AN

ELEMENTARY TEXT-BOOK

ON

STEAM ENGINES AND BOILERS.

FOR THE USE OF STUDENTS IN SCHOOLS
AND COLLEGES.

BY

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ILLUSTRATED WITH DIAGRAMS AND NUMEROUS CUTS
SHOWING AMERICAN TYPES AND DETAILS
OF ENGINES AND BOILERS.

FIFTH EDITION.

NEW YORK:
SPON & CHAMBERLAIN, 123 LIBERTY STREET.
LONDON:
E. & F. N. SPON, 57 HAYMARKET, S. W.
1905.

THE BURR PRINTING HOUSE,
NEW YORK.

PREFACE.

This book is written solely as an elementary text-book for the use of beginners and students in engineering, but more especially for the students in the various universities and colleges in this country.

No attempt has been made to tell everything about any one particular subject, but an attempt has been made to give the student an idea of elementary thermodynamics, of the action of the steam in the cylinder of the engine, of the motion of the steam valve, of the differences between the various types of engines and boilers, of the generation of heat by combustion, and the conversion of water into steam.

Care has been taken not to touch upon the design and proportion of the various parts of engines and boilers for strength; as, in the opinion of the writer, that should come after a general knowledge of the engine and boiler has been obtained.

In the derivation of some of the formulæ in thermodynamics, it has been necessary to use the calculus, but the use of all mathematics higher than algebra and geometry has been avoided as much as possible.

An earnest endeavor has been made to present the subject in a clear and concise manner, using as few words as possible and avoiding all padding.

J. H. Kinealy.

Washington University,
August, 1895.

PREFACE TO THE FOURTH EDITION.

This edition is practically the same as the previous one. The only change made has been to correct some typographical errors.

J. H. KINEALY.

BOSTON, MASS.,
August, 1903.

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CHAPTER I.

ELEMENTARY THERMODYNAMICS.

1. THERMODYNAMICS.— *The Science which treats of the laws and principles according to which work may be converted into heat and, conversely, heat into work is Thermodynamics.*

The whole science is based upon the present conception of "heat as a mode of motion;" and until the present theory of heat was established and accepted, the science was unknown. Up to about the middle of the nineteenth century the usually accepted conception of heat was that it was a material substance that could be made to enter or leave bodies; and according as this substance was present in greater or less quantities the body was more or less hot.

In 1798, Count Rumford was led to assert that heat was a mode of motion and not a material substance, by the results of experiments made while boring cannons. The results of his experiments and his conclusions as to the nature of heat were given in a paper, read before the Royal Society in England, entitled, "An Enquiry Concerning the Source of the Heat which is Excited by Friction."

In 1799, Sir Humphrey Davy, by a series of experiments upon the heating effect of rubbing two pieces of ice together, supported and strengthened the conclusions reached by Count Rumford.

In spite, however, of the work of Rumford, Davy, and others who followed them, it was not until about 1843 that the modern conception of heat as a mode of motion was firmly established, and accepted by physicists.

To Dr. Jules Robert Mayer, of Heilborn, Germany, and Dr. James Prescott Joule, of Manchester, England, is due more than, perhaps, to any others, the honor of firmly establishing the modern conception of heat. Dr. Mayer published, in 1842, an essay on the subject, in which he showed by clear reasoning and analysis that heat could not be a material substance. This was almost immediately followed by the publication, in 1843, of the results of the elaborate experiments made by Joule at Manchester. The results of Joule's experiments proved conclusively the falsity of the materiality of heat, and established the modern conception.

In February, 1850, Prof. John Macquorn Rankine read, before the Royal Society of Edinburgh, a paper, treating of thermodynamics, which, with a paper by Prof. R. Clausius read in the same month and year, before the Berlin Academy, may be considered as the beginning of the work, carried on since then to the present day, upon the Science of Thermodynamics. It is to be noted as a curious coincidence that Rankine, in Scotland, and Clausius, in Germany, each working independently of the other, had reached practically the same conclusions at the same time.

Before leaving this subject it may be well to call attention to the fact that many of the methods of discussion used in thermodynamics, in its application to the study of heat engines, are due to Sadi Carnot, who, while rejecting the modern conception and accepting the materiality of heat, did much work that has been of value in formulating methods of discussion and analysis. To him is due the idea of the cycle process known as the Carnot Cycle, a discussion of which will be given in the proper place.

2. First Law of Thermodynamics.— *Work and heat*

are mutually convertible; and whenever work is produced by heat the quantity of heat consumed is exactly proportional to the amount of work done, and, conversely, by the expenditure of the same amount of work the same quantity of heat may be produced.

The first part of this law was established by the experiments of Rumford and Davy, but it was not until the careful experiments of Joule were made that the second part was assured.

3. WORK, POWER.— *Work is the overcoming of resistance.*

In order that work may be done it is necessary that there should be not only a force but motion also. This must be kept clearly in mind; and it must also be remembered that the element *time* does not enter at all. A man may hold up a weight for any length of time imaginable, but he does no WORK, in the mechanical sense of the word, so long as the weight is held at rest. If, however, he raises the weight, and in so doing overcomes the resistance due to the force of gravity, he does work. Again, if two men carry the same weight of goods up the same stairs, each will do exactly the same amount of work, even if one should carry his weight up the stairs in one-tenth the time that the other does.

The unit of work is the foot-pound (ft.-lb.), which is *the amount of work done by a body in moving through a distance of one foot against a resistance of one pound.*

It follows, from the definition of the unit of work, that the work done by a body in overcoming any resistance is equal to the product of the mean force of resistance and the distance moved through. The force must be expressed in pounds, and the distance in feet. Since, then, work is the product of two factors it may be represented on a diagram as the area of a closed figure, whose mean altitude represents the mean force of resistance, and whose length represents the space or distance moved through. Such a figure may be termed a "Work Diagram."

To illustrate the method of forming a "Work Diagram," let, in Fig. 1, *OX* and *OY* be two lines drawn at right angles to each other, termed axes. *a* represents the position of a body that is doing work. Its distance, *ad*, from *OX*, represents the force against which it is acting; and its distance, *ae*, from *OY*, represents the distance it has moved from its starting point or origin. Now

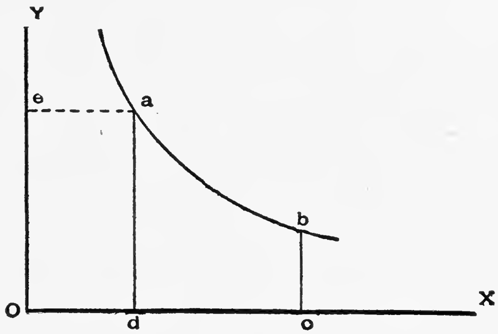


Fig. 1.

suppose the body moves from *a* to *b* along the line *ab*, which is so drawn that at any instant the distance of the body from *OX* represents the force against which it is acting at that instant, and its distance from *OY* represents the distance from its origin at the same instant. When the body gets to *b* it is acting against a force represented by the line *bc;* and it has moved a distance *dc* from *a*. The work done while the body moved from *a* to *b* is the mean force of resistance, F_m, multiplied by the distance *dc*, and is represented by the area of the figure *abcd*.

If the force of resistance had been uniform, while the body moved from *a* to *b*, the line *ab* would have been parallel to *OX*, and the work would have been equal to $ad \times dc$.

In general, the expression for the work done is given by the equation,

$$(1) \qquad \text{Work} = \int_{s_1}^{s_2} F ds$$

Where F is the force of resistance at any instant, and s is the distance from the origin. In order to solve (1) it is necessary to know the law of the path, ab, of the body.

Power involves the element time, and is *the amount of work done in a unit of time.*

In all calculations relating to engines, the power involved is usually so large that the unit of power used is the Horse-Power, H. P., which is *33,000 ft.-lbs. of work done in one minute, or 550 ft.-lbs. in one second.*

To obtain the Horse-Power exerted by an engine, get the work done per minute and divide by 33,000; or, if W represents the work done per minute, the horse-power is

$$(2) \qquad H.\ P. = \frac{W}{33000}$$

4. Unit of Heat.— Rankine defines the unit of heat as "the quantity of heat which corresponds to an interval of one degree of Fahrenheit's scale in the temperature of one pound of pure liquid water, at and near its temperature of greatest density (39.1° Fahrenheit)."

Other writers define it as "the amount of heat required to raise the temperature of one pound of water from 32° to 33° F."

In this work, however, the *Unit of Heat* will be taken as *the amount of heat required to raise the temperature of one pound of water from 62° to 63° Fahrenheit.*

The difference between the three amounts taken as the unit, is, however, so small that in practice it may be neglected.

5. Mechanical Equivalent.—The "*Mechanical Equivalent*" means *the number of foot-pounds of work that is done when one unit of heat is consumed, and is generally designated by the letter J.*

The experiments made by Rumford and Davy were too crude to give any accurate value for the mechanical equivalent, and it was not until the time of Mayer and Joule that a value could be assigned to it with any degree of certainty. Mayer, from certain properties of gases, deduced a theoretical value for J; but it remained for Joule to determine, by a series of carefully conducted experiments extending from 1842 to about 1850, the value of J as 772 ft.-lbs. This value was accepted and used as the correct one until Joule and others showed by later, and perhaps more carefully conducted, experiments, that 772 was probably too small.

Recently, Rowland, of Baltimore, by a series of experiments, in which great care was taken to guard against errors of all kinds, showed the value of the Mechanical Equivalent to be 778 ft.-lbs, the unit of heat being as used in this work. It is probable that 778 ft.-lbs. is nearer the true value of J than 772, and in this work 778 will be assumed as the true value.

6. Application of Heat to Bodies.—Whenever heat is imparted to a body, that is not on the point of changing its state, two effects may generally be observed:—

1st. The temperature of the body rises; its "sensible heat" is increased.

2nd. The body expands; its volume is increased.

There are some exceptions to the general law that the body expands when heated, but in all cases the statement of the law, as given, will suffice if *contraction* be considered as *a negative expansion.*

When heat is supplied to a body, a part of the heat is used simply to increase its temperature, and the remainder is converted into work. The work done may be classed under two heads, internal and external. The internal work is made up of two parts: the first is the work done in effecting that change of the condition of the particles due simply to the increase of temperature; the second is the work done in increasing the volume of the body against the resistance of molecular attraction. The external work is that due to the increase in the volume of the body against the resistance of the pressure of the surrounding air or gases.

The general expression for the heat used in heating a body may, then, be put in the form

$$(3) \qquad H = J\,Q = S + L + W.$$

H is the total heat used, expressed in mechanical units, i. e., foot-pounds; J is the Mechanical Equivalent of heat, 778; Q is the total heat used, expressed in heat units; S is the heat used in simply increasing the temperature of the body; L is the heat used in doing the internal work; W is the heat used in doing external work. S, L, and W are expressed in foot-pounds.

7. SECOND LAW OF THERMODYNAMICS.— Rankine gives this law as follows: "*If the total actual heat of a homogeneous and uniformly hot substance be conceived to be divided into any number of equal parts, the effects of those parts in causing work to be performed are equal.*"

He also says, "This law may be considered as a particular case of a general law applicable to every kind of *actual* energy; that is, capacity for performing work constituted by a certain condition of each particle of a substance, how small soever, independently of the presence of other particles (such as the energy of motion)."

Rankine's statement of the Second Law means simply that a unit of heat is equivalent to a definite amount of work independent of the part or the temperature of the hot body from which it is taken. A unit of heat taken from the inside of a body is equivalent to the same amount of work as a unit of heat taken from the surface; and a unit of heat from a body whose temperature is 1000° is exactly the same as a unit of heat taken from a body whose temperature is 60°.

Clausius agrees with Rankine in his statement of the first law, but as his method of reasoning is different from that of Rankine, his statement of the Second Law, or Second Main Principle as he calls it, is different. It is: "*Heat cannot, of itself, pass from a colder to a hotter body.*"

8. Specific Heat.—There are two specific heats to every body; they may be termed the *apparent specific heat*, and the *real specific heat.*

The apparent specific heat is *the amount of heat required to raise the temperature of one pound of a substance one degree Fahrenheit.*

The apparent specific heat is usually spoken of as simply the "specific heat." It includes not only the amount of heat required to change the *temperature* of the body one degree, but, also, that heat used in doing such internal and external work as may accompany the change of temperature. It is further subdivided into *specific heat at constant volume*, c_v, and *specific heat at constant pressure*, c_p.

Specific heat at constant volume is *the amount of heat required to change the temperature of one pound of a substance one degree when the volume is kept constant.* It includes only such heat as is required for the change of temperature and that part of the internal work due to this change, and excludes all heat required to do work on account of change of volume.

Specific heat at constant pressure is *the amount of heat required to change the temperature of one pound of a substance one degree when the pressure is kept constant.* It includes all the heat required to do the internal and external work due to change of volume, and is, therefore, greater than the specific heat at constant volume.

The real specific heat of a body is *the amount of heat required simply to change the temperature of one pound of a substance one degree, excluding all heat used for internal and external work.*

In the case of perfect gases the internal work done during a change of temperature is zero, and the *specific heat at constant volume* is actually equal to the *real specific heat.* In the cases of solids and liquids, the amount of heat used for internal work when the temperature is changed at constant volume is so small as compared with that required only for change of temperature, that it is usual to consider the *specific heat at constant volume* as equal to the *real specific heat.*

Let K_v represent the specific heat at constant volume, expressed in foot-pounds; K_p the specific heat at constant pressure, in foot-pounds; then, for a perfect gas,

$$(4) \qquad \frac{K_p}{K_v} = \frac{J\,c_p}{J\,c_v} = \gamma^{*} = 1.41, \text{ for air.}$$

9. Absolute Temperature.—Gay-Lussac made a series of experiments to determine the change in volume of a gas when heated under constant pressure, and, as a result of his experiments, found that the volume, V_t, of a gas at a given temperature, t, on the Fahrenheit scale, was always given in terms of its volume, V_0, at 0°, and its temperature t, by the equation

$$(5) \qquad V_t = V_0\,(1 + at),$$

where a is a constant factor, equal to $\frac{1}{461}$, termed the *coefficient of expansion of perfect gases.*

* This symbol is the Greek letter Gamma. It is used by all writers to represent the ratio of K_p to K_v.

If the temperature, t, of the gas is below zero, or negative, then the plus sign in (5) becomes minus.

If a gas is cooled below 0° its volume will be less than V_0; if the cooling is continued, and the gas should always contract in the manner indicated by Gay-Lussac, it is apparent that finally a temperature will be reached where the volume of the gas will become zero. To determine this temperature put, in (5), for V_t its supposed value, and get,

$$O = V_0(1 + at). \quad \text{Whence } t = -\frac{1}{a} = -461.$$

This point, — 461, is the absolute zero, on the Fahrenheit scale, and temperatures counted from it as the starting point are absolute temperatures, and are usually designated by T.

Since the *absolute zero* is 461 Fahrenheit degrees below the *Fahrenheit zero*, any temperature, t, on the Fahrenheit scale may be converted into a temperature, T, on the absolute scale by adding 461 to it. $T = 461 + t$.

Equation (5) may be written in the form

$$(6) \qquad V_t = V_0\left(1 + \frac{t}{461}\right) = V_0\,\frac{(461+t)}{461} = \frac{V_0\,T}{T_0}$$

T_0 is the absolute temperature for 0° F., and is equal to 461; T is the absolute temperature for $t°$ *F.*, equal to $461 + t$.

Equation (6) may be changed to

$$(7) \qquad \frac{V}{T} = \frac{V_0}{T_0} = \text{a constant}.$$

10. Application of Heat to a Perfect Gas.— If a pound of gas be put into a cylinder, closed at one end, in which works, without friction, a piston, the gas will expand and force out the piston until the pressure of the gas inside the cylinder is equal to that of the air on the outside. If now, while the temperature is kept constant,

the volume of the gas is *decreased* by pushing in the piston, the pressure exerted by the gas on the piston will be *increased;* if the volume occupied by the gas is *increased,* the pressure exerted by it on the piston will be *decreased.* The pressure exerted by the gas, while its temperature remains constant, will increase or decrease, as the volume is decreased or increased, according to "Boyle's Law," which is: *the pressure exerted by a gas, whose temperature remains constant, is inversely as its volume.*

In other words, if V_1 and P_1 represent respectively the initial volume and pressure of a gas, at a constant temperature, and V_2 and P_2 its final volume and pressure, the relation existing between V_1, P_1, V_2, and P_2, is

$$(8) \qquad \frac{V_2}{V_1} = \frac{P_1}{P_2}, \text{ or } P_2 V_2 = P_1 V_1 = \text{a constant.}$$

From equation (7), representing the relation between the volume and absolute temperature of a perfect gas under a constant pressure, and equation (8), representing the relation between the volume and pressure of a perfect gas under a constant temperature, is obtained the relation that must always, under all circumstances, exist between the volume, pressure and absolute temperature of a perfect gas. It is given by the equation

$$(9) \qquad \frac{P_2 V_2}{T_2} = \frac{P_1 V_1}{T_1} = R, \text{ a constant.}$$

When the pressures are expressed in pounds per square foot and the volumes in cubic feet, the value of R for *one pound* of air is 53.15. For w pounds of a gas the constant is $w R$.

Let it be supposed that the air in the cylinder, spoken of before, is heated, while the pressure is kept constant at P_1 lbs. per square foot, until the absolute temperature is increased from T_1 to T_2. By definition, the amount of heat supplied to the pound of the gas is, in mechanical units, the change in temperature, $T_2 - T_1$, multiplied by the specific heat at constant pressure, K_p, or

$$(10) \qquad H = K_p (T_2 - T_1).$$

But from (3) it is evident that

$$(11) \qquad H = S + L + W.$$

Where S, the heat required to change only the temperature, is, for a perfect gas, $K_v (T_2 - T_1)$, since the *real specific heat* is equal to the *specific heat at constant volume;* L, the internal work, is, for a perfect gas, equal to zero; and W, the external work, must be the total force exerted on the piston multiplied by the distance it has moved. The total force exerted on the piston is the pressure per square foot, P_1, multiplied by the area, A, of the piston; and if the initial distance of the piston from the bottom of the cylinder is d_1, and the final distance is d_2, the distance moved is $d_2 - d_1$. The external work is, then,

$$W = P_1 A (d_2 - d_1) = P_1 (Ad_2 - Ad_1).$$

But Ad_2 is equal to V_2, the final volume occupied by the gas; and Ad_1 is equal to V_1, the initial volume of the gas. The expression for W is, therefore,

$$W = P_1 (V_2 - V_1).$$

If for S, L, and W_1 are put their values, (11) becomes,

$$(12) \qquad \begin{aligned} H &= K_v (T_2 - T_1) + P_1 (V_2 - V_1) \\ &= K_v (T_2 - T_1) + P_1 V_2 - P_1 V_1. \end{aligned}$$

Since the relation between the initial, and the final pressure, volume, and absolute temperature must satisfy equation (9), we have $\frac{P_1 V_2}{T_2} = R$, and $\frac{P_1 V_1}{T_1} = R$.

Whence, $P_1 V_2 = R T_2$, and $P_1 V_1 = R T_1$

Substituting these values of $P_1 V_2$ and $P_1 V_1$ in (12), and putting for H its value, as given by (10), there is obtained,

$$(13) \qquad K_p (T_2 - T_1) = K_v (T_2 - T_1) + R (T_2 - T_1).$$

From which,

$$(14) \qquad K_p = K_v + R.$$

For a perfect gas, *the specific heat at constant pressure is equal to the specific heat at constant volume plus R.*

11. Isothermal Expansion. — *A body expands or contracts Isothermally, when its volume is increased or decreased in such a way that the temperature remains constant.*

Since the temperature remains constant during isothermal expansion, the term S, representing the heat required to effect a change of temperature of the body, in (3) becomes zero; and the expression for the heat used by the body during isothermal expansion is

$$H = L + W.$$

That is, all the heat given to the body is transformed into work, part of which is internal and part external.

Since, for a perfect gas, the term L, representing the internal work, becomes zero, as has been stated, the expression for heat becomes,

$$H = W.$$

From the general law of a perfect gas, expressed by

(9), it is evident that if the temperature is constant we have,

$$(15) \qquad P_2 V_2 = P_1 V_1 = T_1 R = \text{a constant.}$$

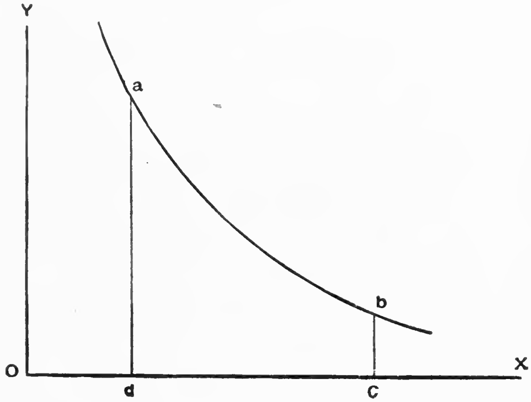


Fig. 2.

Equation (15) represents the law of the change of pressure and volume during isothermal expansion; and the curve represented by the equation, on the work diagram, is called the *Isothermal curve*, which for a perfect gas is the equilateral hyperbola.

In Fig. 2 let OX and OY represent the two axes; the co-ordinates of a, P_1, V_1, the initial pressure and volume of the gas; and the co-ordinates of b, P_2, V_2, the final pressure and volume of the gas. Also, let the curve ab be an isothermal curve, or equilateral hyperbola. The work done by the gas in expanding isothermally from V_1 to V_2, is represented by the area of the diagram *abcd;* and since the heat used is equal to the work done,

$$(16) \qquad H = W = \text{area } abcd = \int_{V_1}^{V_2} P dV = P_1 V_1 \int_{V_1}^{V_2} \frac{dV}{V}$$

$$= P_1 V_1 \; hyp.\ log.\ \frac{V_2}{V_1} \; *$$

* *Hyp. log.* is the usual contraction for "hyperbolic logarithm." Table II is a table of hyperbolic logarithms.

12. Adiabatic Expansion.—*Bodies expand adiabatically when during the expansion, they neither emit nor receive any heat.* This is expressed by making H in equation (3) equal to zero, and the equation then becomes,

$$(17) \qquad 0 = S + L + W.$$

From which

$$(18) \qquad L + W = -S.$$

For one pound of the body S is equal to the specific heat at constant volume, assumed as equal to the real specific heat, multiplied by the difference between the final and initial absolute temperatures, and (18) becomes,

$$(19) \quad L + W = -S = -K_v (T_2 - T_1) = K_v (T_1 - T_2).$$

This equation shows that during adiabatic expansion the temperature of the body falls and that the heat freed by the fall in temperature is converted into work. During adiabatic compression, the work done *on* the body is converted into heat which increases the temperature of the body.

For a perfect gas the internal work, L, done during expansion is equal to zero, so that for a perfect gas (19) becomes,

$$(20) \qquad W = K_v (T_1 - T_2).$$

In order to obtain an expression for the relation that exists between P and V during adiabatic expansion, assume, in Fig. 3, that the curve ab, termed an adiabatic curve, represents the changes of volume and pressure during expansion of one pound of gas; and that the

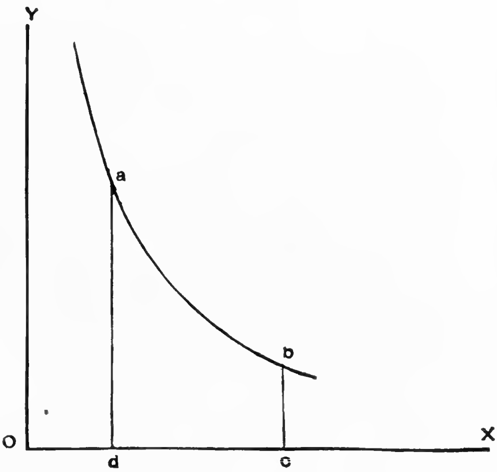


Fig. 3.

area *abcd* represents the work done during adiabatic expansion. Assume the law of the relation of P to V to be represented by the equation

$$P\,V^n = P_2\,V_2{}^n = P_1\,V_1{}^n = \text{a constant.}$$

Where n is an exponent whose value remains to be determined. The work done during expansion is represented by the area *abcd;* and by calculus,

$$\text{(21)} \qquad \text{area } abcd = W = \int_{V_1}^{V_2} P\,dV = P_1 V_1{}^n \int_{V^1}^{V_2} \frac{dV}{V^n}$$

$$= \frac{P_1\,V_1{}^n\,(V_2{}^{1-n} - V_1{}^{1-n})}{1-n} = \frac{P_1\,V_1 - P_2\,V_2}{n-1}.$$

It is already known, from (20), that the work done during adiabatic expansion is

$$\text{(22)} \qquad W = K_v\,(T_1 - T_2).$$

From (9) it is known that $T_1 - T_2 = \dfrac{P_1 V_1 - P_2 V_2}{R}$

Putting this value of $T_1 - T_2$ in (22), we get from (21) and (22),

$$(23) \qquad W = \frac{(P_1 V_1 - P_2 V_2) K_v}{R} = \frac{P_1 V_1 - P_2 V_2}{n-1}$$

From this,

$$n = \frac{R}{K_v} + 1 = \frac{R + K_v}{K_v}.$$

From (14) we have $K_p = R + K_v$; and from (4) we have $\dfrac{K_p}{K_v} = \gamma$. Therefore, $n = \dfrac{K_p}{K_v} = \gamma$.

The relation, then, between P and V during adiabetic expansion is, since n is equal to γ,

$$(24) \qquad PV^{\gamma} = P_2 V_2^{\gamma} = P_1 V_1^{\gamma} = \text{a constant.}$$

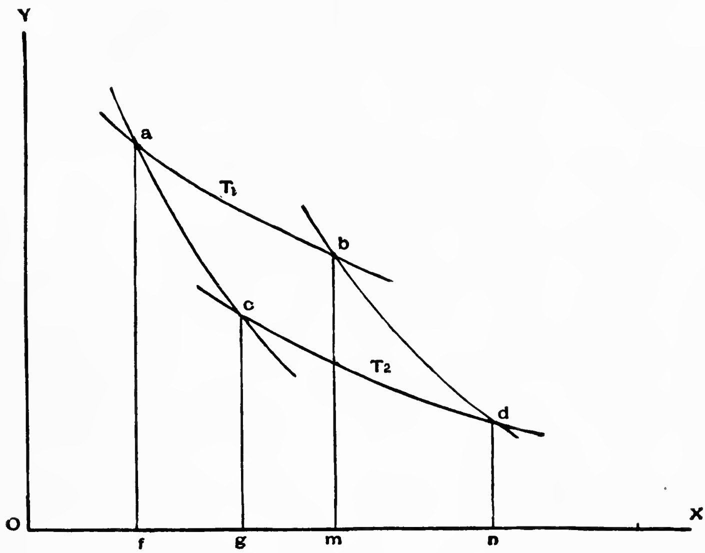


Fig. 4.

A comparison of the equation of the isothermal curve (15), with that of the adiabatic curve (24), makes it apparent that the latter is the steeper curve.

From (9) $\frac{P_2 V_2}{T_2} = \frac{P_1 V_1}{T_1}$, and, therefore,

$$\frac{P_1}{P_2} = \frac{V_2}{V_1} \times \frac{T_1}{T_2}. \text{ But } \frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^\gamma, \text{ from (24).}$$

Therefore,

$$\left(\frac{V_2}{V_1}\right)^\gamma = \frac{V_2}{V_1} \times \frac{T_1}{T_2}, \text{ or}$$

$$(25) \qquad \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

In Fig. 4, let *ab* and *cd* represent the isothermal curves of a perfect gas corresponding to the absolute temperatures T_1 and T_2 respectively. Also, let *ac* and *bd* represent adiabatic curves intersecting the isothermal curves. If now a gas expands so that the relation between P and V is shown by the adiabatic curve *ac*, from T_1 to T_2, the work done will be, from (20), $K_v (T_1 - T_2)$, and will be represented by the area *acgf*. Also, if the gas expands according to the adiabatic curve *bd*, from T_1 to T_2, the work done will be $K_v (T_1 - T_2)$, and will be represented by the area *bdnm*.

Therefore, the area *acgf* is equal to the area *bdnm*.

By inspection of the figure it is seen that, area *abdc* + area *cdng* + area *acgf* = area *abmf* + area *bdnm*.

From which, as area *acgf* = area *bdnm*, we get area *abdc* = area *abmf* — area *cdng*.

If the expansion were carried so far that T_2 became zero, the area *cdng* would also become zero, and we should have area *abdc* = area *abmf*. When T_2 is zero, the lines *cg* and *dn* will be at infinity and each equal to zero. Hence we have, since *acgf* is always equal to *bdnm*, that the work done during adiabatic expansion from P_2 and T_1 to zero temperature is equal to that done during adiabatic expansion from P_1 and T_1 to zero temperature.

13. FUSION.— *When a solid body changes to a liquid it is said to fuse or melt.*

The temperature at which fusion takes place, under standard atmospheric pressure of 14.7 lbs. per square inch or 2116.8 lbs. per square foot, is termed the *fusing point* of the body. During fusion the temperature of the solid and liquid remains constant at the fusing point, so that in the general expression, $H = S + L + W$, for heat used by a body, the term S becomes zero. The expression for the heat to fuse one pound of a body, is, then, $H = L + W$.

The heat required to fuse one pound of a solid, under standard atmospheric pressure, is the latent heat of fusion.

As the change of volume on fusing is small, for most solids, the value of W, the external work, is small; and the greater part of the latent heat of fusion is used in doing *internal* work. In the case of bodies, such as ice, that decrease in volume on fusing, W becomes negative.

14. VAPORIZATION.— *Vaporization is the conversion of a liquid into a vapor.*

The temperature at which vaporization takes place depends not only upon the liquid to be vaporized, but, also, upon the pressure to which it is subjected. During the vaporization the temperature remains constant, if the pressure does not vary.

The temperature at which, under a given pressure, a liquid is converted into a vapor is termed the "boiling point" for the given pressure.

The heat required to vaporize one pound of liquid under a given pressure is the latent heat of evaporation for the given pressure.

Since, during vaporization, the temperature of the liquid remains constant, the expression for latent heat of evaporation is

$$H = l = L + W.$$

As most liquids expand considerably upon being converted into vapor, the value of W, the external work, may be quite large.

15. APPLICATION OF HEAT TO WATER. — When heat is applied to water its temperature rises gradually, and as the specific heat of water increases slightly as the temperature rises, the heat required to raise one pound of water from t_1 to t_2 is somewhat greater than $t_2 - t_1$. For ordinary work, however, it is sufficiently accurate to assume that the specific heat of water is constant, and, therefore, that the heat, in heat units, required to change the temperature of one pound of water from t_1 to t_2 is $t_2 - t_1$. In mechanical units it is $J\,(t_2 - t_1)$. If the heating is continued long enough the temperature of the water will be raised to the boiling point, and the water will be converted into steam.

The latent heat of evaporation, in heat units, of steam is usually denoted by l, and may be approximately calculated by the equation, given by Rankine,

$$l = 1114.4 - 0.7t. \tag{26}$$

t is the temperature, in Fahrenheit degrees, at which the steam is formed.

If (26) is multiplied by J, equal to 778, there will be obtained, l_1, the latent heat of steam expressed in mechanical units :

$$l_1 = J\,l = 867000 - 544.6t. \tag{27}$$

The heat, h, in heat units, required to raise the temperature of one pound of water from t_2 to t_1 and convert it into steam at t_1 is

$$(28) \qquad h = l + t_1 - t_2.$$

Multiply (28) by J and we shall have, h_1, the heat in mechanical units required to raise the temperature of one pound of water from t_2 to t_1 and convert it into steam at t_1.

$$(29) \qquad h_1 = J\,h = J\,[l + t_1 - t_2].$$

The latent heat is used in doing the internal work L, and the external work W. The external work is the product of the pressure per square foot, under which the steam is formed, and the difference between V_w, the volume of one pound of the water and V_s, the volume of one pound of the steam. Assuming that V_w is so small as compared to V_s that it may be considered as zero, and that the pressure per *square inch* under which the steam is formed is p, the external work is, approximately,

$$(30) \qquad W = 144\,p\,V_s.$$

In practical problems relating to the steam engine we usually know the values of p and t_1; but the value of V_s must be obtained from a table made from the results of actual experiments.

Pressure gauges, used to indicate the pressure of steam in steam boilers, are so constructed that they do not show the *absolute* or *true* value of the pressure of the steam, but *show the pressure, per square inch, above that of the atmosphere.* The pressure of the atmosphere is 14.7 (often taken as 15) pounds per square inch. In order, then, to obtain the absolute value, p, of the pressure of the steam it is necessary to add 14.7 to the gauge pressure.

Tables giving the different properties of steam for different pressures are termed Steam Tables; and are all based upon experiments made by Regnault. Table 1, in this work, is a Steam Table.

In the first column is given the *gauge pressure*, per square inch, of the steam, calculated upon the assumption that the atmospheric pressure is 14.7 pounds per square inch. The gauge pressure is given rather than the absolute pressure, as the author considers it the more convenient of the two to use in practice.

In the second column is given the *temperatures*, to the nearest degree, in Fahrenheit degrees, of the steam formed under the different pressures.

In the third column is given, in heat units, the *total heat* above 32; i. e., the total quantity of heat required to raise one pound of water from 32° to the temperature of the steam, and then turn it into steam.

In the fourth column is given, in heat units, the *latent heat* of the steam.

In the fifth column is given the *volume*, in cubic feet, of one pound of the steam.

16. Superheated Steam. — When steam is separated entirely from the presence of water and heated it becomes superheated, and approaches the condition of a gas; the higher its temperature is raised the more and more it departs from the nature of a vapor and approaches that of a gas. The more nearly it assumes the condition of a gas, the more nearly will the equations of a gas apply to it.

CHAPTER II.

THEORY OF THE STEAM ENGINE.

17. THEORETICAL HEAT ENGINE.— A heat engine may be defined as *a machine for converting heat into work.*

In the theoretical heat engine we suppose, what never exists in reality, that we have a machine that works without friction and that is made of materials that will act as perfect conductors or non-conductors of heat, as desired. These suppositions are made in order that the problems may be simplified in the beginning. After we have discussed the theoretical engine we can discuss the real engine, with its attending complications of friction and losses of heat by radiation and conduction.

In all engines, theoretical or otherwise, the heat is brought from its source to the engine by what is termed the *working fluid;* this may be a liquid, a liquid and vapor, or a perfect gas. In the theoretical engine, it is supposed that all the changes through which the *working fluid* passes, during the transformation of heat into work, take place in the engine itself; the working fluid is supposed to always remain in the engine and the same portion of the fluid is supposed to be used over and over again.

18. CYCLE. — The term cycle, as applied to a heat engine, means *a number of consecutive series of changes in the condition of the working fluid, such that the final condition of the fluid, at the end of the series, is, in all respects, the same as was the initial condition at the beginning of the series.*

In every cycle three processes are gone through : —

1. A quantity of heat is given to the working fluid by a hot body.

2. A part of the heat received by the working fluid is given to a cold body.

3. A part of the heat received by the working fluid is transformed into external work.

As the working fluid is, in all respects, in precisely the same condition, as to temperature, pressure, and volume, at the end of the cycle that it was at the beginning, the internal work done during the cycle must amount to zero, and no heat can have been used, therefore, to do internal work. It follows, then, that the total amount of heat given to the working fluid during a cycle, must be equal to the sum of that given by the working fluid to the cold body and that converted into work. Let H be the total quantity of heat received from the hot body; U the quantity given to the cold body by the working fluid; and W that quantity transformed into external work. Then, if all are expressed in mechanical units,

$$H = U + W. \tag{31}$$

Equation (31) may be considered as the principal equation for the cycle of the theoretical heat engine.

H can usually be calculated when the working fluid employed is known, and when the conditions under which the heat is received are known.

W can always be obtained from the *work diagram*, formed by plotting the various changes of pressure and volume of the working fluid during the cycle.

U is sometimes difficult to calculate, but if the working fluid and the conditions under which the cooling takes place are known, there will be no trouble.

As an example of a cycle, assume that the co-ordinates of the point a in Fig. 5 represent the pressure and volume of one pound of the working fluid of a perfect engine.

Let P_1, V_1, and T_1, be the initial pressure, volume, and absolute temperature, respectively, of the fluid.

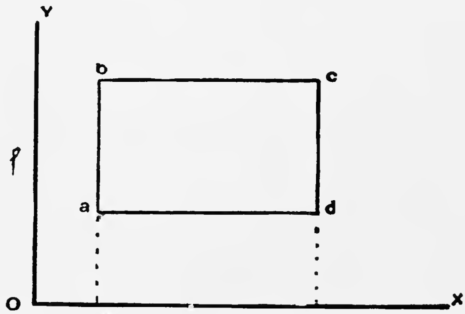


Fig. 5.

Suppose the working fluid is heated at the constant volume V_1 until the pressure is P_2, and the absolute temperature is T_2. The line *ab* will represent the changes of volume and pressure during this process.

Next, let the fluid receive heat at the constant pressure P_2 until its volume is V_2, and its absolute temperature is T_3. *bc* will represent the changes of volume and pressure during this part of the cycle.

Now, let the hot body be removed and a cold one be substituted for it, and let the working fluid be cooled at the constant volume V_2 until the pressure is P_1, and its absolute temperature is T_4. *cd* will represent the changes of volume and pressure during this process.

Let the fluid be further cooled at the constant pressure P_1 until the volume is V_1, and the absolute temperature is T_1. When this has been done, the cycle will have been completed and the fluid will have arrived at its original condition.

The heat given to the working fluid is made up of two parts: —

1. That taken in while being heated at constant volume, equal $K_v\,(T_2 - T_1)$.

2. That taken in while being heated at constant pressure, equal $K_p\ (T_3 - T_2)$.

Therefore,

$$H = K_v\ (T_2 - T_1) + K_p\ (T_3 - T_2).$$

The heat given by the working fluid to the cold body is also made up of two parts: —

1. That emitted while being cooled at constant volume, equal $K_v\ (T_3 - T_4)$.

2. That emitted while being cooled at constant pressure, equal $K_p\ (T_4 - T_1)$.

Therefore,

$$U = K_v\ (T_3 - T_4) + K_p\ (T_4 - T_1).$$

The external work done is represented by the area of the work diagram *abcd*

Therefore,

$$W = (P_2 - P_1)\ (V_2 - V_1).$$

If the values of H, U, and W, as derived, are substituted in (31), we have

$$K_v\ (T_2 - T_1) + K_p\ (T_3 - T_2) = K_v\ (T_3 - T_4) + K_p\ (T_4 - T_1) + (P_2 - P_1)\ (V_2 - V_1).$$

The changes that the working fluid is supposed to pass through during the cycle may be any we choose. In the Carnot cycle, named after Sadi Carnot, the changes are: —

1. The fluid receives heat while expanding isothermally.
2. The fluid expands adiabatically.
3. The fluid is cooled while being compressed isothermally.
4. The fluid is compressed adiabatically.

19. Thermodynamic Efficiency.— This term is used to denote *the ratio of the work done during a cycle, to the total heat taken by the working fluid from the hot body.* The algebraic expression for the efficiency is

$$E = \frac{W}{H} = 1 - \frac{U}{H}. \tag{32}$$

The external work done by a heat engine is equal to the product of the heat used, H, and the efficiency, E, or $W = HE$.

As will be seen later, the Carnot cycle is the most efficient of all possible cycles for any working fluid; and the expression for the efficiency of this cycle is

$$E = \frac{T_1 - T_2}{T_1}. \tag{33}$$

T_1 is the absolute temperature at which the working fluid receives its heat.

T_2 is the absolute temperature at which the working fluid emits its heat to the cold body.

20. Perfect Gas Engine.— The perfect gas engine is supposed to be a theoretical engine using a perfect gas as a working fluid. When a gas is used as the working fluid, all the calculations necessary to determine the heat received or emitted by the gas, during a cycle, can readily be made, since all the properties of a perfect gas are well known.

For instance, in Art. 18, it was shown that for a perfect engine using one pound of any fluid, the expressions for H, U, and W for the cycle used, are

$$\begin{cases} H = K_v\,(T_2 - T_1) + K_p\,(T_3 - T_2). \\ U = K_v\,(T_3 - T_4) + K_p\,(T_4 - T_1). \\ W = (P_2 - P_1)\,(V_2 - V_1). \end{cases} \tag{34}$$

If the fluid had been a perfect gas, it will be easy to show that $H - U = W$, as it ought.

