square of the diameter by .7854, or 3.1416 ÷ 4.

or multiply the square of the radius by 3.1416,
or find half the product of the radius by the circumference,

$$\text{or Area} = .7854 \times (\text{Diameter})^2 = \frac{\pi D^2}{4},$$
$$= 3.1416 \times (\text{Radius})^2 = \pi r^2.$$
$$= \tfrac{1}{2} (\text{Radius} \times \text{Circumference}).$$

Example: Diameter = 10. Then area = .7854 × 100 = 78.54.

LENGTH OF ARC. To find the length of an arc, as A F, having given the corresponding number of degrees and the circumference or diameter:

Rule—Divide the circumference by 360, and multiply by the number of degrees in the arc;

or multiply the number of degrees by .008727 and the product by the diameter,

$$\text{or Length } A\ F = \frac{\text{Circumference}}{360} \times (\text{Number of Degrees})$$

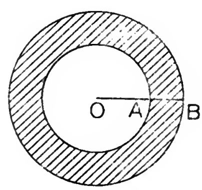
or Length A F = (Number of Degrees) $\times$ .008727 $\times$ Diameter.

Example: Find the length of an arc of 60 degrees in a circle whose diameter is 20.

$$\text{Length} = 60 \times .008727 \times 20 = 10.4724.$$

[11] Circular Ring or Annulus.

The surface lying between two circles, as shown by the shaded part of the figure, is called a *circular ring* or *annulus*.



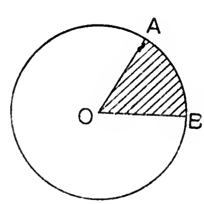
AREA. To find the area of a circular ring, having given the radii of the two circles:

Rule—Find the difference between the areas of the two circles,

$$\text{or Area} = 3.1416\,\overline{OB}^2 - 3.1416\,\overline{OA} = 3.1416\,(\overline{OB} - \overline{OA}^2.)$$

[12] Sector of Circle.

The surface lying between two radii and the corresponding part of the circumference, as shown by the shaded part of the figure, is called the *sector* of a circle, or a circular *sector*.



AREA, A B O. To find the area of the sector of a circle, having given the corresponding number of degrees and the diameter:

Rule—Find the area of the entire circle, divide by 360, and multiply by the number of degrees in the sector;

$$\text{or Area} = \frac{\text{Area of Circle}}{360} \text{ (Number of Degrees in Sector)};$$

or, by proportion, 360° : Number of Degrees in Sector : : Area of Circle : Area of Sector;

or, otherwise, find half the product of the arc by the radius,

$$\text{or Area} = \tfrac{1}{2} (O\ A \times A\ B).$$

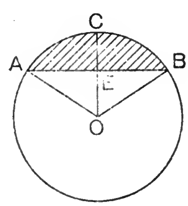
Example: Find the area of a sector of 60 degrees in a circle whose radius is 10.

$$\text{Area of entire circle} = 78.54,$$

$$\text{hence Area} = \frac{78.54}{360} \times 60 = \frac{78.54}{6} = 13.09.$$

[13] Segment of Circle.

A line, such as A B, cutting across a circle is called a *chord.* A part of the surface of a circle between a chord and the circumference, as shown by the shaded part of the figure, is called a *segment of a circle.*



AREA, A C B. To find the area of the segment of a circle, having given the angle, A O B, and the diameter or radius of the circle:

Rule—Find the area of the sector, O A C B, as in [12], and of the triangle O A B, as in [5], and subtract the latter from the former,

$$\text{or Area} = \tfrac{1}{2} (A\ C\ B \times O\ A - O\ E \times A\ B).$$

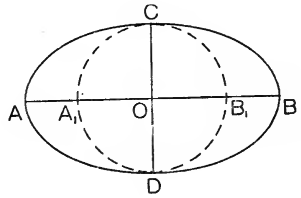
Example: O A = 10, O E = 5, A B = 17.32, A C B = 20.944,

$$\text{then Area} = \tfrac{1}{2} (20.944 \times 10 - 5 \times 17.32) = 61.42.$$

[14] Ellipse.

If the surface of a circle, as shown by the dotted line, be uniformly stretched in one direction (horizontal in the figure) until the diameter, A_1 B_1, becomes equal to A B, the circumference will be changed into a curved line, A C B D, and the figure thus formed is called an *ellipse.* The two lines, A B and C D, are called the *diameters* of the ellipse.

AREA, A C B D. To find the area of an ellipse, having given the two diameters:



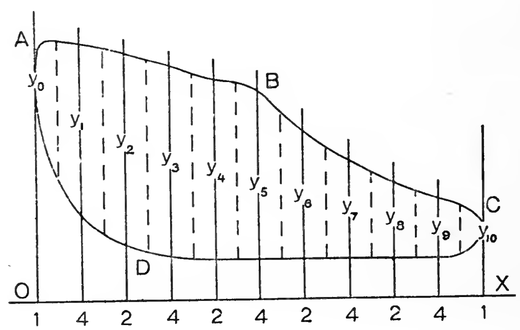
Rule—Multiply the product of the two half diameters by 3.1416,

$$\text{or Area} = 3.1416 \times \text{O C} \times \text{O B}.$$

Example: O C = 5, O B = 8. Then area = 3.1416 × 5 × 8 = 125.664.

[15] Figures With an Irregular Contour.

To find AREA, A B C D, representing, for example, an indicator card. This cannot be done with absolute exactness, but there are a number of rules for finding the approximate area as closely as we may desire.



Divide the base O X into any appropriate number of intervals, usually 10 for an indicator card, and draw lines across the card as shown.

Rule (1)—(*Trapezoidal*). Measure the successive ordinates or breadths on the full line and from their sum subtract one-half the sum of the end ordinates. Multiply the remainder by the length of the interval or by O X ÷ 10, and the product is the area desired;

or calling the breadths y_0, y_1, y_2, etc., we have by formula in this case.

$$\text{Area} = \frac{O\,X}{10}\left(\tfrac{1}{2}y_0 + y_1 + y_2 + y_3 + y_4 + y_5 + y_6 + y_7 + y_8 + y_9 + \tfrac{1}{2}y_{10}\right).$$

As a slightly different and preferable mode of procedure with this rule we may measure the breadths midway between the lines of division as indicated by the dotted lines. Their sum, without modification, is then multiplied by the length of the interval as before.

Any number of spaces as desired may be employed with this rule.

Rule (2)—(*Simpson's or Parabolic*). Measure the ordinates as before.

Take: Once the first and last. Four times every other one beginning with the second. Twice the remaining.

(These multipliers are shown in the figure below the line O X.)

Add the products and multiply by one-third the interval, or in this case by O X ÷ 30; or by formula in this case:

$$\text{Area} = \frac{O\,X}{30}\left(y_0 + 4y_1 + 2y_2 + 4y_3 + 2y_4 + 4y_5 + 2y_6 + 4y_7 + 2y_8 + 4y_9 + y_{10}\right).$$

With this rule the number of spaces must be even.

Rule (3)—(*Durand's*). Measure the ordinates as before. To their sum add one-twelfth the sum of those next the end and subtract seven-twelfths of those at the end. Multiply the result by the interval and the product is the area desired;
or by formula in this case:

$$\text{Area} = \frac{O\,X}{10}\left\{\begin{array}{l} y_0+y_1+y_2+y_3+y_4+y_5+y_6+y_7+y_8+y_9+y_{10} \\ +\frac{1}{12}(y_1+y_9) \\ -\frac{7}{12}(y_0+y_{10} \end{array}\right\}$$

With this rule any number of spaces as desired may be employed.

The measurement and addition of ordinates as in rules (1) and (3) may be quickly effected by means of a strip of paper on

which their lengths are marked off directly from the card and without the use of a scale, joining the end of one to the beginning of the next and thus effecting mechanically the addition desired.

To a reduced scale the strip when marked would resemble the figure, the ordinates being as indicated on the margin. The sum total is then directly found by means of a scale.

Example: Suppose that the ordinates of an irregular area at one-half inch intervals are found by measurement to be as follows in column (1):

(1) Ordinates.		(2) Multipliers.	(3) Products.
y_0	.44	1	.44
y_1	1.42	4	5.68
y_2	1.61	2	3.22
y_3	1.56	4	6 24
y_4	1.51	2	3.02
y_5	1.46	4	5.84
y_6	1.20	2	2.40
y_7	.95	4	3.80
y_8	.71	2	1.42
y_9	.49	4	1.96
y_{10}	.18	1	.18
Sums	11.53		34.20

We then have, according to rule (1):

Sum of ordinates = 11.53
½ sum of ends.. = .31

Difference.... = 11.22
Interval .... = .5

Area = Product = 5.61 square inches.

According to rule (2), we have the multipliers and products as given in columns (2) and (3).

We then have:

$$\text{Area} = \tfrac{1}{3} \times .5 \times 34.20 = 5.70 \text{ square inches.}$$

According to rule (3), we have from column (1):

Sum of ordinates...... = 11.53
$\frac{1}{12}$ sum of next to ends = .16

Sum............... = 11.67
$\frac{7}{12}$ sum of ends........ = .36

Difference........... = 11.31

$$\text{Area} = .5 \times 11.31 = 5.66 \text{ square inches.}$$

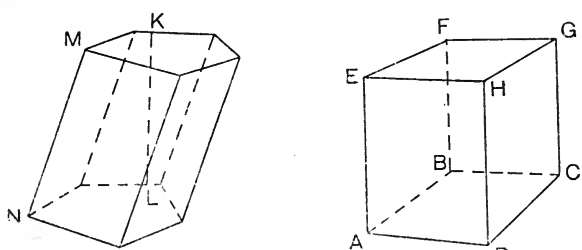
Areas of irregular figures, such as indicator cards, etc., may also be found by an instrument called the *planimeter*. In this instrument a pointer is traced around the contour of the figure, while the area is read off from a wheel, which is given appropriate motion by the movement of the pointer. Such instruments, with instructions for use, may usually be obtained from the makers of steam-engine indicators or from dealers in mathematical instruments.

[16] Prism.

A *prism* is a solid body, bounded by two equal and parallel ends and by three or more sides or faces, forming at their junction a like number of parallel edges.

When the sides are perpendicular to the ends, and are therefore all rectangles, the solid is known as a *right prism*.

When the sides and ends are all square, the solid is known as a *cube*.



a *b*

Surface. To find the surface of any prism, having given the figure complete:

Rule—Find the area of the base (or top) by such of the preceding rules as may be appropriate. For the side or *lateral* surface, multiply the *perimeter* or boundary of the base by the perpendicular or shortest distance between any two corresponding lines of the base and top (as K L in Fig. *a*).

Volume. To find the volume of any prism, having given the base and altitude:

Rule—Multiply the area of the base or end by the altitude or perpendicular distance between the two ends.

Right Prism. In a right prism the preceding general rules hold, but the lateral faces are all rectangles, and the perpendicular distance between their ends, as K L; the length of an edge, as M N, and the altitude or perpendicular distance between the ends, all three become equal.

RIGHT PRISM, WITH RECTANGULAR BASE. In a solid of this character (see Fig. *b*) the preceding rules still hold, but the sides and ends are all rectangles, and the rules may be simplified as follows:

For the LATERAL SURFACE:

Rule—Multiply the perimeter of the base by the altitude,

or Lateral Surface = Length A B C D A $\times$ A E.

For the VOLUME:

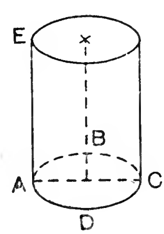
Rule—Multiply together the length and breadth of base and the product by the altitude,

or Volume = A B $\times$ A D $\times$ A E.

Example: A B = 8, A D = 6, A E = 10. Then lateral surface = $(8 + 6 + 8 + 6) \times 10 = 28 \times 10 = 280$, and volume = $8 \times 6 \times 10 = 480$.

[17] Cylinder.

A solid with a circular cross section of constant size is called a *cylinder*. If the center of the top is vertically over the center of the base, the solid is called a *right cylinder*.



LATERAL SURFACE OF RIGHT CYLINDER. To find the lateral surface of a right cylinder, having given the diameter of base and the altitude:

Rule—Multiply the circumference of the base by the altitude, or Lateral Surface = Circumference A B C D $\times$ A E = 3.1416 $\times$ A C $\times$ A E.

Example: A C = 10, A E = 20. Then lateral surface = $3.1416 \times 10 \times 20 = 628.32$.

VOLUME OF RIGHT CYLINDER. To find the volume of a right cylinder, having given the base and altitude:

Rule—Multiply the area of the base by the altitude,

or Volume = Area A B C D $\times$ A E = $.7854\ \overline{AC}^2 \times AE$.

Example: A C = 10, A E = 20. Then volume = .7854 × 100 × 20 = 1,570.8.

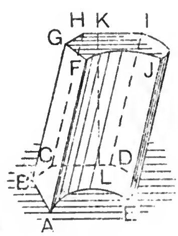
[18] Any Solid With a Constant Section Parallel to the Base, Either Right or Oblique.

Such a solid is the general case of which the prism and cylinder are but special examples. The rules will therefore be similar to those of [16] and [17].

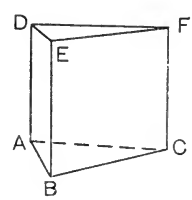
SURFACE. To find the surface of such a solid, having given the figure complete:

Rule—Find the area of the base (or top) by such of the preceding methods as may be appropriate. For the side or lateral surface, multiply the *perimeter* or boundary of the base by the perpendicular or shortest distance between any two corresponding lines of the base and top (as K L in figure *a*):

or Lateral Surface = Length A B C D E A × K L.



a



b

VOLUME. To find the volume of such a solid, having given the figure complete:

Rule—Multiply the area of the base by the perpendicular distance between the base and the top.

or Volume = Area of Base × Vertical Altitude.