To do this, let us refer to Fig. 5, remembering that for a perfect gas $\frac{PV}{T} = R$, and $K_p = K_v + R$, and determine the values of T_1, T_2, T_3, and T_4, in terms of P_1, V_1, P_2, and V_2.

$$T_1 = \frac{P_1 V_1}{R}.$$

$$T_2 = \frac{P_2 V_1}{R}.$$

$$T_3 = \frac{P_2 V_2}{R}.$$

$$T_4 = \frac{P_1 V_2}{R}.$$

Putting these values of T_1, T_2, T_3, and T_4, in the expressions for H and U, we get

$$(35) \quad \begin{cases} H = K_v (P_2 - P_1) \frac{V_1}{R} + K_p (V_2 - V_1) \frac{P_2}{R}. \\ U = K_v (P_2 - P_1) \frac{V_2}{R} + K_p (V_2 - V_1) \frac{P_1}{R}. \end{cases}$$

From (35), $H - U = (P_2 - P_1)(V_1 - V_2) \frac{K_v}{R} +$

$$(P_2 - P_1)(V_2 - V_1) \frac{K_p}{R}$$

$$= (P_2 - P_1)(V_2 - V_1) \left(\frac{K_p - K_v}{R}\right).$$

But $\frac{K_p - K_v}{R} = 1$, from (14), and, therefore, we have proved algebraically that

$$H - U = W = (P_2 - P_1)\ (V_2 - V_1.)$$

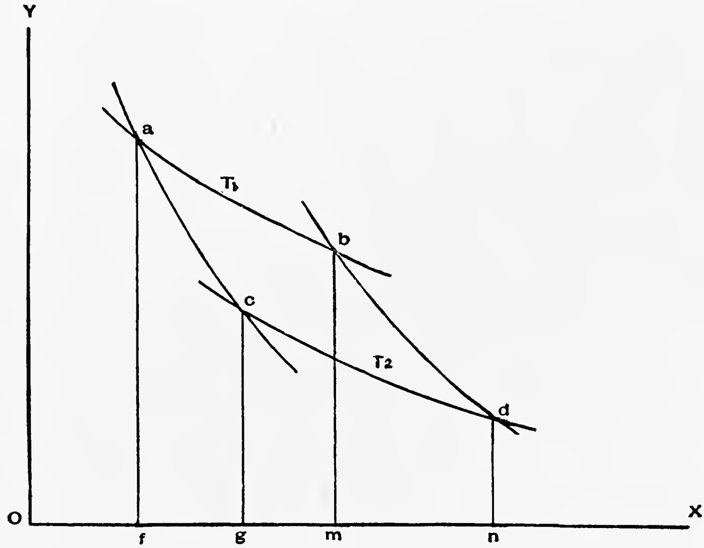


Fig. 6.

If the cycle of work of the perfect gas engine be the Carnot cycle, Fig. 6 represents the diagram of work. The point *a* represents the initial condition of one pound of the gas, where its pressure is P_1, its volume V_1, and its absolute temperature T_1. During the first period of the cycle it expands isothermally from *a* to *b*, where its pressure is P_2, its volume V_2, and its absolute temperature T_1. Next it expands adiabatically from *b* to *d*, where its pressure is P_3, its volume V_3, and its absolute temperature T_2. Then it is compressed, and heat is taken from it, so that it contracts isothermally from *d* to *c*, where its pressure is P_4, its volume V_4 and its absolute temperature T_2. During the last period it is adiabatically compressed from *c* to *a*, thus completing the cycle.

While the gas expanded isothermally from *a* to *b*, the heat given to it was that necessary to do the external work, represented by the area *abmf*, which, from (16), is

$P_1 V_1 \, hyp. \, log. \, \frac{V_2}{V_1}$. Therefore,

$$(36) \qquad H = P_1 V_1 \, hyp. \, log. \, \frac{V_2}{V_1}.$$

The heat emitted by the gas to the cold body, is equal to the work done during isothermal compression from *d* to *c*, represented by the area *cdng*, which, from (16), is

$P_4 V_4 \, hyp. \, log. \, \frac{V_3}{V_4}$. Therefore,

$$(37) \qquad U = P_4 V_4 \, hyp. \, log. \, \frac{V_3}{V_4}.$$

The external work done during the cycle is represented by the area *abdc*. Area *abdc* = area *abmf* + area *bdnm* — area *acgf* — area *cdng*. But, as has been shown in Art. 12, area *acgf* = area *bdnm*, and, therefore, W = area *abdc* = area *abmf* — area *cdng*. Since area *abmf* $= H$, and area *cdng* $= U$, we get from (36) and (37),

$$(38) \qquad W = P_1 V_1 \, hyp. \, log. \, \frac{V_2}{V_1} - P_4 V_4 \, hyp. \, log. \, \frac{V_3}{V_4}.$$

From (9) we have $P_1 V_1 = T_1 R$ and $P_4 V_4 = T_2 R$; and since *ac* and *bd* are adiabatic lines, we have, from (25),

$$\frac{T_1}{T_2} = \left(\frac{V_3}{V_2}\right)^{\gamma-1} = \left(\frac{V_4}{V_1}\right)^{\gamma-1}$$

From this relation we get $\frac{V_3}{V_4} = \frac{V_2}{V_1}$.

Putting for $P_1 V_1$, its value, $T_1 R$, for $P_4 V_4$ its value, $T_2 R$, and for $\frac{V_3}{V_4}$ its value, $\frac{V_2}{V_1}$, we have, from (36), (37), and (38),

$$(39) \quad \begin{cases} H = R\, T_1 \, hyp.\ log.\ \dfrac{V_2}{V_1}. \\ U = R\, T_2 \, hyp.\ log.\ \dfrac{V_2}{V_1}. \\ W = R\,(T_1 - T_2)\, hyp.\ log.\ \dfrac{V_2}{V_1}. \end{cases}$$

The efficiency of the perfect gas engine working according to the Carnot cycle is, from (39),

$$(40) \qquad E = \frac{W}{H} = \frac{T_1 - T_2}{T_1}.$$

Since this expression for E does not involve any *special* function or property of the working fluid, we say:

The efficiency of all heat engines, using any working fluid according to the Carnot cycle, is as given by (40), $\frac{T_1 - T_2}{T_1}$.

As a further proof, let us suppose that the perfect gas engine is used to run in the reverse direction a heat engine using a working fluid that is *more* efficient than the perfect gas. The working fluid in the second engine would *take* heat from the cold body at a temperature T_2; would have work done upon it, instead of doing work; and would *give* heat to the hot body at a temperature T_1. The second engine is transforming *work into heat* instead of *heat into work.* It is evident, therefore, that since the working fluid of the second engine is in precisely the same condition at the end of the cycle that it was at the beginning, the heat given to the hot body must equal the sum of that taken from the cold body and that resulting from the transformation of work into heat. It follows also, that if the second engine is more efficient, as supposed, than the perfect gas engine, it will give to the hot body more heat than the perfect gas engine transforms into work. But as all the work done by the perfect gas engine is used to run the second engine, the two together form a system by means of which heat is either being created or made to pass from a cold to a hot body without any disappearance of energy or change of any kind in the conditions of the working fluids. That is, by an arrangement of machinery, heat is either created or made to pass from a cold to a hot body without any compensation.

This conclusion is contrary to all our experience as to the action of heat, and to our knowledge of the transformation of heat into energy, and, therefore, cannot be considered true. Hence, the second engine is not *more* efficient than the perfect gas engine.

If it were supposed that the second engine ran the perfect gas engine in a reverse direction, the same argument would show that the second engine cannot be *less* efficient than the perfect gas engine.

We are, therefore, forced to conclude that, since the second engine is neither more efficient nor less efficient than the perfect gas engine, the two engines have the same efficiency.

By a method of proof that belongs to a more advanced work than this, it can be shown that the Carnot cycle is the most efficient of all possible cycles. Care must be taken to remember that the efficiency of the heat engine is $\frac{T_1 - T_2}{T_1}$ *only when working according to the Carnot cycle.*

It would be well for the student to work out the expressions for the work done by, and the efficiency of, the perfect gas engine working according to a number of the cycles given under the head of Problems.

21. Perfect Steam Engine. — The perfect steam

engine is a theoretical engine using water and steam as the working fluid.

In the actual steam engine the working fluid is usually considered as being steam only. This is due to the fact that the steam engine really consists of two parts that, in practice, are separated and are considered always as being separate and apart from one another. These parts are the engine, proper, and the boiler. The steam receives its heat while in the boiler, from which it passes to the engine and thère does work; then it leaves the engine and usually passes out and away. The steam drawn from the boiler is replaced by an amount of water necessary to make a quantity of steam equal to that taken away. In some cases, the steam, after leaving the engine, is condensed and returned to the boiler. In such cases, the actual engine approaches nearer to the perfect engine than in any other, as here the same working fluid is used over and over again. In the perfect steam engine the water is supposed to be converted into steam while in the engine, and the engine itself takes the place of both engine, proper, and boiler, in the actual engine. The changes that the steam passes through during one cycle of the perfect steam engine, may be supposed to be exactly the same that it would pass through in an actual engine and boiler together, if the losses of heat due to radiation and conduction are neglected, and, therefore, the work diagram of a cycle of the perfect steam engine will be the same as the work diagram of the real engine.

The cycle, then, that the perfect engine will be assumed to make will be that which approaches nearest to the cycle of the actual engine.

In Fig. 7, let a represent the initial condition of one pound of *water*, just on the point of boiling, whose volume is V_w, pressure per square foot is P_1, and temperature is t_1° F. Let heat be given to the water so that it will be all converted into steam at the temperature $t_1°$ F. The line showing the change in volume will be the line ab, parallel to OX, since the pressure remains constant. At b the volume of the steam will be V_1, the volume of one pound of steam under a pressure per square inch of $\frac{P_1}{144}$. When all the water has been converted into steam, let the hot body be taken away and let the steam expand from b to d. In the actual engine the line of expansion bd is not an adiabatic line, but usually approaches nearer the equilateral hyperbola, unless the expansion is great, whose equation is $P_1 V_1 = P_2 V_2 = PV$. We will suppose, then, in order that the cycle of the perfect steam engine may approach as nearly as possible to that of the real engine, that the steam loses some heat while expanding from b to d, and that bd is an equilateral hyperbola. While expanding from b to d, part of the heat in the steam is being transformed into work, and some of the steam is being condensed; so that, at d the fluid in the engine is a mixture of water and steam at a volume V_2, pressure per square foot P_2, and temperature $t_2°$ F. At d it is supposed that the engine is put in contact with a cold body, corresponding to the condenser of the actual engine, and all the steam is condensed at the uniform temperature $t_2°$ F. The line showing the change in volume during the condensation is dc, parallel to OX, since the pressure is constant. The point c represents one pound of water, whose volume is V_3, under a pressure per square foot of P_2, and at a temperature $t_2°$ F. Now, the engine is again put in contact with the hot body until the temperature of the water is increased from t_2 to t_1, and the pressure raised from P_2 to P_1, thus completing the cycle. The line ac represents the change in volume and pressure during this last period of the cycle, and it is sufficiently accurate to consider it as parallel to OY; the volume at c is equal to the volume at a.

The total quantity, H, of heat given to the water during the cycle is, evidently, the heat required to raise the temperature of the water from t_2 to t_1, plus the latent heat of evaporation at t_1; it can be obtained from Table 1,

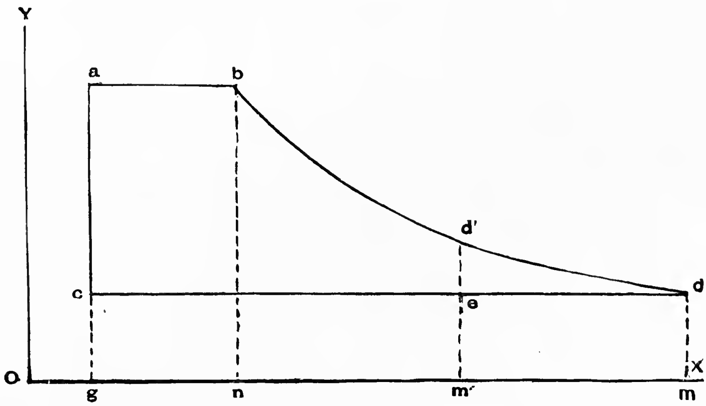


Fig. 7.

in this work, or it may be calculated by using the approxmate formulæ given in Art. 15.

The work, W, done during the cycle is represented in the work diagram by the area *abdc*. From the figure, it is seen that area $abdc =$ area $abng +$ area $bdmn -$ area $dcgm$.

Area $abng = P_1 (V_1 - V_w)$.

$$\text{Area } bdmn = \int_{V_1}^{V_2} P d V = P_1 V_1 \int_{V_1}^{V_2} \frac{d V}{V} = P_1 V_1 \, hyp. \, log. \frac{V_2}{V_1}.$$

Area $dcgm = P_2 (V_2 - V_3)$.

Therefore,

$$W = \text{area } abdc = P_1 (V_1 - V_w) + P_1 V_1 \textit{ hyp. log.} \frac{V_2}{V_1}$$

$$- P_2 (V_2 - V_3).$$

V_w and V_3 are usually so small that they may, without error, be considered as zero, and the expression for W becomes

$$(41) \qquad W = P_1 V_1 \left(1 + \textit{hyp. log.} \frac{V_2}{V_1}\right) - P_2 V_2.$$

The value of U, the quantity of heat emitted by the fluid during the cycle, cannot be calculated directly, as we do not know the quantity of heat emitted during expansion from b to d, nor the exact quantity of steam condensed during compression from d to c. The value of U, however, is given by the expression

$$U = H - W.$$

The efficiency of the engine working according to the given cycle is,

$$(42) \qquad E = \frac{W}{H} = \frac{P_1 V_1 \left(1 + \textit{hyp. log.} \frac{V_2}{V_1}\right) - P_2 V_2}{H}.$$

If the engine had followed the Carnot cycle its efficiency would have been

$$\frac{T_1 - T_2}{T_1}.$$

22. Theoretical Diagram of the Real Engine.—In order that all may fully understand the explanation of the action of the steam in the engine, those who are not

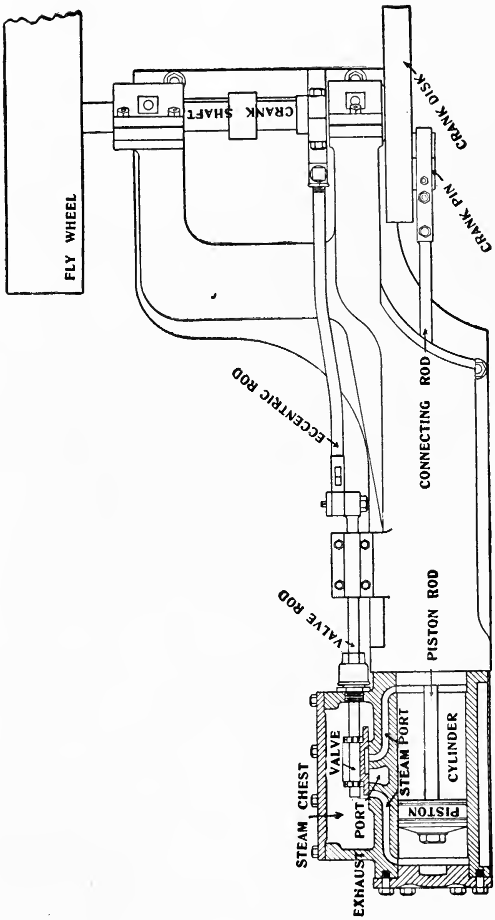

FLY WHEEL
CRANK SHAFT
CRANK DISK
CRANK PIN
ECCENTRIC ROD
CONNECTING ROD
VALVE ROD
PISTON ROD
STEAM CHEST
VALVE
PORT
EXHAUST
STEAM PORT
CYLINDER
PISTON

Fig. 8.

familiar with the names of the different parts of the engine should refer to Figs. 8 and 9.

The steam is taken from the boiler to the "steam chest" of the engine, through the "steam pipe." From the steam chest it passes through the "steam ports" into the "cylinder," and there moves the "piston." The motion of the piston is communicated through the "piston rod" to the "connecting rod," then to the "crank," by means of which the "crank shaft" is given a rotating motion. While the piston moves from one end of the cylinder to the other, makes one "stroke," the crank makes half a revolution.

The "point of cut-off" is that point in the stroke at which the piston is when steam ceases to be admitted to the cylinder: thus, the point of cut-off is at one-quarter stroke, if the steam ceases to be admitted at the instant the piston has finished one-quarter of a stroke.

The action of the steam in the cylinder is as follows: It begins to enter the cylinder when the piston is beginning its stroke, and by its pressure forces the piston forward. As the piston moves forward steam is generated, at a constant pressure, in the boiler, and flows into the cylinder; so that the volume displaced by the piston is kept constantly filled with steam at the boiler pressure. When the piston reaches the point of cut-off the valve closes communication between the steam-chest and the cylinder, and steam can no longer enter the cylinder. From this on to the end of the stroke, the steam expands and drives the piston forward simply by its expansive force. As the piston reaches the end of its stroke the valve opens the exhaust port, and the steam at once rushes out of the cylinder into the place of exhaust, until the pressure in the cylinder becomes about equal to that of the place into which the steam is exhausted. If the engine exhausted into the atmosphere, the pressure of the steam in

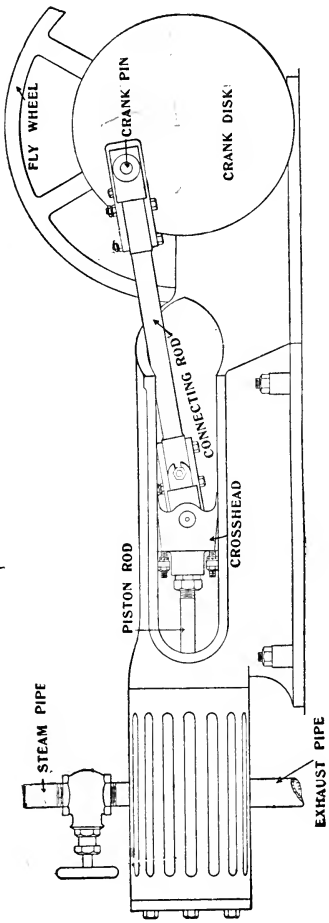

FLY WHEEL
CRANK PIN
CRANK DISK
CONNECTING ROD
CROSSHEAD
PISTON ROD
STEAM PIPE
EXHAUST PIPE

Fig. 9.

the cylinder would drop almost to the atmospheric pressure when the exhaust port is opened. In order to empty the cylinder of the steam remaining in it, the piston is forced back to its original position, against whatever pressure there may be in the place of exhaust, either by the pressure of steam admitted on the other side of the piston or by the momentum of the fly-wheel fixed to the shaft.

To make the diagram of work done by the engine during one forward and backward stroke, assume, in Fig. 10, that *ao* represents the *absolute pressure*, P_1, in pounds per *square inch*, of the steam entering the boiler. Since the steam enters at constant pressure up to the point of cut-off, the line *ab* will represent the volume, V_1, in cubic feet, of the steam admitted to the cylinder at a pressure per *square foot* of 144 P_1.

From the point of cut-off the steam expands until the piston reaches the end of the stroke, when the volume of the steam is, in cubic feet, V_2, and its pressure is P_2 pounds per square inch, as represented by the point *c*. The line of expansion, *bc*, may be assumed as the equilateral hyperbola, whose equation is $P_1 V_1 = P_2 V_2 = P V$.

When the exhaust port is opened, at the end of the stroke, the pressure immediately falls to P_3, as represented at *d*.

On the return stroke, the piston moves against the constant "back pressure" P_3, as represented by the line *de*.

During the forward stroke of the piston, the steam does the work on the *piston* represented by the area *abcfo*; and during the return stroke, the *piston* does the work on the *steam* represented by the area *edfo*. Therefore, the *effective* work done by the steam, for each forward stroke of the piston, is represented by the area of the diagram *abcde*.

From the figure, we see that

$$\text{area } abcfo = \text{area } abgo + \text{area } bcfg,$$

$$\text{area } abgo = 144\ P_1\ V_1,$$

$$\text{area } bcfg = 144 \int_{V_1}^{V_2} P d V = 144\ P_1\ V_1 \int_{V_1}^{V_2} \frac{dV}{V} = 144\, P_1 V_1\, hyp. log. \frac{V_2}{V_1}.$$

Therefore, the work done during a forward stroke is

$$(43) \qquad W_1 = \text{area } abcfo = 144\ P_1\ V_1\ (1 + hyp.\ log.\ \frac{V_2}{V_1}).$$

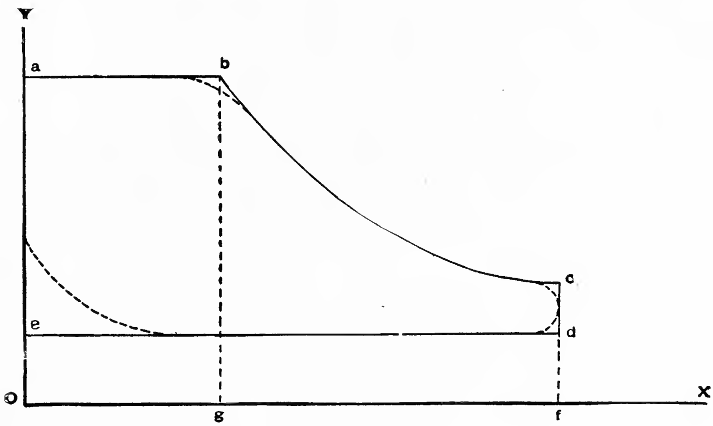


Fig. 10.

If we call P_m the *mean forward pressure* in pounds per square inch on the piston during the forward stroke, then, since the work done is equal to the pressure per square foot multiplied by the volume swept through in cubic feet, we have

$$(44) \qquad W_1 = 144\ P_m\ V_2.$$

From (43) and (44) we get

$$(45) \qquad P_m = \frac{P_1\left(1 + hyp.\ log.\ \frac{V_2}{V_1}\right)}{\frac{V_2}{V_1}}.$$

The ratio $\frac{V_2}{V_1}$ represents the number of times the steam is expanded. If this ratio be denoted by r, the cut-off, $\frac{V_1}{V_2}$, will be $\frac{1}{r}$; and the expression for the mean forward pressure becomes

$$(46) \qquad P_m = \frac{P_1\,(1 + hyp.\ log.\ r)}{r}.$$

On the return stroke, the work, W_2, done by the piston on the steam is that represented by the area *edfo*, and, from the figure,

$$(47) \qquad W_2 = \text{area } edfo = 144\ P_3\ V_2.$$

Since the effective work, W_3, done by the steam on the piston is equal to that done on the forward stroke minus that done on the return stroke, we have,

$$(48) \qquad W_3 = W_1 - W_2 = 144\,P_m\,V_2 - 144\,P_3\,V_2.$$

If we let P_e be the *mean effective pressure* per square inch on the piston, we shall have

$$(49) \qquad W_3 = 144\ P_e\ V_2 = 144\ P_m\ V_2 - 144\ P_3\ V_2.$$

Therefore, from (46) and (49),

$$(50) \qquad P_e = P_m - P_3 = P_1\,\frac{(1 + hyp.\ log.\ r)}{r} - P_3.$$

If A is the area of the piston in *square inches*, and L is the length of *stroke in feet*, then

$$(51) \qquad V_2 = \frac{AL}{144}.$$

Put this value of V_2 in (49) and we get

$$(52) \qquad W_3 = P_e\, L\, A.$$

Now let N be the number of forward *strokes* the engine makes per *minute*. For a double acting engine, one that takes steam on both sides of the piston, N will be equal to twice the number of revolutions made by the engine per minute; and for a single acting engine, N will be equal to the number of revolutions made per minute. Counting the revolutions made by an engine is an easy method of determining the value of N.

The work, W, done per minute by the engine is, evidently, $N\ W_3$, and, therefore, from (52),

$$(53) \qquad W = P_e\, L\ A\ N.$$

If $H.\ P.$ represents the horse power of the engine,

$$(54) \qquad H.\ P. = \frac{W}{33000} = \frac{P_e\, L\ A\ N}{33000}.$$

The diagram *abcde*, in Fig. 10, is termed the theoretical *Indicator Diagram* of the real engine. The actual diagram differs from that in Fig. 10 in that, owing to the friction of the steam in the ports and the mechanical imperfections in the valves, the corners of the diagrams are always more or less rounded. The line *ed* has been drawn as if the back pressure were constant, while as a matter of fact it is not; the exhaust usually closes before all of the steam has been forced from the cylinder, on the return stroke, and thus confines a greater or less quantity of steam in the cylinder and which is compressed as the piston is forced back. The amount of this compression may be great or small, depending upon the type of engine, but the greater it is the greater is made the mean back pressure, P_3. The absolute value of P_3 for engines that exhaust into the atmosphere will vary from 16 to 20 lbs., depending upon the engine; and for condensing engines, those that exhaust into a condenser, the absolute value of P_3 will vary from $3\frac{1}{2}$ to 8 lbs., or even higher. The dotted lines, in Fig. 10, show how the theoretical diagram must be changed to conform to the actual diagram.

23. Clearance.— In the real engine the piston must not touch the end of the cylinder when at the end of the stroke, and there is, therefore, a space of greater or less volume between the piston and the cylinder end. In addition to this volume there is, also, the volume of the steam ports that must be filled by steam. The sum of these two volumes form what is termed the clearance, or clearance volume of the engine.

The clearance is prejudicial to thermodynamic efficiency of the real engine, as it preserves a volume of what might be termed non-active steam subject to condensation; it usually increases the amount of steam required to do a given amount of work, and decreases the number of times the steam is expanded for a given cut-off.

If c represents the volume in cubic feet of the clearance, the volume occupied by the steam at the point of cut-off will be $V_1 + c$; and at the end of the stroke, the volume of the steam will be $V_2 + c$. The real number of times, n, the steam is expanded will be

$$(55) \qquad n = \frac{V_2 + c}{V_1 + c} = \frac{1 + \frac{c}{V_2}}{\frac{V_1}{V_2} + \frac{c}{V_2}} = \frac{r\left(1 + \frac{c}{V_2}\right)}{1 + r\frac{c}{V_2}}$$

That is, the clearance has reduced the number of expansions, r, in the ratio of

$$\left(1 + \frac{c}{V_2}\right) \text{ to } \left(1 + r\frac{c}{V_2}\right).$$

If we call m the ratio of the clearance volume, c, to the volume, V_2, swept through by the piston in one stroke, we shall have $m = \frac{c}{V_2}$, and, from (55),

$$n = \frac{r\left(1 + \frac{c}{V_2}\right)}{1 + r\frac{c}{V_2}} = \frac{r(1+m)}{1+mr}. \tag{55a}$$

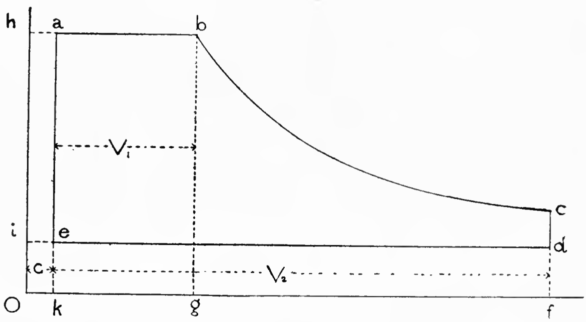


Fig. 11.

In Fig. 11 let $a\ b\ c\ d\ e$ represent the diagram of an engine with a clearance volume c; and let, as before, the expansion line $b\ c$ be an equilateral hyperbola.

The volume occupied by the steam at cut-off, represented by the line $h\ b$, is $V_1 + c = a\ b + h\ a$; and the pressure per square inch during admission is the initial pressure, P_1, represented by the line $b\ g$. The volume occupied by the steam at the end of the stroke, represented by the line $i\ d$, is $V_2 + c = e\ d + i\ e$; and the

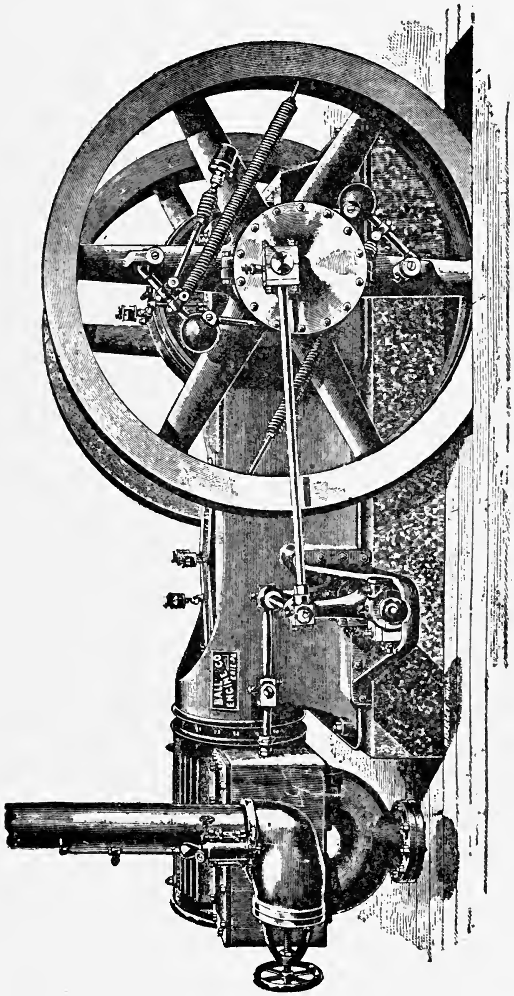


Fig. 12. Ball Engine.

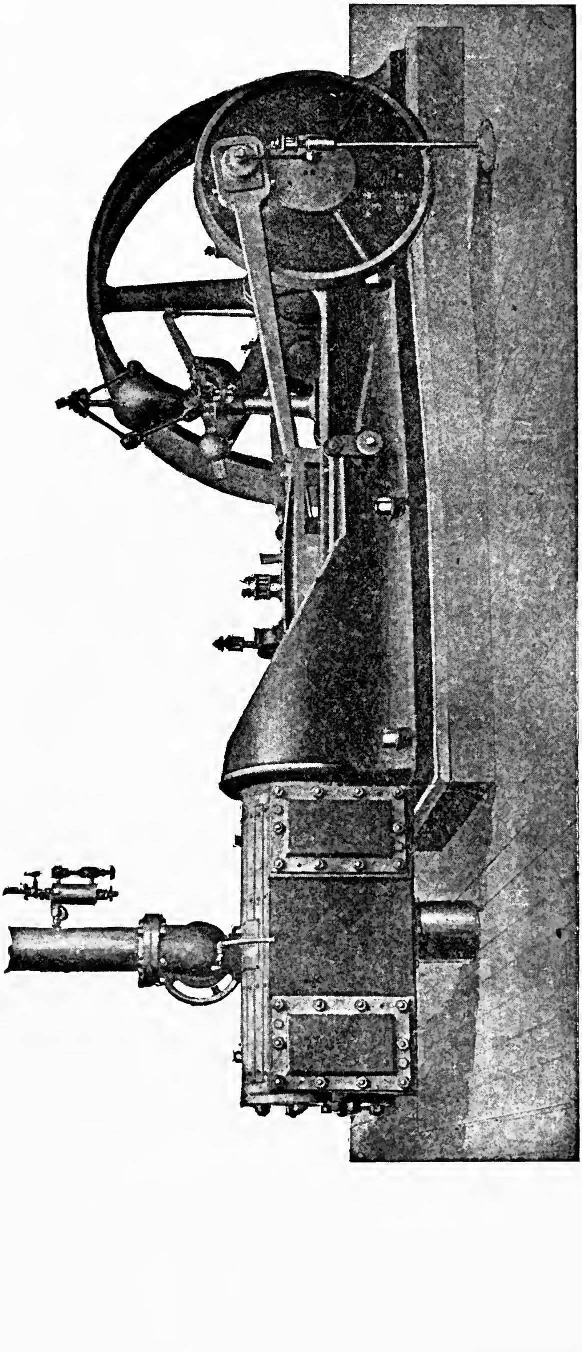


Fig. 13. Porter-Allen Engine.

pressure per square inch against which the piston moves during the return stroke is P_3, represented by the line $d\ f$.

The work done during the forward stroke is represented by the area $a\ b\ c\ f\ k$. By inspection we see that

$$a\,b\,c\,f\,k = h\,b\,g\,o + b\,c\,f\,g - h\,a\,k\,o.$$

$$h\,b\,g\,o = 144\ P_1\ (V_1 + c).$$

$$b\,c\,f\,g = 144 \int_{V_2+c}^{V_1+c} P\,d\,V = 144\ P_1\ (V_1 + c)\ \text{hyp. log.}\ \frac{V_2 + c}{V_1 + c},$$

but $\dfrac{V_2 + c}{V_1 + c} = n$, from (55), and hence

$$b\,c\,f\,g = 144\ P_1\ (V_1 + c)\ \text{hyp. log.}\ n.$$

$$h\,a\,k\,o = 144\ P_1\ c.$$

Therefore

$$a\,b\,c\,f\,k = 144\ P_1\ (V_1 + c)\ (1 + \text{hyp. log.}\ n) - 144\ P_1\ c.$$

If we let P'_m represent the mean forward pressure per square inch, the work done during the forward stroke is 144 $P'_m\ V_2$. Putting this expression equal to the expression for $a\ b\ c\ f\ k$, given above, we have

$$144\ P'_m\ V_2 = 144\ P_1\ (V_1 + c)\ (1 + \text{hyp. log.}\ n) - P_1\ c.$$

From which we have

$$P'_m = P_1\ \frac{(V_1 + c)}{V_2}\ (1 + \text{hyp. log.}\ n) - \frac{P_1\ c}{V_2}$$

But $\dfrac{c}{V_2} = m$, and from (55)

$$\frac{V_1 + c}{V_2} = \frac{V_2 + c}{n\ V_2} = \frac{1 + \dfrac{c}{V_2}}{n} = \frac{1 + m}{n}.$$

Therefore

$$(55b) \qquad P'_m = P_1\ (1 + m)\ \frac{(1 + \text{hyp. log.}\ n)}{n} - P_1\ m.$$

The mean effective pressure per square inch, P'_e, is evidently the mean forward pressure, P'_m, less the back pressure, P_3. Hence

$$(55c)\quad P'_e = P'_m - P_3$$
$$= P_1 (1 + m) \frac{(1 + \text{hyp. log. } n)}{n} - P_1 m - P_3.$$

This expression gives the true mean effective pressure per square inch when clearance is taken into account. The effect of clearance is not only to reduce the number of times the steam is expanded, but also to very materially change the expression for the mean effective pressure, as can be seen by comparing (50) and (55*c*).

24. Efficiency of the Actual Engine.—The thermodynamic efficiency of the actual engine is expressed by the same equation as the efficiency of the perfect steam engine, and is

$$E = \frac{W}{H}.$$

Where W is the work done by a given weight of steam, and H is the total heat in mechanical units, required to raise the temperature of the same weight of water from the initial temperature of the water up to the temperature of the steam and there turn it into steam.

The work done per stroke by an engine is, from (52), $P_e\, L\, A$, and the volume, in cubic feet, of the steam, at the initial pressure, used per stroke is $V_1 = \frac{V_2}{r}$.

From (51), $V_2 = \frac{L\,A}{144}$; and, therefore, the volume of steam used per stroke is

$$(56)\qquad V_1 = \frac{L\,A}{144 r}.$$

If s is the volume in cubic feet of one pound of steam at the initial pressure, as given in Table 1, the weight, S, of steam used per stroke will be

$$(57) \qquad S = \frac{V_1}{s} = \frac{L\,A}{144\,r\,s}.$$

Let h_1, as in (29), be the total quantity of heat, in mechanical units, required to raise the temperature of one pound of water from the initial temperature of the water to the temperature of the steam and there convert it into steam. Then the heat, H_1, required for the weight, S, of steam used per stroke will be, from (57),

$$(58) \qquad H_1 = S\,h_1 = \frac{L\,A\,h_1}{144\,r\,s}.$$

Since the work done per stroke is $P_e\,L\,A$, and the heat used per stroke is H_1, the efficiency of the engine is

$$(59) \qquad E = \frac{P_e\,L\,A}{H_1} = \frac{144\,P_e\,r\,s}{h_1}.$$

As the perfect steam engine had its efficiency decreased by departing from the Carnot cycle, so too, the efficiency of the actual engine is less than that of the perfect engine the more it departs from the cycle of the perfect engine. In the perfect engine the expansion is always continued until the pressure of the steam is that corresponding to the temperature of the feed water. In Fig. 7, let *dm* be the pressure of the steam corresponding to the temperature of the feed water, then the work done by the perfect steam engine would be represented by the area *abdc*, which is greater than the work that would be done by the actual engine, by the area *d′de*.

The actual engine loses a great deal of heat by radiation and conduction, which results in a condensation of

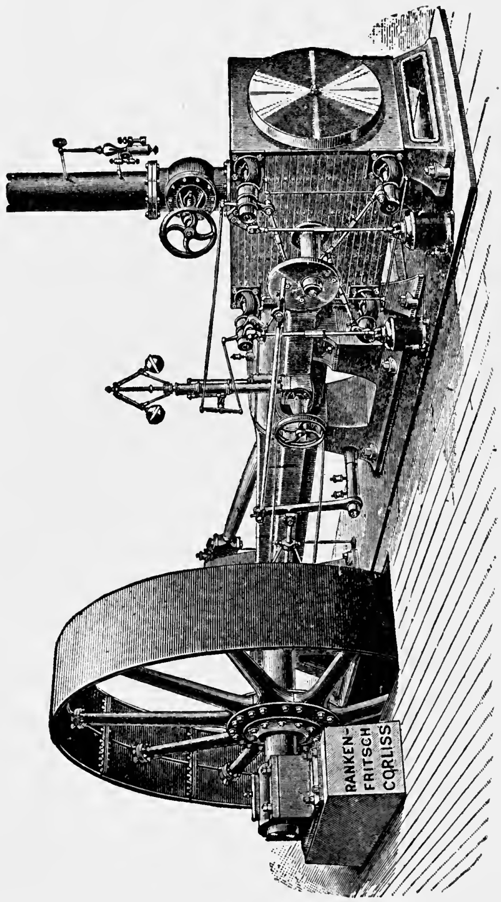


Fig. 14. Ranken-Fritsch Corliss — Rear View.

the entering steam; and the volume of steam that must be applied to the engine is V_1, plus that required for clearance, plus that required to compensate for what is condensed.

A part of the work, resulting from the transformation of the heat of the steam, is used in overcoming the friction of the moving parts of the actual engine; so that the engine has a mechanical efficiency as well as a thermodynamic efficiency, and the two must be carefully distinguished from one another.

CHAPTER III.

TYPES AND DETAILS OF ENGINES.

25. Classification of Engines. — Engine builders usually classify their engines in two great classes as follows: —

1. Condensing Engines.
2. Non-condensing Engines.

Condensing engines are those that exhaust the steam into a condenser; they may be either simple, have but a single cylinder, or compound, have two or more cylinders. While a condensing engine may be a simple engine, most of them are compound and expand the steam a greater number of times than could be well done in a simple engine. The mean back pressure, P_3 in Art. 22, is always less than atmospheric pressure in the case of a condensing engine.

Non-condensing engines are those that exhaust the steam into the atmosphere; and, while they may be either simple or compound, they are usually simple. The mean back pressure in these engines is always greater than the atmospheric pressure, when the engine is running properly.

Engines are sometimes classified according to whether they are used on land or on the ocean, as Land engines and Marine engines.

They may be classified according to the position of the cylinder, as Vertical engines or Horizontal engines.

In this work, all engines will be considered as coming under one of the following heads: —

1. Plain Slide Valve Engines.
2. Automatic High Speed Engines.
3. Corliss Engines.

There are engines on the market that on account of peculiarities of design it would be extremely difficult to

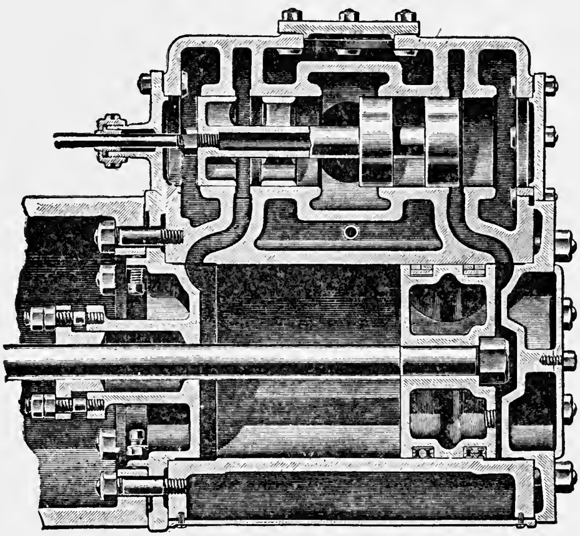


Fig. 15. Horizontal Section.
Cylinder, Dick and Church Engine.

determine exactly under which head they would belong; but such engines must be considered simply as connecting links, in which an endeavor has been made to obtain all the good qualities of the engines of two or more types. Such engines are often, however, unfitted for any special kind of work, as they possess to a certain degree qualities that make them better than is necessary for some work, and yet do not possess the same qualities to a sufficiently high degree to make them well fitted to do other special work.

26. Plain Slide Valve Engines. — These engines are usually plain in appearancc, and use a simple form of *D*-valve. Some are heavy, well made, and made of good material; others are slight in weight, poorly made, and made of the poorest material. To this type of engines belong most, if not all, of the very cheap engines. Engines of this type, however, will stand more hard usage and neglect than perhaps any other on the market; good ones, of course, will stand more than cheap, poor ones. They have rather gone out of fashion at present, and yet, for some conditions and some kinds of work, they are good engines.

Engines of this type have a fixed cut-off, which can only be changed by re-adjusting the engine, and they regulate by throttling the incoming steam so as to reduce the initial pressure in the cylinder. The cut-off usually occurs at about three-fourth stroke; the clearance is about ten to twelve per cent of the volume swept through by the piston; and there is little compression of the exhaust steam. It results, therefore, that as the steam has practically no expansion, these engines are not economical in the use of steam.

The mean back pressure for engines of this type may be taken as about seventeen or eighteen pounds absolute.

In favor of engines of this type it may be said that they are simple in construction; require very little attention; are difficult to put much out of order, and are easily repaired when deranged. They are suitable for out-of-the-way places where facilities for repairs are few, and for places where fuel is cheap and the work to be done is practically constant.

The mechanical efficiency, as well as the thermodynamic efficiency, of engines of this type is often small.

In Fig. 8 is shown a section of an engine of this type, and in Fig. 9 is shown a side view of one.

27. Automatic High Speed Engines.—This type of engines may be considered as the modern type, as it has been evolved since the beginning of the great use of electricity, and, indeed, is the result of the demand for an engine to be used to run electric machinery. With the

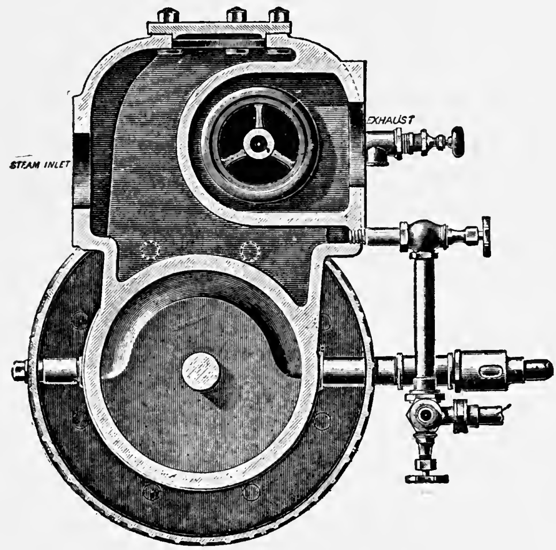


Fig. 16. Cross-section.
Cylinder, Dick and Church Engine.

advent and great use of dynamos there was at once a demand for small engines that could run at a high rotative speed, that would be fairly economical under fluctuating loads, and, above all, would run at a uniform speed under great changes of load. The plain slide-valve type of engines could not be said to satisfy any one of these conditions; the Corliss engine did not satisfy the first condition as to high rotative speed, and, then, they were too large, and, for small plants, occupied too much space.

Engines of this type are termed *high speed* not on account of the speed of the piston, but on account of the number of revolutions they will make per minute. The speed of the engine is kept almost constant by automatically changing the point of cut-off and the amount of compression to suit the various fluctuations of the load. The increase or diminution in the number of revolutions will usually be about one per cent, or less, for a sudden change of from full load to no load, or no load to full load. This increase or diminution of speed, however, will seldom last more than a few revolutions. The number of revolutions made by an engine of the automatic high speed type will depend upon the length of stroke and upon the make; the shorter the stroke the greater the number of revolutions. Ordinarily, the number of revolutions that an engine of this type will make may be obtained by the formula, $N = \frac{1500}{\sqrt[3]{L^2}}$.

Where N is the number of revolutions made per minute; and L, the length of the stroke *in inches*.

On account of the high speed of rotation, engines of this type must have large, ample bearings; all parts must be carefully proportioned, fitted and adjusted, and made of good material. The greatest source of trouble with these engines, which is overheating of bearings, can usually be traced to poor materials and workmanship.

Engines of this type almost invariably use some form of balanced valve, which is automatically made to change the amount of steam admitted to the cylinder in such a manner that the amount of steam admitted is nearly proportional to the work to be done by the engine.

The cylinders of engines of this type are usually of comparatively large diameter and short stroke; the diameter is usually between 0.60 and 0.80 of the stroke, and is often equal to the length of the stroke. These proportions of cylinders mean a short engine for a given horse power, and a comparatively large clearance.

The clearance varies from about 5 to 10 per cent of the volume swept through by the piston; it may usually be

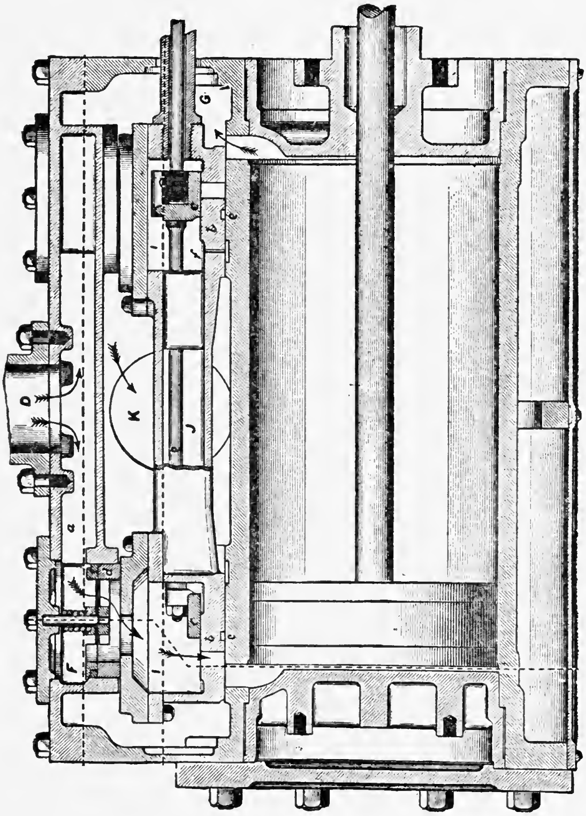


Fig. 17.
Cylinder, Buckeye Engine.

taken as about 10 per cent for those engines having a single valve, and 4 to 6 per cent for those having a system of multiple valves.

The mean back pressure for engines of this type may be taken as about eighteen or twenty pounds absolute.

Most engines of this type have a center crank, although some are made with a side crank.

Summing up, it may be said in favor of engines of this type that they are fairly economical in the use of steam; occupy small space for a given power; regulate well under a fluctuating load; and, as compared to engines of the Corliss type, are of small first cost.

The engines of this type require careful attention to be paid to the bearings, on account of the high speed of rotation, and to the adjustments of the valves and other moving parts. All bearings must be kept in good condition and well lubricated.

Figs. 11 and 12 illustrate engines of this type.

28. Corliss Engines.— Under this head the author includes, in addition to the engines of the pure Corliss type, all of those engines that, even though not having the Corliss valve gear, have more of the characteristics of the Corliss engines than of the engines of the types already described. Engines of this type, while having a high velocity of piston, have a rather slow speed of rotation; even the smaller sizes seldom make more than 100 revolutions per minute. This is due principally to the nature of the valve gearing used to operate the valves. An average value of the number of revolutions made per minute by engines of this type will be given by the equation $N = \frac{800}{\sqrt[3]{L^2}}$, where L is the length of stroke *in inches.*

The cylinders of engines of this type are usually of comparatively small diameter and long stroke. The diameter varies from one-third to two-thirds the length of the stroke, but is usually about one-half the length of the stroke. These proportions of cylinders mean a long engine, occupying much space, for a given horse-power.

Engines of this type usually use a system of multiple valves, which, by means of suitable mechanism, are made to cut-off the steam to suit the requirements of the load, without changing the amount of compression of the exhaust steam. There is usually one steam valve and one exhaust valve for each end of the cylinder, and the governing mechanism changes the action of the steam valve *only*.

In order to preserve a uniform velocity of rotation, the engines of this type not only have the cut-off automati-

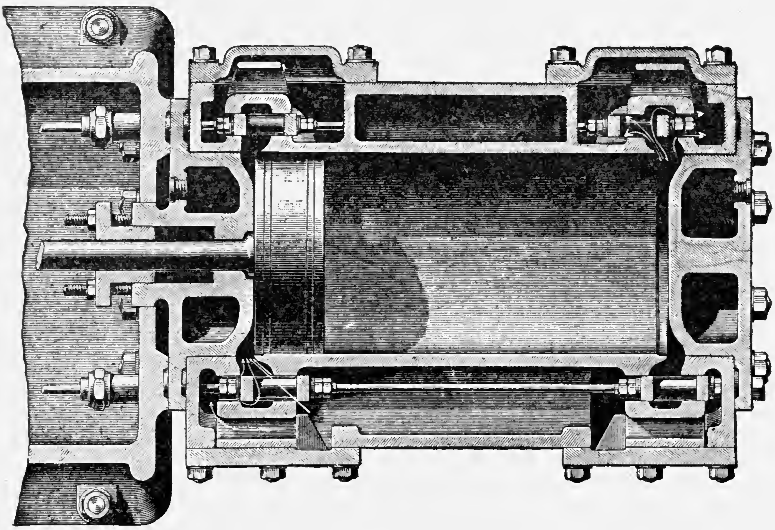


Fig. 18.
Cylinder, Porter-Allen Engine.

cally changed to suit the varying fluctuations of the load, but are provided, also, with large heavy fly-wheels, in which surplus energy is stored when the load is decreased, and from which energy may be drawn when the load is suddenly increased. By the aid of the automatic cut-off valves and the large fly-wheels, the variation in speed of engines of this type may be made as small as desired.

The clearance in engines of this type is usually about 2 per cent of the volume swept through by the piston, although it varies from 1 to 4 or 5 per cent of that volume.

The mean back pressure for engines of this type may be taken as about 16 to 18 pounds absolute.

As the engines of this type make comparatively few revolutions per minute, the bearings are not so apt to get hot as in the case of engines of the automatic high speed type.

This type of engines is the most economical of all types, and for large establishments requiring much power is undoubtedly the best. The engines of this type, however, are of greater first cost, and occupy more space than do engines of the automatic high speed type. They require considerable care and attention, and have a number of small, light parts to be kept in repair and adjustment.

Engines of this type are illustrated by Figs. 13 and 14.

29. Cylinder and Valve Chest.— The cylinders of engines are made of cast iron, with walls sufficiently thick to stand the stress induced by the pressure of the steam, and, also, the straining due to the motion of the piston back and forth. The thickness should be such as to allow at least one reboring. They are all true cylinders inside, but the shape of the outside will depend upon the style of engine and the maker. Most cylinders for short stroke engines overhang the beds of the engines ; some are cast solid with the beds, others are cast separate and bolted on. This last form is perhaps the better, as the cylinder can then be rebored with less trouble.

The steam chest is usually cast with the cylinder, although it is sometimes cast separate and bolted on. Its form and dimensions depend upon the valve, ports, and type of the engine.

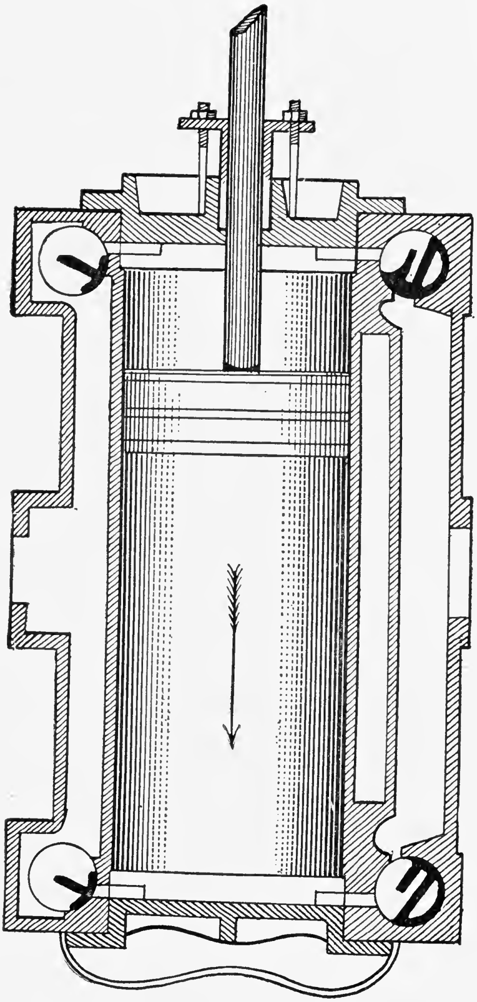


Fig. 19. Section of Cylinder of Corliss Engine.

The ports should be large, with smooth surfaces and without any sharp or abrupt changes in direction. The area of cross-section of the ports should be such that the steam will travel at a velocity between 100 and 150 feet per second when passing into the cylinder. The ports should slope from the cylinder towards the steam chest, so that all water that is condensed in the cylinder may easily drain away.

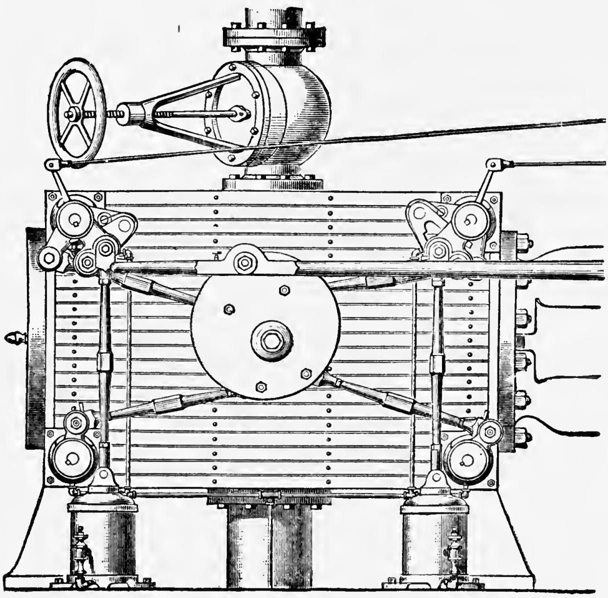


Fig. 20.
Cylinder of Corliss Engine.

All cylinders should be provided with drip cocks, for draining the cylinder and steam chest.

Some of the various methods of inserting the heads, and protecting the cylinders by lagging, are shown in the cuts of cylinders in this work.

30. PISTON.— The main point to be considered in connection with a piston is tightness, as a leaky piston reduces the efficiency of the engine very materially. For engines of the high speed automatic type, where the weight of the reciprocating parts is used to aid in regulating the engine, weight is of an advantage rather than a disadvantage; while in the case of engines of the Corliss type, weight is a disadvantage. The pistons of engines of the automatic high speed type are, usually, much thicker in proportion to the diameter, than those of the Corliss type.

Pistons are usually made tight by using as "packing rings," split cast iron rings that are turned slightly larger in diameter than the bore of the cylinder. When they are in the cylinder, the elasticity of the cast iron keeps the rings pressed out against the cylinder, and thus prevents the passage of the steam. The piston may be a single casting, with grooves into which the packing rings are sprung, or may be built up, as shown in Fig. 21. There, *A* is the "spider;" *C*, the "chunk" or "bull" ring; *D*, the "packing ring;" and *E*, the "follower plate." The bolts marked *O* are for adjusting the bull ring so that it will always run true in the cylinder, even if the center of the piston rod should not coincide with the center of the cylinder.

31. CROSS-HEAD.— The cross-head consists of the body of the cross-head, the "slippers" or bearing surfaces, and the "cross-head pin."

The cross-head is guided in its backward and forward motion by the bearing surfaces of the top and bottom "guides." Center crank engines have usually two top guides and two bottom guides, as the cross-heads are made with two sets of bearing surfaces, one at each end of the cross-head pin, as shown in Fig. 22; while side crank engines have usually one top guide and one bottom guide, as the cross-heads have the general form shown in Fig. 23. For automatic high speed engines, the bearing surfaces are usually plane surfaces, as shown in Fig. 22; while for Corliss engines, they are usually V-shaped, or cylindrical as shown in Fig. 23. Engines are usually run "over," so as to bring the pressure of the cross-heads always on the bottom gudes: that is, the engines are run so that an observer facing an engine with his left hand towards the cylinder, sees the fly-wheel revolve from left to right. The bearing surfaces of the cross-heads are usually made of some anti-friction wearing metal, such as Babbitt metal, and care should be

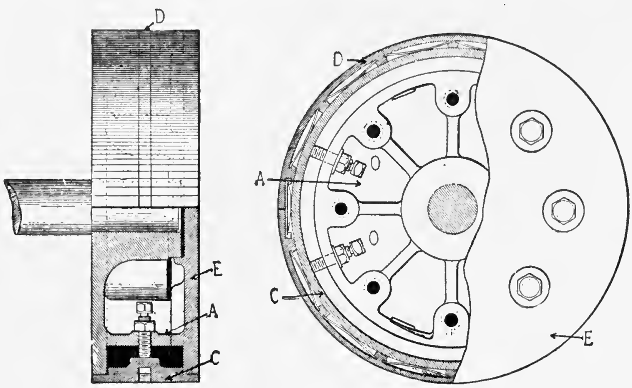


Fig. 21.

taken to keep them well lubricated. It is always best to have some arrangement by means of which adjustment may be made for the wear of the slippers, so as to keep the line of motion of the center of the cross-head pin coincident with the center line of the cylinder. If there is no means of adjusting for the wear, and the line of motion of the center of the cross-head pin is not coincident with the center line of the cylinder, the friction on the cross-head pin and crank pin is increased, and, also, the cross-head will run loose in the guides and cause a knocking noise when the load on the engine is suddenly changed.

The cross-head pin is sometimes cast solid with the body of the cross-head, and then turned up either by hand or, if the shape of the cross-head permits, by machinery; often, however, the pin is made separate and put into the cross-head. This last method gives no advantage, to the user of the engine, over the method of making the pin and cross-head body one casting, unless the pin is put into the body in such a way that it can be removed at any time for returning. Separate pins are usually made of steel. Many engine builders flatten the top and bottom of the cross-head pin in order to reduce the wear. It is

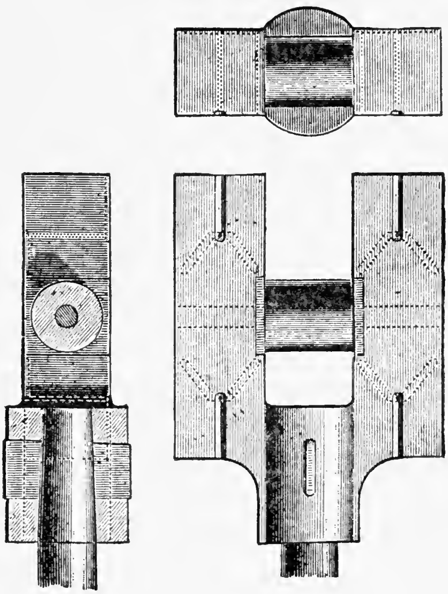


Fig. 22.
Cross-head, Porter-Allen Engine.

doubtful, however, whether this practice attains its object. Ample facilities should be provided for good and proper lubrication of the cross-head pin.

Cross-heads are shown in Figs. 22 and 23.

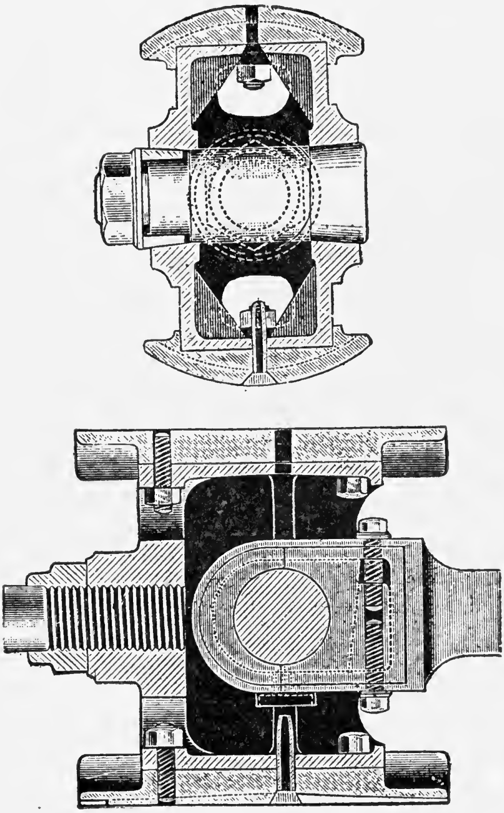


Fig. 23.
Cross-head, Ide Engine.

32. Connecting Rod.— The motion of the cross-head is transmitted to the crank-pin through the connecting rod, which is always made either of wrought iron or steel. If it is assumed that the crank-pin moves with a uniform velocity, the length of the connecting rod will have a marked influence upon the velocity of the cross-head. If the connecting rod could be so arranged that it would always remain parallel to the line of motion of the piston, then the distance that the piston has moved from the end of its stroke, for a given movement of the crank, would always be equal to the distance from the position of the

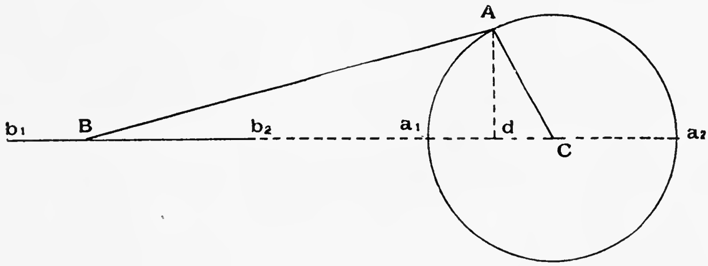


Fig. 24.

crank-pin when on dead center to the foot of a perpendicular let fall from the crank-pin on the line of motion of the piston.

In Fig. 24, let B represent the cross-head of an engine, which being rigidly connected to the piston has the same motion as the piston; and let the line BC represent the line of motion of the piston. Also, let A represent the center of the crank-pin revolving about C and connected to B by the connecting rod BA. Now, if the rod were infinitely long it would always remain parallel to the line BC, and the distance that the piston would have moved while the crank moved from a_1 to A would be equal to the distance $a_1\,d$. Since, however, the connecting rod, BA, does not remain parallel to BC, but is oblique to it, for all positions of the crank-pin except at a_1 and a_2, the distance, $b_1 B$, that the piston actually is from the end of its stroke is not equal to $a_1 d$. As will be shown in Art. 47, it is known that during the forward stroke the distance the piston moves from the end of the stroke, for a given motion of the crank-pin, is greater than it would be if there were no obliquity to the connecting rod; and during the return stroke, the movement of the piston is less than it would be if there were no obliquity.

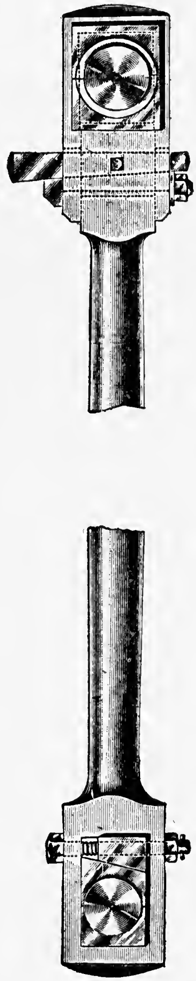


Fig. 25. Connecting Rod, Bates Corliss Engine.

The length of the connecting rod is usually made equal to three times the length of the stroke for Corliss engines, and about two and a half times the length of the stroke for automatic high speed engines.

The cross-section of the connecting rod of a Corliss engine is usually a circle, and that of the connecting rod of an automatic high speed engine is usually a rectangle whose greatest dimension is the depth of the rod.

The connecting rod has at one end the "cross-head pin brasses," and at the other, the "crank-pin brasses." The "brasses" are castings of brass, fastened to the connecting rod in various ways, which form the bearing surfaces of the rod on either the cross-head pin or the crank-pin. In order that, as the brasses wear, the length of the rod, measured from center of cross-head brasses to center of crank-pin brasses, may remain constant, it is necessary to provide a means of taking up the wear.

In Fig. 25 is shown a connecting rod whose cross-head

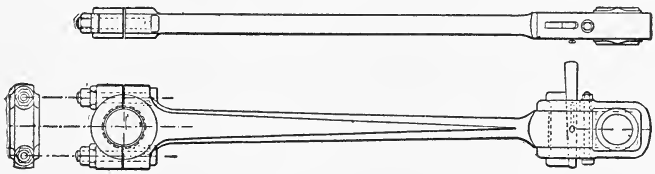


Fig. 26.
Connecting Rod, Porter-Hamilton Engine.

end is of a solid box form, into which the brasses fit; the crank-pin end has the brasses attached to it by means of a "strap," held by a gib and key. The method of taking up the wear of the brasses is indicated in the cut.

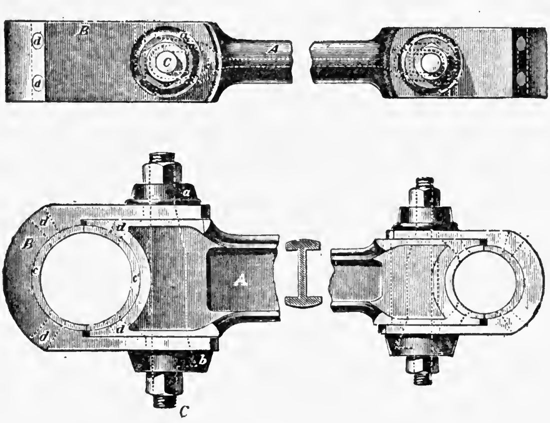


Fig. 27.
Connecting Rod, Woodbury Engine.

In Fig. 26 is shown a connecting rod whose crank end is of the "marine" type, sometimes known as "club ended;" the cross-head end has the strap attached to the rod by a bolt, and a gib and key.

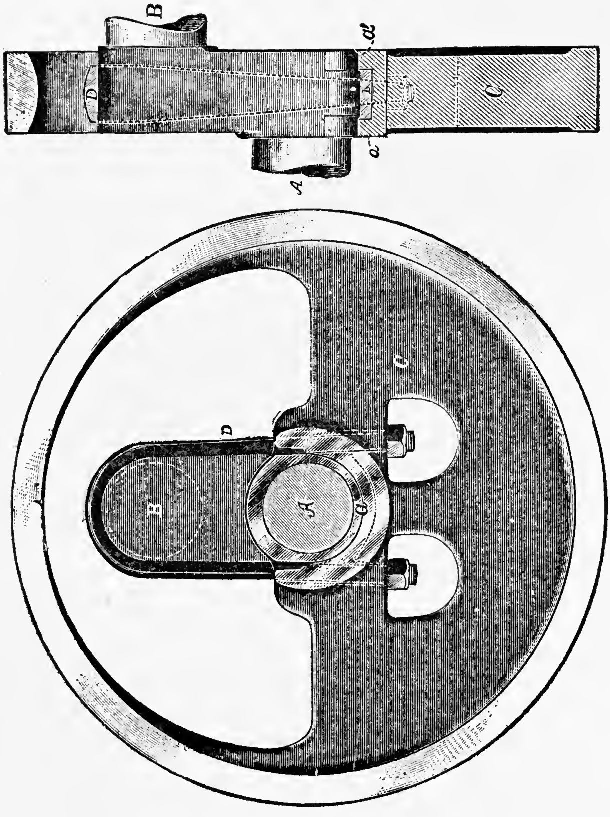


Fig. 28.

Fig. 27 shows a connecting rod whose cross-section is I-shaped. The method of attaching the brasses and taking up the wear is clearly shown.

33. Crank.— Those engines that have the crank between the two main bearings of the shaft are center-crank engines; and those that have the crank on the same side of both the shaft bearings are side-crank engines.

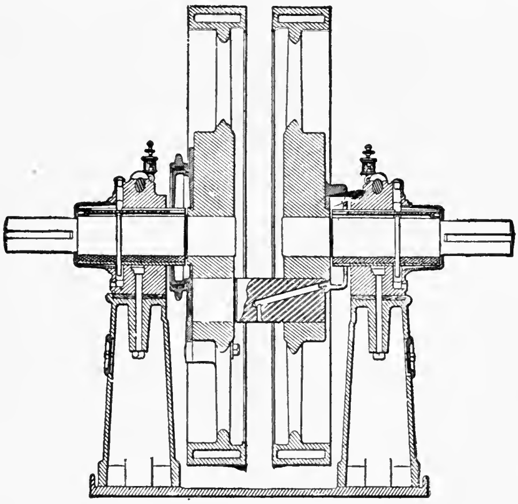


Fig. 29.

Most center-crank engines are of the automatic high speed type, and have the crank and the shaft forged out of one solid piece of iron or steel. Where this is the case it is, usually, customary to fasten to each arm of the crank a cast iron disk provided with a balance weight to balance the weight of the crank and a part of the weight of the connecting rod. The method of fastening these balancing disks to the crank differs for different makes of engines. Some center-crank engines have a built up crank; the shaft is made of two pieces, to the end of each of which is fastened a disk or wheel, and these disks are then fastened together by the crank pin. The disks are, usually, forced on the pieces of the shaft by hydraulic pressure, and then keyed. The pin is, also, forced into the disks by hydraulic pressure.

The diameter of the crank-pin of center-crank engines is almost invariably the same as, or slightly greater

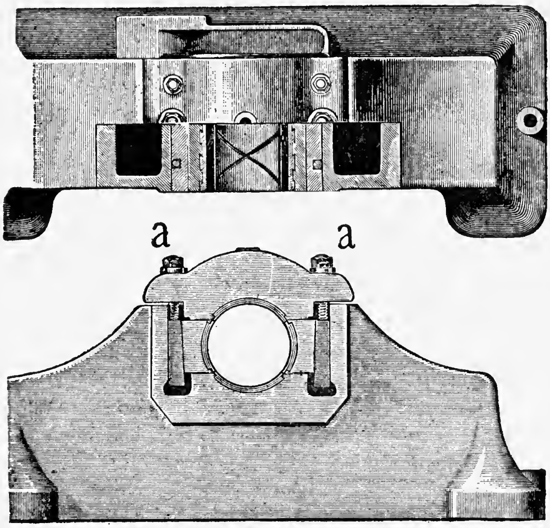


Fig. 30.

than, the diameter of the shaft; and its length is usually equal to that of the cross-head pin.

In Figs. 28 and 29 are shown the forms of center-cranks that have been described. Fig. 28 shows the crank of the Woodbury engine, and Fig. 29 shows the built up center-crank of the Straight Line engine.

Side crank engines may have simply a crank, forced by hydraulic pressure onto the end of the shaft, into which the crank-pin is forced; or they may have a disk, termed the crank-disk, forced onto the end of the shaft, which carries the crank-pin.

Most engines of the Corliss type have simply a crank, while the side-crank engines of the automatic high speed type, usually, have a crank-disk.

The crank-pin of side-crank engines is, usually, about the same size as the cross-head pin. The method of fastening the crank-pin into the crank, or the crank-disk, varies with different makes of engines. Some are simply forced into the hole, provided for them, by hydraulic pressure; others are forced in, and then have a nut put on the back; while others are fastened in by other methods.

It is of the utmost importance that ample provisions be made for the proper lubrication of the crank-pin of all engines, whether side-crank or center-crank.

34. Main Bearings.— It is important that the main bearings of an engine be large, lined with a good wearing metal, and have proper facilities for lubricating. Small bearings, or those not having proper provisions for distributing the oil over the bearings, are apt to give trouble by running too hot and being constantly in danger of cutting.

The caps to the main-bearings are those pieces that go down over the shaft after it is in the bearings. They are sometimes put on in a horizontal position and other times are inclined at an angle of about 30°.

In Fig. 30 is shown a section of the main bearing of the Porter-Allen Automatic Engine. It is made in four parts, viz., the bed, or bottom part; the side boxes, or side parts; and the cap, or top part of the bearing. By screwing up the nuts marked *a*, the wedges may be raised, and the side boxes pressed out and tightened against the shaft.

In Fig. 31 is shown a section of the main bearing of the Porter-Hamilton Engine.

35. ECCENTRIC.— The eccentric is simply a cast iron disk through which the shaft passes and which moves the valve of the engine. In the case of engines of the plain slide valve type, and, also, of the Corliss type, the eccentric is fastened to the shaft either by a key or by a set screw. The advantage of the set screw over the key

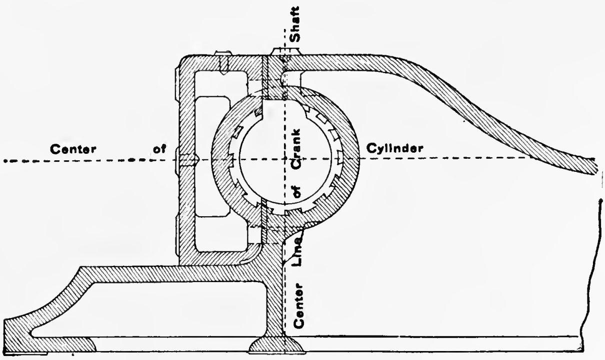


Fig. 31

is that it allows the relative position of the eccentric on the shaft to be changed at will; and the disadvantage is that at times the eccentric may slip and change its relative position without that fact being known.

The distance that the center of the eccentric is from the center of the shaft is its eccentricity. The eccentric is equivalent to a crank whose length is equal to the eccentricity, and takes the place of such a crank. As the shaft is turned, the eccentric turns and moves the valve of the engine back and forth as if it were connected to a crank whose length is equal to the eccentricty of the eccentric.

On most engines of the automatic high speed type, the eccentric is not fastened to the shaft, but the opening through which the shaft passes is larger than the shaft; so that the relative position of the eccentric and, also, the eccentricity can be automatically changed by the gov-

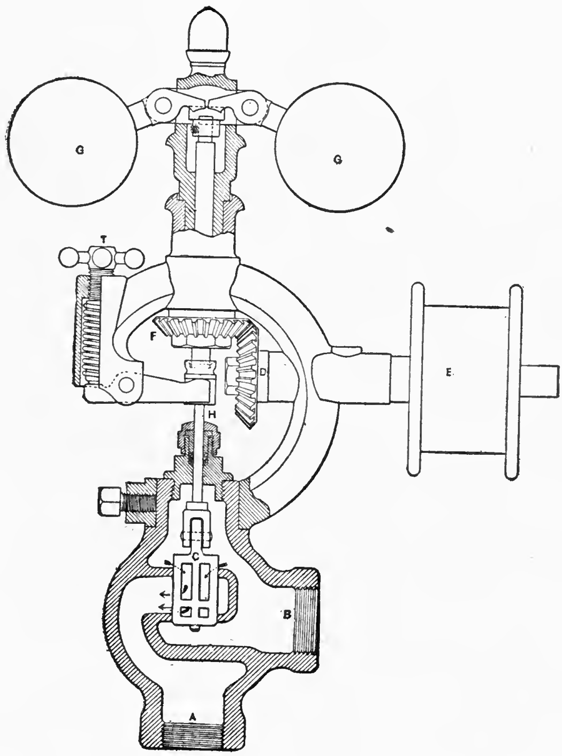


Fig. 32.

erning device. Changing the eccentricity, of course, changes the travel of the valve, and, as we shall see later, this affects the point of cut-off.

36. Governors.—It is impossible to discuss here the governing device of either the automatic high speed type

of engines or the Corliss type, as that involves the discussion of the valve mechanisms that will be given later.

The governor used on engines of the plain slide valve type is a "throttling governor," which is attached to the steam pipe and which decreases the pressure of the entering steam by "throttling," or partially closing an admission valve. In Fig. 32 is shown a section of one of these governors. The opening *A* is connected to the steam chest, and the opening *B* to the steam pipe, so that the steam passes through the valve *C* before entering the engine. The gear wheel *D* is run by means of a belt, from the shaft of the engine to the wheel *E*, and it, in turn, runs the gear wheel *F*, which moves the balls *G*. As the speed of the balls *G* increases, the centrifugal force makes them rise, and in doing so they force down the valve-stem *H*, and partly close the valve *C*. The faster the engine runs, the faster the balls *G* move, and the more the valve *C* is closed; the slower the engine goes, the slower the balls move, and the more the valve *C* is opened. The valve *C*, thus automatically opens wider to admit steam if the engine begins to slow down, and partly closes, thus shutting off the steam, if the engine begins to speed up. By properly adjusting the governor, by means of the screw *T*, the speed of the engine may be fairly controlled within certain limits.

CHAPTER IV.

ADMISSION OF STEAM BY VALVES.

37. Opening and Closing the Ports by the Valve.— As has been explained, the valve is worked by an eccentric which is fastened to the shaft; and the eccentric is equivalent to a crank whose length is equal to the eccentricity of the eccentric. The motions of the eccentric and the valve, therefore, bear the same relations to one another that the motions of the crank and piston do. During one complete revolution of the shaft the valve makes one complete forward and one complete backward motion, and the length of each of these motions is equal to twice the eccentricity of the eccentric. If we neglect the obliquity of the eccentric rod, which changes the motion of the valve in the same way that the obliquity of the connecting rod changes the motion of the piston, the valve will make one-half of its forward, or backward, motion while the eccentric makes a quarter of a revolution, and the relation of the motion of the valve to that of the eccentric will be very much simplified. In all that follows, except when otherwise stated, the obliquity of the connecting rod, and of the eccentric rod, will be neglected; and the motions of the valve and the piston will be discussed as if the rods were of infinite length.

The valve is said to be in "mid-position" when it has reached the middle of its forward or backward motion.

The "travel" of the valve is the total distance that it moves in one direction, either forward or backward, and is equal to twice the eccentricity of the eccentric.

The simplest valve is the plain *D*-valve, shown in Fig. 33. There, *a* represents the steam chest, into which the steam passes from the boiler; *c* represents the exhaust port, through which the exhaust steam passes out of the engine; d_1 and d_2 represent the steam ports, through which the steam passes into the cylinder from the steam chest. In the figure, the valve is supposed to be in mid-position and is shown as lapping over and beyond the ports, at each end, a distance marked *o*; it is also shown as lapping over the ports, towards the inside, the distance marked *i*.

The distance *o* is the *outside* or *steam lap*, which is, the *distance the valve extends over the edges of the steam ports on the outside.*

The distance *i* is the *inside* or *exhaust lap*, which is, *the distance the valve extends over the edges of the steam ports on the inside.*

In Fig. 33, let *C* represent the center of the shaft, and *A*, the center of the eccentric; so that *AC* is the eccentricity of the eccentric, and represents the equivalent crank. When the valve is in mid-position, as shown in the figure, the center of the eccentric is at *B*, and the valve has made half of its travel. While the valve is in this position, it is evident that no steam can enter or leave through either of the ports, d_1 or d_2. If the shaft is revolved right-handed, the valve will move from its mid-position towards the right; and when the center of the eccentric has reached any point such as *D*, the distance that the valve will have moved from mid-position will be equal to the distance, *CE*, from the center of the shaft to the line *DE*, which is drawn perpendicular to *AC*. If, when the center of the eccentric gets to *D*, the distance *CE* is equal to *o*, the outside lap, the steam will be just on the point of entering the port d_1. Now, as the shaft continues to revolve, the valve will continue to move towards the right, and become farther and farther from mid-position. As the port was just about to open when the center of the eccentric was at D, it is evident that when the center of

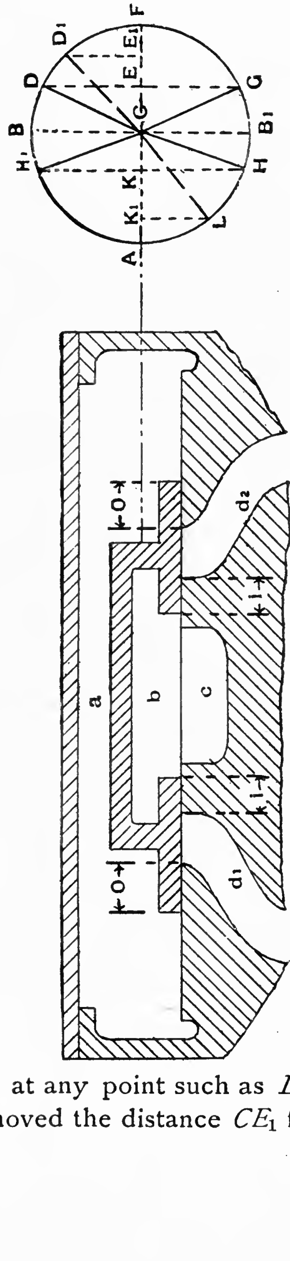


Fig. 33.

the eccentric is at any point such as D_1, beyond D, and the valve has moved the distance CE_1 from mid-position, the port has been opened the amount EE_1, equal to $CE_1 - CE$. When the center of the eccentric gets to the point F, the valve will have reached the end of its travel and will be the distance CF from mid-position; the port will be open the amount EF, equal to $CF - CE$. As the shaft continues to revolve, the center of the eccentric will move from F towards G, and the valve will move towards the left and gradually close the port. When the center of the eccentric has gotten to the point G, such that the distance, CE, of the valve from mid-position is equal to o, the outside or steam lap, the port will be just closed. When the center of the eccentric has reached the point B_1, the valve will again be in mid-position; and when it has gotten to A, the valve will have reached the end of its travel towards the left.