Example: Area of base = 120, altitude = 40. Then volume = 40 × 120 = 4,800.

[19] Wedge.

A right prism with a triangular base, as in *b* above, having two sides equal, as A C and B C, is called a *wedge*.

To find the SURFACE or VOLUME, use the same rules as in [16].

[20] Right Pyramid.

A solid, bounded by a base and by triangular sides meeting in a point or *apex*, as P, is called a *pyramid*. If the base is a regular polygon [8] and the apex is vertically over the center of the base, the solid is called a *right pyramid* (see figure *a* below).

LATERAL SURFACE. To find the lateral surface of a right pyramid, having given the figure complete:

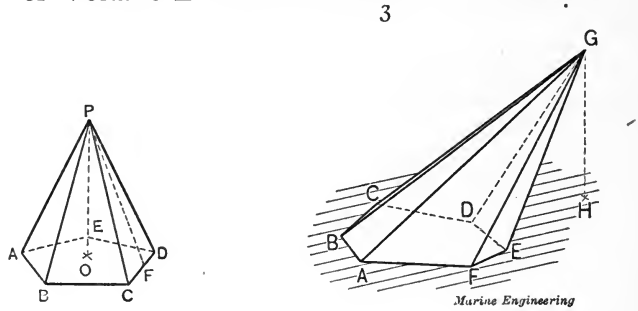
Rule—Take one-half the product of the perimeter or boundary of the base by the perpendicular or shortest distance from the apex to one of the sides of the base, as P F,

$$\text{or Lateral Surface} = \frac{\text{Length } ABCDEA \times PF}{2}.$$

VOLUME. To find the volume of a right pyramid, having given the base and altitude:

Rule—Take one-third of the product of the area of the base by the altitude, as P O,

$$\text{or Volume} = \frac{(\text{Area of Base}) \times (\text{Altitude})}{3}.$$



a *b*

Example: Area of base = 180, altitude = 16. Then volume $= \frac{180 \times 16}{3} = 960.$

[21] General Pyramid.

For definition see [20], also Fig. *b* above.

LATERAL SURFACE. To find the lateral surface of any pyramid, having given the figure complete:

Rule—The surface will consist of a series of triangles, similar or not, according to the nature of the pyramid. These must be computed according to the rules for triangles and the results added.

VOLUME. To find the volume of any pyramid, having given the base and altitude:

Rule—Take one-third the product of the area of the base by the vertical altitude, as G H,

$$\text{or Volume} = \frac{(\text{Area of Base}) \times G\ H}{3}.$$

Example: Area of base $= 48$, $G\ H = 18$. Then volume $= \frac{48 \times 18}{3} = 288$.

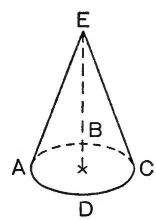
[22] Right Circular Cone.

Any solid having a base with curved or irregular boundary, an apex, and straight sides, is called a *cone* in general. If the base is a circle and the apex is vertically over the center, the solid is called a *right circular cone* (see figure).

LATERAL SURFACE. To find the lateral surface of a right circular cone, having given the base and the slant height:

Rule—Take one-half the product of the circumference of the base by the slant height,

$$\text{or lateral surface} = \frac{\text{Circumference } A\ B\ C\ D\ A \times A\ E}{2}.$$



Example: Diameter $= 10$, $A\ E = 20$. Then surface $= \frac{3.1416 \times 10 \times 20}{2} = 314.16$.

VOLUME. To find the volume of a cone, having given the base and altitude:

Rule—Take one-third the product of the area of the base by the altitude,

$$\text{or volume} = \frac{\text{Area of Base } A\ B\ C\ D \times E\ F}{3}.$$

Example: Diameter $= 8$, $E\ F = 15$. Then volume $= \frac{.7854 \times 64 \times 15}{3} = 251.328$.

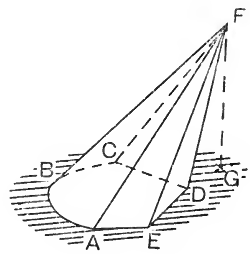
[23] General Cone.

For definition see [22].

VOLUME. To find the volume of any cone, having given the base and vertical altitude:

Rule—Take one-third the product of the area of the base by the vertical altitude, as $F\ G$,

$$\text{or volume} = \frac{\text{Area of base} \times F\ G}{3}.$$



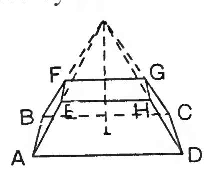
Example: Area of base = 240, $F\ G$ = 40. Then volume = $\frac{240 \times 40}{3} = 3{,}200$.

[24] Frustum of Right Pyramid.

The solid contained between the base of a pyramid and a parallel plane, as $E\ F\ G\ H$, is called the *frustum* of a pyramid.

LATERAL SURFACE. To find the lateral surface of a frustum of a right pyramid, having given the figure complete:

Rule—Add the *perimeters* or boundaries of the base and top, and multiply the sum by one-half the perpendicular or shortest distance between two corresponding lines of the base and top, which we may denote by h.



$$\text{or lat. surface} = \frac{(\text{length } A\ B\ C\ D + \text{length } E\ F\ G\ H) \times h}{2}$$

Example: $A\ B = 10$, $A\ D = 12$, $E\ F = 6$, $E\ H = 7.2$, $h = 5$. Then surface $= \frac{(44 + 26.4) \times 5}{2} = 176$.

VOLUME. To find the volume of a frustum of a right pyramid, having given the base and top and vertical distance between them:

Rule—Add together the areas of the base and top and the square root of their product, and multiply the sum by one-third the vertical altitude, which we may denote by k,

$$\text{or vol.} = \frac{(A\,B\,C\,D + E\,F\,G\,H + \sqrt{A\,B\,C\,D \times E\,F\,G\,H}) \times k}{3}$$

Example: Area $A\ B\ C\ D = 100$, area $E\ F\ G\ H = 42$, $k = 12$. Then volume $= \dfrac{(100 + 42 + \sqrt{4{,}200}) \times 12}{3} = 827.2$.

[25] Frustum of General Pyramid, as [21].

To find the LATERAL SURFACE:

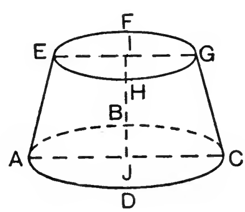
Rule—The surface will consist of a number of trapezoids, similar or not, according to the nature of the frustum. These must be computed according to [4] and the results added.

To find the VOLUME:

Rule—Same as for [24].

[26] Frustum of Right Cone.

The solid contained between the base of a cone and a parallel plane, as $E\ F\ G\ H$, is called the *frustum of a cone*.



LATERAL SURFACE. To find the lateral surface of a frustum of a right cone, having given the base and top and slant height:

Rule—Add the circumference of the base and top, and multiply the sum by one-half the slant height, as $A\ E$,

$$\text{or lat. surface} = \frac{(\text{Cir. } A\,B\,C\,D + \text{Cir. } E\,F\,G\,H) \times A\,E}{2}$$

Example: Diameter $A\ C = 12$, diameter $E\ G = 10$, $A\ E = 8$.

$$\text{Then surface} = \frac{(3.1416 \times 12 + 3.1416 \times 10) \times 8}{2} = 276.47.$$

VOLUME. To find the volume of a frustum of a right cone, having given the base and top and vertical altitude:

Rule—Same as for [24].

Or in slightly different form, we have the following:

Rule—Add together the square of the upper diameter, the square of the lower diameter, and their product, and multiply the sum by the vertical altitude and by the number .2618,

$$\text{or volume} = .2618\ I\ J\ (\overline{E\ G}^2 + \overline{A\ C}^2 + E\ G \times A\ C).$$

Example: E $G = 6$, $A\ C = 8$, $I\ J = 4$. Then volume $= .2618 \times 4\ (36 + 64 + 48) = 155$.

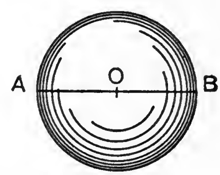
[27] Frustum of General Cone, as [23].

To find the VOLUME:

Rule—Same as for [24].

[28] Sphere.

A solid inclosed by a curved surface, every point of which is equally distant from a point within called the center, is called a *sphere*. The distance, $A\ B$, from one side to the other through the center is called the *diameter*. The distance, $A\ O$, from the



surface to the center is called the *radius*, and is plainly one-half the diameter.

SURFACE. To find the surface of a sphere, having given the diameter:

Rule—Square the diameter and multiply by 3.1416,

$$\text{or Surface} = 3.1416\ (\text{Diameter})^2.$$

Example: Diameter $= 20$. Then surface $= 3.1416 \times 400 = 1{,}256.64$.

VOLUME. To find the volume of a sphere, having given the diameter or radius:

Rule—Multiply the cube of the radius by 4.1888, or multiply the cube of the diameter by .5236,

$$\text{or volume} = 4.1888\ \overline{A\ O}^3.$$

$$\text{or volume} = .5236\ \overline{A\ B}^3.$$

[29] Volumes of Irregular Shape.

Volumes of irregular shape in which the areas of a series of equally-spaced sections may be found.

Rule—Find the areas of a series of equally-spaced cross sections and treat them by the rules given in [15], using areas for ordinates. The result will give the volume desired.

Examples:

(1) Find the volume of an irregular box, 12 feet long; area of one end, 6 square feet; of the other, 15 square feet, and of a section midway between, 10 square feet. The interval is 6 feet.

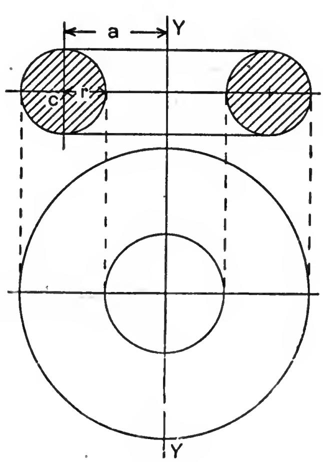
Then with Simpson's rule,

Volume $= 2 \times (6 + 4 \times 10 + 15) = 122$ cubic feet.

(2) Find the volume of a coal bunker, 20 feet long, having cross sections every 4 feet, as follows:

Taking rule (3), in [15], we find the result as below:

	Square Feet.	
A_0 =	40	297
A_1 =	46	8.3
A_2 =	50	305.3
A_3 =	52	55.4
A_4 =	54	249.9
A_5 =	55	4
Sum =	297	999.6; or, say, 1,000 cubic feet.



[30] Volume Generated by Any Area Revolving About an Axis.

To find the surface or volume of such a body, the so-called "Rules of Pappus" are most readily applicable. These may be illustrated by the example of the *Torus* or *Ring* as in the figure.

SURFACE. To find the surface of a ring, having given the necessary dimensions:

Rule—Multiply the length of the generating line by the length of the path followed by its center of gravity.

Example, as in the figure.

The length of the generating line is $2\pi r$.

The length of the path of the center, C, is $2\pi a$.

The surface is $\therefore 2\pi r \times 2\pi a = 4\pi^2 a r$

VOLUME. To find the volume of a ring, having given the necessary dimensions:

Rule—Multiply the generating area by the length of the path traveled by its center of gravity.

Example, as in the figure.

The generating area is πr^2.

The length of the path of the center, C, is $2\pi a$.

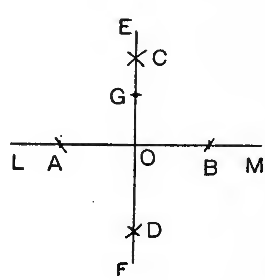
The volume is $\therefore 2\pi^2 a r^2$.

The same general rules apply, no matter what the form of the generating area, and they will often be found of use in solving problems not readily treated in any other manner.

Sec. 10. PROBLEMS IN GEOMETRY.

[1] At Any Point in a Straight Line to Erect a Perpendicular.

Let O be the given point in the line $L\ M$. Then take points A and B such that $O\ A = O\ B$. From A and B as centers with



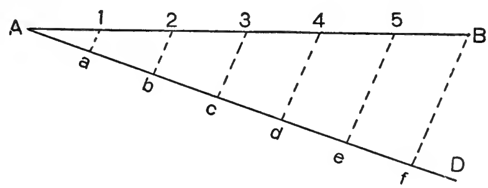
any radius greater than $A\ O$ or $O\ B$, describe arcs cutting in C or D, or if preferred, in both. Then a line drawn through O and C or O and D will be perpendicular to $L\ M$, or the line may be drawn through C and D, in which case it will also contain O and be perpendicular to $L\ M$ as before.

[2] To Bisect* the Distance Between Two Points.

In the figure for problem [1] let A and B be the two points. Then finding points C and D as in [1] it is plain that the point O determined by drawing the line $C\ D$ will be the middle point between A and B as desired, and that we shall have $A\ O = O\ B$.

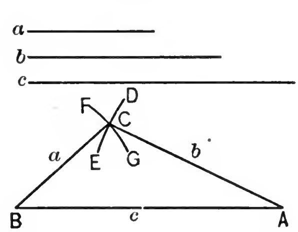
[3] To Find the Center from Which to Pass an Arc of Given Radius Through Two Given Points.

In the figure for problem [1] let A and B be the points. Then finding points C and D as in [1] it is plain that any point in the indefinite line $E\ F$ will be at equal distances from A and B. Hence from A or B as a center and with the given radius cut the line $E\ F$ in a point as G. This is the point desired.