It is evident, from the figure, that no motion of the valve towards the left of mid-position can open the port d_1 to the space a of the steam chest; and, as the valve is always to the left of mid-position while the center of the eccentric is anywhere on the arc $B_1 AB$, it follows that the port d_1 cannot be open to the steam chest while the center of the eccentric moves from B_1 to B. As it has been shown that the port d_1 is open only while the center of the eccentric moves over the part DG of the arc BFB_1, it follows that, during one revolution of the shaft, the port d_1 is open only during the time that the center of the eccentric moves from D to G.

It is evident, from the figure, that the larger is the outside lap, equal to the distance CE, the smaller is the arc DG; and the smaller is the lap, the larger is the arc DG. If the outside lap were zero, the port would remain open during half a revolution of the eccentric, or while the center of the eccentric traveled over the arc BFB_1.

Referring again to Fig. 33, let us discuss the opening and the closing of the port d_1 to the exhaust port c, during one revolution of the shaft.

From the figure, it can be seen that no movement of the valve towards the right of the mid-position will open d_1 to c, and, therefore, at no time during the motion of the center of the eccentric from B to B_1 is the port d_1 in communication with c. When the center of the eccentric gets to B_1 the valve is again in midposition, and as it continues to revolve, the valve moves towards the left. When the center of the eccentric gets to the point H, such that the distance CK is equal to, i, the inside or exhaust lap, the port d_1 is just on the point of opening to the exhaust port, c. As the motion of the eccentric continues, the port becomes more and more open to c; and when the center of the eccentric has reached any point such as L, the port d_1 is open to c the amount KK_1, equal to $CK_1 — CK$. When the center of the eccentric gets to A, the valve is at the end of its travel towards the left; and as the center continues to revolve, the valve moves toward the right, and the opening of the port d_1 to the exhaust port, c, becomes smaller and smaller until, when the center is at H_1, where the distance CK is equal to the exhaust lap, the communication of d_1 with c is closed.

From what has been said, it is now easy to follow the various openings and closings of d_1 during one revolution of the shaft. Let us start with the valve in mid-position, and the center of the eccentric at B, and consider the shaft as revolving right-handed. When the center of the eccentric gets to D, the port d_1 is opened to the steam space a, so that steam may enter the cylinder, and it continues open until the point G is reached. During the movement of the center of the eccentric from G to H the port d_1 is closed, and no steam can enter from a nor leave by the exhaust port, c; but at H the port d_1 is opened to c, so that the steam may leave the cylinder, and it continues open until H_1 is reached.

It is usual to say *the steam port is open,* when d_1 is open to *a;* and to say *the exhaust port is open,* when d_1 is opened to *c.*

By an analysis similar to that made for the port d_1, it can be shown that, during one revolution of the shaft, the port d_2 is, also, open once for the admission of steam and once for its exhaust; the difference between the two ports is that d_2 would be open for the admission of steam during the semi-revolution $B_1 AB$, instead of the semi-revolu- BFB_1, and would be open for its exhaust during the semi-revolution BFB_1, instead of the semi-revolution $B_1 AB$.

38. Relative Movements of the Piston and Valve.—In Fig. 34, the piston is shown at the left-hand end of the cylinder, ready to begin its stroke toward the right; and the valve is shown in such a position that the steam port is open by the amount marked l. The shaft is represented by two points, C and C_1. C is the center of the crank shaft; CD is the crank; and the circle $DGHF$ is the circle described by the crank-pin during one revolution. C_1 represents, also, the center of the shaft, and the circle $A_1D_1G_1H_1$ is the circle described by the center of the eccentric during one revolution of the shaft. Of course, C and C_1 ought to coincide, as they both represent the center of the same shaft; they are put one above the other, as shown, simply to avoid confusion in the drawing, and for the sake of clearness.

The steam port is open the amount l, and the steam is entering the cylinder to the left of the piston. The center of the eccentric is at the point D_1, and the crank-pin is on the dead center at D. The shaft is supposed to revolve right-handed, as in Art. 37.

The distance l is the *lead,* which may be defined as *the amount the steam port is open when the piston is at the beginning of its stroke.*

As the crank and the eccentric are both fixed to the shaft, they must preserve the same relative positions; and the center of the eccentric will always be ahead of the crank-pin, by the angle $D_1\ C_1\ A_1$.

The angle $B_1\ C_1\ D_1$ is the *angle of advance,* equal to $D_1\ C_1\ A_1 - 90$, and is defined as *the angle the center line of the eccentric makes with a perpendicular to the line of motion of the piston, when the crank is on the dead center.*

If we make $C_1\ N$ equal to the steam lap, and $C_1\ K$ equal to the exhaust lap, and draw the lines $F_1\ G_1$ and $L_1\ H_1$ perpendicular to $A_1\ C_1$ and intersecting the circle at the points F_1, G_1, L_1, and H_1, we shall have, from the reasoning of Art. 37, the point, F_1, at which the center of the eccentric is when the steam port is opened; the point, G_1, at which the steam port is closed; the point, H_1, at which the exhaust port is opened; and the point, L_1, at which the exhaust port is closed. Since $C_1\ N$ is equal to the steam lap, o, of the valve, and $C_1\ R$ is the distance the valve is from mid-position when the center of the eccentric is at D_1, it follows that the lead, l, is equal to $C_1\ R - C_1\ N$; and $C_1\ R = o + l$. And as $C_1\ R$ is the distance the valve has moved from mid-position while the eccentric turns through the angle $B_1\ C_1\ D_1$, it follows that the angle of advance is *the angle through which the eccentric must be turned in order to move the valve from mid-position a distance equal to the steam lap plus the lead.*

Therefore, for a given steam lap of valve and a given eccentricity, the greater the angle of advance is made, the greater becomes the lead; and the less the angle of advance is made, the less becomes the lead. If the angle of advance is equal to the angle $B_1\ C_1\ F_1$, the lead would be zero and the valve would be said to be "blind;" if the angle of advance should be less than the angle $B_1\ C_1\ F_1$, $C_1\ R$ would be less than $C_1\ N$, and the lead would be negative. Again, for a given lead and eccentricity, the

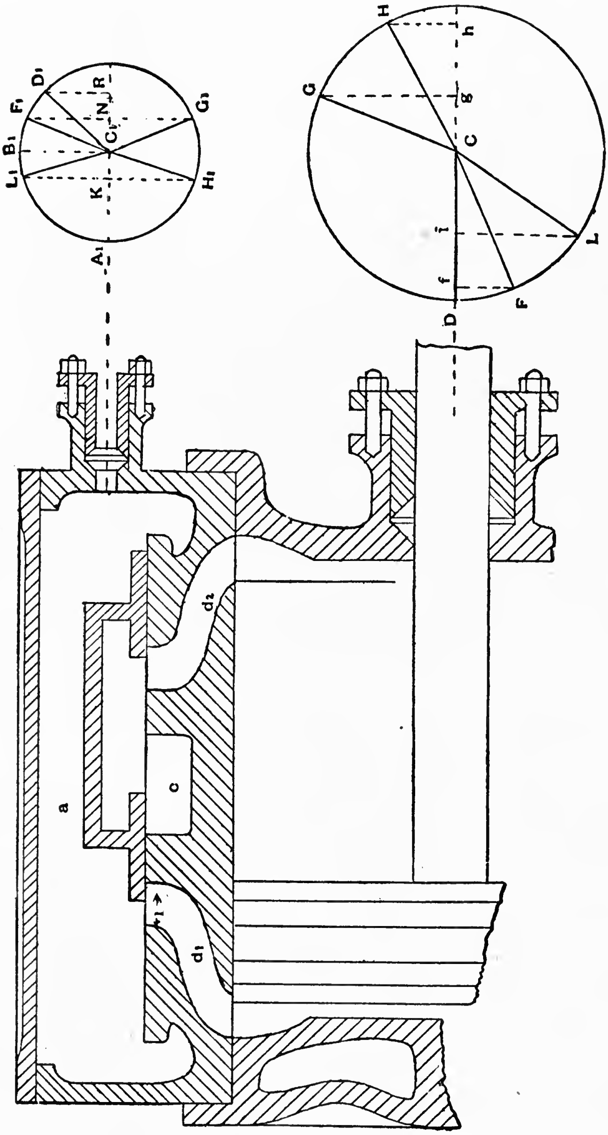


Fig. 34.

angle of advance will be increased, if the steam lap is increased; and be decreased, if the steam lap is decreased.

In finding the position of the crank for different positions of the eccentric, we start with the position, D, of the crank when the eccentric is at D_1, and, as their positions relative to one another are fixed, we know that they must both turn through the same angle in the same time. The position, G, of the crank-pin when the steam port is closed, is obtained by making the angle DCG equal to the angle $D_1 C_1 G_1$. From the figure it is seen that the greater the angle of advance becomes, for a valve with a given steam lap, the less becomes the angle $D_1 C_1 G_1$ and its equal, DCG, and, therefore, the earlier will be the point of cut-off; also, the larger the steam lap, the less is the angle $F_1 C_1 G_1$, and, therefore, for a given lead, the less will be the angle $D_1 C_1 G_1$ and its equal, DCG, and the earlier will be the cut-off. From this follows a general proposition, which always holds for the slide valve, as follows, *the greater the angle of advance, for a given eccentricity and steam lap, the greater will be the lead, and the earlier will be the cut-off; the greater the steam lap, for a given eccentricity and lead, the greater will be the angle of advance, and the earlier will be the cut-off.*

To find the position, H, of the crank-pin when the exhaust port opens, or release occurs, make GCH equal to $G_1 C_1 H_1$, or make DCH equal to $D_1 C_1 H_1$. Anything that will make the angle $D_1 C_1 H_1$ or its equal, DCH, smaller will make the release occur earlier. Now, increasing the angle of advance throws D_1 farther towards H_1, and decreases the angle $D_1 C_1 H_1$; increasing the exhaust lap, $C_1 K$, throws H_1 farther towards A_1, and increases the angle $D_1 C_1 H_1$. It follows, therefore, *that, for a given eccentricity and exhaust lap, increasing the angle of advance makes the release occur earlier; and, for a given angle of advance and eccentricity, increasing the exhaust lap makes the release occur later.*

The position, L, of the crank-pin when compression begins, or the exhaust port closes, is best obtained by making the small angle $D\,C\,L$, below DC, equal to the small angle $D_1\,C_1\,L_1$. Anything that causes the angle $D_1\,C_1\,L_1$ to be large, will make the point L be farther from D, and will make the compression begin earlier. The angle $D_1\,C_1\,L_1$ may be increased by making the angle of advance, $B_1\,C_1\,D_1$, greater or by increasing the exhaust lap, thus throwing L_1 farther towards A_1. It follows, then, *that, for a given eccentricity and exhaust lap, an increase in the angle of advance makes the compression begin earlier; and, for a given eccentricity and angle of advance, an increase in the exhaust lap makes the compression begin earlier.*

Finally, the position, F, of the crank-pin when admission begins, or the steam port is opened, is obtained by laying off, below $D\,C$, the angle $D\,C\,F$ equal to the angle $D_1\,C_1\,F_1$. It is evident, from the figure, that, *for a given eccentricity and steam lap, an increase in the angle of advance makes the admission occur earlier; and, for a given eccentricity and angle of advance, an increase in the steam lap makes the admission occur later.*

It will be noticed, from what has been explained, that, for a given eccentricity, steam lap, and exhaust lap, an increase in the angle of advance makes the lead greater, and makes the ports open and close earlier.

When the positions of the crank-pin are known for the different positions of the valve, it is very easy to determine the positions of the piston, provided the obliquity of the connecting rod is neglected, since the distance the piston is from the beginning of its stroke is always equal to the distance from the dead center to the perpendicular let fall, on the line of motion of the piston, from the center of the crank-pin. Thus, when the crank-pin is at G, the piston is at the distance Dg from the beginning of its

Since *G* is the position of the crank-pin when cut-off takes place, *g* is the "point of cut-off," and *Dg* expressed as a fraction of the stroke is the "cut-off."

That point in the stroke at which the piston is when the exhaust port is opened, is called the "point of release;" that point in the stroke at which the piston is when the exhaust port is closed, is the "point of compression;" and that point in the stroke at which the piston is when the steam port is opened, is called the "point of admission." The "point of cut-off" has already been defined.

The positions of the piston at the times of the opening and closing of the ports are indicated on the drawing. *g* is the point of cut-off; *h*, the point of release; *i*, the point of compression; and *f*, the point of admission. We are now able to trace the motion of the piston and note the action of the steam. The piston starts at the left end of its stroke and moves towards the right, with steam entering the cylinder during the whole time; at *g* the steam is cut-off, and while the piston moves to *h*, from *g*, the steam remaining in the cylinder is being expanded. At *h* release occurs; the steam begins to leave the cylinder and continues to leave until the piston gets to *i*, on its return stroke. At *i* the exhaust port is closed, and the steam remaining in the cylinder is compressed while the piston moves from *i* to *f*, on the return stroke. At *f* the steam port is opened, and steam begins to enter the cylinder.

The same kind of analysis can be followed out for the steam entering the cylinder to the right of the piston. The point of cut-off, of release, of compression, and of admission, will be the same distance from the end of the stroke; everything will be the same except that the points of *cut-off and release* will occur while the piston is moving from *right to left*, and the points of *compression and admission* will occur while the piston is moving from *left to right.*

39. Balanced Slide Valve.— In order to avoid the friction of the plain *D*-valve, such as shown in Figs. 33 and 34, where the full pressure of the steam presses the valve against its seat, and makes it difficult to move, some form of balanced valve is generally used on automatic high speed engines. A balanced valve not only requires less power to move it, but also wears less than an unbalanced valve.

The general form of balanced slide valve in common use is the "Straight Line" valve, or some modification of it. This valve is shown in Figs. 35 and 36. It will be seen that the valve is simply a flat casting, *a*, with open-

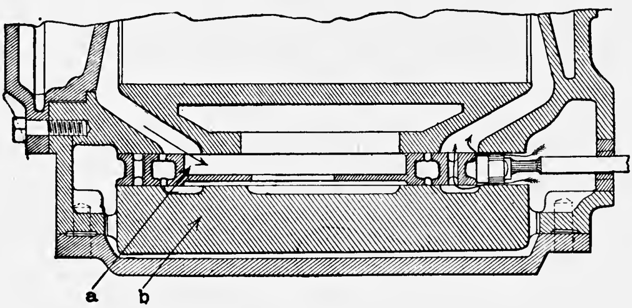


Fig. 35.

ings through it, that moves back and forth between the valve seat and a "cover-plate," *b*. The cover-plate does not rest directly on the valve but on "distance-pieces," *c*, at the top and bottom of the valve. The cover-plate is sometimes kept in place by means of springs, interposed between it and the steam chest cover; other times the valve is not set exactely vertical, but is slightly inclined, so that the weight of the valve and cover-plate is sufficient to keep it in place. The valve shown in Figs. 35 and 36 has no special means of correcting for the wear of the valve; the only way to do this is to reduce the thickness of the distance pieces, *c*. Some valves are provided with wedges, by means of which the distance pieces may be set out and the cover-plate lifted any desired distance from the valve. Fig. 35 shows steam being admitted to the right-hand end of the cylinder and being exhausted from the left-hand end, as indicated by the arrows.

40. Piston Valve.— The piston valve consists simply of a piston working in a cylinder through which the ports

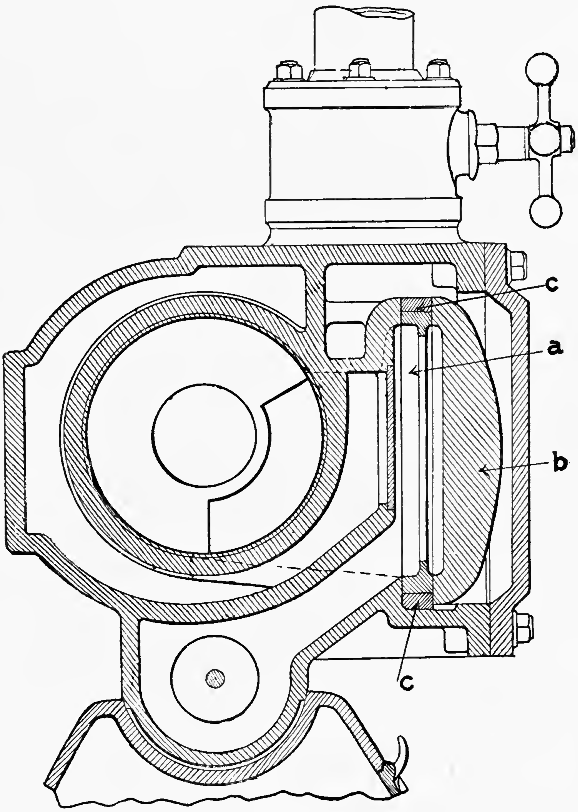


Fig. 36.

are cut. The length of the port is equal to the circumference of the piston, less the width of such ribs as may extend across the port. This form of valve is perfectly balanced, but it is difficult to keep it tight and prevent it from leaking. When steam is first turned on to an engine having a piston valve, care must be taken not to start it up until the steam chest has got thoroughly hot, as the valve is very likely to become hot before the cylinder in which it works, and to expand and stick, and perhaps cause a breakage somewhere.

The piston used for the valve is sometimes made tight by the use of cast iron packing rings, other times it is simply turned to a steam tight fit with the cylinder in which it works.

In Fig. 15 is shown a form of piston valve where the cylinder in which the valve fits is made with thin walls and is surrounded by live steam, so that it heats quicker than the valve and there is not so much danger of the valve sticking. The figure shows steam being admitted to the right-hand end and exhausted from the left-hand end.

41. Multiple Admission Valve.— With the advent of the automatic high speed engine, there came a demand for a single valve which could be made to cut-off early in the stroke and that would give a large port-opening with a small travel. To meet these requirements, the multiple admission valve was devised. Valves of this type are so made that the opening of the port is not, as shown in Art. 37, equal to the distance the valve is from mid-position minus the steam lap, but is equal to two times this distance for a double admission valve, and four times it for a quadruple admission valve.

The "Straight Line" valve, shown in Figs. 35 and 36, is a double admission valve.

42. Meyer Valve.—The Meyer valve consists of two valves, one riding on top of the other; and the advantages it has over the single valves are that the clearance may be made smaller, and the point of cut-off may be changed as desired without in any way changing the points of release or compression, something which cannot be done with a single valve. The valve shown in Fig. 17 is a modified form of a Meyer valve. It consists of the main

valve *b*, moved backwards and forwards by a hollow valve stem, and the small auxiliary valve *c*, which rides on *b*, and is moved by a small valve stem working inside of the hollow valve stem which moves *b*. In order that steam may enter the cylinder, the port in *b* must not be covered by *c*, and it must be over the port in the cylinder. In the figure steam is being admitted into the left-hand end of the cylinder and exhausted from the right-hand end, as indicated by the arrows. *D* is the steam supply pipe, and *K*, the exhaust pipe. The cut-off is regulated entirely by the small, riding valve *c*, while the admission, release, and compression are regulated entirely by the main valve *b*.

43. Corliss Valve.—The Corliss valve is a cylindrical valve, but instead of having a reciprocating motion in the direction of its axis, it has an oscillating motion about its axis. In Figs. 19 and 37 are shown Corliss valves. It will be seen, in Fig. 19, that there are four valves in all, two steam valves at the top of the cylinder, and two exhaust valves at the bottom. The valves are not fastened to the stem by which they are moved, but the stem is flattened and simply lies in the valve. The valves are always made so that the steam pressure comes on top of them and the pressure of the steam presses them down on their seats. The moving mechanism of the valves is quite complicated, and has many small parts that must be kept in order.

The clearance is reduced by making the ports very short and placing the valve close to the cylinder.

The Corliss valve permits a regulation of the point of cut-off without any change in the release or compression.

As there are separate steam and exhaust ports, the exhaust steam does not pass out through the same port through which the hot, live steam enters. Whether or not this is any advantage, and conduces to the economy of the engine, is a somewhat undecided question. In Fig.

37 is shown the mechanism by means of which the valves of the Corliss engine are worked.

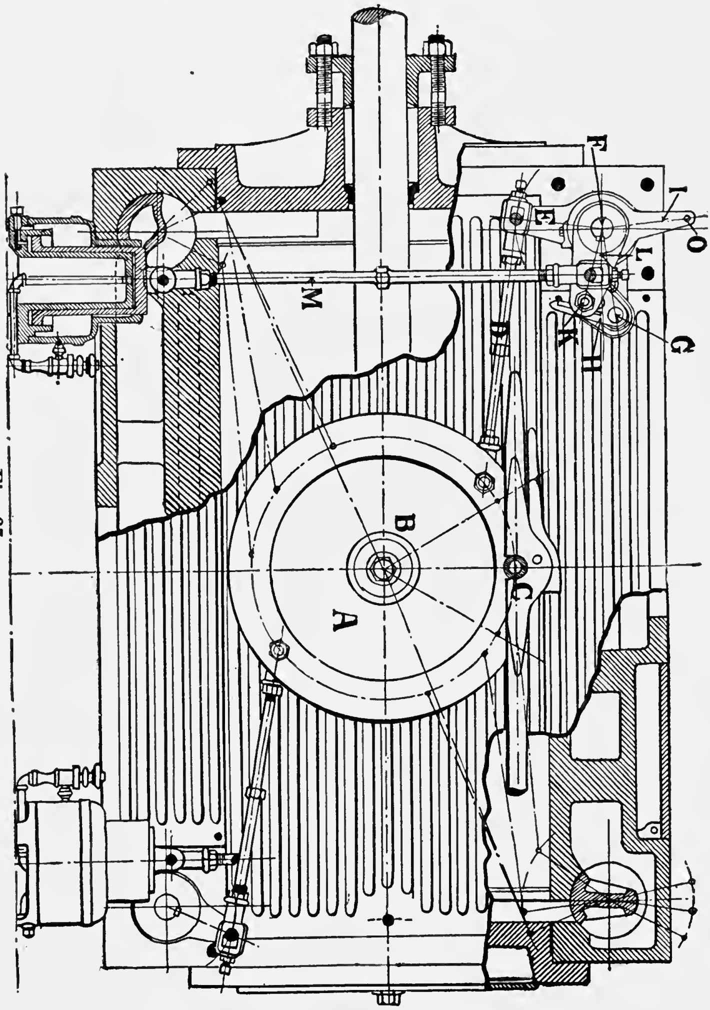


Fig. 37.

The "wrist-plate," *A*, is made to oscillate about the pin *B*, by means of the "reach-rod," *C*, which engages with the "wrist-pin." The wrist-plate is connected by the rod *D* to the bell crank, *E*, that oscillates about the valve stem *F*. At the farther end of *E* is the pin *G* which carries the *V*-shaped lever H. The inner end of *H* is kept pressed against the cam *I* by means of a spring; and the outer end has a hook which engages with a steel block fastened, by means of the bolt *K*, to an arm, *L*, rigidly attached to the valve stem *F*. The dash pot rod, *M*, is, also, attached to the arm *L*. The cam *I* has a projection on it, and is moved backward or forward by means of the governor rod, not shown in the figure, that is attached to the pin *O*.

When the wrist-plate is turned right-handed the crank *E* is turned left-handed, and the hook on *H* engages with the block on *K*, and thus lifts the lever *L* and opens the valve. When *L* is lifted, the dash-pot rod, *M*, is lifted. After the lever *L* has been lifted to a certain distance, the inner end of *H* strikes the projection on the cam *I*, which turns *H* about the pin *G*, so that the hook is released from the block *K*. As soon as this takes place, the arm *L* is made to fall, by the weight of the dash-pot piston and the pressure of the air on top of the piston, and thus close the valve. The function of the dash-pot is to close the valve; and it is so arranged that by means of a small valve a greater or less vacuum may be maintained under its piston.

The governor changes the position of the cam *I* so that the block *K* is disengaged early or late, as required to govern the engine, from the hook on *H*.

The exhaust valves have no disengaging mechanism, but are simply made to oscillate backward and forward by means of a rod connecting them to the wrist-plate.

Owing to this peculiar method of closing the valve, the speed of rotation of the engine cannot be great, as the dash-pot piston must have time in which to fall. The writer has known of but few cases where the number of revolutions has exceeded one hundred per minute, and in most cases the engines were small.

The advantages claimed for Corliss valves are: —

1. They permit of a regulation of the cut-off without any change in the release or compression.

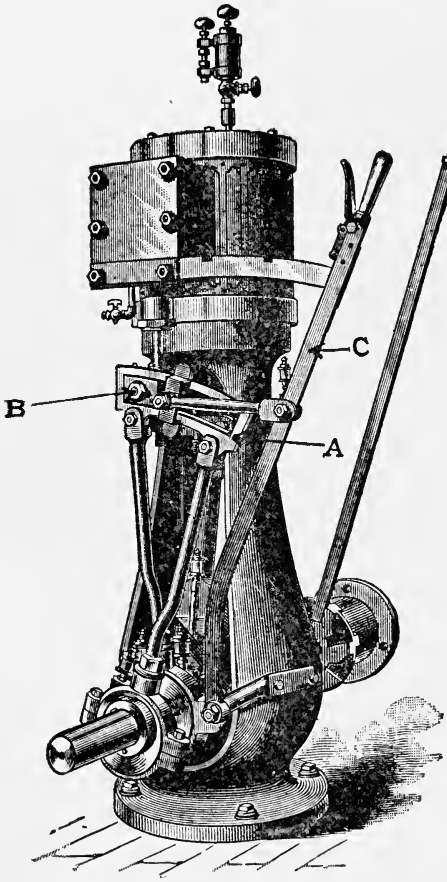


Fig. 38

2. Short ports, and, consequently, small clearance volume.

3. Separate steam and exhaust ports, reducing condensation.

4. Quick, sharp, motion of the valve when cutting off steam.

44. Link Motion. — On marine engines, locomotives, and some few land engines, it is necessary to have some device by which the direction of rotation of the crank-shaft may be changed. The mechanism usually used is the link motion. The engine is provided with two eccentrics, keyed to the crank-shaft, each of which is connected by an eccentric rod to the end of a link. A block, connected by a suitable mechanism to the valve, slides along a groove cut in the link. When the block is at one end of the link it has all the motion of the eccentric connected to that end, and very little of the motion of the other eccentric; when the block is in the middle of the link it has a little motion, backward and forward, that is the result of the motions of both eccentrics. One of the eccentrics is the "forward" eccentric, and the other is the "backward" eccentric. When the motion of the valve, on account of the position of the block, is due more to the "forward" than to the "backward" eccentric, the engine runs forward; and when influenced more by the "backward" than the "forward" eccentric, the engine will run backward. The position of the block in the link may be changed by moving the block and keeping the link in the same position; or by keeping the block at rest and moving the link, as is done in the Stephenson link motion.

In Fig. 38 is shown a small vertical engine with a link motion. *A* is the "*link*;" *B* is the "*block*," that, in this case, is fastened directly to the valve stem; *C* is the "*reversing lever*," by means of which the link is moved so that the position of the block in it may be changed.

CHAPTER V.

VALVE DIAGRAMS.

45. Zeuner Valve Diagram.— A valve diagram is a diagram that will show, at once, the steam lap, exhaust lap, lead, distance the valve is from mid-position, and, also, the amount the port is open for a given position of the crank. By means of valve diagrams, all the various problems connected with the motion of valves may be solved. There are several systems of valve diagrams, each of which is considered better than the others by those who use it; and as, in the opinion of the author, the Zeuner diagram is better for all uses than any other, it will be used in this work.

In Fig. 39, let AA' represent the stroke of an engine drawn to any desired scale, and the circle ABA', the path of the center of the crank pin, or the crank-circle. Also, let nm be the travel of the valve drawn to any desired scale, which may or may not be the same as the scale of AA'. The distance On will be equal to the eccentricity of the eccentric, and the circle $nA_1 m$ will be the path of the center of the eccentric. When the crank-pin is at A, the center of the eccentric will be at A_1, and YOA_1 will be the angle of advance. Now, if the crank moves from AO to any position as BO, the center of the eccentric will move from A_1 to B_1, and the angle AOB will be equal to the angle A_1OB_1. Draw B_1b perpendicular to the line AA'; then, neglecting the obliquity of the eccentric rod, when the crank is in the position BO, the valve will be moved from mid-position a distance equal to Ob. To find the distance Ob by the method just described, it was necessary to draw three lines, OB, OB_1, and B_1b, and to make the angle A_1OB_1 equal to the angle AOB.

Draw OB' so that the angle $B'OA'$ will be equal to the angle BOA, and $B'\,O$ will be the position of an imaginary crank which starts from $A'O$ when the real crank starts from AO and moves with the same velocity as, but in

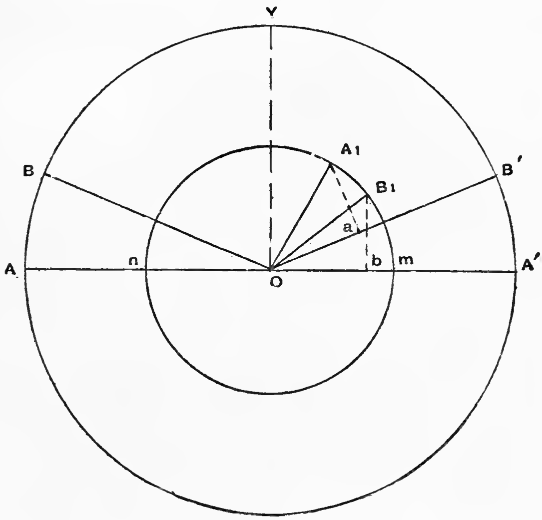


Fig. 39.

the opposite direction to, the real crank. Draw A_1a perpendicular to OB'. In the two right triangles, OB_1b and OA_1a, we have OA_1 equal OB_1, and the angle B_1Ob equal the angle A_1Oa. Therefore, the two triangles are equal, and Oa is equal to Ob: or Oa is equal to the distance the valve is from mid-position when the crank has moved through the angle AOB, equal to $A'OB'$. From this, it is seen that if, instead of considering the real crank, we consider the motion of an imaginary crank revolving in the opposite direction to, but with the same velocity as, the real crank, the distance that the valve is from mid-position, for a given angular motion of the crank, may be obtained by drawing only two lines, OB' and A_1a.

Since the position of the point A_1 is fixed, for a given eccentricity and angle of advance, and the angle A_1aO is a right angle, *the point* a *will always fall upon the circumference of the circle drawn upon* OA_1 *as a diameter.* This gives us, then, an extremely simple means of obtaining the distance the valve is from mid-position after the crank has moved through any given angle.

In Fig. 40, let AA', as before, represent the stroke of the engine, and $ADBA'$ the path of the crank-pin; also, let the angle YOA_1 be the angle of advance; and let OA_1 be, to any desired scale, the eccentricity of the eccentric, equal to half the travel of the valve. Upon OA_1, as a diameter, draw the "valve circle" OaA_1. From the preceding paragraph, it follows that if OB represents any position of the imaginary crank, at any instant, Ob will represent the distance the valve is form mid-position at that instant. That is, by drawing one line we determine, at once, the angle the crank has turned through and the distance the valve is from mid-position. Care must be taken to remember that OB does not represent the *real crank* of the engine, but an *imaginary crank* that revolves with *same velocity* as the real crank, but in the *opposite* direction.

Neglecting the obliquity of the connecting rod, the distance the piston would have moved from the end of its stroke, while the crank moved through an angle equal to $A'OB$, would be B_1A'. And if, as we supposed, the real crank moved in the direction from A to Y, the piston would have moved the distance B_1A' from the left-hand end of the stroke and not from the right-hand end.

If we describe the arc dc, with O as a center and a radius equal to the outside or steam lap of the valve, the distance the steam port is open, when the crank has moved through the angle equal to BOA', will be equal to $Ob - Ob'$; that is, it will be equal to the distance the valve is from mid-position minus the steam lap.

If the lines OC and OD be drawn through c and d, respectively, OC will represent the position of the imaginary crank when steam begins to enter the cyclinder, and OD its position when steam is cut-off; because, for those positions, the distance the valve is from mid-position is equal to the steam lap.

When the crank is on the dead center, the imaginary crank is at OA', and the distance the valve is from mid-position is Oa, so that the lead is $a'a$.

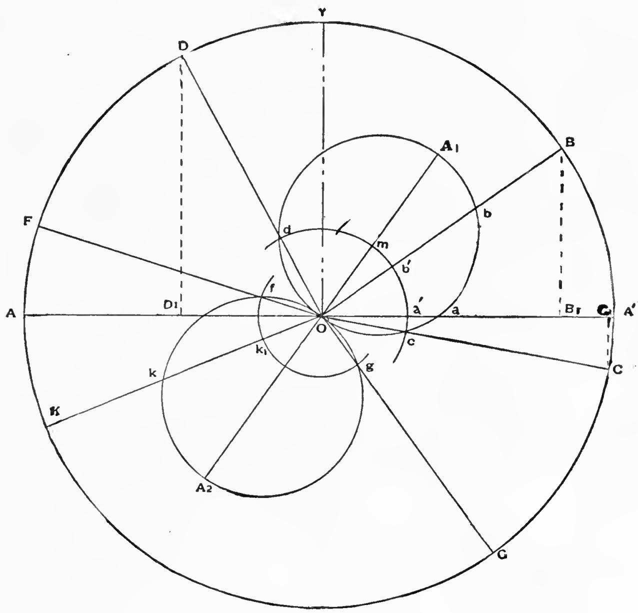


Fig. 40.

The valve is open its maximum distance when the imaginary crank is in the position of the line OA_1, and then the valve is at the right-hand end of its travel.

If the line DD_1 is drawn perpendicular to AA', the distance $A'D_1$ will be the distance the piston is from the beginning of the stroke at the point of cut-off.

If the line OA_1 be continued to A_2, so that OA_2 is equal to OA_1, and another valve circle be drawn on OA_2 as a diameter, the exhaust port may be discussed. Draw the arc fg, with O as a center and a radius equal to the exhaust lap. Then, for any position of the imaginary crank, such as OK, the exhaust port is open the distance kk_1, equal to the distance, Ok, the valve is to the left of mid-position minus the exhaust lap, Ok_1.

If OF and OG are drawn through the points f and g, respectively, OF will be the position of the imaginary crank when release occurs, and OG its position when compression begins; since, for those positions, the distance the valve is from mid-position is equal to the exhaust lap.

46. Valve Diagram Problems.— By assuming a number of the variables in the valve diagram, in Fig. 40, as known, various problems can be made up, all of which can be solved by the proper use of the valve diagram. The solution of every problem will necessitate a good, clear, understanding and knowledge of the relation of the various parts of the diagram to one another. For the sake of making the student familiar with the use of the valve diagram and to give him practice in the use of it, a number of the most important problems likely to be met with in practice will be solved in detail.

Problem 1. Given the *point of admission, the point of cut-off,* and the *travel of the valve;* find the *angle of advance,* the *steam lap,* and the *lead.*

Referring to Fig. 40, we see that since Od is equal to Oc, being the radii of the steam lap circle, the arc Od is equal to the arc Oc, and, therefore, the arc dA_1 is equal to the arc cA_1. That is, the line OA_1 bisects the angle between the positions of the imaginary crank at admission and at cut-off. The construction, therefore, is as follows: —

In Fig. 41, let AA_1 be the stroke of the engine drawn to any desired scale; A_1B_1, the distance from the end of the stroke to the point of admission; and $A_1 C_1$, the distance from the beginning of the stroke to the point of cut-off. On $A_1 A$ as a diameter, construct the crank circle ACA_1B. Draw the lines $B_1 B$ and $C_1 C$ perpendicular to AA_1 and intersecting the crank circle at B and C, respectively. Now draw OB and OC, and they will represent the positions of the imaginary crank at admission and cut-off, respectively. Draw Om, bisecting the angle COB, and lay off Om, according to any desired scale, equal to the given eccentricity of the eccentric, or half

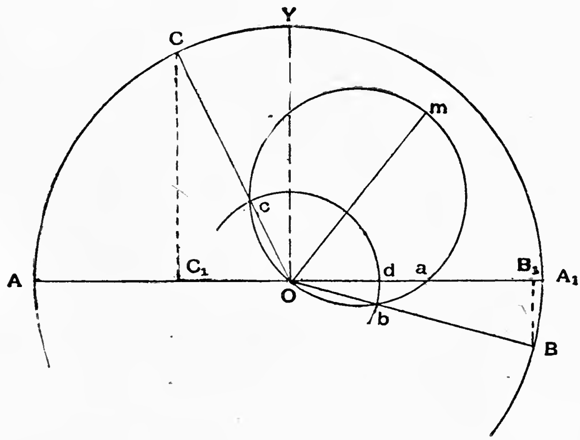


Fig. 41.

the travel of the valve. m will be the position of the center of the eccentric when the real crank is in the position OA; and the angle YOm will be the required angle of advance.

On Om as a diameter, draw the valve circle intersecting OC at c, OA at a, and OB at b. With O as a center and a radius equal to Oc, equal Ob, draw the arc cdb intersecting OA at d. cdb is an arc of the steam lap circle; Ob is the steam lap; and da is the lead.

Problem 2. Given the *point of admission*, the *point of* *cut-off*, and the *steam lap*; find the *angle of advance*, the *eccentricity*, and the *lead*.

An inspection of Fig. 40 shows that the valve circle passes through the points of intersection of the steam lap circle with the lines showing the position of the imaginary crank at admission and at cut-off. Hence the following construction: —

In Fig. 42, let AA_1 be the stroke of the engine, drawn to any scale, and the circle ACA_1B, the crank-circle; A_1B_1, the distance of the point of admission from the end of the stroke; and A_1C_1, the distance of the point of cut-off from the beginning of the stroke. Draw BB_1 and

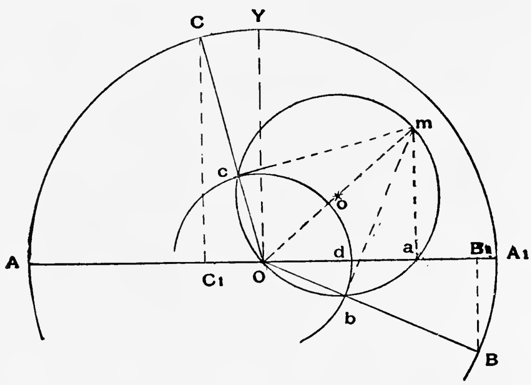


Fig. 42.

CC_1 perpendicular to AA_1 and intersecting the crank-circle at the points B and C, respectively. Draw OB and OC to represent the position of the imaginary crank at admission and cut-off, respectively. With O as a center and a radius, Ob, equal, on any desired scale, to the steam lap, draw the steam lap circle cutting OB and OC at b and c, respectively. Now pass a circle through the points c, O, and b, and it will be the required valve circle. The center of the valve circle may be found by the ordinary method of finding the center of a circle that shall pass through three given points; or, if desired, the diameter Om of the required circle may be obtained by drawing, at c, the line cm perpendicular to OC and continuing it until it meets, at m, the line bm drawn perpendicular to Ob at b, and then connecting the points O and m.

The center of the eccentric is at m when the crank is on the dead center; the angle YOm is the angle of advance; the line Om is the required eccentricity, equal to half the travel of the valve; and da is the required lead.

Problem 3. Given the *point of admission*, the *point of cut-off*, and the *lead*; find the *angle of advance*, the *eccentricity*, and the *steam lap*.

As explained in Problem 1, the diameter of the valve circle always bisects the angle between the positions of the imaginary crank at admission and at cut-off. From Fig. 42, we see that the lead, da, is equal to $Oa - Od$; and, since Od is equal to Ob, we have $da = Oa - Ob$. But from the triangle Oma, we have $Oa = Om \cos. mOa$; and from the triangle Omb, we have $Ob = Om \cos. mOb$. Therefore, $da = Oa - Ob = Om\,[\cos. mOa - \cos. mOb]$.

Since the angles mOa and mOb are constant for a given admission and cut-off, it follows that da will vary directly as Om. Therefore, to solve the problem proceed as follows: —

In Fig. 43, let AA_1 be the stroke of the engine; $A\,C\,A_1\,B$, the crank circle; OB and OC, the positions of the imaginary crank, obtained as in the preceding problems, at admission and cut-off, respectively. Bisect the angle COB by the line Om, and the angle YOm will be the angle of advance.

On the line Om take any point, such as m, and from it draw md perpendicular to $O\,B$, and ma perpendicular to OA. With O as a center and a radius Od, draw an arc cutting OA_1 at e. ea would be the lead for an eccentricity equal to Om. As has been shown, the lead is directly proportional to the eccentricity, and the given lead is to *ea* as the required eccentricity is to *Om*. Therefore, make *Of* equal to *ea*, and *Og* equal to the given lead. Connect the points *f* and *m*, then, through *g*, draw *gn* parallel to *fm* and intersecting *Om* in the point *n*. *On* is the required eccentricity.

Draw, through *n*, the line nd_1 parallel to *md* and intersecting *OB* at d_1, Od_1 will be the required steam lap.

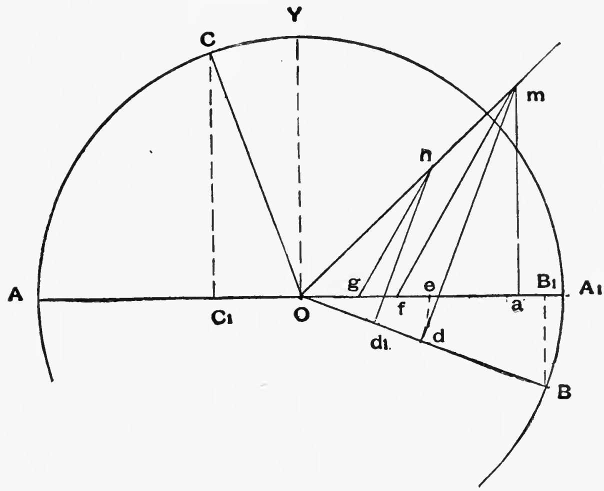


Fig. 43.

Having found the angle of advance and the eccentricity, the valve circle may be drawn if desired.

Problem 4. Given the *point of cut-off*, the *lead*, and the *maximum opening of the port*; find the *angle of advance*, the *eccentricity*, and the *steam lap*.

This problem is met with in steam engine designing more than perhaps any other relating to the valve, and it is probably the most difficult to solve exactly. It is usually solved by trial and approximation. The exact solution is usually difficult for the student, because the reasons for the different steps in the construction of the diagram are seldom fully understood. In order to explain thoroughly the principles involved, let us suppose that in Fig. 44 the problem has been solved, and let us determine the relations that exist between the different parts of the diagram. In the diagram, Om is the eccentricity; Ob, the steam lap; ba, the lead; and dm, the maximum opening of the steam port. $A_1 C_1$ is the distance from the beginning of the stroke to the point of

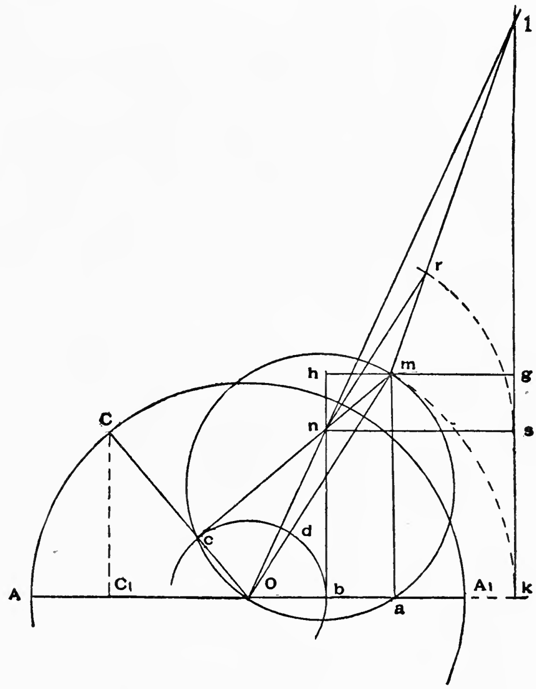


Fig. 44.

cut-off, and OC is the position of the imaginary crank at cut-off. Draw am, and, since the arc Oam is a semicircle, am will be perpendicular to OA. Also draw cm, and, since Ocm is a semi-circle, it will be perpendicular to OC. Draw bh at right angles to OA; it will intersect cm at n. Through m, draw hm parallel to OA and intersecting bh at h. hm will be equal to ba, the lead. Now, with O as a center and a radius equal to Om, draw the arc mk intersecting OA at k; then draw kg at right angles to OA and intersecting hm at g. hg will be equal to bk, equal to dm, the maximum opening of the steam port. Now, since Oc is equal to Ob, and the angles ncO and nbO are right angles, if we draw the line On it will bisect the angle, cnb, between the line nb, at right angles to the imaginary crank on dead center, and the line cm, at right angles to the imaginary crank at cut-off.

Continue On until it cuts gk at l, and then draw lm. Draw ns parallel to hg and intersecting kl at s. With n as a center and a radius equal to ns, draw an arc cutting lm at r. Draw nr, and it will be parallel to Om, as may be proved as follows : —

In the two similar triangles lns and lOk, we have $\frac{ln}{lO}=\frac{ns}{Ok}=\frac{nr}{Om}$. But ln and nr are sides of the triangle lnr; also lO and Om are sides of the triangle lOm. Therefore, since the triangles lnr and Olm have their sides proportional and the angle nlm common to both, they must be similar; and the line nr is parallel to Om.

In the same way it can be shown that if the lines rs and mk be drawn they will be parallel.

Knowing the relations that have been shown to exist between the various parts of the diagram, the solution of the problem becomes as follows :—

Let AA_1, in Fig. 45, represent the stroke of the engine, drawn to any scale, and ACA_1, the crank circle; $A_1 C_1$, the distance the piston is from the beginning of its stroke at the point of cut-off; and OC, the position of the imaginary crank at cut-off.

Make Od, according to any desired scale, equal to the given lead; and Oa, equal to the given maximum opening of the steam port.

Through O and a, respectively, draw Ob and af perpendicular to OA. Through d, draw cd at right angles to

OC. Bisect the angle *ceb* by the line *O'e*, and prolong it until it intersects *af* at *f*.* Connect *f* and *d* by the line *fd;* and draw *eg* parallel to *OA* and intersecting *fa* at *g*. With *e* as a center and a radius *eg*, draw the arc *gh* cutting *df* at *h*. Connect the points *e* and *h*, and then draw *dO'* parallel to *eh* and intersecting *O'f* at *O'*.

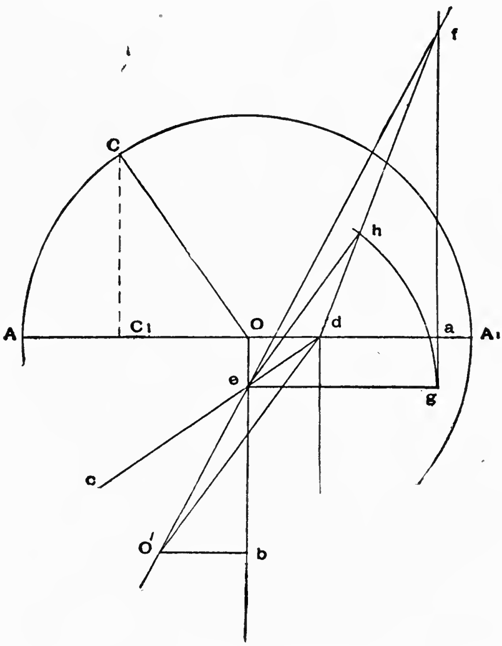


Fig. 45.

The angle *Oeh* is the required angle of advance ; and *O'd* is the required eccentricity. Draw *O'b* parallel to *OA* and intersecting *Oe* at *b*. *O'b* is the steam lap.

A line through *d* parallel to a line through *h* and *g* will pass through the intersection of *fa* and *O'b* prolonged.

* The rest of the solution is simply the solution of the well-known geometrical problem: *Given two lines and a point, to find a point on one of the lines which is equally distant from the other line and the given point.* *fe* and *fa* are the given lines and *d* is the given point *O'*, the required point on *fe*, is equally distant from *d* and the line *fa* prolonged.

Problem 5. Given the *angle of advance*, the *eccentricity*, and the *point of compression*; find the *exhaust lap* and the *point of release.*

In Fig. 46, let AA_1 be the stroke of the engine, drawn to any desired scale, and the circle ABA_1, the crank circle. Draw the line OB, making the angle YOB equal to the given angle of advance. Lay off AD_1 equal to the distance of the point of compression from the beginning of the return stroke, and, then, draw $D_1 D$ at right angles to AA_1. Connect O and D, and OD will be the position of the imaginary crank at the beginning of compression.

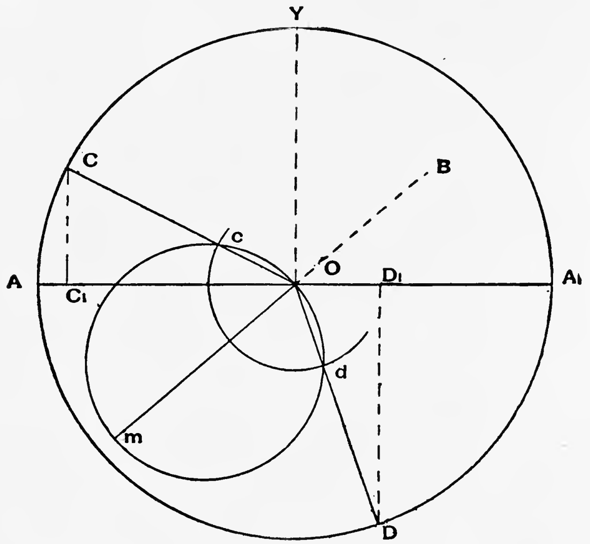


Fig. 46.

Continue OB to m, so that Om is equal to the given eccentricity, and on Om as a diameter draw the valve circle $Ocmd$, intersecting the line OD at d. Od is the required exhaust lap.

With O as a center and a radius equal to Od, draw an arc cutting the valve circle at c. Through c, draw the line OC, and it will represent the position of the imaginary crank when release takes place. Draw CC_1 perpendicular to AA_1, and A_1C_1 will be the distance of the point of release from the beginning of the stroke.

Problem 6. Given the *point of compression*, the *point of release*, and the *eccentricity*; find the *angle of advance* and the *exhaust lap*.

An inspection of Fig. 46 will show that the line Om bisects the angle COD, which is formed by the lines indicating the positions of the imaginary crank at release and at compression. Hence the construction for the solution of the problem is as follows: —

In Fig. 46, let AA_1 be the stroke of the engine, and let the circle ACA_1 be the crank circle. Let OC be the position of the imaginary crank at release, and OD its position at the beginning of compression. Draw OB bisecting the angle COD, and on it make Om equal to the given eccentricity. On Om, as a diameter, draw the valve circle cutting OC at c, and OD at d.

The angle YOB is the required angle of advance, and Oc, equal to Od, is the required exhaust lap.

47. Effect of the Obliquity of the Connecting Rod on the Point of Cut-off. — In Fig. 47, let $A_1 A_2$ represent the stroke of an engine; O, the center of the crank shaft; OA, the crank; and AA_1, the length of the connecting rod. Suppose the crank to revolve as indicated by the arrow.

Make Ab equal to A_1B_1, and draw bB' perpendicular to AA' and cutting the crank circle at B'. If the obliquity of the connecting rod were neglected, the crank would be in the position OB' when the piston has moved through the distance A_1B_1 on its forward stroke. With B_1 as a center and a radius equal to AA_1, describe an arc cutting the crank circle at B. OB will be the actual position of the crank when the piston has gotten to B_1. It makes no difference where B_1 is taken; the construction will always show that, on the forward stroke of the engine, the obliquity of the connecting rod makes the actual position of the crank lag behind the position it would occupy if there were no obliquity: that is, in order that the crank shall turn through a given angle AOB', the piston must move

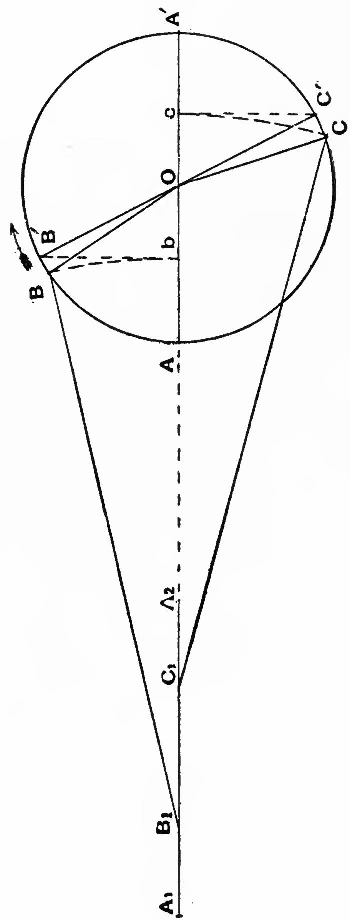


Fig. 47.

through a greater distance than the distance, Ab, that it would move through if there were no obliquity. It follows, therefore, that on the forward stroke the actual distance of the piston from the beginning of the stroke at the point of cut-off is greater than indicated by the valve diagram, by an amount depending upon the obliquity of the connecting rod. In other words, the obliquity of the connecting rod makes the cut-off occur later, on the forward stroke, than indicated by the valve diagram.

Let C_1 be any position of the piston on the return stroke. Make cA' equal to $C_1 A_2$, and draw cC' perpendicular to AA' and intersecting the crank circle at C'. OC' would be the position of the crank, when the piston is at C_1 on the return stroke, if there were no obliquity to the connecting rod. With C_1 as a center and a radius equal to AA_1 draw an arc cutting the crank circle at C. OC will be the actual position of the crank when the piston is at C_1 on the return stroke. The construction shows that, for a given movement of the piston on the return stroke, the obliquity of the connecting rod makes the crank keep in advance of the position it would be in if there were no obliquity; that is, in order that the crank shall turn through a given angle $A'OC'$, the piston must move through a less distance than the distance, $A'c$, that it would move through if there were no obliquity. It follows, then, that, on the return stroke, the actual distance of the piston from the beginning of its stroke at the point of cut-off is less than indicated by the valve diagram, by an amount depending upon the obliquity of the connecting rod. In other words, the obliquity of the connecting rod makes the cut-off occur earlier, on the return stroke, than indicated by the valve diagram.

From what has been said, it is evident that if the steam laps of a valve be made as determined by the valve diagram, and the valve be set with equal lead on the head end and crank end of the cylinder, the point of cut-off will occur earlier on the return stroke than on the forward stroke, unless there is some special means of equalizing the points of cut-off. The usual manner of equalizing the points of cut-off is to set the valve with a somewhat greater lead on the head end of the cylinder than on the crank end. The exact amount that the lead on the head end ought to be made greater than that on the crank end, depends upon the obliquity of the connecting rod, the eccentricity of the eccentric, and the lead on the crank end. It may be determined by making a valve diagram for each end of the cylinder, and using in each diagram the positions of the imaginary crank as determined by taking into account the obliquity of the connecting rod.

48. Swinging Eccentrics. — Automatie high speed engines regulate by changing the angle of advance or the eccentricity of the eccentric, or both, and thus change the point of cut-off. As no changes in the angle of advance or the eccentricity can affect the dimensions of the valve, the steam lap and the exhaust lap must always remain the same for the same valve.

In order to understand how the cut-off and lead will be affected by a change in the eccentricity and angle of advance, for a given eccentric and a given valve, let us refer to the valve diagram in Fig. 48. There, ABA_1 is the crank circle; OB is the position of the imaginary crank at cut-off; OC is the position of the imaginary crank at admission; YOm is the angle of advance; Om is the eccentricity; and Ob, equal to Oc, is the constant steam lap. m is the position of the center of the eccentric when the real crank is on the dead center; and ad is the lead. The line md is perpendicular to Od, since the angle Odm is inscribed in a semi-circle.

Now, suppose the center of the eccentric to be shifted to m_1 from m; then the angle of advance will be YOm_1, and the eccentricity will be Om_1. By the change we have decreased the angle of advance and increased the eccentricity. The new valve circle, drawn on Om_1 as a diameter, intersects the lap circle at the points b_1 and c_1;

OB_1 is the new position of the imaginary crank at cut-off; and OC_1 is the new position of the imaginary crank at admission.

By making, then, the angle of advance less and the eccentricity greater, we have made the cut-off occur later. If we should consider m_1 the original position of

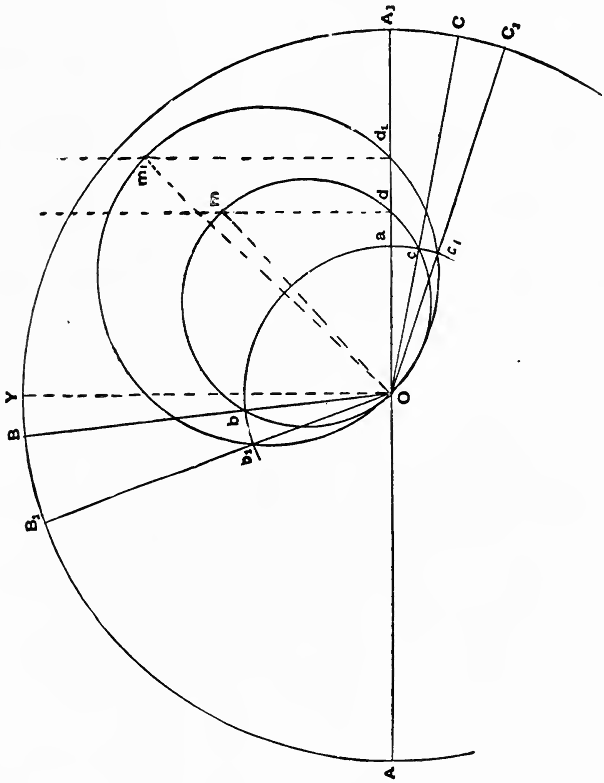


Fig. 48.

the center of the eccentric and m its final position, we see that by increasing the angle of advance and decreasing the eccentricity, we make the cut-off occur earlier. We obtain, therefore, the following propositions:—

1. To make the cut-off occur *later*, make the angle of advance *less* and the eccentricity *greater*.

2. To make the cut-off occur *earlier*, make the angle of advance *greater* and the eccentricity *less*.

It now remains to see what effect is produced on the lead by changing the angle of advance and the eccentricity. In Fig. 48 it is seen that changing the center of the eccentric from m to m_1 has changed the lead from ad to ad_1. Draw m_1d_1 and, since the angle Od_1m_1 is inscribed in a semi-circle, it will be perpendicular to OA_1. Since ad_1 is greater than ad, the line m_1d_1 lies to the right of md. Therefore, it is seen that *if, when the cut-off is changed, the lead becomes greater, the center of the eccentric will lie to the right of a line drawn through its original position perpendicular to the center line of the engine.*

If the lead had remained constant, ad_1 would be equal to ad and the line m_1d_1 would coincide with md. Therefore, it follows that, *if the lead remains constant when the cut-off is changed, the center of the eccentric will remain on a line drawn through its original position perpendicular to the center line of the engine.*

If ad_1 were less than ad, the lead would be less than before and the line m_1d_1 would lie to the left of md. That is, *if the lead is decreased when the cut-off is changed, the center of the eccentric will lie to the left of a line drawn through its original position perpendicular to the center line of the engine.*

Very few engines preserve a constant lead under varying cut-offs, owing to the difficulty of making the center of the eccentric have a straight-line motion, as it must in order that m_1 may always fall on md.

On most engines of the automatic high speed type the eccentric is swung about a pin outside the shaft, so that, as the angle of advance is changed, the center, m, of the eccentric moves in the arc of a circle whose center is the center of this suspending pin. Making the eccentric swing about the pin effects a continual variation in the lead, as the point of cut-off is changed, and the way in which the lead varies depends upon the relative positions of the center of the suspending pin, the center of the shaft, and the center of the eccentric. Usually, although not always, the lead is made to decrease as the cut-off becomes later. Often, the lead is made zero for cut-off at one-quarter stroke, negative for points of cut-off later than one-quarter, and positive for points of cut-off earlier than one-quarter. In such cases, the center of the shaft is between the center of the suspending pin and the center of the eccentric.

The position of the center of the suspending pin is found by assuming three required positions of the center of the eccentric, and finding the center of a circle that will pass through these positions. The center of this circle will be the required center of the suspending pin.

In Fig. 49 is shown a diagram of a governor similiar to that used on the Straight Line engine. *R* is the eccentric, which, as shown, is carried by the frame *T*, and which has an opening in it through which the shaft passes. The eccentric and frame swing about the pin *S*, on the governor wheel. When the engine is cutting off at its *latest*, the center of the eccentric is at *n;* and when the engine is cutting off at its *earliest*, the center of the eccentric is at *a*. The center of the eccentric is shown in Fig. 49 as at *n*, and the eccentricity is the distance of *n* from the center of the shaft. When the engine is run, the centrifugal force of the weight *C* tends to make it move farther from the center of the shaft. When *C* moves, it moves about the pin *O*, and, by means of the link *H*, makes the frame, *T*, and the eccentric, *R*, move about *S* as a center; so that the center of the eccentric moves from *n* towards *a*. When *C* is moved outward by its centrifugal force it will bend the spring *E*, to which it is connected by the band *P*, until the resistance to bending of *E* is equal to the moving force acting on *C;* then the eccentric will be at rest, with its center somewhere between *n* and *a*.

If, on account of an *increased* load, the speed of the engine should be *decreased*, the centrifugal force of *C* would become *smaller*, and the spring would pull *C* towards the center of the shaft, and move the center of the eccentric towards *n;* thus, the cut-off would

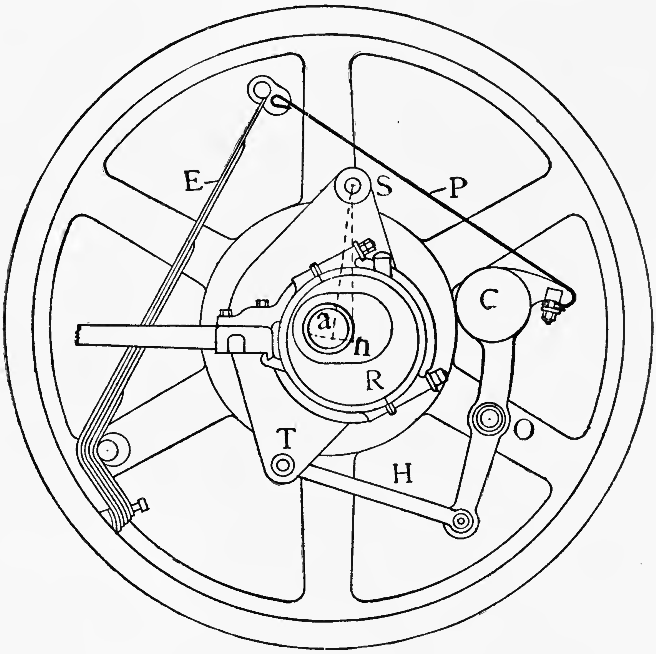


Fig. 49.

be made later and an increased amount of steam would be admitted into the engine to make it go faster. So, also, if, on account of a *decrease* in the load, the speed of the engine should *increase*, the centrifugal force would become greater, and *C* would move farther from the center of the shaft, thus moving the center of the eccentric toward *a*, making the cut-off earlier, and reducing the amount of steam admitted to the engine.

CHAPTER VI.

INDICATORS AND INDICATOR CARDS.

49. Indicators.—The indicator is an instrument by means of which the actual work diagram of the steam in the cylinder of an engine is automatically drawn on a piece of paper. The diagram obtained by the use of the indicator is termed an "indicator card."

There are several indicators, which differ from one another in their details only, for sale on the market.

In Fig. 50 is shown a view of the Crosby indicator, made by the Crosby Steam Gauge & Valve Co., Boston, Mass. It consists of the "drum" *A*, on which the paper for the card is held by clips *a*; the cylinder *F*, in which works a steam tight piston connected to the piston rod *G*; and a lever *K*, which carries a pencil *c*, at its free extremity. The motion of the engine is reduced by a suitable reducing motion and, by means of a cord *D*, is communicated to the drum *A*. As the piston moves forward, the drum is turned in one direction by the pull on *D*, and as it moves back, on its return stroke, the drum is turned in the opposite direction by means of a strong spring inside of it, which is shown in the sectional view of the indicator given in Fig. 51. It is evident that if, during the backward and forward motion of the drum, the pencil *c* had been kept at rest and pressed against the paper, it would have marked on there a line, parallel to the base of the drum, whose length would be proportional to the stroke of the engine.

The steam enters the cylinder *F*, and presses against the piston and makes it rise; and it, in turn, makes the pencil *c* rise. As seen in Fig. 51, the piston in the cylinder is kept down by a spring that must be compressed before the piston can rise. The springs used to keep the piston down are numbered and named according to the number of pounds pressure per square inch required to raise the pencil *c* through one inch: thus, a No. 40 spring, or a 40 lb. spring, is a spring that, when in the indicator, will require an effective pressure of 40 lbs. per square inch to make *c* rise one inch.

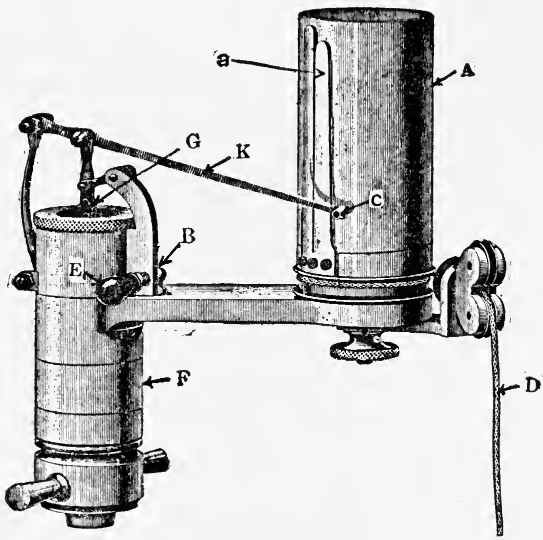


Fig. 50.

By means of the system of levers, shown in Fig. 50, connecting *K* to the frame of the indicator and to the piston rod *G*, the pencil *c* is made to move in a straight line parallel to the axis of the drum *A*. It is in the system of levers for making *c* move in a straight line, that indicators on the market differ most.

The handle *E* can be so adjusted by turning it to the right or left, that when it is pressed forward, so that its inner end strikes against the stop *B*, the pencil *c* will press with any desired pressure against the paper on the drum *A*. When the pencil is pressed against the paper it makes a line, every point of which represents, at once, the position of the piston in its stroke and the pressure of the steam at the same instant. The position of the piston is indicated by the distance of the point from the ends of the card; and the pressure of the steam is referred to the "atmospheric line," obtained by shutting off the cylinder of the indicator from the cylinder of the engine, putting it in communication with the air, and then pressing the pencil against the paper.

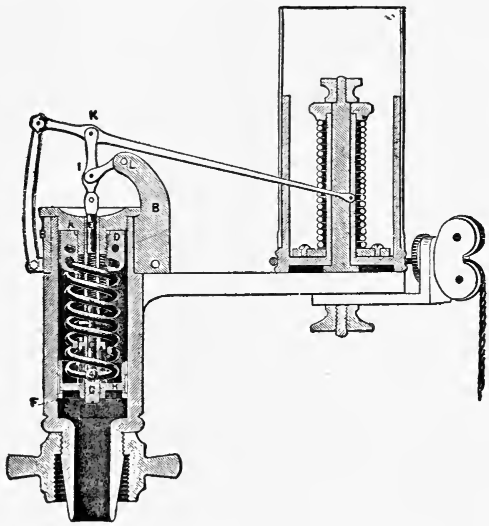


Fig. 51.

50. Adjustments and Connections of Indicators.— In order that the results obtained by the use of an indicator may be of value, it is necessary that the various parts should be in adjustment and act as they are designed to act. If an indicator is tested, and it is in adjustment, it will be found that :—

1. The pencil will move in a straight line parallel to

the axis of the drum; and the line obtained by moving the pencil and keeping the drum at rest, will be at right angles to the line obtained by keeping the pencil at rest and moving the drum.

2. For equal amounts of increase in the pressure on the piston, the *pencil* will rise equal distances.

The motion of the pencil must be adjusted by the maker of the instrument; and if it is not correct the instrument should not be used for important work, as the cards obtained from it would be distorted, and would be apt to give a wrong impression of the real action of the steam in the cylinder of the engine.

The springs used in the cylinder of the indicator should always be tested hot, so that, when being tested, they will be as nearly as possible in the same condition as when in actual use. It is not necessary that the springs should be exact, provided the error is constant, and can be determined.

It is important that the friction of the pencil be as little as possible and that the play, or back lash, in the joints of the levers, for producing the straight line motion of the pencil, should be as small as possible. The friction of the pencil on the paper must be reduced to the minimum, by adjusting the pencil so that it presses against the paper with sufficient force to just make a mark and no more.

The piston of the indicator should be a nice fit in the cylinder, and it is preferable to have it too loose rather than too tight. The fit will be about right if the piston will be moved down the cylinder by its own weight, when the spring is removed. The piston, and all the moving parts attached to it, should be as light as is consistent with strength.

The drum of the indicator should be light and should move easily on its axis. Its cross-section should be a perfect circle; and its axis of rotation should coincide with the axis of the cylinder. The tension of the spring in the drum should be regulated so that the inertia of the drum will not lengthen the cards too much; it should be greater for high speed, than for low speed engines.

The indicator is connected to the cylinder of the engine by a piece of half-inch pipe; and, while much difference of opinion seems to exist as to whether or not the card obtained with a short connection, having as few bends as possible, is materially different from that obtained with a long connection, having several bends, it is, undoubtedly, true that the long connections do no good; and, therefore, the connections should be as short as possible. Where the load on an engine fluctuates through wide ranges, it is almost impossible to determine with any degree of accuracy, from the cards of an engine, whether or not the valves are properly adjusted, unless cards are taken at the same time from both ends of the engine. To do this, it is necessary to have two indicators, one at each end of the cylinder; they should be so arranged that when the pencil of the one is pressed against its paper, the pencil of the other will, also, be pressed against its paper. Where two indicators cannot be arranged to work together, so that cards may be taken simultaneously from each end of the cylinder, and a single indicator is used, it should be connected to the ends of the cylinder so that it will not be necessary to change its position in order to take a card from either end of the cylinder. The best method of making the connections for a single indicator is to connect both ends of the cylinder to a single pipe, along the side of the cylinder, and in the middle of this pipe put a three-way cock, to which the indicator may be attached. Makers of indicators make special three-way cocks for indicator connections.

51. Reducing Motions.— As the diameters of the drums of indicators are, usually, either $1\frac{1}{2}$ or 2 inches,

their circumferences will be about 5 or 6 inches; and, as the length of the card taken on an indicator must be considerably shorter than the circumference of the drum, the cards will usually be 3 inches long for a drum $1\frac{1}{2}$ inches in diameter, and about 4 inches long for a drum 2 inches in diameter. The drum is connected to some point on the engine that, by a suitable "reducing motion," makes the drum move through a distance equal to the desired length of the indicator card. The motion of this point must be such that the drum of the indicator will make one complete movement in one direction, during the same time that the engine makes one stroke; also, the ratio of the velocity of turning of the drum, at any instant, to the velocity of the piston, at the same instant, ought to be a constant quantity. If this last requirement is not fulfilled, the card will be distorted, shortened up at some places and lengthened out at others, so that an event which occurs at, say, one quarter of the stroke of the engine, will be shown on the card as occurring either before or after one quarter stroke. To test the accuracy of a reducing motion, put the engine on dead center, so that the piston is just beginning its stroke, and mark the position of the pencil on the indicator card. Now divide the distance through which any chosen point on the cross-head moves, during one revolution, into a number, such as four or eight, of equal parts, and make a mark at each point of division. Move the piston forward until the chosen point on the cross-head coincides with the first division mark, and mark the position of the pencil on the drum; then move the cross-head to the next division mark, and mark, again, the position of the pencil on the drum. Continue moving the cross-head forward one division, and marking the corresponding motion of the drum, until the piston has made one stroke. Take the card off the drum and determine whether or not the distance between any two successive marks is always the same; if it is, the reducing motion is correct, but if it is not, the reducing motion is defective.

The commonest form of reducing motion is the "pendulum motion," shown in Fig. 52. It consists of a bar, *A*, slotted at one end, and suspended by a pin, *B*, at the other end. A pin, *c*, fastened to the cross-head, fits in the slot at the lower end of the bar; and a pin, *D*, is fastened to the bar. The string from the drum of the indicator is tied to *D*. As the engine moves backward and forward, the pin *c* makes the bar oscillate about *B* as a center. At any instant, the ratio of the velocity of the

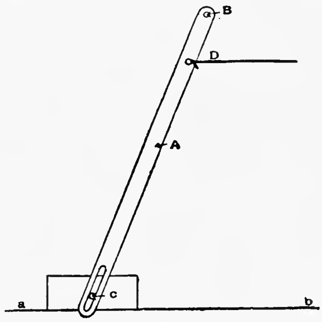


Fig. 52.

drum to the velocity of the piston is equal to $\frac{BD}{Bc}$. As Bc changes for different positions of the piston, this ratio is not a constant one. The point *D* moves in the arc of a circle whose center is *B*; and, therefore, the direction of the string, leading from *D* to the drum of the indicator, is constantly changing. This makes a slight error in the motion of the drum. If Bc is not made less than twice the length of the stroke of the engine and, when in mid-position, is perpendicular to the direction of the motion of the piston, and the string is lead off perpendicular to Bc in mid-position, it will be found that the errors of the motion will be small, and the motion will give fairly good results.

The length of the card given by this motion is equal to the length of the stroke of the engine multiplied by the length of *BD* and divided by the length of *Bc*.

The "Brumbo Pulley," shown in Fig. 53, is a motion devised to overcome the errors of the simple pendulum motion. It is more elaborate then the simple pendulum motion. It consists of a link, *A*, which is suspended by a pin, *B*, and to which is fastened an arc, *D*, whose center coincides with that of the pin *B*. The lower end of *A* is connected to the cross-head, by the link *C*. It is usual to make the link *A*, when in mid-position, perpendicular to the line of motion of the piston.

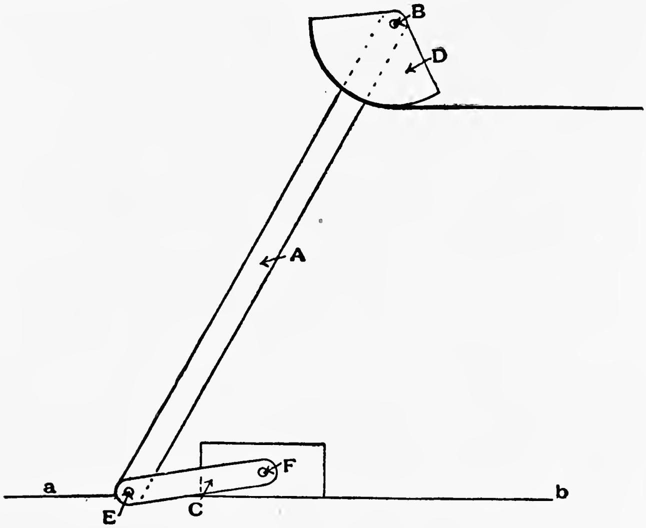


Fig. 53.

By assuming the length of the links *A* and *C*, and plotting the required motion of the drum, for given motions of the piston, a form of arc may be obtained that will give a perfectly correct motion to the drum of the indicator. It is usual, however, to make *D* the arc of a circle; and, then, the motion is not exact, but the error due to the obliquity of the string leading from *D* is avoided, as its direction is not changed.

In Fig. 54 is shown the "pantograph" used on long stroke engines. It gives a perfect motion if properly used.

The instrument is made of a number of light wooden links joined together as shown. When used, the pivot B is fastened, by means of the thumb screw on its end, to any convenient support. The pin A is dropped into an opening either in the cross-head itself or in a piece fastened to the cross-head; so that, while B remains stationary, A has the motion of the cross-head. The cord leading to the indicator is fastened to a pin, E, in the cross-head bar DC, and should be lead off parallel to the line of motion of the piston.

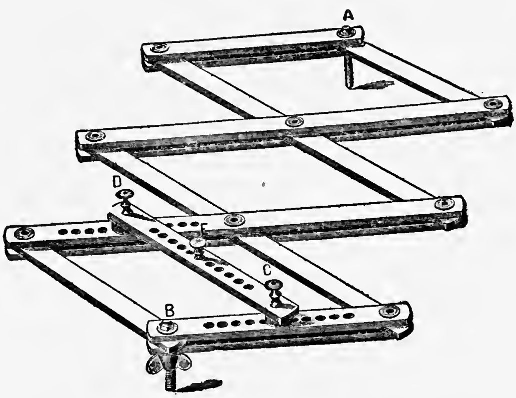


Fig. 54.

This instrument may be used either in a horizontal or a vertical position, whichever is the more convenient, and will give equally good results in either position.

The length of the card depends upon the ratio of the distance BE to the distance BA, and upon the stroke of the engine. For a given position of the cross-bar, BC, and the pin, E, in it, the ratio of BE to BA is a constant one, no matter how long the instrument may be stretched. The position of the cross-bar, DC, may be changed by changing the holes in which the pins D and C are placed. The bar DC must always be parallel to the left-hand bar passing through B. The pin, E, on the cross-bar, DC, must be placed so that it is on the line BA. The joints of the instrument must be kept tight and well lubricated.

The length of the card given by this instrument is equal to the length of the stroke of the engine multiplied by the length of BE and divided by the length of BA.

In Fig. 55 is shown one of the many forms of "reducing wheels," used in connection with high speed engines. Its construction is evident from the figure. The cord from the large drum leads to the cross-head of the engine, and that from the small drum leads to the indicator.

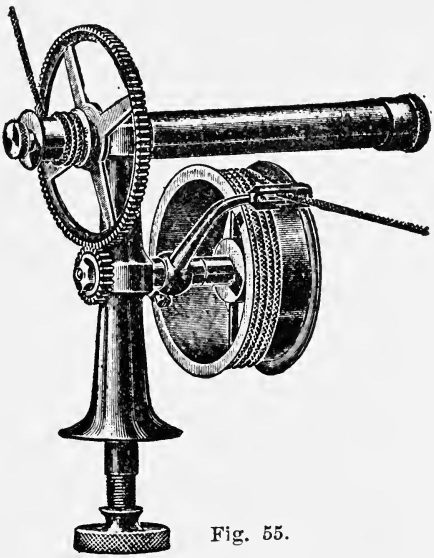


Fig. 55.

52. Cord for Indicator. — The cord used for transmitting the motion of the reducing mechanism to the drum of the indicator, should be a good quality of strong, cotton, cord that has but little stretch. Where the distance from the reducing mechanism to the indicator is great, it is preferable to use good steel wire instead of cord.

In leading off from the reducing motion, the cord should always run, for a short distance at least, parallel to the direction of motion of the piston of the engine. If this is not done, there will be an error due to the obliquity of the cord.

In changing the direction of the cord it should be passed over small guide pulley-wheels, made for that purpose.

It is customary to have a loop in the end of the leading, cord in which may be caught the hook that is usually attached to the end of the cord fastened to the drum of the indicator. By unfastening the hook from the loop, the indicator will be disconnected from the engine and stopped, and the card on the drum may be changed; by catching the hook in the loop, the indicator may be put in motion again.

53. Taking the Indicator Card.— To take a card, turn steam on the indicator and wait until it has got thoroughly warm; connect the drum to the reducing motion; open the communication between one end of the cylinder of the engine and the cylinder of the indicator; press the pencil against the paper, and hold it there while the engine makes *one* revolution; then shut off the indicator from the engine, and at once take the atmospheric line. If the indicator is connected so that a card may be taken from both ends of the engine, take a card from one end, then, as rapidly as possible, shut off that end and take the card from the other end, before taking the atmospheric line.