[4] To Divide a Given Line Into a Given Number of Equal Parts.

Let $A\ B$ denote the given line. Draw a line $A\ D$ at a small angle with $A\ B$, and lay off upon it as many equal divisions $A\ a$, $a\ b$, $b\ c$, etc., as it is desired $A\ B$ shall have. These divisions should be so chosen that the total length $A\ f$ shall not widely differ from $A\ B$. Next draw $B\ f$ and then a series of parallels through the points of division $a\ b\ c$, etc. The points where these lines cut $A\ B$ will give the points of division desired.



[5] To Construct a Triangle, Having Given the Three Sides.

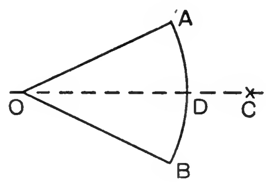
Let $a\ b\ c$ denote the sides. Take $A\ B = c$, and from A as

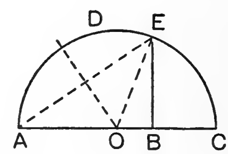
* To *bisect* a geometrical quantity means to divide it into two equal parts.

center and with b as radius, describe an arc $D\ E$, and similarly from B as center and with a as radius describe an arc $F\ G$. These arcs intersect in C, and drawing lines $A\ C$, $B\ C$, the construction is completed.

[6] To Bisect a Given Arc or Angle.

Let $A\ O\ B$ denote the angle and $A\ B$ the arc. From A and B as centers, and with any radius greater than half the distance between A and B, describe arcs intersecting in some point C. Then a line $O\ C$ will bisect the angle $A\ O\ B$ and at D the arc $A\ B$.





[7] To Construct a Mean Proportional* Between Two Given Lines.

Let the two lines be denoted by $A\ B$ and $B\ C$ placed end to end. Find the center O of the line $A\ C$, and describe a semi-circle $A\ D\ C$. Draw $B\ E$ at right angles to $A\ C$. Then $B\ E$ is the desired mean proportional, and we have:

$$A\ B : B\ E :: B\ E : B\ C$$
$$\text{or, } A\ B \times B\ C = \overline{B\ E}^2$$

[8] To Construct a Fourth Proportional† to Three Given Lines.

Let a, b and c denote the three lines, and let the desired proportion be:

$$a : b :: c : (\ )$$

---

* A mean proportional between two quantities a and c is a third quantity b, such that we have:

$$a : b :: b : c$$
$$\text{or, } b^2 = a\,c$$
$$\text{or, } b = \sqrt{a\,c}.$$

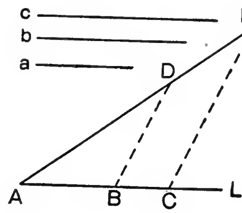
See also Sec. 6.

† A fourth proportional to three quantities a, b and c is a quantity d, such that we have:

$$a : b :: c : d.$$

See also Sec. 6.

Lay off $A\ B = a$ and $A\ C =$ b. Then at any convenient angle lay off $A\ K$ and take on it a distance $A\ D = c$. Draw $B\ D$ and $C\ E$ parallel to it. Then $A\ E$ is the fourth term desired and we have:



$$A\ B : A\ C :: A\ D : A\ E$$
$$\text{or, } A\ B \times A\ E = A\ C \times A\ D$$

[9] To Construct a Square Equivalent in Area to a Given Rectangle.

Find by problem [7] a mean proportional between the sides of the rectangle, and this will be the side of the square desired.

[10] To Construct a Square Equivalent in Area to a Given Triangle.

Find by problem [7] a mean proportional between the base and half the altitude, or between the altitude and half the base. This will be the side of the square desired.

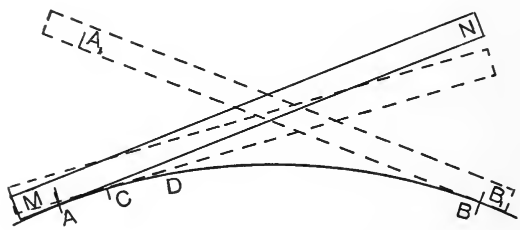
[11] With One Given Side, to Construct a Rectangle Equivalent to a Square.

This is equivalent in [7] to having given $A\ B$ the side of the rectangle and $B\ E$ the side of the square. Find by the use of the construction in problem [1] a point O on $A\ B$ at equal distances from A and E and describe the semi-circle. Then $B\ C$ is the remaining side of the rectangle desired.

[12] To Find the Length of an Arc of a Curve.

Let $A\ C\ D\ B$ denote the given arc. Take a strip of paper $M\ N$ and lay with an edge just neatly tangent to the curve at A. Mark a point opposite A on the strip. Then, placing the pencil at C, a point near where the curve and edge of the paper strip begin to separate, bear down slightly and rotate the paper about C as a center until the edge is tangent at C or at a point slightly beyond. Then move the pencil along to a point D and repeat, and so on until the strip has been thus rolled along the curve to

B. The distance A_1 B_1, between the point opposite *A* on the strip at the start and the point opposite *B* at the end, will be very close to the true length of the curve. A little experience will enable the points *C D*, etc., to be so chosen that the error will be very small. A check on the operation may be obtained by reversing

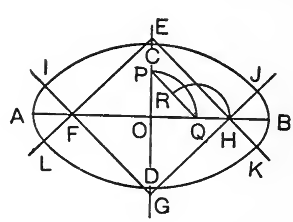


the process and rolling the paper back to the original position. If the point A_1 comes again to A' it shows that no slip has been made, and the distance found may be accepted as a very close approximation to the true length of the curve.

[13] To Construct an Ellipse.

Of the many methods available, three are given as follows:

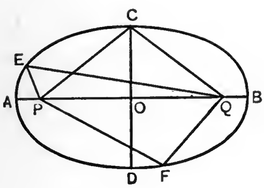
(1) Given the two diameters *A B* and *C D*. Take *B Q* equal to *O C* and then *O P* equal to *O Q*. Draw *P Q* and find *R* its middle point. Then take *Q H* = *Q R*, and *O E*, *O F*, *O G* all

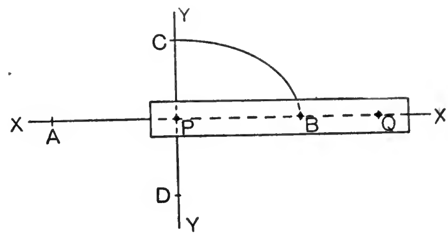


equal *O H*. Through *E*, *F*, *G* and *H* draw lines as shown. Then with *H* and *F* as centers draw arcs *J K* and *I L*, and with *E* and *G* as centers draw arcs *L K* and *I J*. These arcs join and complete the contour. While this method is only approximate and does not give a true ellipse, it answers very well for draughting purposes where a good representation of an ellipse is all that is desired.

(2) This method is exact in principle. Given *A B* and *C D* the two diameters as before. With *C* as center and *C Q* = *O B* as a radius find the points *P Q*. These are the *foci* of the ellipse.

Then adjust a thread $P\ C\ Q$ secured at P and Q and of a length $P\ C + C\ Q = A\ B$. A pencil carried around in the bight of this thread, as shown in different positions at E, C, F, will trace the ellipse desired.

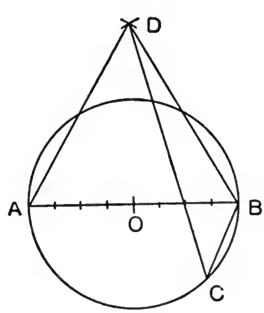




(3) This method is also exact in principle. Given $A\ B$ and $C\ D$ the two diameters as before. Prepare a strip of cardboard or thin wood with holes P, B, Q, such that $P\ B$ equals one-half $A\ B$ and $B\ Q$ equals one-half $C\ D$. Then move the *trammel*, as it is called, so that P shall always move on the vertical $Y\ Y$ and Q on the horizontal $X\ X$. The point B will then trace the ellipse desired. If points on the curve only are required, this method is readily applied.

[14] To Construct Any Regular Polygon (Approximate).

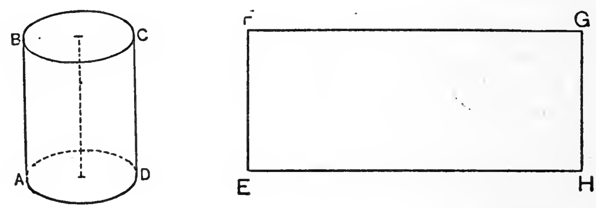
Let $A\ B$ denote any diameter of the given circle within which the polygon is to be inscribed. Divide $A\ B$ into as many parts as there are to be sides in the polygon. From A and B as



centers and radius $A\ B$ describe arcs cutting in D. From D draw a line $D\ C$ through the second of the points of division. Then $B\ C$ is the side of the polygon desired within a very small error. For the square or hexagon, or when the number of sides is 4 or 6, the construction is exact.

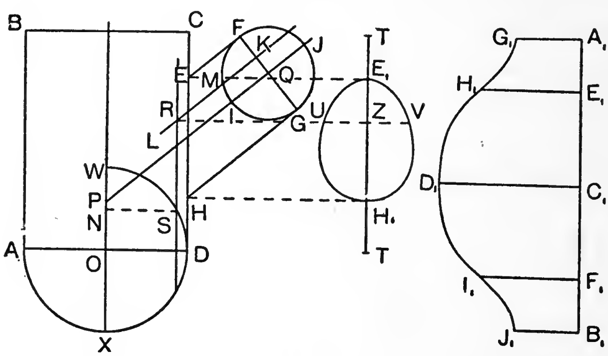
[15] To Develop the Surface of a Cylinder.

Let $A\ B\ C\ D$ denote the cylinder. Lay off $E\ F$ = the altitude and $E\ H$ = the circumference of the base, $= \pi \times$ diam. $= 3.1416 \times$ diam. Then the rectangle $E\ F\ G\ H$ represents the development desired.



[16] To Develop the Surface of a Cylinder Which is Intersected by Another Cylinder, the Two Axes Being in the Same Plane.

The developed form of the intersection is the only part requiring special notice. Let $A\ B\ C\ D$ and $E\ F\ G\ H$ denote the two cylinders. Draw any line $T\ T$ to denote the element $C\ D$ in the developed surface of $A\ B\ C\ D$. Then the developed form of the intersection will be symmetrical about $T\ T$. Project E and



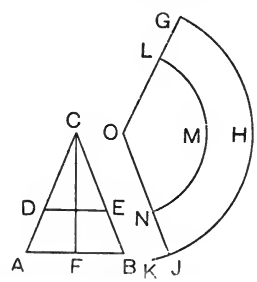
H over to E_1 and H_1 for the top and bottom of the curve. Then to find intermediate points proceed as follows: Draw any line $K\ L$ parallel to $P\ Q$ and denoting the edge of a plane perpendicular to the paper and cutting both cylinders. On $F\ G$ as diameter describe the circle as shown, and on $A\ D$ the semi-circle $W\ D\ X$. Make $N\ O$ equal to $M\ K$ and project over to S, thence up to R and then over to $T\ T$. Rectify the arc $S\ D$ and lay off $Z\ U$ and $Z\ V$ each equal to the rectified length. Then will U and V be points on the curve as desired. Other points may be found in a similar manner and the curve filled in.

To develop the form of the smaller cylinder proceed as follows:

Let $C_1 D_1$ denote in the development the element $H G$. Lay off $A_1 B_1 =$ the circumference $I F J G$. Then for the points corresponding to the plane $K L$ take $C_1 E_1 = C_1 F_1 =$ the developed length of the arc $G M$. Then draw $E_1 H_1$ and $F_1 I_1$ each equal to $K R$. This will give two points, H_1 and I_1, and others may be found similarly and the curve filled in as shown.

[17] To Develop the Surface of a Cone.

Let $A B C$ denote the cone. With $A C$ as radius and any point O as center, draw an arc, $G H K$. Then lay off the circumference of the base $A B$ ($= 3.1416 \times A B$) on a strip of paper, and lay off this length by rolling as in problem [12] from G to some point J. Then the arc $G H I =$ circumference of $A B$ and the sector $O G H J$ is the developed surface of the cone.

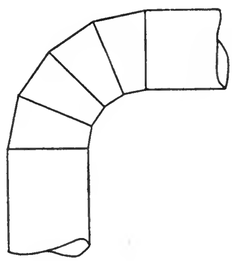


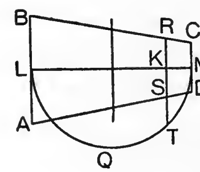
[18] To Develop the Surface of the Frustum of a Cone.

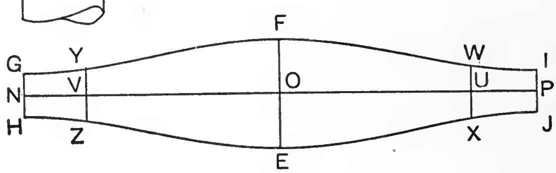
Referring to the figure for the preceding problem, let $A D E B$ denote the frustum. Then, after proceeding as in that problem, take next a radius $O L = C D$ and describe the arc $L M N$. Then will the sector $O L M N$ represent the surface of the cone $C D E$, while the strip $L G H J N M$ represents that of the frustum.

[19] To Develop the Segments of an Elbow.

These are portions of a cylinder cut by oblique planes. Let $A B C D$ denote such a segment. Draw $L M$ perpendicular to $A B$ and construct the semi-circle $L Q M$. $L M$ may be placed at any convenient location between $B C$ and $A D$. In the development let $E F$ denote the element $A B$. Make $F O = B L$ and then draw perpendicular to $E F$ lines $N O = O P =$ each to the semi-circumference $L\ Q\ M$. Then $N\ P$ is the development of $L\ M$. Lay off $N\ G = P\ I = M\ C$ as shown. Then to find intermediate points on $G\ F\ I$ take any line $R\ S$ and project down to T. Develop $L\ Q\ T$ and lay off the developed length at $O\ U$ and $O\ V$. Then make $V\ Y$ and $U\ W$ each equal to $K\ R$, and Y and







W will be points on the development of $B\ C$. Other points may be found in a similar manner and the development of $B\ C$ completed as shown by the curve $G\ F\ I$. The curve $H\ E\ J$ as the development of $A\ D$ may be found in an entirely similar manner, and if $B\ C$ and $A\ D$ are equally inclined to the elements of the cylinder, $L\ M$ will naturally be located midway between them, and $H\ E\ J$ will be symmetrical with $G\ F\ I$ about $N\ P$, and both may thus be found at the same time by laying off above and below $N\ P$ the distances determined as above shown.

Sec. 11. PHYSICS.

[1] Acceleration Due to Gravity.

In engineering computations there frequently enters a quantity known as the *acceleration of gravity* or the *acceleration due to gravity*, and denoted by the symbol g. This is the change per second which the gravity or attraction of the earth is able to bring about in a freely falling body. For engineering purposes its value is usually taken at 32.16 or 32.2.

[2] Specific Gravity.

The specific gravity of a given substance is the relation between the weights of equal volumes of the given substance, and of some standard substance, usually water. Thus a specific gravity of 8 means that, volume for volume, the given substance is 8 times as heavy as water.

[3] Heat Unit.

Heat is measured in terms of a unit defined as the amount of heat required to raise 1 lb. of water 1 deg. in temperature at 62 deg. Fahrenheit, or from 62 deg. to 63 deg.

[4] Specific Heat.

The specific heat of a substance is the relation between the amount of heat required to raise it 1 deg. at the given temperature and under given conditions as to pressure or volume, and the unit of heat as just defined. Thus a specific heat of .32 means that under the given conditions it will require to raise the temperature 1 deg., .32 of the heat necessary to raise 1 lb. of water from 62 deg. to 63 deg.

[5] Expansion of Metals.

Nearly all substances expand with the addition of heat, and usually with nearly equal amounts per degree rise of temperature, especially where the substance is not near its melting or boiling point. The following table gives the *coefficient of linear* or *length expansion* for various substances. This is the expansion in unit length for 1 deg. Fahr. rise of temperature.

Substance.	Coef.	Substance.	Coef.
Aluminum	.0000123	Iron, cast.	.0000056
Brass, cast	.0000096	Iron, wrought	.0000065
Brass, drawn	.0000105	Lead	.0000157
Brick	.0000031	Mercury	.0000333
Bronze	.0000099	Steel, cast	.0000064
Bismuth	.0000098	Steel, wrought	.0000069
Concrete	.0000080	Tin	.0000116
Copper	.0000089	Zinc	.0000141
Glass	.0000045		

To find the expansion in the length of any bar for any given rise in temperature we have the following:

Rule—Multiply the coefficient taken from the table by the number of degrees, and this by the length of the bar, and the product is the expansion desired.

Example: What is the expansion of a steel bar 20 ft. long between 60 deg. and 350 deg. Fahr.?

Ans.: $.0000069 \times 290 \times 20 = .04$ ft. $= .48$ in.

The coefficient for area or surface expansion is taken twice that for linear expansion, and that for cubic or volume expansion is taken three times that for linear expansion.

Example: What is the increase in area between 60 deg. and 300 deg. Fahr. in a copper sheet having an area of 54 sq. ft.?

Ans.: $.0000178 \times 240 \times 54 = .231$ sq. ft. $= 33.3$ sq. in.

What is the increase in volume between 100 deg. and 200 deg. Fahr. in a piece of brass having a volume of 2·1-2 cu. ft.?

Ans.: $.0000288 \times 100 \times 2.5 = .0072$ cu. ft. $= 12.44$ cu. in.

Sec. 12. MECHANICS.

[1] It is a general law of nature that all bodies tend to remain unchanged as regards their condition of rest or relative motion. A body at rest does not move unless caused to do so by some outside agency. A body in motion continues to move until it is brought to rest by outside agencies such as friction, resistance of the air, or of water, etc.

[2] Force.

Any agency which changes or tends to change the condition of a body as regards its rest or relative motion is called a *force*. For engineering purposes force is measured by the *pound* or *ton* as unit. In marine engineering the ton, unless otherwise stated, is usually of 2,240 lb.

[3] Specification of a Force.

A force has three characteristics or particulars:

(1) Its line of direction, as north and south.

(2) The way it acts in that line, as north.

(3) Its magnitude.

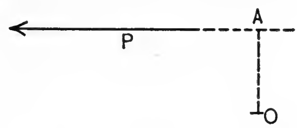
A force may therefore be completely represented by a line *A B* of length to represent the magnitude, and drawn in the direction of the line of action of the force.

Thus the force *A B* would mean a force represented in amount according to some scale by the length *A B*, and acting along the line *A B from A to B*. The direction of action is also frequently denoted by an arrow point, thus:

[4] Moment of a Force.

This is the product of the magnitude or measure of the force multiplied by the perpendicular distance from its line of action to a given reference point.

Thus in the figure let the full line represent a force, P denote



its measure, and O the reference point. Then $P \times O\,A$ is the moment of the force P about the point O. In the term *moment of a force* a point of reference is therefore always implied.

[5] Resultant.

The resultant of a system of forces two or more in number with their lines of action all meeting at a common point, is the single force which represents the combined action or result of the system.

[6] Work.

Work is done when a force (or resistance, as it may be called in such case) is overcome; as for example when a weight is lifted or a ship is forced through the water. Work is measured by the product of the resistance by the distance through which it is overcome. The two essential factors of work are therefore *force* or *resistance* on the one hand, and *motion* or *distance* on the other. The unit of work is the *foot pound* or the work done in raising one pound weight one foot in height. Thus if 16 lb. be lifted 20 ft. the work done is $20 \times 16 = 320$ ft. lb.

[7] Power.

Work in itself is independent of the time required to do it, and depends simply on *resistance* and *distance*. Power means a certain amount of work performed in a given time. The common unit is the *horse power*, which is 33,000 ft. lb. of work performed in one minute. The added element involved in power should not be forgotten. Thus 33,000 ft. lb. of work performed in 1 hour would not mean one horse power, but only $\frac{1}{60}$ of such amount, while 33,000 ft. lb. of work performed in 1 second would mean 60 horse power. Likewise 550 ft. lb. per second represents one horse power as also 1,980,000 ft. lb. per hour. To find the horse power in any given case we have therefore the following:

Rule—Find the foot pounds of work performed per minute and divide by 33,000.

Example: An engine in one-half hour performs 118,800,000 ft. lb. work. What is the horse power?

$$\text{Horse power} = \frac{118{,}800{,}000}{30 \times 33{,}000,} = 120$$

From the above general expressions for work and power there come two forms of especial interest to the engineer. These relate to the work done by or on a fluid in a cylinder with a moving piston, as in a steam engine or a pump.

Let $A =$ the area of the cylinder in square inches.

$p =$ the average working pressure per square inch.

$L =$ the length of stroke in feet.

$N =$ the number of revolutions or double strokes per minute.

Then $pA =$ the average total pressure or load on the piston. This is the *force* factor.

$2LN =$ the distance per minute assuming the engine or pump to be double acting. This is the *distance* factor.

Then:

Work per mt. $= (pA) \times (2LN)$ ft. lbs. . . . . (a)

or as it is frequently written by changing the order of the factors,

Work per mt. $= 2pLAN$ ft. lbs. . . . . (a_1)

To reduct this to horse power we simply divide by 33,000 and have:

$$\text{Horse power} = \frac{2pLAN}{33{,}000} \quad \ldots\ldots\ldots\ldots (a_2)$$

Now for the second form let us arrange the factors thus:

Work per mt. $= (p) \times (2LNA)$.

Then multiply p by 144 and divide A by the same number. This will not change the value and we shall have:

$$\text{Work per mt.} = (144\,p) \times \left(\frac{2LN\,A}{144}\right)$$

The first factor ($144p$) is the pressure per square foot. Also $A \div 144$ is the area of the piston in square feet while $2LN$ is the distance it moves through per mt. measured in feet. Hence $2LN\,A \div 144$ is the volume swept through per mt. Hence we have for work the following form:

Work per mt. = (pressure per unit area) $\times$ (volume swept through per mt.) . . . . (b)

In this form it must be noted that the unit area and the volume must both refer to the same unit, and since work is measured in foot lbs. this unit must be the foot. Hence we may write more definitely:

Work per mt. = (pressure per square foot) × (volume swept through in cubic feet)............................(b_1)

In a still more general sense when a liquid is moved under pressure we may put *volume moved* or *change of volume* for the second factor and thus write:

Work per mt. = (pressure per square foot) × (volume moved in cubic feet)..................................(b_2)

[8] Energy.

This is the capacity for performing work, and depends on special conditions of motion or location. For convenience energy is considered of two kinds.

Kinetic Energy is the energy possessed by a body in virtue of a condition of motion. As we know, such a body resists an attempt to stop it, and it will overcome a certain resistance through a certain distance before being brought finally to rest. This kind of energy is measured by the formula $E = \frac{W v^2}{2 g}$ where W is the weight in pounds, v is the velocity in feet per second, and g is the acceleration due to gravity or 32.2. Since energy is directly convertible into work it must be really similar in character to work, and we may therefore speak of so many foot pounds of energy just as well as of so many foot pounds of work.

Potential Energy is the energy possessed by a body in virtue of its location or condition relative to the forces acting on it. Thus a weight lifted to the top of a building has potential energy relative to the street because it could do work if allowed to move downward. Similarly a compressed spring or a compressed gas has potential energy because either, if properly allowed to return to the condition toward which it tends, will perform work. Potential energy is measured by the work which could be thus performed, or by its equal the work which must be done upon the body in order to produce the given condition; as for example the work done in lifting the weight to the top of the building, or in compressing the spring or gas. Potential energy is therefore measured directly in foot pounds.

[9] Conservation of Energy.

It is a fact of universal experience that energy can neither be destroyed nor created. The seeming appearance or disappearance of energy or work is always the result of a change of form. Energy may exist in a variety of forms, as (1) Mechanical, (2) Thermal, (3) Electrical, (4) Chemical, and *when there is an increase in any one form there must be a decrease in the other forms of exactly the same total amount,* and likewise *when there is a decrease in any one form there must be an increase in the other forms also of exactly the same total amount.* There may be a like exchange between kinetic and potential energy, the one increasing as the other decreases, and *vice versa.* Thus with mechanical energy if there is no change to other forms we shall find that the sum of the kinetic and potential energies is always the same, and that one increases as the other decreases and *vice versa.* In this view work simply appears as an attendant upon the exchange of energy, or more definitely, as a measure of the exchange. Again if we fix our attention upon one body, its changes of total energy measure the work which it receives or gives out. If its energy increases it has had work done upon it. If its energy decreases it has given out work to some other body.

[10] Statics.

If the forces which act on a body are properly related or balanced, the body remains at rest. The conditions necessary to the realization of this state of rest under the action of forces form the subject of *statics.*

[11] Dynamics.

If the forces are not so related, the body does not remain at rest and motion results. The amount and nature of such motion and its relation to the system of forces form the subject of *dynamics.*

[12] Propositions in Statics.

Following are a few simple propositions in statics given without proof:

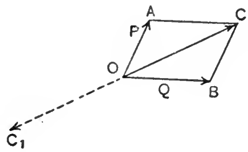
(1) A force may be transferred along its line of action without changing its effect.

(2) Two forces equal and directly opposite will balance or produce equilibrium.

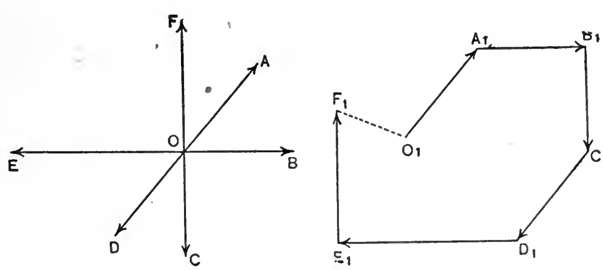
(3) PARALLELOGRAM OF FORCES. If two forces whose lines of action meet in a point O are represented in amount and direc-

tion by the lines $O\ A$ and $O\ B$, then will the resultant of these two forces be represented also in amount and direction by the diagonal $O\ C$ of the parallelogram $O\ A\ C\ B$ erected on $O\ A$ and $O\ B$ as adjacent sides.

(4) A force $O\ C_1$ represented by the diagonal $O\ C$ reversed will balance $O\ C$ or the resultant, and therefore will balance P and Q.



(5) Polygon of Forces. Let there be a system of forces represented as in (*a*). In (*b*) starting at any point O_1 and $O_1\ A_1$ equal and parallel to $O\ A$. Then from A_1 as starting point draw $A_1\ B_1$ equal and parallel to $O\ B$, and so on, drawing finally $E_1\ F_1$ equal and parallel to $O\ F$. Then will the closing line $O_1\ F_1$ in direction and amount represent the resultant of the system of forces, while $O_1\ F_1$ reversed, or $F_1\ O_1$ will represent similarly the balancing force of the system—that is, the single force which will balance the system and with it produce equilibrium. In the

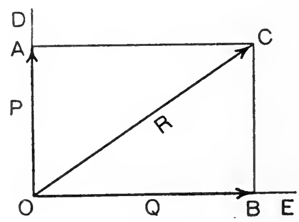


a *b*

construction in (*b*) the order in which the forces are taken is indifferent, but we have here supposed them taken in regular order to the right, beginning with $O\ A$ and ending with $O\ F$.