While one man is taking the indicator card, another ought to be getting the number of revolutions of the engine.

54. To Determine the Horse-Power from the Indicator Card.— The indicator card gives us the real

"work diagram" of the steam in the cylinder, and, as has been explained in Article 22, the area of this diagram represents the work done, by the steam on the engine, each time steam enters the cylinder. Therefore, if we get the area of the diagram in square inches and divide it by the length of the diagram in inches, we shall obtain the mean height of the diagram. If this mean height be multiplied by the number of the indicator spring we shall get the mean effective pressure, P_e, per square inch, of the steam

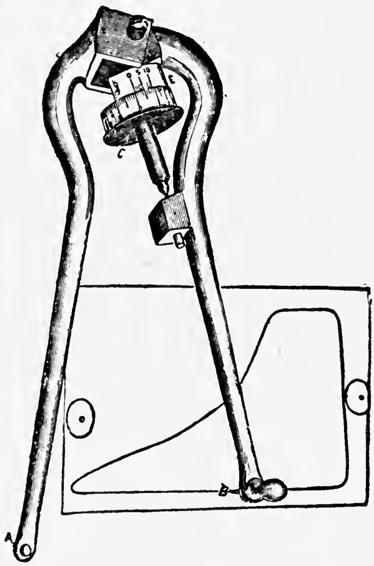


Fig. 56.

on the piston; and if we put this value of P_e in the equation for the horse-power of an engine, as given by (54) of Art. 22, we get

$$H.\ P. = \frac{P_e\ L\ A\ N}{33000}.$$

As explained in Art. 22, L is the stroke of the engine in feet; A, the area of the piston in square inches; and N, the number of times per minute the engine takes steam. If the engine is double acting, N is equal to the number of strokes, or twice the number of revolutions made per minute. P_e, for a double acting engine, ought to be taken as the mean of the values of P_e derived from the cards from both ends of the cylinder.

The best method of determining the area of an indicator card is to use the Amsler Planimeter, shown in Fig. 56. The card is fastened to a drawing board, or the smooth top of a table, and the point A of the instrument pressed into the drawing board or table, so that it cannot move. The tracing point, B, of the instrument is now put at any convenient point on the line of the card, and the reading of the scale on the wheel C is determined by means of the venier E, which enables one to read to the hundredth part of a square inch. The tracing point, B, is now moved around, always in a right-handed direction, on the line of the card until it returns to the point from which it was started. The scale on the wheel is again read, and the difference between the last reading and the first reading, of the scale on C, is the area of the diagram in square inches. Measure the length of the diagram; then divide the area of the diagram by the length of the diagram, and the result multiplied by the number of the indicator spring, is the mean effective pressure, P_e, to be used in calculating the horse-power.

The price of planimeters is now so small that it is rather unusual for any person who has an indicator to be without a planimeter, but for the sake of those who do not have one, the method of obtaining the value of P_e without the use of the planimeter is given. In Fig. 57, let AB represent the atmospheric line of the card. Through A, draw AC in any convenient direction. Take any convenient, small distance, AD, on AC. From D lay off successively nine equal distances, DF, FG, GH, etc., to N; and make each of these distances equal to *twice* the length of AD. Now lay off NC equal to AD, and draw BC. Through the points D, F, G, H, etc., on AC, draw lines parallel to BC and intersecting AB at the points d, f, g, h, etc. Through the points d, f, g, h, etc., draw lines, *11*, *22*, *33*, *44*, etc., perpendicular to AB and, each, intersecting the bounding line of the card in two points.

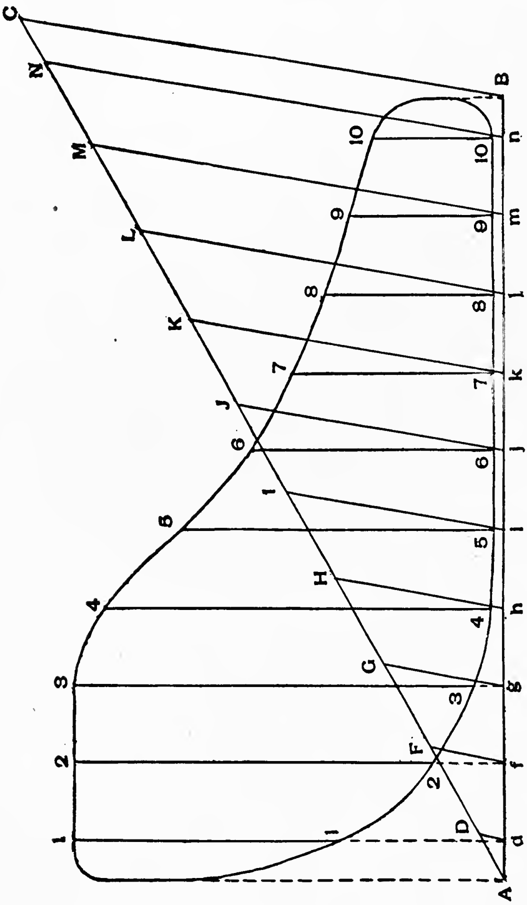


Fig. 57.

Get the sum of the distances *11*, *22*, *33*, *44*, etc., and divide it by ten; multiply this quotient by the number of the indicator spring, and the result will be the value of mean effective pressure, P_e, to be used in determining the horse-power of the engine.

55. To Find the Ratio of Clearance of the Engine from the Indicator Card — In Fig. 58 we have a card whose atmospheric line is AB. That part of the bounding line of the card from 1 to 2 is made during admission of the steam to the cylinder; that part from 2 to 3 is made during expansion of the steam, after cut-off at 2; that part from 4 to 5 is made during the return stroke, while the exhaust valve is open; and that part from 5 to 6 is made during compression, after the exhaust valve

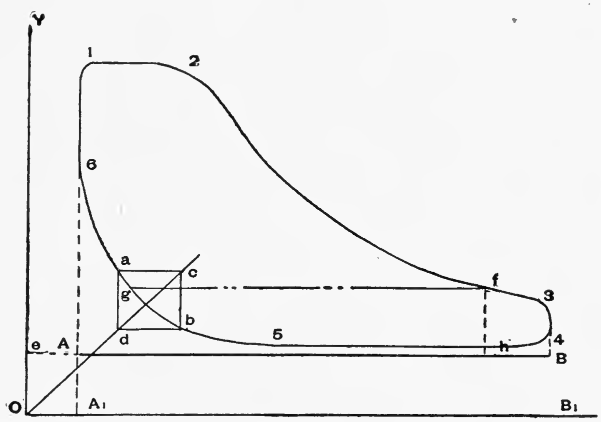


Fig. 58.

closed at 5. The part from 3 to 4 is made at the end of the stroke, when the pressure suddenly falls, from the final pressure of the steam after expansion, to almost the atmospheric pressure; and the part from 6 to 1 is made when the steam begins to enter the cylinder, just before the beginning of the forward stroke.

To find the ratio of the clearance volume to the volume swept through by the piston during one stroke, draw AA_1 perpendicular to the atmospheric line; and lay off AA_1 so that, according to the scale of pressures of the card, it will be equal to the atmospheric pressure, 14.7 lbs. per square inch. Through A_1, draw A_1B parallel to AB.

Now take any two points, such as a and b, on the compression line *56*. Through a draw a line parallel to AB, and continue it until it intersects, at c, a line drawn through b perpendicular to AB; also, through a draw a line perpendicular to AB, and continue it until it intersects, at d, a line drawn through b parallel to AB. Draw cd and continue it until it intersects A_1B_1 at O, the "origin" of the card.

Through O, draw OY perpendicular to AB and intersecting it at e. Then, the ratio of Ae to AB will be the ratio of the volume of clearance, at that end of the cylinder from which the card was taken, to the volume swept through by the piston in one stroke. This construction is based upon the supposition that the curve 56 is an equilateral hyperbola. The ratio obtained may, or may not, be the same for both ends of the cylinder.

56. TO FIND THE WEIGHT OF STEAM USED PER HOUR PER HORSE-POWER.— Take any point, f, near the end of the expansion line, and through it draw the line fg parallel to AB and intersecting the compression line at g.

Let the ratio obtained by dividing the length of fg by the length of AB be denoted by X.

The volume of the steam, at the pressure by the gauge equal to the pressure corresponding to hf, that is used per stroke is equal to the volume swept through by the piston, in one stroke, multiplied by X. Since the volume, in cubic feet, swept through by the piston per stroke is $\frac{LA}{144}$, the volume of steam used per stroke is $\frac{XLA}{144}$.

If the number of strokes made per minute is N, the number made per hour will be 60 N, and the volume of steam, at a gauge pressure corresponding to hf, used per hour will be

$$V = \frac{60\,X\,L\,A\,N}{144} = \frac{5\,X\,L\,A\,N}{12} \qquad (60)$$

Let s be the volume, in cubic feet, of one pound of steam at the gauge pressure corresponding to the pressure hf, as obtained from Table 1; then the weight of steam used per hour by the engine will be

$$(61) \qquad S = \frac{V}{s} = \frac{5\ X\ L\ A\ N}{12\ s}.$$

From (54), we have that the horse-power of the engine is

$$H.\ P. = \frac{P_e\ L\ A\ N}{33000}.$$

The weight of steam used per hour per horse-power will be, from (61),

$$(62) \qquad W = \frac{S}{H.\ P.} = \frac{165000\ X}{12\ s\ P_e} = \frac{13750\ X}{s\ P_e}$$

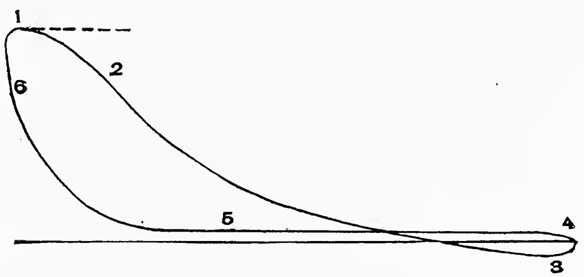
X and P_e are obtained from the indicator card, and s is obtained from Table 1.

57. Interpretation of the Action of the Valves from the Appearance of the Indicator Card.— In studying the cards shown in this article, the student must remember that each card is drawn to illustrate some special defect, which is made very prominent and which may be much less prominent on a card from an engine.

Wire-drawing or Throttling is indicated by the admission line gradually dropping, from the beginning of the stroke to the point of cut-off, below a line drawn parallel to the atmospheric line through the point indicating the initial pressure of the steam, as shown at *12* in Fig. 59. Wire-drawing is due, in the case of non-throttling engines, to the valve not opening far enough to admit the steam to the cylinder, or to bad and poorly designed ports. In the case of throttling engines, the wire-drawing is due to the action of the governor, and is always to be expected.

Too Great an Expansion is indicated by the expansion line extending down below the initial back pressure and forming a loop, as shown at *34* in Fig. 59. The pressure of the steam at the end of the stroke is less than the initial back pressure, and when the exhaust valve is opened the pressure in the cylinder is raised instead of being lowered.

The area of the loop at 345, must be subtracted from the area *1256*. The small loop means negative work; since, while the loop is being formed, the forward pressure



F g. 59.

on the piston is less than the back pressure. In measuring the area of such a card with the planimeter, the instrument will, of itself, subtract the area of the small loop, if the tracing point be moved down and around over the expansion line just as the pencil moved when the loop was formed.

Early Admission, which is a result of too much lead, is indicated by the line 61, in Fig. 60, slanting backwards, and by the sharp, backward-pointing beak at 1.

Early Release is indicated by the line 34, in Fig. 60, slanting forward, and by the sharp, forward-pointing beak at 4.

In the case of engines with single valves, early release will, if the valve is properly made, always accompany early admission, but in the case of engines that have separate steam and exhaust valves, early release need not accompany early admission.

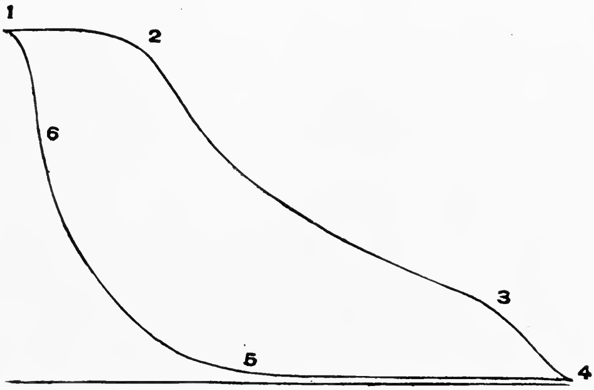


Fig. 60.

In the case of single valve engines, early admission and early exhaust may be corrected by turning the eccentric backward on the shaft, so as to reduce the angle of advance.

When there are separate exhaust and steam valves, the manner of correcting early admission, or early exhaust, depends upon the valve gear.

Late Admission, which is a result of too little lead, is indicated by the line 61, in Fig. 61, slanting forward, and by the sudden rounding at 6.

Late Release is indicated by the line 34, in Fig. 61, slanting backward, and by the sharp, beak-like point at 3.

When the exhaust and admission are controlled by the same valve, late release will always, if the valve is properly designed, accompany late admission; both may be corrected by turning the eccentric forward on the shaft, so as to make the angle of advance greater.

Where there are separate exhaust and steam valves, late exhaust need not accompany late admission, and the manner of correcting either will depend upon the valve gear.

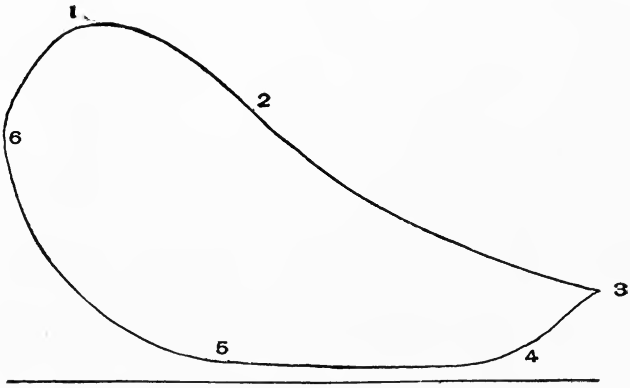


Fig. 61.

Too Great Compression is shown by the compression line being carried above the initial line of pressure and forming a loop, as shown at 1 in Fig. 62.

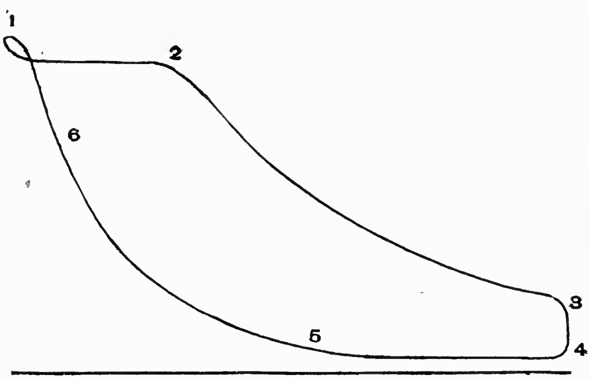


Fig. 62.

This can sometimes be remedied by turning the eccentric backward, so as to make the angle of advance less. Where there are separate steam and exhaust valves, the exhaust valves must be changed so that they will not close so early in the return stroke.

Leak During Compression is indicated by a break in the compression line, as shown at 67 in Fig. 63.

In case a card such as shown in Fig. 63 is obtained, the piston should be tested for tightness. To do this, remove the head of the cylinder, after the steam has been shut off, and block the engine in such a position that the steam port admitting steam to the crank end of the cylinder is open; then turn on the steam and note whether or not

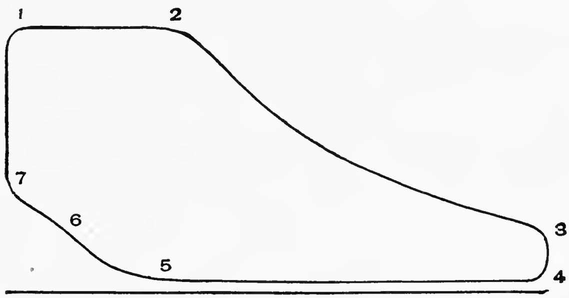


Fig. 63.

any passes the piston. If the piston leaks it should be made tight at once.

If the piston is tight and all the drip cocks are closed tight, a break in the compression line, as shown at 67 in Fig. 63, indicates a leak from the cylinder into the exhaust port.

Leak During Expansion. In order to determine whether or not there is any leakage into, or out of, the cylinder during expansion, it is necessary to draw the theoretical expansion line as follows:—

In Fig. 64, let AB be the atmospheric line; AA_1 be equal to the atmospheric pressure, according to the scale of pressures of the card; and let O be the "origin" of the card, found from the compression line by the method described in Art 55.

Take any point, *c*, on the expansion line, close to the point of cut-off; through it draw *Cc* parallel to *AB*, and *cD* perpendicular to *AB*. Care must be taken to choose the point *c* in such a position that there can be no doubt but that the steam valve is closed.

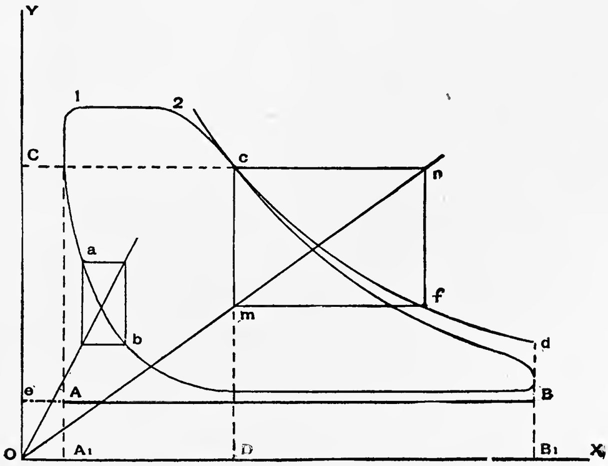


Fig. 64.

Through *O* draw any line intersecting *Cc*, at *n*, and *cD*, at *m*. Draw through *n* a line perpendicular to *AB*, and continue it until it intersects, at *f*, a line drawn through *m* parallel to *AB*.

The point *f* is a point on an equilateral hyberbola, here assumed to be the theoretical expansion line. By repeating the construction, any number of points may be found on the desired curve; and the curve may afterwards be drawn in.

Let *cd* be the theoretical expansion line. If the expansion line of the indicator card is *below* the theoretical line, it indicates condensation or leakage out of the cylinder, or both; while, if the expansion line of the card is *above* the theoretical line, it indicates leakage of steam past the steam valve into the cylinder.

Where the number of times the steam is expanded is large, the expansion line of the card will always fall *below* the theoretical line. This is due to the fact that, for a large number of expansions, the expansion line is not an equilateral hyperbola, as has been assumed in this work.

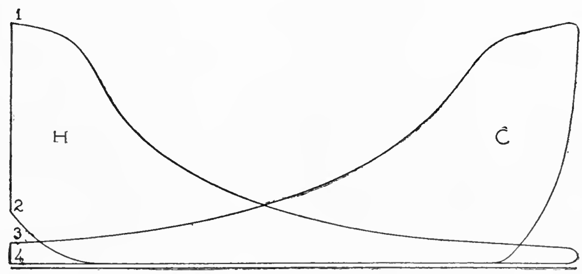


Fig. 64a.

Indicator Cord too Long. When the cord connecting the drum of the indicator to the reducing motion is too long the drum is allowed to move back and come to rest before the piston gets to the head end of its stroke. The effect of this on the card is as shown in Fig. 64a. On the card from the head end of the cylinder, marked *H*, there appears the vertical line 12; and on the card from the crank end of the cylinder, marked *C*, there appears the vertical line 34.

CHAPTER VII

COMPOUND ENGINES AND CONDENSERS.

58. Compound Engines. — While any engine in which the steam is used in more than one cylinder is a compound engine, the term has come to be restricted, by custom, to those engines in which the steam is used in but two cylinders, a high pressure cylinder and a low pressure cylinder. The steam enters the high pressure cylinder on leaving the boiler, and there expands a certain number of times; after leaving the high pressure cylinder the steam passes into the low pressure cylinder, where it is expanded more. By using two cylinders for a given number of expansions, the number of expansions in each cylinder is made much less than it would be if there were but one cylinder, and, therefore, the range of pressures in each cylinder is made less. As the range of temperatures, of expanding steam, depends upon the range of pressures, the use of two cylinders, to obtain a given number of expansions, will reduce the range of temperatures in each cylinder.

The losses in an engine are always of two kinds, thermodynamic and mechanical.

The thermodynamic losses are due to radiation of heat from the walls of the cylinder, and the heating of the metal of the cylinder; they bring about an increase in the comsumption of steam, per horse power per hour, by condensing the steam in the cylinder. It has been found that the condensation of steam in a cylinder is decreased by well lagging the cylinder with non-conducting materials; by increasing the speed of rotation of the engine, thus decreasing the length of time each particle of steam is in contact with the metal of the engine; by reducing the number of expansions in the cylinder, thus reducing the range of temperatures of the steam during expansion; and, finally, by decreasing the radiating surface of the cylinder.

The mechanical losses are due to the friction of the moving parts, and are decreased by good workmanship and materials in construction; by reducing the number of moving parts; and by care and good management while running.

The whole object of compounding is to reduce the amount of steam used, per horse-power per hour, by reducing the losses in the engine. As the lagging of a simple engine can be made just as good and efficient as that of a compound engine, there can be no reduction of losses on that account; the speed of rotation of the simple engine may be just as great as that of the compound, and, therefore, there can be no reduction on that account; two cylinders expose more surface for radiation than one, and therefore, compounding increases the losses due to this; but the range of expansion in each cylinder of a compound engine is less than it would be in a single cylinder engine of the same total number of expansions, and there is, then, a reduction of loss in the compound engine on this account.

The mechanical losses may be, and usually are, greater in the compound engine of the same power than they are in the single cylinder engine.

Summing up then, the compound engine tends to increase the thermodynamic losses by increasing the radiating surface, and also tends to increase the mechanical losses by increasing the number of moving parts; but it tends to decrease the thermodynamic losses by decreasing the range of temperatures of the steam in the cylinders, by reducing the number of expansions in each.

When the reduction of losses is greater than the increase, the compound engine should be used. With low pressures and a small number of expansions, the single cylinder engine is more economical than the compound engine. It is probably safe to say that for pressures under sixty pounds, by the gauge, the single cylinder condensing engine is more economical than a compound engine; but for pressures above sixty pounds, by the gauge, the compound engine is more economical. The higher the pressure and the greater the number of expansions, the greater is the economy of the compound engine as compared to the single cylinder engine. In the case of non-condensing engines the single cylinder engine is probably more economical for all pressures under ninety pounds, by the gauge, but above this it is probable that the compound engine is the better to use.

As the single cylinder engine has its limit, at which it becomes less economical than the compound engine, so the compound engine becomes less economical than the triple expansion engine, where the steam is expanded successively in three cylinders, for pressures greater than about one hundred and twenty pounds by the gauge.

Up to about four expansions, it is probable that the single cylinder engine is more economical than the compound engine; for from four to six expansions, there is not much choice between the two engines, so far as economy is concerned; for from six to ten expansions, the compound engine is the more economical. For a greater number of expansions than ten, it will usually be found better to use a triple expansion engine.

Compound engines are usually classified under two heads, viz., Tandem Compound, Cross-compound.

59. Tandem Compound Engines.— Tandem compound engines are those which have the two cylinders placed one in front of the other. There are two pistons, one for each cylinder; one piston rod; one connecting rod; and one crank. The steam flows as directly as possible from

the high pressure cylinder into the low pressure cylinder, and the connecting pipes are usually made small. There is no "receiver," or vessel into which the steam flows, between the high pressure cylinder and the low pressure cylinder.

The tandem compound engine occupies less space than the cross-compound.

60. CROSS-COMPOUND ENGINES. — The two cylinders of the cross-compound engine are placed side by side, and there is for each cylinder, a separate piston rod, connecting rod, cross-head, and crank. The steam often passes into a "receiver" after leaving the high pressure cylinder, and there remains until it passes into the low pressure cylinder. The cranks are usually set at an angle of 90° with one another; so that when the high pressure piston is at the end of the stroke, the low pressure piston is in mid-position and has half a stroke to complete before it is ready to take steam; it is on this account that it is always necessary to have a receiver for cross-compound engines whose cranks are at 90°.

The cross-compound engine occupies more space than the tandem compound engine, but by having two piston rods, two connecting rods, and two cranks, it really becomes equivalent to two engines connected to the same shaft, each of which need be only about half the power of the single cylinder engine to do the same work. And, therefore, all the parts of the cross-compound engine may be lighter and smaller than the same parts on either a single cylinder engine or a tandem compound engine.

The twisting effort of the crank shaft is made more uniform by having the cranks at 90°, as when one piston is exerting its maximum effort the other is exerting a much less effort, and, therefore, the engine becomes easier to govern.

61. Ratio of Cylinders of Compound Engines.— The ratio of the volume of the low pressure cylinder to the volume of the high pressure cylinder may be denoted by K, and, since the strokes of the two cylinders of a compound engine are usually the same, we have $K = \frac{D^2}{d^2}$. Where D is the diameter of the low pressure cylinder, and d the diameter of the high pressure cylinder. The value of K ought to be made to vary with the pressure of the steam and the number of times the steam is expanded. Designers usually try to make the value of K such that the " drop," or fall in pressure of the steam between the high pressure cylinder and the low pressure cylinder, shall be small, when the work of the engine is divided so that it is the same in each cylinder.

The value of K varies from $2\frac{1}{4}$ to 4 for automatic high speed engines; and from 3 to $4\frac{1}{2}$ for engines of the Corliss type; it is always equal to the quotient obtained by dividing the total number of times the steam is expanded by the number of times it is expanded in the high pressure cylinder.

62. The Horse-power of Compound Engines.— In determining the horse-power of a compound engine from which indicator cards have been taken, work up the card for each cylinder just as if it were the card from a single cylinder engine; and having found the mean effective pressure for each cylinder, calculate the horse-power of each by the method explained, in Art. 54, for a single cylinder engine. The sum of the horse-powers developed in the two cylinders will be the horse-power of the engine.

In calculating the horse-power that a compound engine ought to develop, for given conditions as to absolute steam pressure, total number of expansions of the steam, number of strokes per minute, diameters of low and high pressure cylinders, and length of stroke, proceed as if there were but one cylinder, of the same size as the low pressure cylinder, in which the total expansion of the steam took place. If the horse-power obtained by assuming that all the work was done in the low pressure cylinder be multiplied by a factor, whose value depends upon the type of the compound engine, the result will be equal to the horse-power the compound engine ought to develop under the given conditions.

Let A be the area, in square inches, of the low pressure piston; L, its length of stroke, in feet; N, the number of strokes made by the low pressure piston per minute; E, the total number of times the steam is expanded in the compound engine; and C, a factor whose value depends upon the type of the compound engine. The horse-power of the engine will be given by the equation

$$(63) \qquad H.\ P. = \frac{P_e L A N C}{33000}.$$

P_e is the mean effective pressure, which, by substituting E for r in (50), is found to be

$$(64) \qquad P_e = P_1 \frac{(1 + hyp.\ log.\ E)}{E} - P_3.$$

P_1 is equal to the gauge pressure of the steam in the boiler plus 14.7. P_3 is the average back pressure, and may be assumed to be between two and a half and five pounds per square inch, for condensing engines.

The value of the factor C will vary from 0.75 to 0.90 for automatic high-speed engines, and from 0.80 to 0.90 for engines of the Corliss type.

63. CONDENSERS.— Condensers are usually associated in the minds of most people with compound engines,

although there is no reason why a condenser should not be used with a single cylinder engine, provided there is a good supply of water.

There are two kinds of condensers in use at present, viz., the jet condenser and the surface condenser.

The jet condenser is more generally used on land, because it is simpler; not so apt to get out of repair; and, where the water is good, gives as good results as the surface condenser, if not better. When the jet condenser is used, the exhaust steam from the engine enters the "condensing chamber," where it comes in contact with the condensing water, in the form of spray, and is condensed. The condensed steam, together with the condensing water, is pumped out of the "condensing chamber" into the "hot well," by means of a pump called the "air pump." Part of the water in the hot well is pumped back into the boiler by the "feed pump," and the remainder is allowed to run to waste.

The condensing water is sprayed, when it enters the condensing chamber, by means of a rose fixed on the end of the injection pipe, or by means of a perforated plate, called the "spattering plate," fixed inside of the condensing chamber.

The principal parts of a jet condenser are, the condensing chamber, the air pump, and the hot well.

The air pump of jet condensers is sometimes worked by the engine, being connected to it by a belt, but is often provided with an independent steam cylinder of its own. Condensers whose air pump is worked independent of the engine are known as "independent condensers." Condensers of this class will pump their own condensing water; but they ought not be expected to draw water more than 12 or 15 feet, as there are a great number of joints about a condenser that are often difficult to keep tight, and the water flows into the condensing chamber against whatever pressure there may be there. The smaller the height through which the air pump lifts the condensing water, the better will the condenser work.

In Fig. 65 is shown a section of an "independent condenser." *A* is the steam inlet; *B* is the water inlet; *F* is the condensing chamber; *G* is the air pump, worked by the steam cylinder *K*; and *J* is the outlet to the hot well.

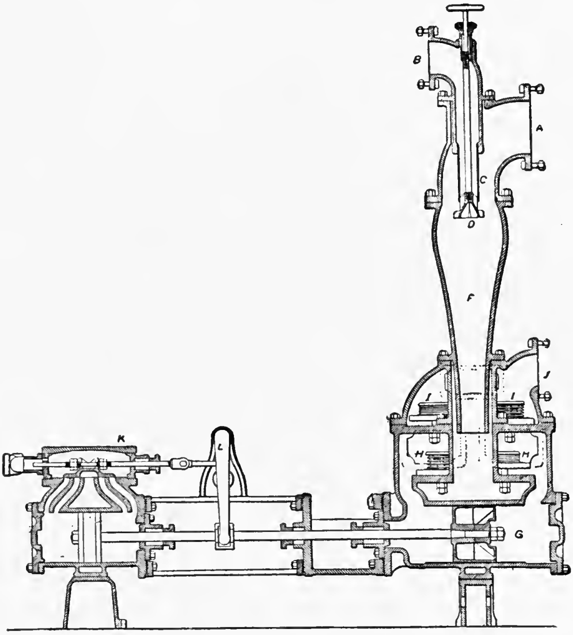


Fig. 65.
Worthington Independent Condenser.

When the surface condenser is used, the condensing water passes through a number of small copper or composition tubes, on the outside of which the steam to be condensed circulates. The steam coming in contact with the cold surface of the tubes is condensed and falls to the bottom of the condenser, from where it is pumped, by the "air-pump," into the hot well. The condensing water is either drawn or forced through the condenser tubes by the "circulating pump."

The surface of the tubes in contact with the steam forms the condensing surface of the condenser. The

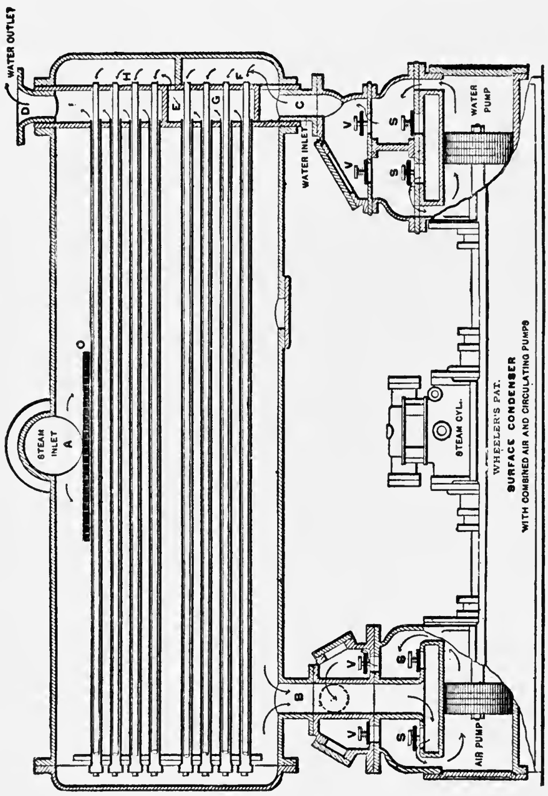


Fig. 66.

tubes are always of very small diameter, and the metal of which they are made is as thin as is consistent with safety. Sometimes, instead of the steam circulating on the outside of the tubes, it passes through them, and the condensing water circulates on the outside.

The principal parts of the surface condenser are: the condensing surface, being the surface of a great number of very small tubes; the air pump; the circulating pump; and the hot well.

The great point in favor of the surface condenser is the fact that the steam, after being condensed, is returned to the boiler and used again without coming in contact with the condensing water. The same water for making steam is used over and over in the boiler, without any new feed water except what is necessary to supply the loss due to leakage of the boiler and the various connections from the boiler to the engine and condenser. As but little fresh feed water is needed, the boiler may be kept very free from scale and sediment. This, of course, is a great point in its favor for use at sea, or where the water is bad and likely to form scale in the boiler.

The main objection to the surface condenser was that when the tubes got hot they expanded, and then were likely to become loose and leak. They were, also, likely to "creep;" that is, work out of the bearing at one end. Both of these objections have been overcome in the Wheeler condenser, shown in Fig. 66, by fastening the tubes at one end only.

An explanation of this condenser is unnecessary as everything is clearly shown in the figure.

64. EFFECT OF THE CONDENSER ON THE POWER OF THE ENGINE.— Condensing the exhaust steam from an engine diminishes the back pressure by creating a partial vacuum behind the piston. This vacuum is generally spoken of as being so many "inches of mercury;" each inch of mercury representing a diminution of about half a pound in the back pressure, and therefore, a corresponding increase in the mean effective pressure on the piston.

It is seldom that the vacuum maintained by a condenser will exceed 26 inches of mercury, while the usual vacuum will be about 24 inches. It is usual to assume that the difference between the back pressure without the condenser and the back pressure with the condenser, is equal to the pressure corresponding to the vacuum maintained by the condenser.

The equation for the horse-power of an engine is, from (54),

$$H.\ P. = \frac{P_e\ L\ A\ N}{33000}.$$

Now, if we suppose that the speed of the engine is to be the same after the condenser is attached, as it was before, L, A, and N will be constants and the horse-power will vary directly as P_e varies. From equation (50) we have

$$(65) \qquad P_e = P_1\,\frac{(1 + hyp.\ log.\ r)}{r} - P_3.$$

If P_3 is the average back pressure without the condenser; P'_3, the average back pressure with the condenser; P_e, the mean effective pressure without the condenser; P'_e, the mean effective pressure with the condenser; $H.\ P.$, the horse-power of the engine without the condenser; and $H'.\ P'.$, the horse-power with the condenser; we have

$$(66) \qquad \frac{H'.\ P'.}{H.\ P.} = \frac{P'_e}{P_e} = \frac{P_1\left(\dfrac{1 + hyp.\ log.\ r}{r}\right) - P'_3}{P_1\left(\dfrac{1 + hyp.\ log.\ r}{r}\right) - P_3}.$$

Equation (66) shows how many times greater the horse-power of the engine is with a condenser than it is without a condenser.

Suppose that it is desired that the power of the engine should not be increased, but that it should remain the same, and that the cut-off should change so that the engine will use less steam with the condenser than without it. Let r be the number of times the steam is expanded without the condenser, and r', the number of times it is expanded with the condenser.

In this case we have $H.\ P. = H'.\ P'.$, and, therefore, $P_e = P'_e$, or

$$(67) \qquad P_1 \frac{(1 + hyp.\ log.\ r)}{r} - P_3 = P_1 \frac{(1 + hyp.\ log.\ r')}{r'} - P'_3.$$

From this we get

$$(68) \qquad \frac{1 + hyp.\ log.\ r'}{r'} = \frac{1 + hyp.\ log.\ r}{r} - \frac{(P_3 - P'_3)}{P_1}.$$

This equation can be solved by trial, using Table 2. If there were no clearance to the engine and V were the volume in cubic feet of the cylinder of the engine, the volume of the steam used per stroke without the condenser would be $\frac{V}{r}$; and the volume used with the condenser would be $\frac{V}{r'}$. The fraction of saving would be

$$(69) \qquad y = \frac{\frac{V}{r} - \frac{V}{r'}}{\frac{V}{r}} = 1 - \frac{r}{r'}.$$

From y, given by (69), must be subtracted the fraction obtained by dividing the quantity of steam required to run the condenser by $\frac{V}{r}$.

If, instead of increasing the work done by the engine or changing the cut-off, it should be desired that the engine should do the same work with the condenser, and cut-off at the same point, that it did without the condenser, but use a lower absolute pressure of steam in the boiler, we would have, since $P_e = P'_e$ and the value of r is not changed,

$$(70) \qquad P_1 \frac{(1 + hyp.\ log.\ r)}{r} - P_3 = P'_1 \frac{(1 + hyp.\ log.\ r)}{r} - P'_3.$$

P'_1 is the absolute boiler pressure that is carried when the condenser is used.

From (70) we have

$$(71) \qquad P'_1 = P_1 - \frac{(P_3 - P'_3)\,r}{1 + hyp.\,log.\,r}.$$

65. Amount of Condensing Water Required.— The number of pounds of water required to condense one pound of steam depends upon the temperature of the steam when it leaves the engine, and upon the initial and final temperatures of the condensing water. The temperature of the steam when it leaves the engine depends upon the absolute pressure, P_2, of the steam at the end of the forward stroke, just before the exhaust valve is opened. The expression for this final absolute pressure may for all ordinary purposes be taken as $P_2 = \frac{P_1}{r}$. P_1 is the initial absolute pressure of the steam, and r is the number of times the steam is expanded.

The gauge pressure of the steam, or pressure above the atmosphere, when it enters the condenser is $P_2 - 14.7$.

Let l be the latent heat of one pound of steam at a gauge pressure of $P_2 - 14.7$, and t_1 the corresponding temperature. The values of l and t_1 may be obtained from Table I. Also, let t_2 be the initial temperature of the condensing water, and t_3 the final temperature.

The heat given out by one pound of the steam when it condenses will, evidently, be $l + t_1 - t_3$; and the heat taken up by every pound of the condensing water will be $t_3 - t_2$; therefore, the number of pounds, W, of water required to condense one pound of steam from the engine will be,

$$(72) \qquad W = \frac{l + t_1 - t_3}{t_3 - t_2}.$$

t_3 should be taken as 110°, for the ordinary work of condensers.

The value of t_2 depends upon the source of supply of the water and the climate of the location of the engine; it will be greater in summer than in winter. It will usually be safe to take the value of t_2 as 80°.

CHAPTER VIII.

HEAT AND COMBUSTION OF FUEL.

66. Steam Making.—The steam used in an actual engine is made in an apparatus that is often spoken of as the boiler or the boiler plant. It consists of three main parts, each, in a manner, dependent upon the other two, and yet in many ways distinct from them. These parts are, the furnace, the boiler proper, and the chimney.

In Fig. 67 is shown a section of a furnace and boiler such as is in common use everywhere in this country. The various parts are lettered so that their relations to one another may be seen at once.

The part termed the furnace is the part in which the heat, afterwards converted by the engine into work, is generated by the combustion of fuel.

The boiler is simply a closed vessel which contains the water of which the steam used in the engine is formed. The boiler may be of any shape or size.

The chimney is the part which carries off the products of combustion.

The fuel is put in the furnace on the grate, and is there burned. During the combustion of the fuel heat is generated; a part of this heat is given directly to the boiler, by radiation from the hot fuel, and a part is carried off by the gases generated by the combustion. These gases pass out of the furnace into the chimney, and from there they pass into the air. On the way from the furnace to the chimney, the hot gases are made to come in contact with the boiler; and as the boiler is cooler than the gases, a part of the heat they contain is given up to it. The heat thus obtained by the boiler is transmitted to the water, which is gradually heated and, finally, converted into steam.

The dimensions and proportions of the furnace depend

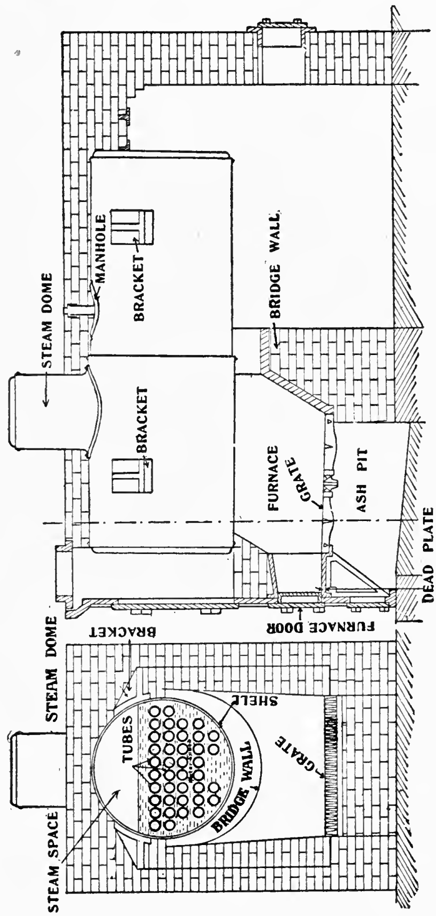


Fig. 67.

upon the heat required by the boiler per unit of time, the kind of fuel, and the type or kind of furnace.