It is readily seen that this proposition is a generalization of (3) extended to cover the case of any system of forces. It follows from this proposition that if any system of forces may be represented as in (*b*) by the sides of a completely closed polygon, then such system will produce equilibrium, for the resultant in such case would be zero. Again in such case any force may be considered as the *balancing force* for the system composed of all the others, and any force reversed may be considered as the *resultant* of the system composed of all the others.

(6) COMPONENTS. In the figure the two forces P and Q are at right angles. In such case they are known as the *components,* or, more correctly, the *rectangular components,* of their resultant R along the lines $O\,D$ and $O\,E$. In general the component $O\,B$ of any force $O\,C$ along any line $O\,E$ is found by drawing from C a line $C\,B$ perpendicular to $O\,E$, thus determining the length $O\,B$.



(7) CONDITIONS FOR EQUILIBRIUM. The conditions for the equilibrium of any body are as follows:

(a) The sum of all the components of all the forces acting on the body taken along any line, or more particularly along any pair of lines at right angles, must balance.

(b) Taking any point as origin, the sum of the moments of all the forces tending to turn the body in one direction about this origin must equal or balance the corresponding sum in the other direction.

If all the forces act through a single point, only the first of these conditions is necessary. If instead they act through different points of a body, both conditions are necessary.

(8) PARALLEL FORCES. The *resultant* of a system of parallel forces is the algebraic sum of the forces.

The *center* of a system of parallel forces is a point such that if a force equal and opposite to the resultant be here applied, the whole system will be maintained in equilibrium.

Or otherwise, it is the point at which the resultant of the whole system may be considered as acting.

The *center of gravity* of a body is the center of the system of sensibly parallel forces due to the attraction of gravitation.

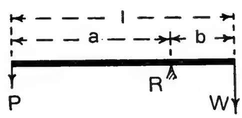
Or otherwise, it is the point at which the entire weight of the body may be considered as centered, or through which it may be considered as acting.

Or otherwise, it is the point of the body which, if supported, the whole body will be supported in equilibrium, and perfectly free to turn into any position.

[13] Mechanical Powers.

LEVER. A lever consists essentially of a bar which supports a weight or applies a force at one point by means of a force applied at another point, the bar in the meantime being supported and turning about a third point called the fulcrum. According to the relation of these three points, levers are divided into three classes as below:

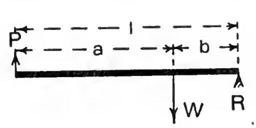
LEVER OF THE FIRST CLASS. In this the fulcrum R is be-



tween the points of application of the forces P and W. The following proportions and equations apply to this case:

$$P : W :: b : a \text{ or } P\,a = W\,b$$
$$P : P + W :: b : l \text{ or } P\,l = (P + W)\,b$$
$$W : P + W :: a : l \text{ or } W\,l = (P + W)\,a$$
$$P = \frac{b}{a} W = \frac{b}{l}(P + W) \qquad a = \frac{W}{P} b = \frac{W}{P + W} l$$
$$W = \frac{a}{b} P = \frac{a}{l}(P + W) \qquad b = \frac{P}{W} a = \frac{P}{P + W} l$$

LEVER OF THE SECOND CLASS. In this the weight lifted or resultant force W is between the applied force P and the fulcrum



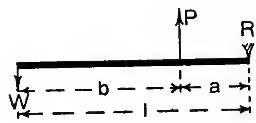
R. The following proportions and equations apply in this case:

$$P : W :: b : l \text{ or } P\,l = W\,b$$
$$P : (W - P) :: b : a \text{ or } P\,a = (W - P)\,b$$
$$W : (W - P) :: l : a \text{ or } W\,a = (W - P)\,l$$
$$P = \frac{b}{l} W = \frac{b}{a}(W - P) \qquad a = \frac{W - P}{P} b = \frac{W - P}{W} l$$
$$W = \frac{l}{b} P = \frac{l}{a}(W - P) \qquad b = \frac{P}{W} l = \frac{P}{W - P} a$$

Lever of the Third Class. In this the applied force P is between the weight lifted or resultant force W and the fulcrum R. The following proportions and equations apply to this case:

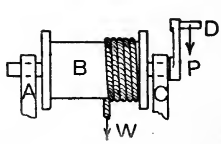
$$P : W :: l : a \quad \text{or} \quad P\,a = W\,l$$
$$(P - W) : W :: b : a \quad \text{or} \quad (P - W)\,a = W\,b$$
$$(P - W) : P :: b : l \quad \text{or} \quad (P - W)\,l = P\,b$$

$$P = \frac{l}{a} W = \frac{l}{b}(P - W) \qquad a = \frac{W}{P} l = \frac{W}{P - W} b$$
$$W = \frac{a}{l} P = \frac{a}{b}(P - W) \qquad b = \frac{P - W}{W} a = \frac{P - W}{P} b$$



An ordinary crowbar, a pair of scissors, an air pump lever, are all examples of a lever of the first class. A pair of nut crackers, an oar (the water being the fulcrum), and often many of the levers about the starting and drain gear of an engine, are examples of a lever of the second class. The forearm (the elbow being the fulcrum), or a ladder when raised against a house, are examples of the third class.

Windlass and Crank. In this device a barrel B is carried on an axle supported in bearings at A and C, and operated by a crank D. The weight W may then, by means of a rope wound on the barrel, be raised or lowered by the action of a force P



applied at the crank. The following proportions and equations give the relations between the various quantities concerned:

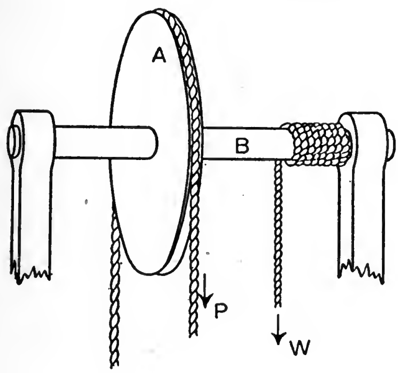
$$P : W :: r : R \text{ or } P\,R = W\,r$$
$$P = \frac{r}{R} W \qquad W = \frac{R}{r} P$$
$$r = \frac{P}{W} R \qquad R = \frac{W}{P} r$$

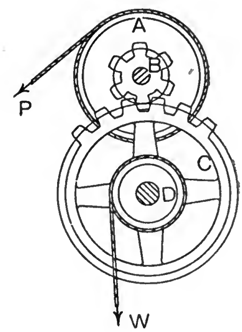
r = radius of barrel.
R = radius of crank.

Wheel and Axle. This device is the same as the preceding, except that the wheel A takes the place of the crank. The same proportions and equations apply as for the windlass and crank above. Illustrations of the principles involved in this and the preceding figures are found in all forms of windlasses, deck winches, etc.

r = radius of barrel.

R = radius of wheel.





Geared Hoist. This device is similar to the two preceding, with the addition of gearing between the force P and the weight W. The following equations apply to this case:

$$\frac{P}{W} = \frac{r_1\, r_2}{R_1\, R_2}$$

$$P = \frac{r_1\, r_2}{R_1\, R_2} W$$

$$W = \frac{R_1\, R_2}{r_1\, r_2} P$$

Most deck winches are illustrations of a simple geared hoist.

R_1 = radius of A

r_1 = radius of B

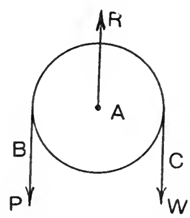
R_2 = radius of C

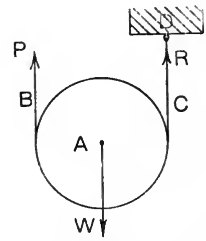
r_2 = radius of D

Single Fixed Pulley. A is a pulley or sheave supported from R and turning about its center. B C is a single rope led over the pulley, to one end of which the force P is applied, and to the other end of which the weight W is attached. The following equations apply to this case:

$$P = W$$
$$R = P + W = 2\,P$$
$$\text{Velocity of } W = \text{velocity of } P$$

A single whip used for raising light weights is an illustration of this purchase.

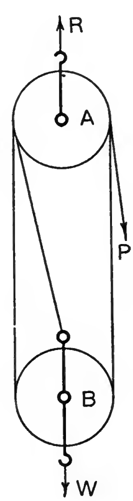




Single Movable Pulley. *A* is a pulley or sheave to the frame of which is attached the weight *W*. *B C* is the rope rove around the sheave, having one end made fast to the support *D*, while to the other is applied the force *P*. The following equations apply to this case:

$$W = 2\,P$$
$$R = P = \frac{W}{2}$$
$$\text{Velocity of } W = 1\text{-}2 \text{ velocity of } P$$

Tacks and sheets on light sails are illustrations of this form of purchase.



Luff Tackle. In this purchase there are two sheaves at *A* and one at *B*. The rope is led as shown from the frame of *B* up around one of the sheaves A, then down around the sheave B and up over the other sheave A to the point P, where the power is applied. The following equations apply to this case:

$$W = 3\,P$$
$$R = 4\,P$$

If upper block is fixed,

Velocity of W = 1-3 velocity of P.

If lower block is fixed,

Velocity of R = 1-4 velocity of P.

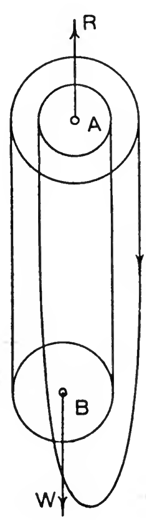
In order to obtain the greatest advantage with this purchase, therefore, B should be the fixed block.

A Pair of Blocks, as in the Luff Tackle Figure, with any Number of Sheaves in Either Block.

$\frac{W}{P}$ = total number of ropes at the lower block, passing through and attached.

$\frac{R}{P}$ = total number of ropes at the upper block, passing through and attached.

Thus in the figure the number of ropes at the lower block is 3 and the number at the upper block is 4, which, according to the rule, would give the same relations between P, R and W as in the equations above.



Differential Pulley. In this purchase there are two sheaves at A fastened together, or made in one piece, and one sheave at B. A rope or chain is rove as shown in the figure, and the force is applied at P while the weight W is supported from the lower block. The following equations apply to this case:

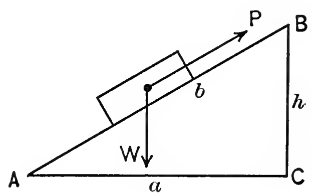
R = radius of larger upper pulley
r = radius of smaller upper pulley

$$\text{Then} \quad \frac{W}{P} = \frac{2\,R}{R-r}$$

$$\text{or} \quad W = \frac{2\,R}{R-r}\,P$$

$$\text{Velocity of} \quad W = \frac{R-r}{2\,R}\,(\text{velocity of}\,P)$$

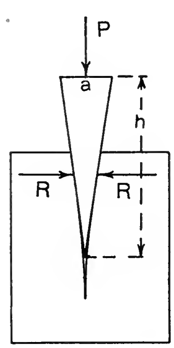
The differential pulley is commonly found in all engineer's outfits on board ship.



INCLINED PLANE.

$$\frac{W}{P} = \frac{b}{h} \quad \text{or} \quad W = \frac{b}{h}\,P$$

$$\text{and} \quad P = \frac{h}{b}\,W$$



WEDGE.

$$\frac{R}{P} = \frac{h}{a} \quad \text{or} \quad R = \frac{h}{a}\,P$$

$$\text{and} \quad P = \frac{a}{h}\,R$$

SCREW.

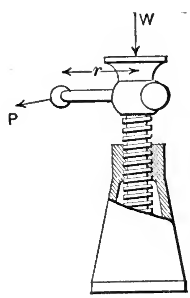
p = pitch of screw
P = force applied at radius r
W = pressure exerted

$$P : W :: p : 2\pi r$$

$$\text{or } P : W :: p : 6.2832\, r$$

$$\text{or } W = \frac{6.2832\, r}{p} P$$

$$P = \frac{p}{6.2832\, r} W$$



Examples of the applications of the last three figures will be too familiar to need special mention.

[14] Examples in Mechanics.

We give below the solutions of a few simple examples as illustrations of the preceding principles of mechanics. In all cases the effects of friction are omitted.

(1) In a lever of the first class as shown, $a = 48$ and $b = 8$. With a pull P of 160 lb., what weight W can be raised?

$$W = \frac{a}{b} P = \frac{48}{8} \times 160 = 960 \text{ lb.}$$

(2) In a lever of the second class as shown, $l = 72''$ and $b = 12''$. What force P will be required to raise a weight W of 600 lb.?

$$P = \frac{b}{l} W = \frac{12}{72} \times 600 = 100 \text{ lb.}$$

(3) In a lever of the third class as shown, $l = 40''$ and $W = 30$ lb. Where must a force P of 80 lb. be located so as to maintain equilibrium?

$$a = \frac{W}{P} l = \frac{30}{80} \times 40 = 15''$$

(4) The dimensions of a windlass and crank as illustrated are as follows: Radius of crank $= 14''$. Radius of barrel $= 4\ 1\text{-}2''$. What weight can be raised with a force of 60 lb. applied at the crank?

$$W = \frac{R}{r} P = \frac{14}{4\frac{1}{2}} \times 60 = 186\ 2\text{-}3 \text{ lb.}$$

(5) With a wheel and axle as illustrated, the diameter of the wheel is 6′ and of the axle 10″. What force P will be required to hoist a weight W of 600 lb.?

$$P = \frac{r}{R} W = \frac{5}{36} \times 600 = 83\ 1\text{-}3 \text{ lb.}$$

(6) The dimensions of a geared hoist as illustrated are as follows: Diam. of $A = 24''$; number of teeth in $B = 16$; number of teeth in $C = 96$; diam. of $D = 10''$. What weight W can be hoisted if $P = 100$ lb.?

Since the diameters and radii of wheels are in the same ratio as their numbers of teeth, we have:

$$W = \frac{R_1 R_2}{r_1 r_2} P = \frac{12 \times 48 \times 100}{8 \times 5} = 1440 \text{ lb.}$$

(7) With a single movable pulley as illustrated, what weight can be raised with a pull P of 90 lb.?

$$W = 2P = 2 \times 90 = 180$$

(8) With a luff tackle purchase as illustrated, what force P will be required to raise a weight W of 372 lb., and what will be the load at R?

$$W = 3P \text{ or } P = W \div 3 = 372 \div 3 = 124$$
$$R = 4P = 4 \times 124 = 496$$

(9) The dimensions of a differential pulley as illustrated are as follows: Larger diameter, 13″; smaller diameter, 11″. With a pull P of 80 lb., what weight W can be raised?

$$W = \frac{2R}{R - r} P = \frac{2 \times 6\frac{1}{2} \times 80}{6\frac{1}{2} - 5\frac{1}{2}} = 1040 \text{ lb.}$$

(10) An inclined plane as illustrated has dimensions as follows: Slant length, $b = 72''$; height, $h = 18''$. With a pull P of 40 lb., what weight W can be moved up the plane?

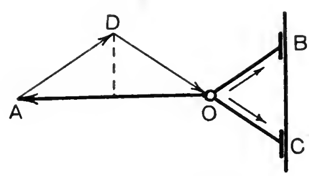
$$W = \frac{b}{h} P = \frac{72}{18} \times 40 = 160 \text{ lb.}$$

(11) A wedge as illustrated has the following dimensions: Back, $a = 4''$; length, $h = 26''$. What resistance R can be overcome by a force P of 216 lb.?

$$R = \frac{h}{a} P = \frac{26}{4} \times 216 = 1404 \text{ lb.}$$

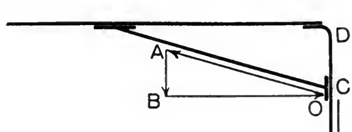
(12) A screw as illustrated has the following dimensions: Pitch $p = 1\text{-}4''$; radius $r = 12''$; force $P = 60$ lb. What pressure W can be exerted?

$$W = \frac{6.2832 \times 12 \times 60}{\frac{1}{4}} = 18095.6$$



(13) Given a boiler brace $O\,A$ with crowfoot or forked attachment to the plate $B\,C$. With a known load on $O\,A$, required the load on $O\,B$ and $O\,C$.

Evidently the three forces on $O\,A$, $O\,B$ and $O\,C$ keep the joint O in equilibrium. If represented according to the polygon of forces [12] (5), they must form a closed triangle. This is represented by $O\,A\,D$, where $O\,A$ represents the force on the brace, $A\,D$ that on $O\,B$ and $D\,O$ that on $O\,C$. Hence if $O\,A$ is laid down to some convenient scale to represent the load on the brace, then $A\,D$ and $D\,O$ respectively will, according to the same scale, represent the loads on $O\,B$ and $O\,C$.

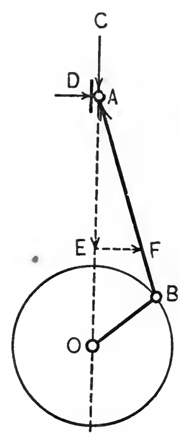


(14) Given a boiler brace $O\,A$ oblique to the shell $C\,D$. With a known load in the direction $B\,O$, required the load on $O\,A$.

The point of attachment O is again maintained in equilibrium by the action of the three forces, one along $O\ A$, one in the direction $B\ O$, and a third C existing as a tension in the plate. The triangle of forces in this case is represented by $O\ B\ A$. Hence if $O\ B$ is laid off to any scale to represent the known load, then will $O\ A$ represent to the same scale the resulting load on the brace.

(15) Let $C\ A\ B\ O$ represent the moving parts of an engine. With a known piston load acting along $C\ A$, required the resultant loads on the connecting rod and on the guide.

The point A is kept in equilibrium by the action of the three forces, one acting down along $C\ A$, one acting up the rod along $B\ A$ and one acting from the guide along $D\ A$. It is the two latter which are required. The triangle of forces in this case is



represented by $A\ E\ F$. Hence if $A\ E$ is laid off to represent to any convenient scale the known piston load, then to the same scale will $A\ F$ and $E\ F$ represent the loads on the connecting rod and crosshead respectively. It thus appears that the load on the connecting rod is in general greater than that on the piston rod, and that it is greater in the same ratio that any length $A\ F$ is greater than the corresponding distance $A\ E$.

(16) Let the diagram represent a davit $C\ D$ supporting a weight W and braced by a stay $A\ B$. With a given weight W, required the tension on the stay and the forces at the foot of the davit.

The tension T may be represented by its two components P and Q, and the reaction at C by two components N and R. In this case we require the use of both conditions of equilibrium, and without giving all details we will simply write the equations and the resulting values of the forces.

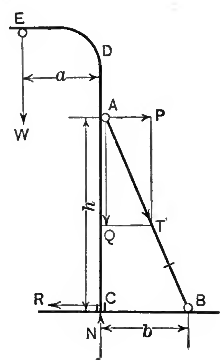
Equating vertical forces $W + Q = N$

Equating horizontal forces $P = R$

Taking moments about C $Wa = Ph$

We also have $P = \frac{b}{l} T$

and $Q = \frac{h}{l} T$



From these equations we find :

$$P = \frac{a}{h} W$$

$$Q = \frac{a}{b} W$$

$$T = \frac{l\,a}{b\,h} W$$

$$R = \frac{a}{h} W$$

$$N = \left(\frac{a+b}{b}\right) W$$

Answers to Examples in Part II.

Sec. 1. COMMON FRACTIONS.

[2] $\frac{17}{6}$, $\frac{60}{7}$, $\frac{199}{16}$, $\frac{107}{12}$, $\frac{205}{16}$, $\frac{788}{45}$, $\frac{36139}{144}$.

[3] $5\frac{11}{13}$, $1\frac{3}{8}$, $2\frac{87}{101}$, $2\frac{2}{3}$, $1\frac{1}{4}$, $1\frac{2}{9}$, $3\frac{1}{7}$, $40\frac{4}{19}$.

[4] $\frac{1}{3}$, $\frac{5}{11}$, $2\frac{5}{12}$, $\frac{1}{2}$, $\frac{14}{27}$, $\frac{21}{121}$, $1\frac{3}{11}$, 3, $\frac{1}{3}$.

[6] $1\frac{5}{42}$, $1\frac{1}{8}$, $2\frac{4}{9}$, $2\frac{3}{10}$, $15\frac{23}{48}$, $\frac{7}{24}$, $\frac{37}{48}$, $4\frac{121}{144}$, $\frac{59}{432}$, $\frac{1}{64}$.

[7] $1\frac{1}{3}$, 2, $2\frac{2}{11}$, $3\frac{1}{2}$, $9\frac{1}{2}$, 7, $2\frac{1}{4}$.

[8] $\frac{3}{14}$, $\frac{2}{27}$, $\frac{3}{17}$, $\frac{3}{35}$, $\frac{4}{23}$, $\frac{5}{6}$, $\frac{7}{48}$.

[9] $\frac{4}{7}$, $\frac{7}{20}$, $\frac{36}{77}$, $\frac{2}{27}$, $\frac{15}{68}$, $1\frac{2}{3}$, $9\frac{1}{3}$, $1\frac{1}{3}$, $\frac{17}{78}$, $1\frac{1}{5}$, $\frac{128}{161}$, $\frac{4}{17}$.

[10] $\frac{5}{12}$, $10\frac{5}{8}$, $\frac{4}{5}$.

Sec. 2. DECIMAL FRACTIONS.

[5] .4, 2.4, 24.4, .072, .1307+, 4.5, 2.6, 1.1951+, .001875, .007368+, 3.2857+.

[8] 3.808, 3466.892, .21888, 93978.89. .0294, .000504.

[9] .4531+, 11.334+, 0000006., 600, 4000, 500.

MISCELLANEOUS EXERCISES.

(1) One ton of coal is found to contain 300 pounds of ashes. What percent is combustible and what percent ashes? (See Part II, Sec. 3).

Ans. 86.6% and 13.4%.

(2) In a certain boiler it requires 1,120 heat units to evaporate one pound of water. With the addition of a feed heater this is reduced to 1,030. Find the percent of saving. (See Part II, Sec. 3).

Ans. 8.04%.

(3) A ship steams 2,800 miles in 11 days. How long will it require to steam 740 miles at the same rate? (See Part II, Sec. 6).

Ans. 2.9 days.

(4) A given engine with a reduced mean effective of 36 pounds, stroke 33 inches and revolutions of 120 develops 1,200 I. H. P. With revolutions 140 and stroke 30 inches what would be the reduced mean effective for 1,300 I. H. P. (See Part II, Sec. 6 [2]).

Ans. 36.77.

(5) A propeller at 90 revolutions and 24 per cent slip gives a speed of 12 miles per hour. With 100 revolutions and 26 per cent slip what speed will be attained? (See Part I, Sec. 80; Part II, Sec. 6).

Ans. 12.98.

(6) How many gallons of oil will be contained in a tank of rectangular form, 4 feet long, 27 inches wide and 2 feet high? (See Part II, Sec. 4 [5], Sec. 9 [16]).

Ans. 134.6.

(7) An oil tank 55 inches long has trapezoidal cross sections. The two parallel sides are 28 inches and 40 inches, and the distance between them is 40 inches. Find the capacity in gallons. (See Part II, Sec. 4 [5], Sec. 9 [4] [16]).

Ans. 323.8.

(8) What is the capacity of a pump in gallons per mt. if the cylinder is 9 inches in diameter, 10 inches stroke and 60 strokes per mt.? (See Part II, Sec. 4 [5], Sec. 9 [17]).

Ans. 165.2.

(9) A coal bunker of rectangular form is 18 feet fore and aft, 40 feet wide by 13 feet deep. Find the capacity in tons, allowing 42 cu. ft. per ton. (See Part I, Sec. 11 [7]; Part II, Sec. 9 [16]).

Ans. 223.

(10) A coal bunker of irregular form has three cross sections, as follows: The first is a rectangle 12 ft. by 18 ft. The second is a trapezoid with parallel sides 12 and 10 ft. by 16 ft. between them, and the third is a trapezoid with parallel sides 12 and 8 ft. by 14 ft. between them. These sections are 24 feet apart. Find the capacity in tons of 44 cu. ft. each.

Solution. The areas of the three sections are as follows: 216, 176, 140. Then by Simpson's rule, Part II, Sec. 9 [15] [29] the volume is bound as follows: $V = (216 + 4 \times 176 + 140) \div 3 \times 24$, or Volume $= 8{,}480$ cu ft.

Then tons $= 8{,}480 \div 44 = 192.7$.

(11) An oil can in the form of a frustum of a cone is 11 inches high, and the diameters of the base and top are respectively 5 and 4 inches. Find the capacity. (See Part II, Sec. 9 [26]).

Ans. .76 gallon.

(12) Will a boiler 60 in. diam., ½ in. thickness of plate stand as much pressure as a boiler 48 in. diam., $\frac{7}{16}$ in. thickness of plate? (See Part I, Secs. 19, 63).

Ans. No.

(13) Find the required weight for a safety valve whose diam. is 4″, fulcrum 3″, length of lever 34″, weight of lever 12 lbs., weight of valve and stem 6 lbs., steam pressure 65 lbs. (See Part I, Sec. 61).

Ans. 65.5 pounds.

(14) What would be the safe working pressure of a boiler

1.08 in. thickness of plate, tensile strength 60,000 lbs. per sq. in. diameter of boiler 12 ft.? (See Part I, Sec. 19).

Ans. 150 pounds per square inch for single riveting or 180 pounds per square inch for double riveting.

(15) The diameter of a boiler shell is 15 feet. The working pressure desired is 180 pounds per square inch. The strength of the material is 60,000 pounds per square inch. Find the thickness with double riveting. (See Part I, Sec. 19).

Ans. 1.35 inches or say $1\frac{3}{8}$.

(16) The diameter of a steel screw stay bolt at bottom of thread is $1\frac{1}{4}$ inches. The area supported by each is 49 square inches. Find the pressure allowed. (See Part I, Sec. 19).

Ans. 200 pounds per square inch.

(17) What would be the thickness of plate required for the same boiler as in (16) the spacing being 7 by 7 inches, and the bolts being simply riveted over on ends? (See Part I, Sec. 19).

Ans. 17-32 in.

(18) For other problems in boiler bracing see Part I, Sec. 62.

(19) Temperature of feed 160°. Steam pressure 180 pounds gauge. Quality of steam 97 per cent. Thermal value of coal 13,800 thermal units per pound. Efficiency of boiler .68. Find the pounds of water evaporated per pound of coal into steam of the given quality. (See Part I, Sec. 58).

Ans. 8.76 pounds.