The dimensions and proportions of the boiler depend upon the amount of steam required by the engine per unit of time, the conditions under which the steam is generated, and the type of the boiler.

The dimensions and proportions of the chimney depend upon the kind of fuel, the amount used by the furnace per unit of time, and the temperature at which the hot gases pass off.

67. Steam Required per Hour.— In all problems relating to boilers it is necessary to know, as a basis upon which to design the furnace, the boiler and the chimney, the number of pounds of steam required per hour and the conditions under which it must be made.

If we are designing a boiler to supply steam for an engine of given dimensions and power, using steam at a given initial gauge pressure, we may calculate the number of pounds of steam that will be used per hour by the engine and to this add a per cent to cover leakage and condensation, and thus obtain the number of pounds of steam the boiler will probably be called upon to supply per hour.

From (57) of Art. 24, we have that the weight of steam used per stroke by an engine is

$$S = \frac{LA}{144\,r\,s}.$$

If N is the number of strokes made by the engine per minute, the weight, W_1, of steam used by the engine per minute will be

$$W_1 = N\,S = \frac{LAN}{144\,r\,s}. \tag{73}$$

To make allowances for waste from various sources, for the amount of steam used by the pumps, and for that condensed in the engine, the amount of steam the boiler ought to be designed to supply per minute may be taken as $\frac{3}{2}\,W_1$. As the steam required to be supplied per hour is 60 times that required per minute, the expression for, W, the number of pounds of steam required to be supplied by the boiler per hour is

$$(74) \qquad W = \frac{60 \times 3}{2} W_1 = \frac{90\ LAN}{144\ r\ s} = \frac{5\ LAN}{8\ r\ s}.$$

68. Heat Required per Hour.—Having assumed or determined the number of pounds of steam required per hour, it is next necessary to determine, if it is not already known, the pressure by the gauge, and the initial temperature of the "feed-water," or water entering the boiler.

Let P be the pressure per square inch, by the gauge, of the steam in the boiler; H, the total heat of evaporation above 32°, in heat units, of one pound of steam at the gauge pressure P; t, the initial temperature of the feed-water.

H must be taken from Table I; while t depends upon the source of supply of the feed water, and upon considerations that will be discussed later.

It is evident that, since H is the heat required to raise the temperature of one pound of water from 32° to the temperature of the boiling point corresponding to the gauge pressure P and then to turn the water into steam, the heat required to raise one pound of water from a temperature t to the boiling point corresponding to P and then turn it into steam, will be $H - (t - 32)$. The heat required, then, to evaporate W pounds of water per hour, under the given conditions, will be

$$(75) \qquad H_1 = W\left[H - (t - 32)\right].$$

From equation (75), it is evident that, for a given value of W, H_1 will be smaller as we make H smaller and as we make t larger, and hence the latter should always be as large as possible.

The value of H depends upon the pressure by the gauge at which the steam is formed and upon nothing else; and as is seen by an inspection of Table 1, the higher is the pressure of the steam in the boiler, the greater is the value of H. It is also seen, however, that the value of H increases very slowly as the pressure increases. Thus, the value of H corresponding to a pressure of 75 lbs., by the gauge, is 1179.4, and the value of H corresponding to a pressure of 150 lbs., by the gauge, is 1193.5; so that, while the pressure has been increased by 75 lbs., H has been increased by but 14.1 heat units.

The value of t depends upon the source of supply of the feed-water and upon whether or not the feed-water is heated before it is forced into the boiler. It is customary to force the feed-water through a "feed-water heater" before it enters the boiler. As feed-water heaters will be discussed farther along, it will suffice to say that they usually consist of a number of tubes, surrounded by exhaust steam from the engine, through which the feed-water is forced before it enters the boiler. The water, while passing through the tubes of the heater, has its temperature raised by the heat imparted to it by the exhaust steam. The heat in the exhaust steam would be lost if it were not taken by the feed-water; so that the feed-water heater, by raising the temperature of the feed-water, is a heat saving appliance, and a valuable adjunct to any engine and boiler plant, where there is exhaust steam escaping into the atmosphere.

The greater t is made, the less will be the value of H_1 for given values of W and H.

Instead of speaking of the number of heat units a boiler will require per hour it is customary to speak of "the equivalent water from and at 212°" that it will evaporate per hour.

The equivalent water from and at 212° is the number of pounds of water that *could be evaporated* by the expenditure of the same number of heat units actually used by the boiler, if the water entered the boiler at 212° and was converted into steam at a temperature of 212°.

Since the heat required to convert one pound of water at a temperature of 212°, into steam at 212° is equal to the latent heat of water at atmospheric pressure, about 966 heat units, it is seen that the expression for "the equivalent water from and at 212°," W_0, is

$$(76) \qquad W_0 = \frac{H_1}{966} = W \frac{[H - (t - 32)]}{966}.$$

The factor $\frac{H - (t - 32)}{966}$ is called the "factor of evaporation," and may be defined as, *the factor by which the water actually evaporated by a boiler must be multiplied in order to reduce it to "equivalent water from and at 212.°"*

In Table 3 will be found factors of evaporation for different gauge pressures of steam and different temperatures of feed water.

Equation (76) gives us a means of determining the heat required per hour for a boiler, when we know the equivalent water from and at 212° required to be evaporated per hour. As will be seen later, boilers are often assumed as being able to evaporate 34½ lbs. of water from and at 212° per hour per horse-power. Upon this assumption, $W_0 = 34\frac{1}{2}\,B$, where B is the horse-power of the boiler, and the expression for H_1 becomes,

$$(77) \qquad H_1 = 966\ W_0 = 33327\ B.$$

69. Fuel Required per Hour.— The number of pounds of fuel required to supply the heat necessary for the boiler per hour, depends upon the heat developed by the combustion of one pound of the fuel and upon the amount of heat that is lost, in various ways, by the furnace,

the boiler and the chimney. If we take the amount of heat developed by the combustion of one pound of the fuel and from this quantity subtract the amount that is lost, we shall obtain the quantity of heat used by the boiler per pound of fuel; and the total quantity of heat required per hour divided by the quantity used per pound of fuel will give us the number of pounds of fuel that must be burned per hour in the furnace.

It is evident, therefore, that it is extremely important that we should know the amount of heat developed by the complete combustion of one pound of the fuel in the furnace.

Combustion may be defined as a rapid oxidation, accompanied by the evolution of light and heat.

In all fuels there are certain elements that will not burn, but which remain after combustion and form ash; and there are other elements that are in the fuel in such small quantities that their presence may be neglected. The principal elements in all fuels, whether gaseous, liquid, or solid, are carbon, hydrogen and oxygen.

The carbon may be present either in a free, uncombined state or in combination with a part of the hydrogen.

The hydrogen is always present either in combination with the oxygen or with a part of the carbon. We always assume that a part of the hydrogen is in combination with *all* of the oxygen, and that the rest is in some sort of combination with part of the carbon.

It is generally assumed that the oxygen in a fuel is in combination with a part of the hydrogen, and is present as water. The oxygen, of course, does not burn, but simply reduces the amount of hydrogen that is available for combustion.

Upon combustion, the carbon in a fuel may form one of two compounds:—

1. If the combustion is complete, every atom of the carbon will take up, and enter into combination with, two

atoms of oxygen and form carbon dioxide, or carbonic acid gas as it is sometimes called, whose chemical symbol is CO_2.

2. If there is a lack of oxygen and the combustion is not complete, every atom of the carbon will combine with one atom of oxygen and form carbon monoxide, whose chemical symbol is CO.

It has been determined by experiments that when one pound of carbon is completely burned, so as to form carbon dioxide, there is evolved, by the combustion, 14,500 heat units; * also, that when one pound of carbon is burned to form carbon monoxide, there is evolved 4,400 heat units. Thus there is a difference of 10,100 heat units between the amounts of heat evolved by the complete and the partial combustion of one pound of carbon. It is customary, in all discussions as to the heat of combustion of fuels, to assume that all the carbon in the fuel is completely burned to carbon dioxide.

Hydrogen, when burned, enters into combination with oxygen, in the proportion of two atom of hydrogen to one of oxygen, and forms water, whose chemical symbol is H_2O.

It has been determined by experiments that one pound of hydrogen, on being burned, will evolve 62,032 heat units, or about 4.28 times as many heat units as are evolved by the complete combustion of one pound of carbon. When one pound of hydrogen burns it unites with eight pounds of oxygen; so that, with the oxygen present in any fuel there is always united one-eighth of its weight of hydrogen. If, then, we subtract from the total weight of hydrogen present in a fuel, one-eighth of the weight of the oxygen, the remainder will be the weight of free hydrogen in the fuel, or the weight of hydrogen that will be burned.

To obtain the theoretical amount of heat that will be

* The heat evolved by the complete combustion of one pound of carbon varies slightly with the source from which the carbon is obtained, and recent experiments have shown that it is probably nearer 14,600 than 14,500.

evolved by the combustion of one pound of fuel, it is necessary for us to first learn, from a chemical anaylsis, the weights of carbon, hydrogen, and oxygen in one pound of the fuel. The weight of carbon multiplied by 14,500, will give the number of heat units that will be evolved by the complete combustion of the carbon; and the weight of *free* hydrogen, equal to the total hydrogen less one-eighth of the weight of the oxygen, multiplied by 14,500 times 4.28, will give the heat that will be evolved by the combustion of the hydrogen in the fuel. The sum of the heats evolved by the carbon and by the free hydrogen will be the total heat evolved by the combustion of one pound of the fuel. Putting what has been said in mathematical language, we see that the expression for the theoretical amount of heat, h, evolved by the complete combustion of one pound of fuel is

$$(78) \qquad h = 14500\left[C + 4.28\left(H' - \frac{O}{8}\right)\right].$$

C is the weight, in pounds, of carbon in one pound of the fuel. H' is the weight, in pounds, of hydrogen in one pound of the fuel. O is the weight, in pounds, of oxygen in one pound of the fuel.

Owing to the fact that, in most fuels, there is always a small quantity of substances, other than carbon and hydrogen, that burn and give off more or less heat, and that a part of the total heat evolved is used in decomposing the elements before they can burn, the theoretical amount of heat obtained by the use of equation (78) is not exactly equal to the heat actually evolved by the combustion of one pound of the fuel. Equation (78), however, may be used when no other means is at hand for determining the amount of heat evolved by one pound of a fuel.

The principal fuels used in boiler furnaces are wood and coal.

Wood is seldom used, on account of the expense, except in special establishments where the refuse consists largely of shavings, saw-dusts, and pieces of wood that must be got rid of. In such cases, of course, it is much better and cheaper to use this refuse as fuel than it is to buy coal. Wood burns rapidly and with a bright flame, but does not evolve much heat. It is customary to consider one pound of wood equivalent to 0.4 pounds of coal.

Coal is more extensively used as a fuel, in boiler furnaces, than any other substance. Although the mining engineer classifies coal into several groups or classes, it will suffice for us to consider all coal used in boiler furnaces as either anthracite or bituminous coal.

Anthracite coal is a hard, dense coal containing a large per cent of carbon and a small per cent of volatile matter; it is slow to ignite and burns at a high temperature with little or no visible flame.

Bituminous coal is somewhat soft and easily broken; it usually contains from 20 to 50 per cent of volatile matter; it ignites easily and burns freely with quite a flame.

Coke is the residue obtained after distilling off the gases from certain kinds of bituminous coals; it is not very dense, but contains a high per cent of carbon.

In Table 4 is given the heat developed by the complete combustion of one pound of various fuels.

The loss of heat by a furnace, boiler, and chimney may be ascribed to four causes: —

1. Incomplete combustion.
2. Radiation.
3. Hot gases escaping out of the chimney.
4. Dropping of fuel through the grate into the ash-pit.

The loss due to incomplete combustion is the most serious of all losses. It may be that, when the coal is put into the fire, there is not a sufficient amount of air to burn the volatile gases that pass off, so that the greater part of them will not be burned. Or, it may be, that owing to a lack of air, the carbon is not completely burned to carbon dioxide, but is burned only to carbon monoxide. If the carbon is not completely burned, there is a loss of 10,100 heat units for every pound of carbon converted into carbon monoxide.

The loss by radiation may be reduced by a careful and correct setting of the furnace and boiler, and by taking precautions to have just as little hot surface exposed as possible.

The loss due to the hot gases escaping up the chimney may be estimated if we know the temperature, t_1, of the air entering the furnace and the temperature, t_2, of the gases entering the chimney. The specific heat of one pound of chimney gases may, without serious error, be taken as equal to that of ordinary atmospheric air, or 0.24; so that every pound of gas escaping out of the chimney carries off 0.24 $(t_2 - t_1)$ heat units.

The heat carried off by the gases in the chimney cannot be said to be wasted, unless there is some fault in the design of the boiler plant, as this heat is used to produce the draft. It will be seen later that the amount of air entering the furnace depends upon the draft of the chimney, which, in turn, depends upon the height of the chimney and upon the temperature of the air outside, and that of the gases inside of the chimney. The temperature of the gases in the chimney should be sufficient to produce the required draft, and no more.

The loss due to dropping of fuel through the grate ought not to be large, if the grate is properly proportioned and care is exercised in firing the furnace. Some loss, of course, is always bound to occur, but when this loss is large there should be a change either of the grate or fireman, or perhaps of both.

The sum total of all the heat lost in the various ways, per pound of the coal consumed, will amount to from 20 to 50 per cent of the total heat of combustion. For the best boiler plants, where care is exercised in firing, the amount of heat used per pound of coal may be taken as from 70 to 80 per cent of the heat of combustion; for good boiler plants the amount may be taken as from 60 to 70 per cent of the heat of combustion; and for poorly designed plants, poorly fired, the amount will be from 40 to 60 per cent of the total heat of combustion.

Therefore, if K represents the fraction of the total heat of combustion that is utilized, the heat utilized per pound of coal will be, from equation (78),

$$(79)\quad h_0 = Kh = K14500\left[C + 4.28\left(H' - \frac{O}{8}\right)\right].$$

If, now, we divide the total heat, H_1, required per hour, as given in (75) or (77), by the heat utilized, h_0, per pound of coal we obtain the amount of coal, F, that it will be necessary to burn per hour in the furnace.

Therefore, the expression for F is

$$(80)\qquad F = \frac{H_1}{h_0}$$

70. Air Required for Combustion.—The air admitted to a furnace for the combustion of a fuel is a mechanical mixture, consisting principally of oxygen and nitrogen; these gases are present in the proportion, by weight, of 23 per cent of oxygen and 77 per cent of nirtogen; by volume, the proportion in which they are present is 20 per cent of oxygen and 80 per cent of nitrogen. The oxygen, only, is used in combustion; the nitrogen is inert, and so far as aiding combustion is concerned, is useless.

From chemistry we learn that when two substances unite chemically they always do so in a certain fixed proportion, by weight. It is known that one pound of hydrogen always requires eight pounds of oxygen for its complete combustion into water, H_2O; also, that one pound of carbon requires $\frac{4}{3}$ of a pound of oxygen for its combustion to carbon monoxide, CO, and $\frac{8}{3}$ of a pound of oxygen for its combustion to carbon dioxide, CO_2.

Therefore, if we assume complete combustion of the carbon and of the free hydrogen, the number of pounds of oxygen required per pound of fuel will be,

$$(81) \qquad O_1 = \frac{8}{3} C + 8 \left(H' - \frac{O}{8} \right).$$

C is the total carbon, H' the total hydrogen, and O the total oxygen in one pound of the fuel.

Now, since, as has been said, there is only 23 per cent, by weight, of oxygen in the air, the weight of air, A, required to supply O_1 pounds of oxygen will be, $A = \frac{O_1}{0.23}$. Putting for O_1 its value, and neglecting fractions, we get the following expression for the pounds of air required for the complete combustion of one pound of fuel.

$$(82) \qquad A = 12\,C + 36 \left(H' - \frac{O}{8} \right).$$

Ordinarily, we may assume that 12 pounds of air will be needed for the complete combustion of one pound of *coal;* and as one pound of air at 32° occupies a volume of 12½ cubic feet, the volume of air, at 32°, required for the complete combustion of one pound of coal may be taken as 150 cubic feet.

It has been found that in the boiler furnace there is always needed more air than is actually necessary for the combustion of the fuel, in order to dilute the gases of combustion and to make sure that every particle of hydrogen and carbon will come in contact with the amount of oxygen necessary to burn it. The amount of air for dilution, as the surplus air is called, depends upon how intimately the air for combustion and the combustible gases are mingled and mixed. It has been found that it is advantageous to have the air enter in a number of streams rather than in a large body, and that the higher the velocity of the entering air the less the quantity required for dilution. Experience has shown that in the case of natural or chimney draft, the amount of air required for dilution will be about equal to the amount required for complete combustion; while in the case of forced draft, the amount of air required for dilution will be about equal to one-half that required for combustion.

Therefore, we may say that for chimney draft it is necessary to supply 24 pounds of air to the furnace for each pound of coal burned; and for forced draft, 18 pounds of air are required per pound of coal. Of course, the greater the quantity of air we supply to the furnace, over and above that actually required for combustion, the greater will be the loss of heat due to the temperature of the escaping gases. Again, the greater the quantity of air supplied per pound of coal burned the larger must be the chimney to carry off the gases from the furnace.

It is evident, therefore, that the amount of air supplied to the furnace should be no more than the quantity actually necessary for proper combustion.

71. Rate of Combustion.—By the rate of combustion is meant the number of pounds of fuel that is burned per square foot of grate surface per hour.

There are two limits to the rate of combustion, a maximum and a minimum.

The maximum rate depends upon the kind of fuel and the force of the draft; and where the draft is great enough to supply the amount of air required for combustion, the maximum limit will be reached only when the draft becomes so great as to blow the fuel off of the grate bars. It is evident that this limit will depend somewhat upon the density of the fuel. It is probable that the greatest rate of combustion has been attained in locomotives, where a rate of about 120 pounds of anthracite coal has been reached. Of course, this is not a rate that is continued for any great length of time.

The minimum rate of combustion depends upon the kind and nature of the fuel and the construction of the grate of the furnace; it is the rate at which it is possible to keep a bright clear fire just on the point of burning through in places, and so admitting a body of cool air to chill the furnace. For anthracite coal, the minimum rate of combustion in boiler furnaces is about 4 pounds; and for bituminous coal, it is about 10 pounds.

The rate of combustion, with chimney draft, for anthracite coal, will vary from 7 to 20 pounds, the average being about 12 pounds; for bituminous coal, the rate will vary from 12 to 40 pounds, the average being about 20 pounds.

The whole tendency of modern practice is towards forced draft and high rates of combustion.

72. The Furnace.—In Fig. 67 the furnace under the boiler is shown in section. It will be seen that the "grate bars" rest on the "bridge wall" at the back end, and on the "dead plate" at the front end. Where the grate is long, it is customary to make it up of two lengths of "grate bars" supported at the middle by a "bearing bar."

The grate bars are made of cast iron and of different shapes for different kinds of fuels. There are quite a number of patented grate bars on the market, for each of which the inventor claims certain advantages.

In Fig. 68 is shown a view of a common form of grate bar. The bars are made single or double, in order that, by using a number of single and double bars, grates of any desired width may be built up. There are lugs, marked *A* in Fig. 68, which prevent the bars from being

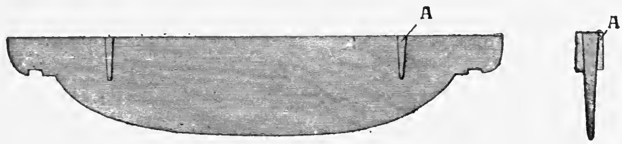


Fig. 68.

packed too close together, and which cause the formation of air spaces, through which the air for combustion enters from the "ash-pit." The area of the openings for the admission of air between the grate bars, ought to be made to depend, somewhat, upon the kind of coal burned, but is usually about one-half the total area of the grate.

Where very fine coal is used the area of the openings should be less than where coarser coal is burned, in order that there may not be a great waste by the dropping of coal, through the openings, into the ash-pit.

In Fig. 69 is shown a somewhat different style of grate bar that is quite extensively used.

The grate bars are seldom made longer than four feet; and grates are seldom made longer, measured from the

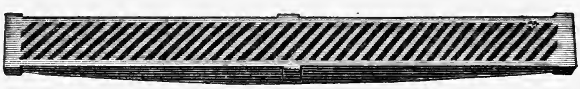


Fig. 69.

furnace door to the bridge wall, than seven feet. If a grate is longer than seven feet it becomes almost impossible to fire and stoke it properly, and the end next to the bridge is very apt to be useless, if not detrimental. The grate is usually built with a slope, from the front of the furnace towards the bridge wall, of from one-half to one inch fall for every foot in length.

The dead-plate at the front of the grate is sometimes made quite large, although it is usually rather small.

The doors of the furnace should be made double, and should have perforations in them. The perforations are for the admission of air, which aids in the combustion of the gases passing off from the fuel on the grate and, also, cools the door and prevents it from burning out. The inner part, or lining, of the door is to prevent the outer part, or door proper, from being too highly heated by the heat radiated from the burning fuel on the grate.

The furnace shown in Fig. 67 is what is termed an external furnace, since it is exterior to the boiler. Sometimes, however, the furnace is contained in the boiler itself, as shown in Fig. 73, when it is termed an internal furnace.

73. Firing the Furnace.—The term "firing" is applied to the work of putting the fuel in the furnace, and keeping the fire in a clean, bright condition. To the uninitiated it would seem as if the whole thing to be done, in feeding a furnace, would be to open the furnace door and throw the fuel in on the grate; it has been found, however, that in order to preserve a good, hot, fire, it is best to adopt some system of firing.

There are three systems in common use, each of which has its advocates. These systems are, the spreading, the alternate, and the coking.

In the spreading system, the fresh charge of coal is spread in a layer over the whole area of the grate. This is perhaps the most common system, and if the fire is fed frequently, with small quantities of fresh fuel at each charge, it will give good results. If, however, the fire is fed at rather long intervals, with a large quantity of fuel at each charge, the fire will be chilled every time fresh fuel is put on it. This chilling of the fire will result in the incomplete combustion of the gases in the furnace and a loss of heat.

The alternate system can only be used to advantage with a wide grate. In this system, the fresh fuel is put alternately on each side of the grate in sufficient quantities to cover about one-half of the whole surface. The whole area of the grate is never covered, at any one time, with fresh fuel; so that, the whole fire is never chilled. The gases that pass off from the fresh fuel on one side of the grate, come in contact with the surplus hot air coming through that side of the grate on which there is no fresh fuel, and are burned. This system of firing does not require such care and watchfulness on the part of the fireman, as does the spreading system, but as has been said, it can be used to advantage only in the case of rather broad furnaces.

The coking system is the system that has been adopted in all mechanical stoking devices. In it, the fresh charge of fuel is put just inside the door of the furnace, on the dead plate, and allowed to remain there until the greater part of the most volatile gases are driven off; then the coal is pushed farther back into the furnace, where a part of it is burned and all of the gases are driven off, and a fresh charge is put on the dead plate. Each succeeding charge pushes the charges preceding it further towards the end of the furnace, and the charges are put in at such intervals that each will be completely burned during its passage from the dead plate to the bridge wall. The gases, evolved from the charge on the dead plate, are obliged to pass over the hot bed of fire and come in contact with the surplus air coming through the back end of the grate; and, as both the gases and the air are at a high temperature, there is a strong probability that all the gases will be burned. This method of firing works equally well with bituminous coal and anthracite coal,

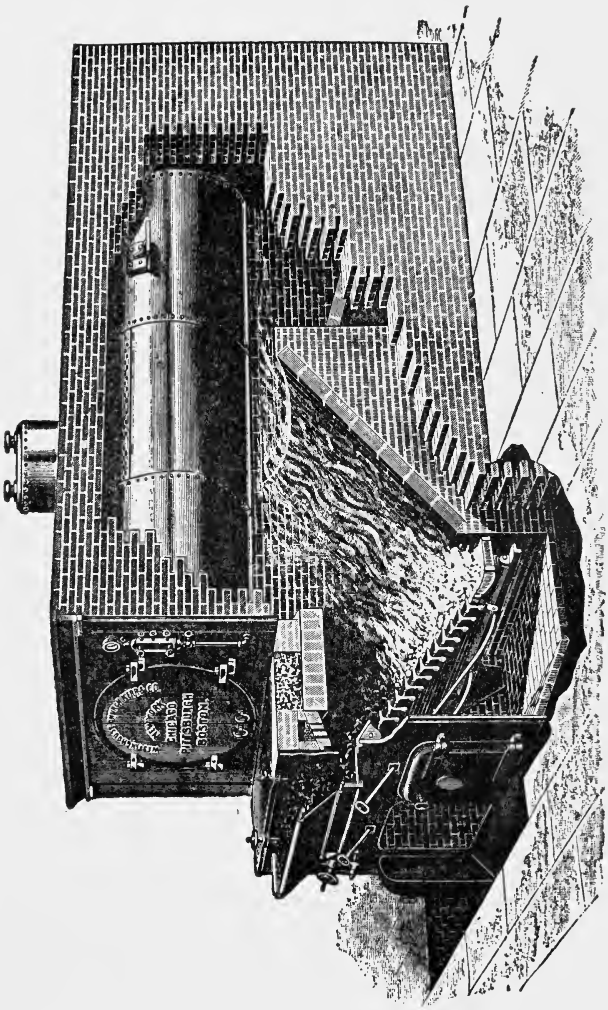


Fig. 70.

but is of more value where bituminous coal is used, on account of the large per cent of volatile gases such coal contains.

It is impossible to say that any one of the three systems of firing is better than another; any one will give good results if it is properly carried out, and any one is better than no system at all. To get good results as to evaporation and rate of combustion, it is absolutely necessary that care be exercised in the firing; good results cannot be obtained by careless, bad firing, where the coal is thrown into the furnace in any way and in large quantities at a time.

The fire should be kept bright, and free from dirt and clinkers, and of as nearly a uniform thickness over the whole grate as possible. A bright, clean fire will always give better results than one that is dirty, and full of clinkers and ash.

The thickness of the bed of coals has quite a marked influence upon the economy of the combustion. The best thickness will depend largely upon the fuel to be burned, but it may safely be said that it should not be allowed to be less than six inches for a good, hot fire. It is seldom that the thickness of the fire is allowed to be greater than twelve inches. Experiments, with the same coal, have shown that a fire six inches thick gave poorer results, as to evaporation, than a nine-inch fire; and the nine-inch fire gave poorer results than a twelve-inch fire. Care should be taken to see that the fire never burns through in spots, leaving a portion of the grate uncovered by hot coals.

74. Mechanical Stokers.— There are several forms of mechanical stokers on the market, and almost all of them feed the coal, from a hopper, onto the front part of the furnace, where it is partially burned, and from where it is gradually made to move back along the grate to the

bridge wall. The rate of feeding of the fuel and the rate of combustion must be such that all of the coal will be burned on its way from the furnace door to the bridge wall. If the rate of feeding is too great, the fuel will not be completely burned when it reaches the end of the grate, and a part of it will be forced into the ash pit, and be lost; while if the rate of feeding be too small, the part of the grate next to the bridge wall will not be completely covered with live coals, and cold air will leak through and chill the gases on their way to the chimney.

The grates of almost all mechanical stokers are usually inclined at a considerable angle to the horizon, and the coal is made to move from the dead-plate to the bridge wall by a movement of the grate bars. This movement is generally derived from a shaft that is rotated by a small engine.

It is probable that the Roney Mechanical Stoker is one of the most used and best known mechanical stokers in this country. Fig. 70 shows the Roney Mechanical Stoker as applied to the ordinary return tubular boiler. In Fig. 71 is shown the stoker in detail.

Referring to Fig. 71, it is seen that the grate is inclined from the front of the furnace towards the bridge wall; and that the grate bars are arranged as steps, with their lengths at right angles to the direction of the length of the boiler. The coal is fed into the "coal hopper," and from there is pushed onto the "dead plate;" the coal falls onto the front grate bars and is made to move from one grate bar to another, towards the bridge wall, by an oscillating motion of the grate bars. This motion of the grate bars is derived from the "rocker bar," which is moved back and forth by the "connecting rod.' The "connecting rod" derives its motion from the "agitator," that is connected, by means of the "link," to the "disk-crank." The disk-crank is rotated by the shaft to which it is fastened. By the motion of the rocker arm the grate bars

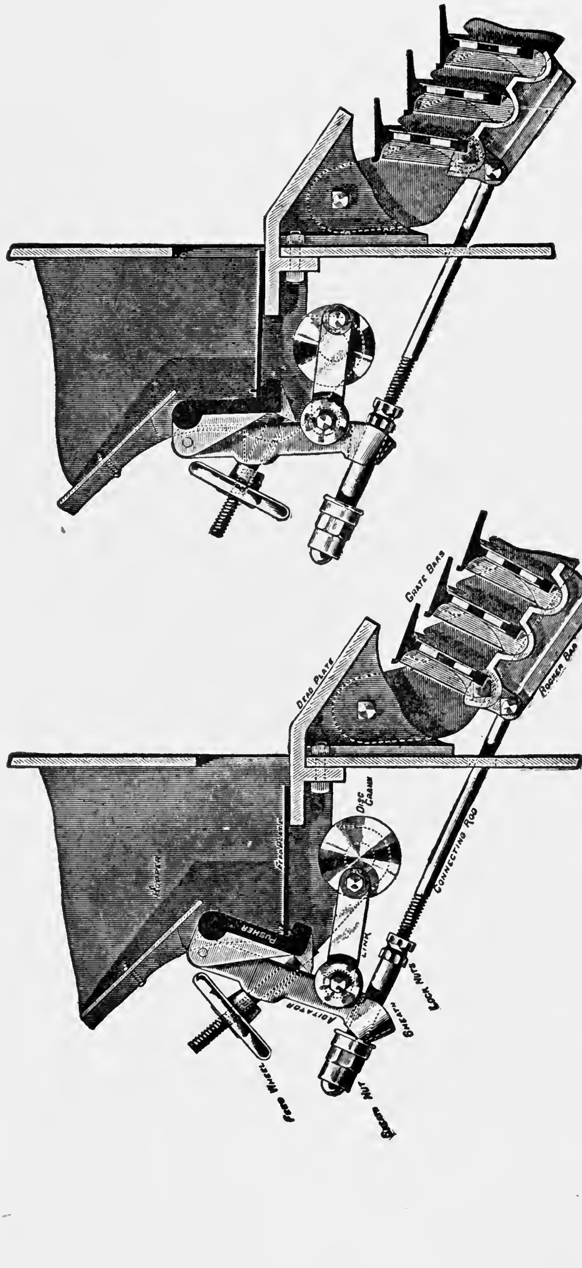

Grate Bars
Dead Plate
Rocker Bar
Disc Crank
Connecting Rod
Link
Pusher
Agitator
Sheath
Lock Nuts
Feed Wheel

Fig. 71.

are made to assume an inclined position, and then a stepped position. When the bars are in the inclined position, the coal tends to slide down the grate towards the bridge wall. When the agitator is moved out towards the end of its stroke it strikes a nut on the end of the connecting rod and moves the rod until the grate bars assume the inclined position. As the agitator moves inward, it pushes on the pusher and forces coal from the hopper onto the grate and, also, forces the coal on the grate down the grate bars, which remain in their inclined position until the agitator strikes the inside nut on the connecting rod. Thus, it is seen that during about half the time of one revolution of the disk crank, the coal on the grate may move towards the bridge, and during the other half of the time it is at rest on the grate. By means of the "feed wheel," the pusher may be adjusted so that it will be moved forward during any desired part of the forward motion of the agitator. The amount of coal fed to the furnace depends largely upon the adjustment of the pusher, although in some cases the weight of the coal, in the hopper and on the grate, will cause a movement of the coal along the grate. By adjusting the lock nuts on the connecting rod, the movement of the grate bars may be adjusted. Also, everything may be made to occur quicker or slower by running the disk crank at a high or low speed.

75. HAWLEY DOWN DRAFT FURNACE.— One of the latest, successful, improvements in furnaces for boilers may be said to be the down draft system of the Hawley Down Draft Furnace Co. In this system, two grates are used, one above the other, as shown in Fig. 72.

The upper grate is made up of a number of tubes, *C*, connected to the drums *A* and *B*. The drums, *A* and *B*, are connected to the boiler, so that there is always a circulation of water through the drums and through the tubes. The bottom grate, *D*, is of the ordinary kind.

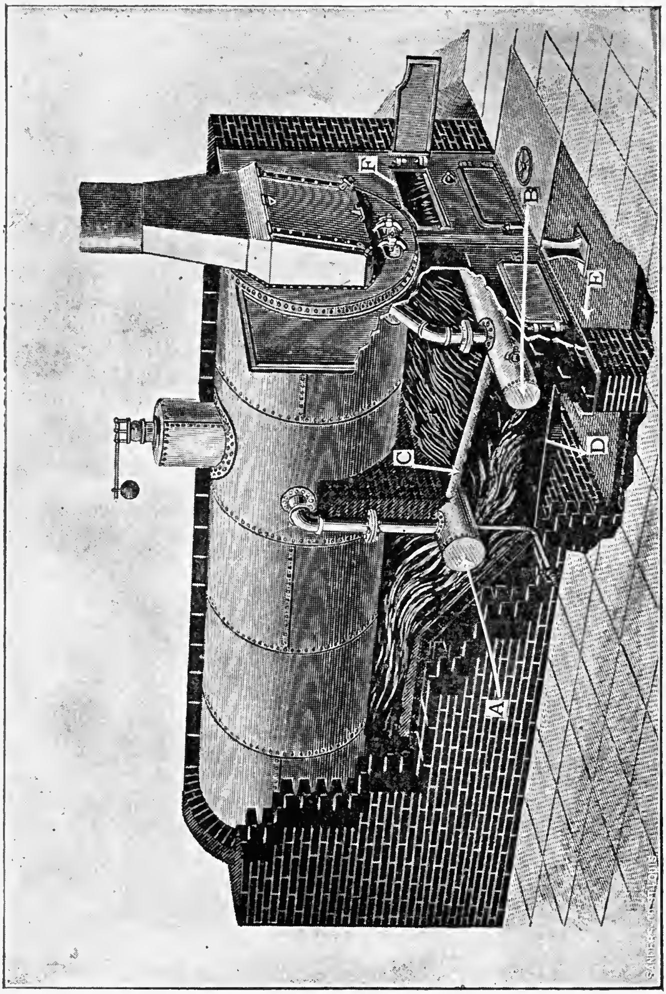


Fig. 72.

All the fresh fuel is put on the upper grate, through the doors *F*, and air is admitted, also through *F*, over the fuel. By the arrangement of the furnace, the air and gases of combustion are made to pass down *through* the upper grate in order to reach the chimney. All the fine particles of coal, and partially burned coal, drop between the upper grate bars and fall onto the lower grate; so that, the lower grate is always covered with a thick layer of hot coals. Air is admitted through the ash pit, and it passes up through the hot coals on the lower grate; a part of this air is used in burning the coals on the lower grate; the remainder comes in contact with the hot, combustible, gases from the upper grate and burns them.

This furnace has given good results as a preventer of smoke.

CHAPTER IX.

BOILERS.

76. Types of Boilers.— Before considering the different types, it will be well to define certain terms used in referring to boilers.

The grate surface is the area of the grate of the furnace of the boiler.

The heating surface is the area of the surface of contact of the hot gases with the boiler, while on their way from the furnace to the boiler chimney.

The shell is the main vessel in which is contained the water and steam.

The water space is the volume of that part of the boiler occupied by the water.

The steam space is the volume of that part of the boiler occupied by the steam.

All classifications of boilers are based, generally, upon peculiarities in the design of the shell, or in the relative position of the heating surface and grate surface. Boilers are, also, often classified according to the place where they are to be used. Among the different classifications may be mentioned the following: marine boilers, land boilers; upright boilers, horizontal boilers; internally fired boilers, externally fired boilers; fire-tube boilers, water-tube boilers.

A marine boiler is one that is used on vessels; and a land boiler is one that is used on land.

An upright boiler is one having the axis of the shell vertical; and a horizontal boiler is one having the axis of the shell horizontal.

An internally fired boiler is one that has its furnace inside the shell of the boiler; and an externally fired boiler is one that has its furnace exterior to the shell.

A fire-tube boiler is one a part of whose heating surface is the *internal* surfaces of a number of tubes *surrounded* by water; and a water-tube boiler is one a part of whose heating surface is the *external* surfaces of a number of tubes *filled* with water.

77. Old Types of Boilers.—Under this head has been included those forms of boilers that were first used, and that are still used to a large extent in England, but

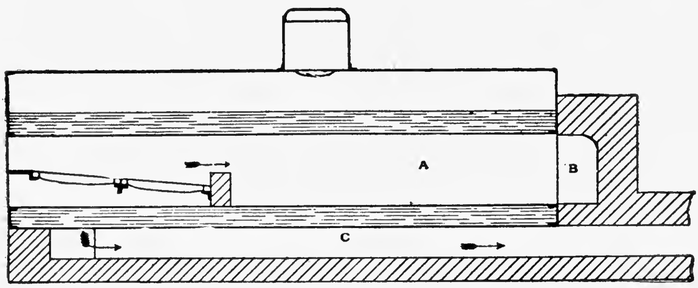


Fig. 73.

that are seldom met with in this country. The principal among these may be said to be the Cornish boiler and the Lancashire boiler.

The Cornish boiler is an internally fired boiler, with a cylindrical shell. The furnace is contained in a large cylindrical flue running the whole length of the boiler from front to rear. In Fig. 73 is shown a longitudinal section of a Cornish boiler, and in Fig. 74 is shown a cross-section of one. The furnace is in the flue *A*, and the products of combustion pass through *A* to the rear of the boiler, where they divide into two portions, and return through the passages *B* to the front of the boiler, they then enter *C* and pass through it to the chimney.

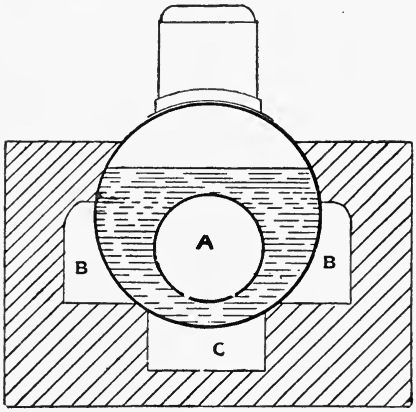


Fig. 74.

The Cornish boiler has, always, but a single furnace and a single flue.

The Lancashire boiler, like the Cornish boiler, is an internally fired boiler, but it differs from the Cornish boiler in having two furnaces and, for a part of its length, two flues. The two flues of the Lancashire boiler are united into a single flue near the back end of the boiler. The distinguishing features of the Lancashire boiler are the two furnaces and the two flues uniting in the single flue a short distance from the back end of the boiler.

The Cornish and Lancashire boilers built to-day are almost always provided with "Galloway tubes" across the flues. These are shown in Fig. 75, and are simply tubes, *a*, of the shape of a truncated cone, placed in an inclined position across the large flue *b*. They facilitate the circulation of the water in the boiler and increase the heating surface.

In many of the modern forms of Cornish or Lancashire boilers, the gases pass *under* the boiler from the

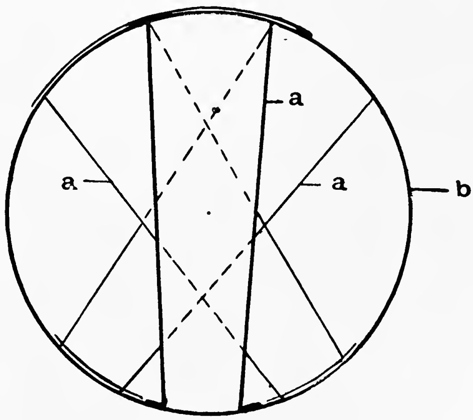


Fig. 75.

rear towards the front, instead of along the sides, and pass along the sides on the way to the chimney. That is, in Figs. 73 and 74, the gases pass to the front through the passage *C*, after leaving *A*, and then pass off to the chimney through the passages *B*.