(20) With a coal consumption of 1.80 per I. H. P. per hour how much coal will be required in the bunkers of a ship making a 10-day trip, the I. H. P. being 1,800 and a margin of 12% being allowed for emergencies? (See Part I, Sec. 59).

Ans. 388.8 tons.

(21) A corrugated furnace has a diameter of 44 inches and thickness of 1-2 inch. Find the pressure allowed. (See Part I, Sec. 19).

Ans. 170 or 159, according to the style of corrugation.

(22) Find the necessary thickness of a copper steam pipe for 200 pounds working pressure, the diameter of the pipe being 9 inches. (See Part I, Sec. 19).

Ans. .2875 inch.

(23) With cut off at 62 per cent of stroke and clearance of 12 per cent, what is the expansion ratio? (See Part I, Sec. 68).

Ans. 1.51.

(24) The I. H. P. is 2,100, pitch of propeller 18 feet, revolutions 100. Find the indicated thrust in pounds. (See Part I, Sec. 71).

Ans. 38,500 pounds.

(25) What is the weight of a rectangular piece of boiler plate 14 feet long, 5 feet 6 inches wide and 1 1-8 inch thick? (See table page 30).

Ans. 12,474 cu. in. = 3,530 pounds.

(26) Given speed of ship 12 knots, revolutions 110, slip of propeller 20 per cent, find the pitch. (See Part I, Sec. 80 [1]).

Ans. 13.8 ft.

(27) Allowing an Admiralty constant of 280, what would be the I. H. P. required for a displacement of 8,600 tons with a speed of 10 knots? (See Part I, Sec. 82).

Ans. 1,500.

(28) Given speed of ship 18 knots, revolutions 110, pitch 20 feet. Find the slip ratio. (See Part I., Sec. 80 [1]).

Ans. 17.1 per cent.

(29) Given the pitch of a propeller 54 inches, revolutions 360, slip 21 per cent. Find the speed in miles per hour. (See Part I., Sec. 80 [1]).

Ans. 14.54.

(30) Given the diameter of the rolling circle of a paddle wheel 30 feet, slip 28 per cent., revolutions 24. Find the speed in miles per hour.

Ans. 18.5.

(31) Given displacement 120 tons, I.H.P. 420. Allowing an Admiralty-Constant of 160, what speed may be expected? (See Part I., Sec. 82).

Ans. 14.03 knots.

(32) Given I.H.P. = 33,000, D = 20,000, speed = 23 knots. Find the Admiralty-Constant. (See Part I., Sec. 82).

Ans. 272.

(33) Allowing an Admiralty-Constant of 110, what would be the I.H.P. required for a displacement of 4 tons at a speed of 7 knots? (See Part I., Sec. 82).

Ans. 7.86.

(34) On a trial trip over a course of two nautical miles, suppose the observations as follows:

Run North 2 m. 36 sec. Revolutions 648.

Run South 2 m. 24 sec. Revolutions 604.

Pitch of propeller 24 feet. Find average speed and slip of propeller. (See Part I., Sec. 84).

Ans. 24 knots, 19 per cent.

(35) Given the ship in example (27). With the same Admiralty-Constant, what percentage increase in power would be required for 30 per cent. increase in displacement? (See Part I., Sec. 82).

Ans. 19 per cent.

(36) Given the ship in example (27). With an Admiralty-Constant of 250, what percentage increase in power would be required for a 30 per cent. increase in speed? (See Part I., Sec. 82).

Ans. 146 per cent.

Vacuum Gauge, Inches of Mercury.	Absolute Pressure. lbs. per square inch.	Temperature Fahrenheit.	Total Heat above 32° F. In the Water h Heat-units.	Total Heat above 32° F. In the Steam H Heat-units.	Latent Heat $L = H - h$. Heat-units.	Relative Volume. Vol. of Water at 39° F. = 1.	Volume Cu. ft. in 1 lb. of Steam.	Weight of 1 cu ft. Steam lb.
29.74	.089	32	0	1091.7	1091.7	208 80	3333.3	.00030
29.67	.122	40	8	1094.1	1086.1	154330	2472.2	.00040
29.56	.176	50	18	1097.2	1079.2	107630	1724.1	.00058
29.40	.254	60	28.01	1100.2	1072.2	76370	1223.4	.00082
29.19	.359	70	38.02	1103.3	1065.3	54660	875.61	.00115
28.90	.502	80	48.04	1106.3	1058.3	39690	635.80	.00158
28.51	.692	90	58.06	1109.4	1051.3	29290	469.20	.00213
28.00	.943	100	68.08	1112.4	1044.4	21830	349.70	.00286
27.88	1	102.1	70.09	1113.1	1043.0	20623	334.23	.00299
25.85	2	126.3	94.44	1120.5	1026.0	10730	173.23	.00577
23.83	3	141.6	109.9	1125.1	1015.3	7325	117.98	.00848
21.78	4	153.1	121.4	1128.6	1007.2	5588	89.80	.01112
19.74	5	162.3	130.7	1131.4	1000.7	4530	72.50	.01373
17.70	6	170.1	138.6	1133.8	995.2	3816	61.10	.01631
15.67	7	176.9	145.4	1135.9	990.5	3302	53.00	.01887
13.63	8	182.9	151.5	1137.7	986.2	2912	46.60	.02140
11.60	9	188.3	156.9	1139.4	982.4	2607	41.82	.02391
9.56	10	193.2	161.9	1140.9	979.0	2361	37.80	.02641
7.52	11	197.8	166.5	1142.3	975.8	2159	34.61	.02889
5.49	12	202.0	170.7	1143.5	972.8	1990	31.90	.03136
3.45	13	205.9	174.7	1144.7	970.0	1846	29.58	.03381
1.41	14	209.6	178.4	1145.9	967.4	1721	27.50	.03625
Gauge Pressure lbs. per sq. in.	14.7	212	180.9	1146.6	965.7	1646	26.36	.03794
0.304	15	213.0	181.9	1146.9	965.0	1614	25.87	.03868
1.3	16	216.3	185.3	1147.9	962.7	1519	24.33	.04110
2.3	17	219.4	188.4	1148.9	960.5	1434	22.98	.04352
3.3	18	222.4	191.4	1149.8	958.3	1359	21.78	.04592
4.3	19	225.2	194.3	1150.6	956.3	1292	20.70	.04831
5.3	20	227.9	197.0	1151.5	954.4	1231	19.72	.05070
6.3	21	230.5	199.7	1152.2	952.6	1176	18.84	.05308
7.3	22	233.0	202.2	1153.0	950.8	1126	18.03	.05545
8.3	23	235.4	204.7	.7	949.1	1080	17.30	.05782
9.3	24	237.8	207.0	1154.5	947.4	1038	16.62	.06018
10.3	25	240.0	209.3	1155.1	945.8	998.4	15.99	.06253
11.3	26	242.2	211.5	.8	944.3	962.3	15.42	.06487
12.3	27	244.3	213.7	1156.4	942.8	928.8	14.88	.06721
13.3	28	246.3	215.7	1157.1	941.3	897.6	14.38	.06955
14.3	29	248.3	217.8	.7	939.9	868.5	13.91	.07188
15.3	30	250.2	219.7	1158.3	938.9	841.3	13.48	.07420

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Gauge Pressure, lbs. per sq. in.	Absolute Pressure, lbs. per square inch.	Temperature Fahrenheit.	Total Heat above 32° F. In the Water h Heat-units.	Total Heat above 32° F. In the Steam H Heat units.	Latent Heat L, $= H - h$ Heat-units.	Relative Volume Vol. of Water at 39° F. = 1.	Volume Cu. ft. in 1 lb. of Steam.	Weight of 1 cu. ft. Steam lb.
16.3	31	252.1	221.6	.8	937.2	815.8	13.07	.07652
17.3	32	254.0	223.5	1159.4	935.9	791.8	12.68	.07884
18.3	33	255.7	225.3	.9	934.6	769.2	12.32	.08115
19.3	34	257.5	227.1	1160.5	933.4	748.0	11.98	.08346
20.3	35	259.2	228.8	1161.0	932.2	727.9	11.66	.08576
21.3	36	260.8	230.5	.5	931.0	708.8	11.36	.08806
22.3	37	262.5	232.1	1162.0	929.8	690.8	11.07	.09035
23.3	38	264.0	233.8	.5	928.7	673.7	10.79	.09264
24.3	39	265.6	235.4	.9	927.6	657.5	10.53	.09493
25.3	40	267.1	236.9	1163.4	926.5	642.0	10.28	.09721
26.3	41	268.6	238.5	.9	925.4	627.3	10.05	.09949
27.3	42	270.1	240.0	1164.3	924.4	613.3	9.83	.1018
28.3	43	271.5	241.4	.7	923.3	599.9	9.61	.1040
29.3	44	272.9	242.9	1165.2	922.3	587.0	9.41	.1063
30.3	45	274.3	244.3	.6	921.3	574.7	9.21	.1086
31.3	46	275.7	245.7	1166.0	920.4	563.0	9.02	.1108
32.3	47	277.0	247.0	.4	919.4	551.7	8.84	.1131
33.3	48	278.3	248.4	.8	918.5	540.9	8.67	.1153
34.3	49	279.6	249.7	1167.2	917.5	530.5	8.50	.1176
35.3	50	280.9	251.0	.6	916.6	520.5	8.34	.1198
36.3	51	282.1	252.2	1168.0	915.7	510.9	8.19	.1221
37.3	52	283.3	253.5	.4	914.9	501.7	8.04	.1243
38.3	53	284.5	254.7	.7	914.0	492.8	7.90	.1266
39.3	54	285.7	256.0	1169.1	913.1	484.2	7.76	.1288
40.3	55	286.9	257.2	.4	912.3	475.9	7.63	.1311
41.3	56	288.1	258.3	.8	911.5	467.9	7.50	.1333
42.3	57	289.1	259.5	1170.1	910.6	460.2	7.38	.1355
43.3	58	290.3	260.7	.5	909.8	452.7	7.26	.1377
44.3	59	291.4	261.8	.8	909.0	445.5	7.14	.1400
45.3	60	292.5	262.9	1171.2	908.2	438.5	7.03	.1422
46.3	61	293.6	264.0	.5	907.5	431.7	6.92	.1444
47.3	62	294.7	265.1	.8	906.7	425.2	6.82	.1466
48.3	63	295.7	266.2	1172.1	905.9	418.3	6.72	.1488
49.3	64	296.8	267.2	.4	905.2	412.3	6.62	.1511
50.3	65	297.8	268.3	.8	904.5	406.6	6.53	.1533
51.3	66	298.8	269.3	1173.1	903.7	400.8	6.43	.1555
52.3	67	299.8	270.4	.4	903.0	395.2	6.34	.1577
53.3	68	300.8	271.4	.7	902.3	389.8	6.25	.1599
54.3	69	301.8	272.4	1174.0	901.6	384.5	6.17	.1621
55.3	70	302.7	273.4	.3	900.9	379.3	6.09	.1643
56.3	71	303.7	274.4	.6	900.2	374.3	6.01	.1665
57.3	72	304.6	275.3	.8	899.5	369.4	5.93	.1687
58.3	73	305.6	276.3	1175.1	898.9	364.6	5.85	.1709
59.3	74	306.5	277.2	.4	898.2	360.0	5.78	.1731

Gauge Pressure, lbs. per sq. in.	Absolute Pressure, lbs. per square inch.	Temperature Fahrenheit.	Total Heat above 32° F. In the Water h Heat-units.	Total Heat above 32° F. In the Steam H Heat-units.	Latent Heat $L = H - h$. Heat-units.	Relative Volume. Vol. of Water at 32° F. = 1.	Volume. Cu. ft. in 1 lb. of Steam.	Weight of 1 Cu. ft. Steam lb.
60.3	75	307.4	278.2	.7	897.5	355.5	5.71	.1753
61.3	76	308.3	279.1	1176.0	896.9	351.1	5.63	.1775
62.3	77	309.2	280.0	.2	896.2	346.8	5.57	.1797
63.3	78	310.1	280.9	.5	895.6	342.6	5.50	.1819
64.3	79	310.9	281.8	.8	895.0	338.5	5.43	.1840
65.3	80	311.8	282.7	1177.0	894.3	334.5	5.37	.1862
66.3	81	312.7	283.6	.3	893.7	330.6	5.31	.1884
67.3	82	313.5	284.5	.6	893.1	326.8	5.25	.1906
68.3	83	314.4	285.3	.8	892.5	323.1	5.18	.1928
69.3	84	315.2	286.2	1178.1	891.9	319.5	5.13	.1950
70.3	85	316.0	287.0	.3	891.3	315.9	5.07	.1971
71.3	86	316.8	287.9	.6	890.7	312.5	5.02	.1993
72.3	87	317.7	288.7	.8	890.1	309.1	4.96	.2015
73.3	88	318.5	289.5	1179.1	889.5	305.8	4.91	.2036
74.3	89	319.3	290.4	.3	888.9	302.5	4.86	.2058
75.3	90	320.0	291.2	.6	888.4	299.4	4.81	.2080
76.3	91	320.8	292.0	.8	887.8	296.3	4.76	.2102
77.3	92	321.6	292.8	1180.0	887.2	293.2	4.71	.2123
78.3	93	322.4	293.6	.3	886.7	290.2	4.66	.2145
79.3	94	323.1	294.4	.5	886.1	287.3	4.62	.2166
80.3	95	323.9	295.1	.7	885.6	284.5	4.57	.2188
81.3	96	324.6	295.9	1181.0	885.0	281.7	4.53	.2210
82.3	97	325.4	296.7	.2	884.5	279.0	4.48	.2231
83.3	98	326.1	297.4	.4	884.0	276.3	4.44	.2253
84.3	99	326.8	298.2	.6	883.4	273.7	4.40	.2274
85.3	100	327.6	298.9	.8	882.9	271.1	4.36	.2296
86.3	101	328.3	299.7	1182.1	882.4	268.5	4.32	.2317
87.3	102	329.0	300.4	.3	881.9	266.0	4.28	.2339
88.3	103	329.7	301.1	.5	881.4	263.6	4.24	.2360
89.3	104	330.4	301.9	.7	880.8	261.2	4.20	.2382
90.3	105	331.1	302.6	.9	880.3	258.9	4.16	.2403
91.3	106	331.8	303.3	1183.1	879.8	256.6	4.12	.2425
92.3	107	332.5	304.0	.4	879.3	254.3	4.09	.2446
93.3	108	333.2	304.7	.6	878.8	252.1	4.05	.2467
94.3	109	333.9	305.4	.8	878.3	249.9	4.02	.2489
95.3	110	334.5	306.1	1184.0	877.9	247.8	3.98	.2510
96.3	111	335.2	306.8	.2	877.4	245.7	3.95	.2531
97.3	112	335.9	307.5	.4	876.9	243.6	3.92	.2553
98.3	113	336.5	308.2	.6	876.4	241.6	3.88	.2574
99.3	114	337.2	308.8	.8	875.9	239.6	3.85	.2596
100.3	115	337.8	309.5	1185.0	875.5	237.6	3.82	.2617
101.3	116	338.5	310.2	.2	875.0	235.7	3.79	.2638
102.3	117	339.1	310.8	.4	874.5	233.8	3.76	.2660

Gauge Pressure, lbs. per sq. in.	Absolute Pressure, lbs. per square inch.	Temperature Fahrenheit.	Total Heat above 32° F. In the Water h Heat-units.	Total Heat above 32° F. In the Steam H Heat-units.	Latent Heat L, $= H - h$. Heat-units.	Relative Volume. Vol. of water at 39° F. = 1.	Volume Cu. ft. in 1 lb. of Steam.	Weight of 1 cu. ft. Steam, lb.
103.3	118	339.7	311.5	.6	874.1	231.9	3.73	.2681
104.3	119	340.4	312.1	.8	873.6	230.1	3.70	.2703
105.3	120	341.0	312.8	.9	873.2	228.3	3.67	.2724
106.3	121	341.6	313.4	1186.1	872.7	226.5	3.64	.2745
107.3	122	342.2	314.1	.3	872.3	224.7	3.62	.2766
108.3	123	342.9	314.7	.5	871.8	223.0	3.59	.2788
109.3	124	343.5	315.3	.7	871.4	221.3	3.56	.2809
110.3	125	344.1	316.0	.9	870.9	219.6	3.53	.2830
111.3	126	344.7	316.6	1187.1	870.5	218.0	3.51	.2851
112.3	127	345.3	317.2	.3	870.0	216.4	3.48	.2872
113.3	128	345.9	317.8	.4	869.6	214.8	3.46	.2894
114.3	129	346.5	318.4	.6	869.2	213.2	3.43	.2915
115.3	130	347.1	319.1	.8	868.7	211.6	3.41	.2936
116.3	131	347.6	319.7	1188.0	868.3	210.1	3.38	.2957
117.3	132	348.2	320.3	.2	867.9	208.6	3.36	.2978
118.3	133	348.8	320.8	.3	867.5	207.1	3.33	.3000
119.3	134	349.4	321.5	.5	867.0	205.7	3.31	.3021
120.3	135	350.0	322.1	.7	866.6	204.2	3.29	.3042
121.3	136	350.5	322.6	.9	866.2	202.8	3.27	.3063
122.3	137	351.1	323.2	1189.0	865.8	201.4	3.24	.3084
123.3	138	351.8	323.8	.2	865.4	200.0	3.22	.3105
124.3	139	352.2	324.4	.4	865.0	198.7	3.20	.3126
125.3	140	352.8	325.0	.5	864.6	197.3	3.18	.3147
126.3	141	353.3	325.5	.7	864.2	196.0	3.16	.3169
127.3	142	353.9	326.1	.9	863.8	194.7	3.14	3190
128.3	143	354.4	326.7	1190.0	863.4	193.4	3.11	.3211
129.3	144	355.0	327.2	.2	863.0	192.2	3.09	.3232
130.3	145	355.5	327.8	.4	862.6	190.9	3.07	.3253
131.3	146	356.0	328.4	.5	862.2	189.7	3.05	.3274
132.3	147	356.6	328.9	.7	861.8	185.5	3.04	.3295
133.3	148	357.1	329.5	.9	861.4	187.3	3.02	.3316
134.3	149	357.6	330.0	1191.0	861.0	186.1	3.00	.3337
135.3	150	358.2	330.6	.2	860.6	184.9	2.98	.3358
136.3	151	358.7	331.1	.3	860.2	183.7	2.96	.3379
137.3	152	359.2	331.6	.5	859.9	182.6	2.94	.3400
138.3	153	359.7	332.2	.7	859.5	181.5	2.92	.3421
139.3	154	360.2	332.7	.8	859.1	180.4	2.91	.3442
140.3	155	360.7	333.2	1192.0	858.7	179.2	2.89	.3463
141.3	156	361.3	333.8	.1	858.4	178.1	2.87	.3483
142.3	157	361.8	334.3	.3	858.0	177.0	2.85	.3504
143.3	158	362.3	334.8	.4	857.6	175.0	2.84	.3525
144.3	159	362.8	335.3	.6	857.2	174.9	2.82	.3546
145.3	160	363.3	335.9	.7	856.9	173.9	2.80	.3567

Gauge Pressure, lbs. per. sq. in.	Absolute Pressure, lbs. per. square inch.	Temperature Fahrenheit.	Total Heat above 32° F.		Latent Heat $L. = H - h$ Heat-units.	Relative Volume. Vol. of Water at 39° F. = 1.	Volume Cu. ft. in 1 lb. of Steam.	Weight of 1 cu. ft. Steam lb.
			In the Water h Heat-units.	In the Steam H Heat-units.				
146.3	161	363.8	336.4	.9	856.5	172.9	2.79	.3588
147.3	162	364.3	336.9	1193.0	856.1	171.9	2.77	.3609
148.3	163	364.8	337.4	.2	855.8	171.0	2.76	.3630
149.3	164	365.3	337.9	.3	855.4	170.0	2.74	.3650
150.3	165	365.7	338.4	.5	855.1	169.0	2.72	.3671
151.3	166	366.2	338.9	.6	854.7	168.1	2.71	.3692
152.3	167	366.7	339.4	.8	854.4	167.1	2.69	.3713
153.3	168	367.2	339.9	.9	854.0	166.2	2.68	.3734
154.3	169	367.7	340.4	1194.1	853.6	165.3	2.66	.3754
155.3	170	368.2	340.9	.2	853.3	164.3	2.65	.3775
156.3	171	368.6	341.4	.4	852.9	163.4	2.63	.3796
157.3	172	369.1	341.9	.5	852.6	162.5	2.62	.3817
158.3	173	369.6	342.4	.7	852.3	161.6	2.61	.3838
159.3	174	370.0	342.9	.8	851.9	160.7	2.59	.3858
160.3	175	370.5	343.4	.9	851.6	159.8	2.58	.3879
161.3	176	371.0	343.9	1195.1	851.2	158.9	2.56	.3900
162.3	177	371.4	344.3	.2	850.9	158.1	2.55	.3921
163.3	178	371.9	344.8	.4	850.5	157.2	2.54	.3942
164.3	179	372.4	345.3	.5	850.2	156.4	2.52	.3962
165.3	180	372.8	345.8	.7	849.9	155.6	2.51	.3983
166.3	181	373.3	346.3	.8	849.5	154.8	2.50	.4004
167.3	182	373.7	346.7	.9	849.2	154.0	2.48	.4025
168.3	183	374.2	347.2	.1	848.9	153.2	2.47	.4046
169.3	184	374.6	347.7	1196.2	848.5	152.4	2.46	.4066
170.3	185	375.1	348.1	.3	848.2	151.6	2.45	.4087
171.3	186	375.5	348.6	.5	847.9	150.8	2.43	.4108
172.3	187	375.9	349.1	.6	847.6	150.0	2.42	.4129
173.3	188	376.4	349.5	.7	847.2	149.2	2.41	.4150
174.3	189	376.9	350.0	.9	846.9	148.5	2.40	.4170
175.3	190	377.3	350.4	1197.0	846.6	147.8	2.39	.4191
176.3	191	377.7	350.9	.1	846.3	147.0	2.37	.4212
177.3	192	378.2	351.3	.3	845.9	146.3	2.36	.4233
178.3	193	378.6	351.8	.4	845.6	145.6	2.35	.4254
179.3	194	379.0	352.2	.5	845.3	144.9	2.34	.4275
180.3	195	379.5	352.7	.7	845.0	144.2	2.33	.4296
181.3	196	380.0	353.1	.8	844.7	143.5	2.32	.4317
182.3	197	380.3	353.6	.9	844.4	142.8	2.31	.4337
183.3	198	380.7	354.0	1198.1	844.1	142.1	2.29	.4358
184.3	199	381.2	354.4	.2	843.7	141.4	2.28	.4379
185.3	200	381.6	354.9	.3	843.4	140.8	2.27	.4400
186.3	201	382.0	355.3	.4	843.1	140.1	2.26	.4420
187.3	202	382.4	355.8	.6	842.8	139.5	2.25	.4441
188.3	203	382.8	356.2	.7	842.5	138.8	2.24	.4462

Gauge Pressure, lbs. per sq. in.	Absolute Pressure, lbs. per square inch.	Temperature Fahrenheit.	Total Heat above 32° F. In the Water h Heat-units.	Total Heat above 32° F. In the Steam H Heat-units.	Latent Heat L. $= H - h$. Heat-units.	Relative Volume. Vol. of water at 39° F. = 1.	Volume Cu. ft. in 1 lb. of Steam.	Weight of 1 cu. ft. Steam, lb.
189.3	204	383.2	356.6	.8	842.2	138.1	2.23	.4482
190.3	205	383.7	357.1	1199.0	841.9	137.5	2.22	.4503
191.3	206	384.1	357.5	.1	841.6	136.9	2.21	.4523
192.3	207	384.5	357.9	.2	841.3	136.3	2.20	.4544
193.3	208	384.9	358.3	.3	841.0	135.7	2.19	.4564
194.3	209	385.3	358.8	.5	840.7	135.1	2.18	.4585
195.3	210	385.7	359.2	.6	840.4	134.5	2.17	.4605
196.3	211	386.1	359.6	.7	840.1	133.9	2.16	.4626
197.3	212	386.5	360.0	.8	839.8	133.3	2.15	.4646
198.3	213	386.9	360.4	.9	839.5	132.7	2.14	.4667
199.3	214	387.3	360.9	1200.1	839.2	132.1	2.13	.4687
200.3	215	387.7	361.3	.2	838.9	131.5	2.12	.4707
201.3	216	388.1	361.7	.3	838.6	130.9	2.12	.4728
202.3	217	388.5	362.1	.4	838.3	130.3	2.11	.4748
203.3	218	388.9	362.5	.6	838.1	129.7	2.10	.4768
204.3	219	389.3	362.9	.7	837.8	129.2	2.09	.4788
205.3	220	389.7	362.2*	1200.8	838.6*	128.7	2.06	.4852
215.3	230	393.6	366.2	1202.0	835.8	123.3	1.98	.5061
225.3	240	397.3	370.0	1203.1	833.1	118.5	1.90	.5270
235.3	250	400.9	373.8	1204.2	830.5	114.0	1.83	.5478
245.3	260	404.4	377.4	1205.3	827.9	109.8	1.76	.5686
255.3	270	407.8	380.9	1206.3	825.4	105.9	1.70	.5894
265.3	280	411.0	384.3	1207.3	823.0	102.3	1.64	.6101
275.3	290	414.2	387.7	1208.3	820.6	99.0	1.585	.6308
285.3	300	417.4	390.9	1209.2	818.3	95.8	1.535	.6515
335.3	350	432.0	406.3	1213.7	807.5	82.7	1.325	.7545
385.3	400	444.9	419.8	1217.7	797.9	72.8	1.167	.8572
435.3	450	456.6	432.2	1221.3	789.1	65.1	1.042	.9595
485.3	500	467.4	443.5	1224.5	781.0	58.8	.942	1.062
535.3	550	477.5	454.1	1227.6	773.5	53.6	.859	1.164
585.3	600	486.9	464.2	1230.5	766.3	49.3	.790	1.266
635.3	650	495.7	473.6	1233.2	759.6	45.6	.731	1.368
685.3	700	504.1	482.4	1235.7	753.3	42.4	.680	1.470
735.3	750	512.1	490.9	1238.0	747.2	39.6	.636	1.572
785.3	800	519.6	498.9	1240.3	741.4	37.1	.597	1.674
835.3	850	526.8	506.7	1242.5	735.8	34.9	.563	1.776
885.3	900	533.7	514.0	1244.7	730.6	33.0	.532	1.878
935.3	950	540.3	521.3	1246.7	725.4	31.4	.505	1.980
985.3	1000	546.8	528.3	1248.7	720.3	30.0	.480	2.082

*The discrepancies at 205.3 lbs. gauge are due to the change from Dery's to Buel's figures.

Appendix

Questions on the Subject Matter of Part I, With Page References.

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