78. Return Fire-tube Boilers.— Boilers of this type are usually spoken of as "return tubular boilers," and are more used in this country than those of any other type.

In Figs. 67 and 76 are shown views of a boiler of this type.

These boilers, as seen from the cuts shown, are cylindrical, externally fired, boilers. The gases pass from the furnace under the boiler to the "back connection," and

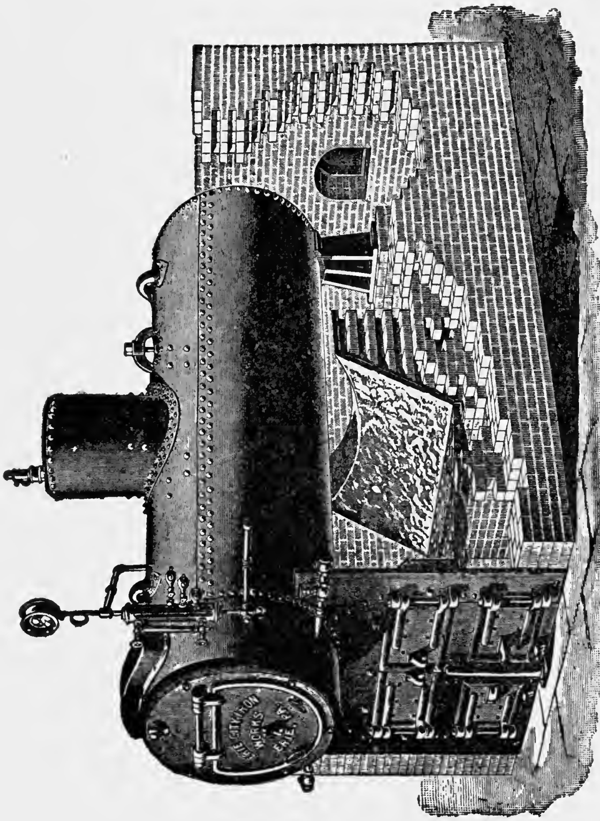


Fig. 76.
Return Fire-tube Boiler.

from there pass to the front through a number of tubes or flues. If the tubes are of greater diameter than six inches, they are usually spoken of as flues. The number of tubes varies with the diameter of the shell of the boiler and the diameter of the tubes.

The heating surface of a return tubular boiler consists of that part of the surface of the shell in contact with the gases; of the *outside surface* of the tubes, the surface in *contact with the water;* and that part of the ends, or heads, of the boiler with which the gases come in contact. The area of the surface of the shell that is in

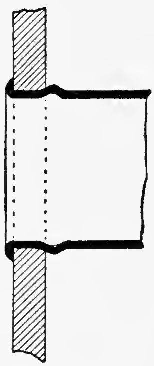


Fig. 77.

contact with the gases depends upon the setting of the boiler; although it is customary to assume, what is not always true, that two-thirds of the area of the shell is exposed to the hot gases. It is usually customary to neglect, as unimportant, the parts of the heads in contact with the gases. It is evident, then, that the heating surface of the shell will be obtained by multiplying two-thirds the outside circumference of the shell, in feet, by the length, in feet; and the heating surface of the tubes will be obtained by multiplying the *outside* cylindrical surface of one tube by the number of tubes in

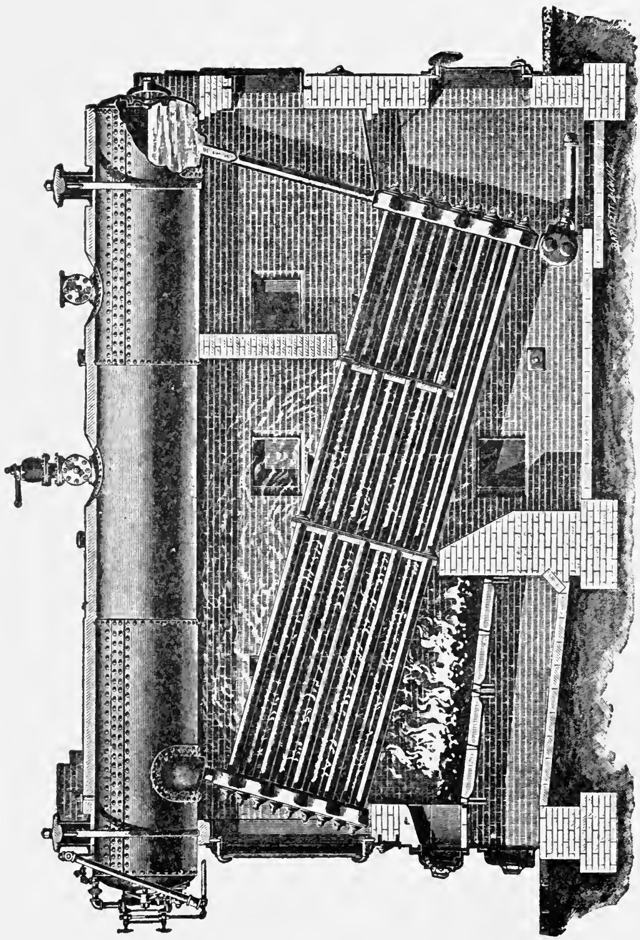


Fig. 78.
Babcock and Wilcox Boiler.

the boiler. Therefore, let D denote the diameter, in feet, of the shell of the boiler; l, the length in feet; d, the *outside* diameter, in inches, of each tube; and N, the number of tubes. Then the heating surface of the shell will be, $3.1416 \times \frac{2}{3} Dl$; and that

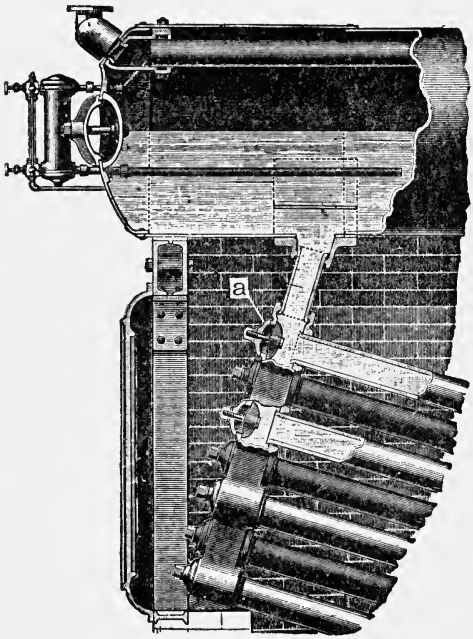


Fig. 79.

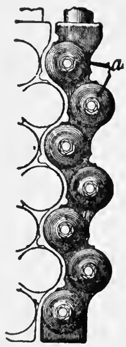


Fig. 80.

of the tubes will be $\frac{3.1416\ dlN}{12}$. The total heating surface, S, will be

$$S = 2.1\ Dl + 0.262\ dlN. \tag{83}$$

The tubes used in return tubular boilers are sold according to their external diameter; and the diameters in most

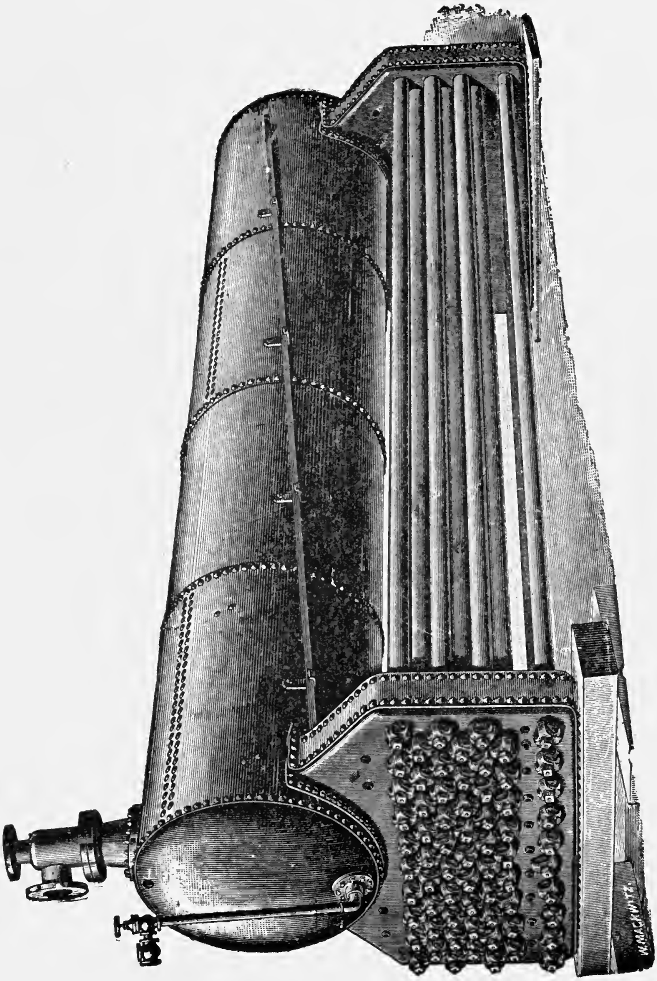


Fig 81. Heine Boiler.

common use are 3, $3\frac{1}{2}$, and 4 inches. The diameter of the tube used in a boiler is somewhat limited by its length, as the length should not exceed sixty times the diameter.

The diameters of the shells of return tubular boilers vary from about 30 inches to 84 inches; and the lengths of the shells will vary from about 6 feet to 20 feet, although the usual lengths are from 12 to 16 feet.

Boilers whose diameters do not exceed 72 inches, and whose lengths are not greater than 18 feet, may be obtained with the bottom of the shell made of a single sheet of iron or steel, and the top part made of one or more sheets. The advantage of a single sheet on the bottom of an externally fired boiler, is that there are then no rivets to come in contact with the hot gases.

The ends of tubes less than six inches in diameter are fastened in the heads of the boilers by "expanding" them, as shown in Fig. 77, and those six inches, or more, in diameter are riveted to the heads.

79. Water-tube Boilers. — Boilers of this type are not as much used as those of the return fire-tube type, but they are becoming more and more extensively used every day. The first cost of these boilers is usually much greater than that of boilers of other types. The heating surface of these boilers is made up of such a variety of surfaces that it is impossible to give any general rule for determining it, that will be applicable to all kinds of water-tube boilers. It must suffice to say that the heating surface is the total surface of those parts of the boiler in contact with the hot gases.

It is impossible to illustrate and explain all the different varieties of water-tube boilers, but they may be distinguished into three classes, each of which may be illustrated.

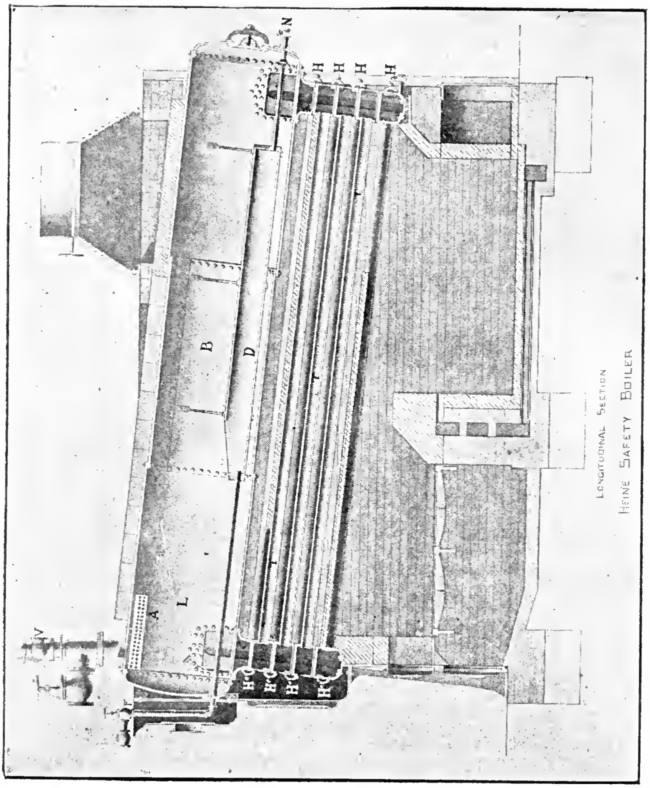


Fig. 82.

In the first class are included those water-tube boilers that have a number of tubes fastened together into sets by common headers or legs, and these common headers connected to a drum containing water and steam. The form of the headers, connecting the ends of the tubes, depends upon the make of the boiler and the number of tubes connected into one set. From all boilers of this class, it is possible to remove one set of tubes without in any way injuring the other sets. It is probable that the greater number of all the water-tube boilers on the market belong to this class.

As an example of this class of water-tube boilers, the Babcock and Wilcox boiler is shown.

Fig. 78 shows a section of this boiler; Fig. 79 shows an enlarged view of the connection of the tubes to the headers, and of the connection of the headers to the water and steam drum. In Fig. 80 is shown a front view of one of the headers. It will be seen that each header, both at the front and the rear, is entirely independent of the others, and that by taking off the hand-hole covers, *a* in Figs. 79 and 80, one of which is placed on the headers opposite the end of each tube, the tubes may be examined, or taken out if desired.

In some forms of water-tube boilers, of this class, the headers are so arranged that the water from the lower tubes must flow through the headers of the upper tubes before it can enter the water and steam drum.

The Heine boiler will serve to illustrate what might be termed the second class. In this class would be included all those water-tube boilers that have the ends of all the tubes fastened into one common header at each end of the boiler.

Fig. 81 shows a perspective of a Heine boiler ready for shipment; Fig. 82 shows a section; and Fig. 83 shows the method of fastening the tubes to the header, and the details of the hand holes, one of which is placed opposite the end of each tube. In Fig. 83, *T* represents the ends of the tubes fastened to the header; *O*, hollow stay-bolts bracing the walls of the header; and *J*, *h*, *H*, and *b*, parts of the hand-hole cover.

In the third class of water-tube boilers, would be included those that have two or more large drums connected by nearly vertical water-tubes.

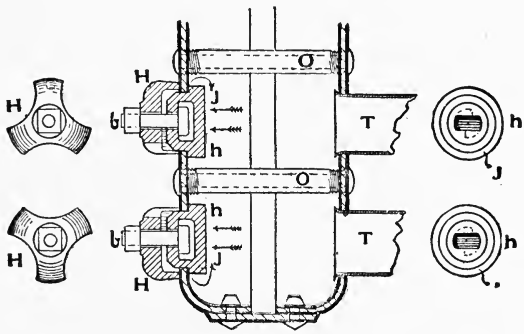


Fig. 83.

This class is illustrated by the Stirling boiler; a section of which is shown in Fig. 84, and a half-front elevation in Fig. 85.

80. Vertical Boilers.— Until of late years almost all vertical boilers made were of small size, but now large ones are being made and used. These boilers are usually what might be termed internally fired; and they are liked on account of the small floor area occupied by them.

In Fig. 86 is shown a half-section and half-elevation of a small vertical boiler, such as is in common use in this country.

The heating surface in these boilers consists of the sur-

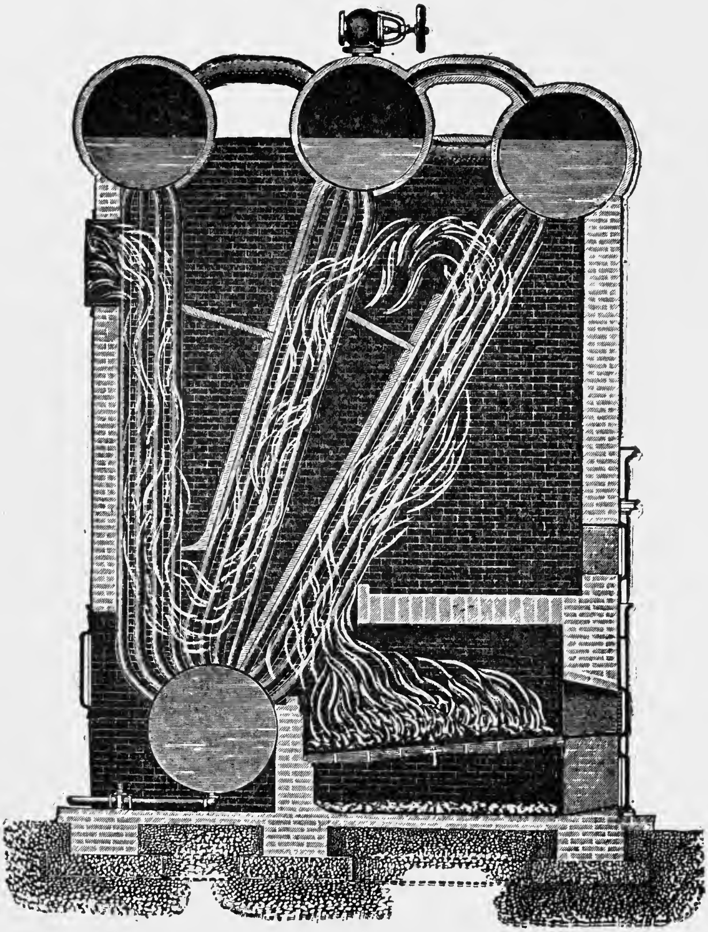


Fig. 84.
Stirling Boiler.

face of the furnace and of the outside surface of the tubes through which the gases pass. The tubes of these small, vertical boilers are usually of 2 or 2½ inches outside diameter.

81. Marine Boilers.— While any boiler used on a vessel is, or ought to be, called a "marine boiler," custom has generally confined the name to boilers similar to that show in section in Fig. 87, and in half-elevation and half-section in Fig. 88.

These boilers are always internally fired; they make steam rapidly and occupy but a small amount of space. The tubes used in marine boilers are usually about 2½ inches in external diameter.

82. Rating of Boilers.— Most boilers are usually rated as being of a given number of horse-power. By the term horse-power, when applied to a boiler, is meant the horse-power of the engine to which the boiler is capable of supplying steam. Of course, it is at once evident that the power of a boiler will vary between very wide limits, depending upon the efficiency in the use of steam of the engine to which it is attached.

In order that there might be some uniformity in the rating of the boilers tested at the Centennial Exposition, at Philadelphia, in 1876, the judges decided that one boiler horse-power should mean thirty pounds of water per hour, evaporated from an initial temperature of 100° F., under a boiler pressure of seventy pounds by the gauge. This is equal to 34½ lbs., per hour, of equivalent water from and at 212°. This standard of rating has become almost universally adopted, and one horse-power for a boiler may be considered as the evaporation of 34½ pounds of water per hour from and at 212°, or its equivalent.*

* This has been adopted as the standard boiler horse-power by the American Society of Mechanical Engineers.

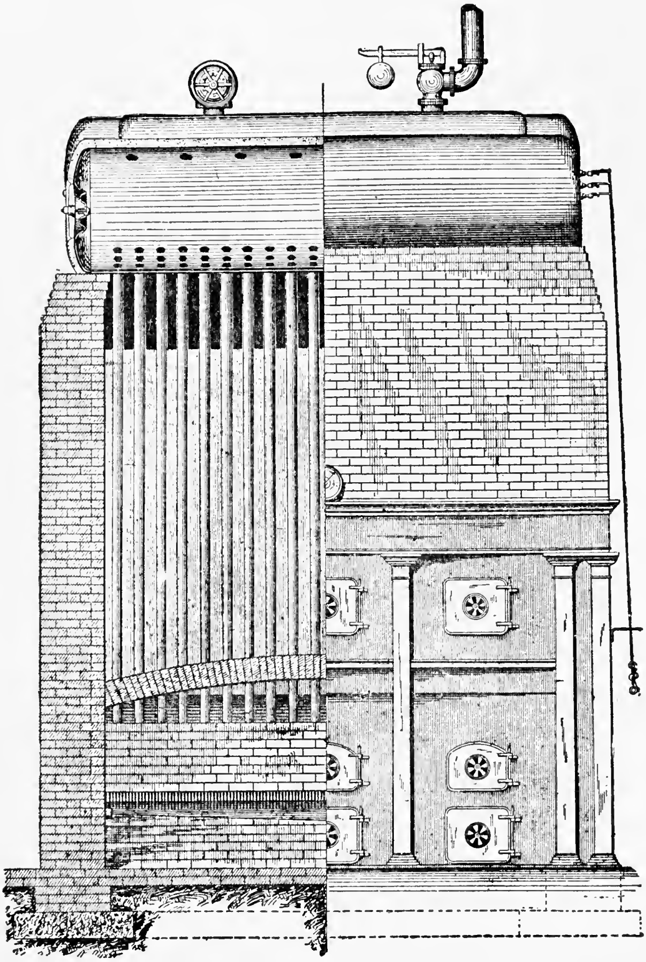


Fig. 85
Stirling Boiler.

When, however, boilers are sold and no test is made to determine the amount of steam they will make, they cannot be rated according to the standard of $34\frac{1}{2}$ pounds of water per hour from and at 212°; and manufacturers usually rate them according to the number of square feet of heating surface. There is no uniformity among manufacturers as to the number of square feet of heating surface that shall be necessary for one horse-power, nor is any distinction made as to the difference in efficiency of the different parts of the heating surface. Some manufacturers of return tubular boilers rate their boilers on the basis of $12\frac{1}{2}$ square feet of heating surface per horse-power, while others rate their boilers on a basis of 15 square feet of heating surface per horse-power.

Manufacturers of water-tube boilers usually rate their boilers on a basis of 10 or 11 square feet of heating surface per horse-power.

Vertical boilers are usually rated upon a basis of 12 square feet of heating surface per horse-power.

When comparing the prices asked by different manufacturers for boilers of the same rated horse-power, it is necessary to compare carefully the areas of the heating surfaces, in order to determine whether or not the boilers are rated on the same basis.

83. Appendages to a Boiler.—Under this head are included pressure gauges, water gauges, gauge cocks, safety valves, feed-water heaters, and other small parts of a boiler that need some short description.

Pressure Gauges.— The most common form of pressure gauge is shown in Figs. 89 and 90. In Fig. 90 the pressure gauge is indicated by *a*. It has a dial face graduated to show pressure in pounds per square inch *above* the atmosphere, so that when the pressure in the boiler is simply that of the atmosphere the gauge will indicate zero pounds. Fig. 89 shows a pressure gauge with the dial

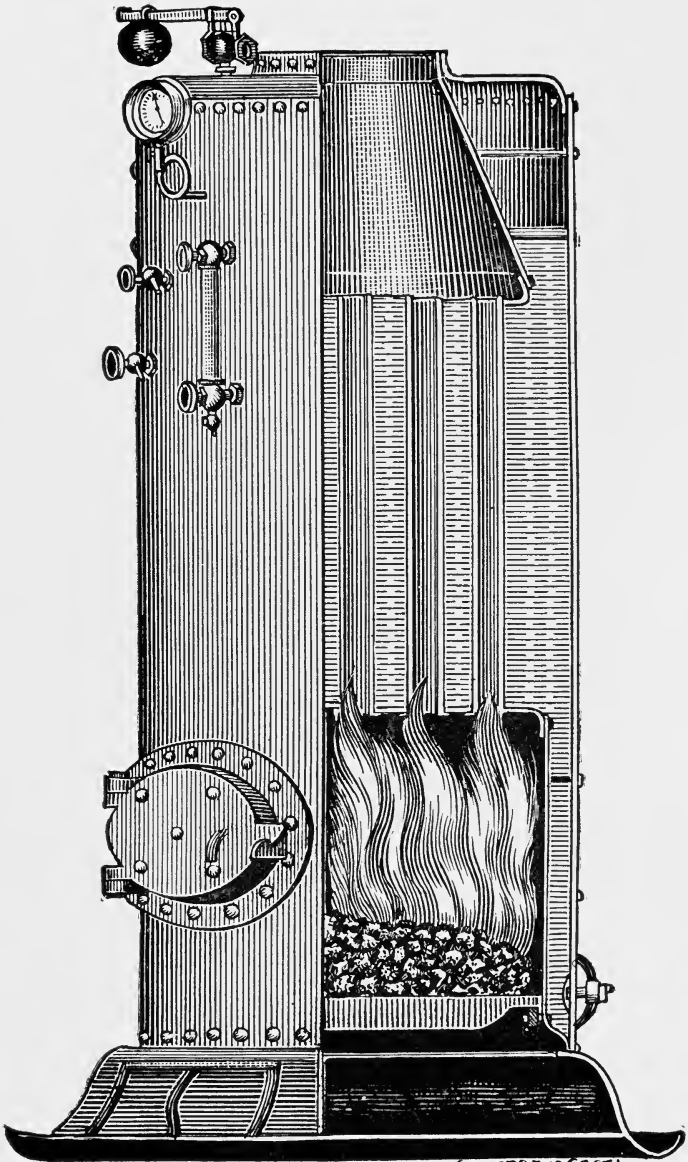


Fig. 86. Vertical Boiler.

face removed, so that the inside mechanism can be seen. The steam enters, through *a*, the flexible, bent tube *b*, shown in section at *e*, and by its pressure tends to straighten the tube. As the tube straightens, it moves the arc *c*; which in turn moves the hand *d* by means of a

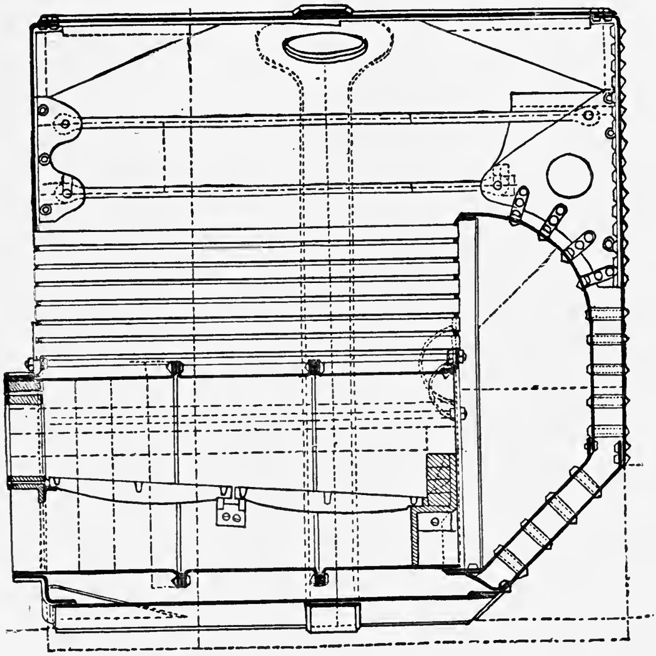


Fig. 87.
Marine Boiler.

small pinion fastened to the axis of *d* and gearing with *c*. By properly adjusting the position the hand *d*, and the strength of the tube *b*, the gauge may be made to correctly indicate pressures.

Syphon.— The syphon is simply a bent piece of $\frac{1}{4}$ inch pipe, shown at *b* in Fig. 90, to which the gauge is always fastened. The syphon is used in order to keep the tube of the pressure gauge filled with water, and thus

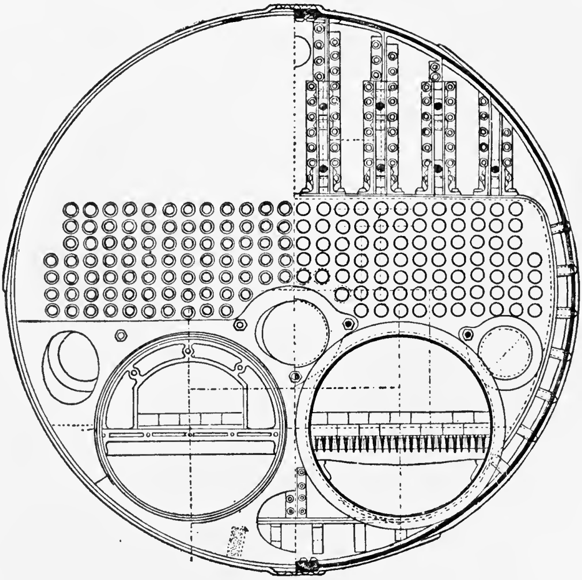


Fig. 88.

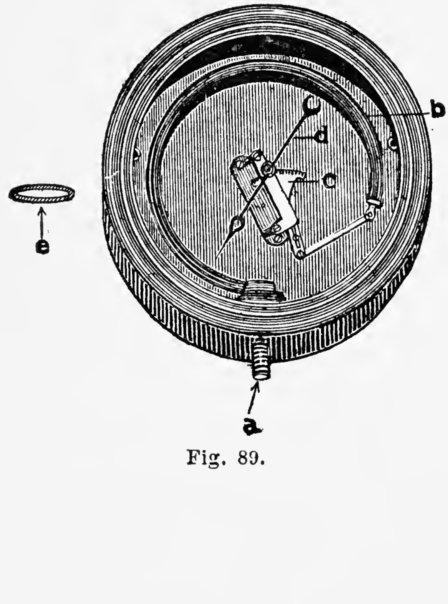


Fig. 89.

prevent the very hot steam from coming in direct contact with it.

Water Column.— The water column is a casting to which is fastened a number of the small appendages to a boiler. In Fig. 90, *A* indicates the water column. To it is fastened the pressure gauge *a*, the gauge cocks *c*, and the water gauge *d*. The upper part of *A* is connected to the *steam space* of the boiler, and the lower part to the *water space;* so that the water stands at the same level in the water column that it does in the boiler.

Gauge Cocks.— The gauge cocks *c*, in Fig. 90, are for determining the position of the water line in the boiler. There are usually three gauge cocks, about three inches apart. The water line should be just about the middle cock, so that if the upper cock is opened, steam will escape; if the middle cock is opened, a mixture of steam and water will escape; and if the lower cock is opened, water will escape. Boilers are sometimes provided with two sets of gauge cocks, one fastened to the water column and one fastened directly to the shell of the boiler.

Water Gauge.— The water gauge or, as it is sometimes called, the water glass, is indicated by *d*, in Fig. 90. It is a glass tube communicating with the steam space of *A* at the top, and with the water space at the bottom. If the tube is open and not choked at any point, the level of the water in the water gauge will be the same as that of the water in the boiler; so that the water in the gauge will enable one to see at a glance where the level of the water stands in the boiler. The glass tube has, usually, about twelve inches exposed to view; and the middle gauge cock is at about the middle of the glass tube. The lower end of the glass tube is about on the level of the tops of the tubes of the boiler, in the case of a return fire-tube boiler.

Safety Valve.— A safety valve is a valve which, when

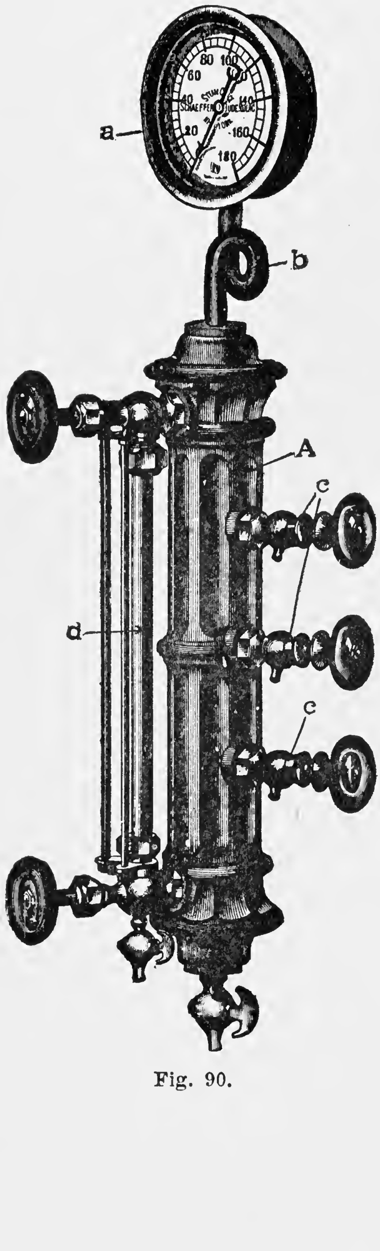


Fig. 90.

the pressure of the steam becomes equal to a given amount depending upon the setting of the valve, is opened by the pressure of the steam, and allows some of the steam to "blow off" into the atmosphere; it thus prevents the pressure in the boiler from becoming too great. There should be at least one safety valve on every boiler, and it is desirable to have two safety valves to every boiler, in order that if, for any reason, one of the valves should fail to act the other would act.

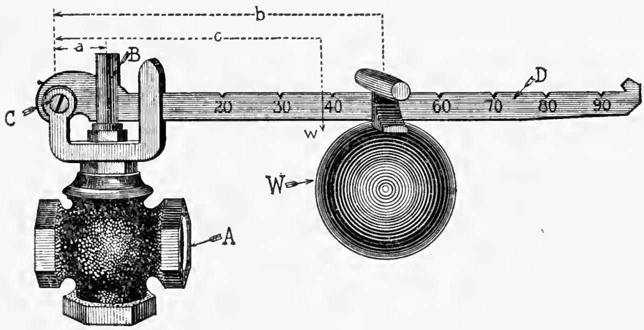


Fig. 91.

There are two kinds of safety valves in common use on boilers generating steam for steam engines, the lever safety valve and the pop safety valve.

In Fig. 91 is shown a lever safety valve. It consists of a valve, in the body *A*, to which is a spindle *B*, that presses against a lever, *D*. The lever is free to swing about the pin *C*, and carries a poise, *W*, whose position on the lever may be changed at will. The steam presses against the bottom of the valve, in the body *A*, and tends to force the spindle, *B*, upwards. In order that the spindle may rise and allow the valve to open, the lever must be moved about the pin, *C*, as a center; but the weight of the lever, *D*, and the poise, *W*, tends to keep the lever from moving. It is evident that the valve will not open until the moment of the force acting on the valve becomes equal to the sum of the moments of the weight of the lever and the weight of the poise.

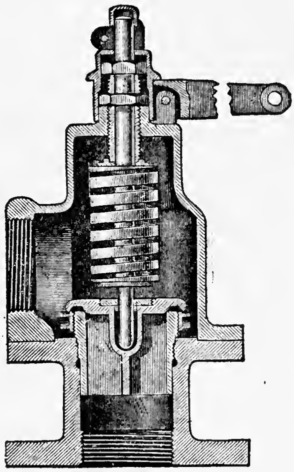


Fig. 92.

Let, in Fig. 91, *V* be the area of the valve in square inches; *P*, the pressure of the steam, by the gauge, at which the valve will blow off; *a*, the distance, in inches, from the center of *C* to the center of the spindle *B*; *b*, the distance, in inches, to the center of the poise; *c*, the distance, in inches, to the center of gravity of the lever; *w*, the weight, in pounds, of the lever; *W*, the weight, in pounds, of the poise; and *m*, the weight, in pounds, of the valve.

Then, it is evident from Fig. 91, that

$$PVa = cw + bW + ma \tag{84}$$

From (84) we get

$$P = \frac{cw}{Va} + \frac{bW}{Va} + \frac{m}{V}. \tag{85}$$

In a given safety valve, the only thing that can be changed is the distance, b, that the poise is from the center of the pin C; and the greater b is made, the higher will be the pressure at which the valve will blow off.

A "pop" safety valve is a safety valve in which the valve is held down on its seat by a spring, instead of a lever and poise. In Fig. 92 is shown a section of a "pop" safety valve such as is manufactured by the Consolidated Safety Valve Co. The valve is set to blow off at different pressures by adjusting the tension of the spring by means of the nuts at the top.

The only objection to a pop safety-valve is the noise it makes when it opens.

Feed-Water Heater. — We have already shown how the amount of heat required to evaporate a pound of water is reduced by increasing the temperature of the feed water before it enters the boiler. Whenever it is possible, the exhaust steam of a non-condensing engine should be used to heat the feed-water, instead of being allowed to pass off into the atmosphere. The apparatus in which the feed-water is heated before entering the boiler is termed a "feed-water heater."

In Fig. 93 is shown a National feed-water heater. As seen in the cut, the feed-water is made to pass through a coil of pipe, before entering the boiler, that is surrounded by the exhaust steam.

Sometimes a feed-water heater is made to serve the double purpose of heating the feed-water, and of catching all the sediment and impurities that would otherwise be deposited in the boiler, when the water has been heated as hot as it becomes in the heater.

Feed Pipe. — The feed pipe is the pipe through which the water enters the boiler. It is often attached to the shell of the boiler near the back, or to the bottom of the front end. Neither of these positions, however, is to be recommended, as in either case the comparatively cool feed-water impinges upon the hottest part of the shell.

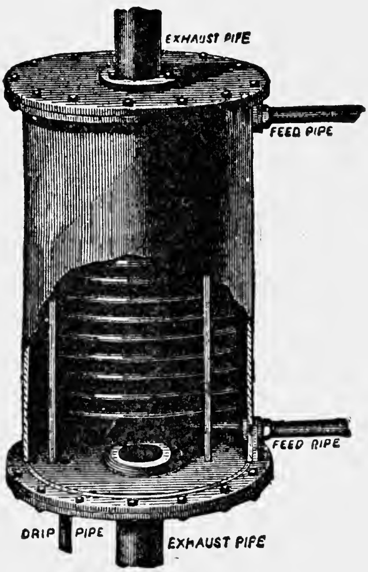


Fig. 93.

It is better that the feed pipe should be attached to the front end of the boiler a few inches below the water line, and be carried back into the boiler; and the water should be allowed to escape from the pipe through small perforations in it, rather than through the open end.

Blow-off Pipe.— Every boiler should have a blow-off pipe, by means of which it may be emptied of the water it contains.

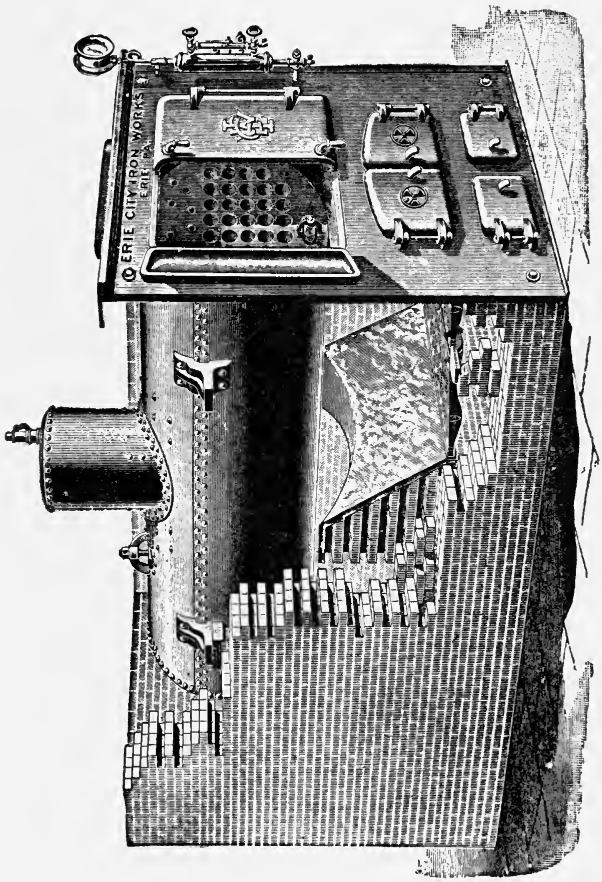


Fig. 94.
Return Fire-tube Boiler.

Valves.— The steam pipe, for taking steam from the boiler to the engine, should have a valve close to the boiler.

The feed pipe should have two gate valves, close to the boiler, with a check valve between them.

The blow-off pipe should not have a valve on it, but should have a good plug-cock.

Feeding Apparatus.— The feed-water is usually forced into the boiler by means of a pump, called a feed-water pump, although the injector is often used.

84. Settings of Boilers.— The setting of a boiler means the general arrangement of furnace, boiler, and chimney relative to one another, and the manner in which the furnace and boiler are inclosed and built in. Of course, the setting will depend largely upon the type and construction of the boiler, but for the ordinary return fire-tube boiler there are two recognized standard settings, viz., the full-arch front setting, and the half-arch front setting.

Fig, 76 shows a perspective view of a return fire-tube boiler with a half-arch front setting; and Fig. 94 shows one with a full-arch front setting.

When two or more boilers are set side by side, with common front and rear walls, they form what is termed a "battery" of boilers. All the boilers of a battery may, or may not be connected to the same chimney.

The setting for return fire-tube boilers, recommended by the Hartford Steam Boiler Inspection and Insurance Company, described in *The Locomotive* for February, 1895, is shown in Figs. 95, 96 and 97.

In this setting the furnace is supposed to be lined with fire-bricks, and the walls are made very thick. The company, in its description of the setting, says: —

"The width of the furnace in the settings advocated by this company is six inches less than the diameter of the boiler. Beginning just above the grate, the side walls batter at such angle as to make them 3" clear of the boiler at the center, where the walls project inward and close against the boiler. This batter gives greater stability to the walls, and another special feature of it is, that it allows the heated gases to rise without impinging against the walls of the setting, and they flow away from

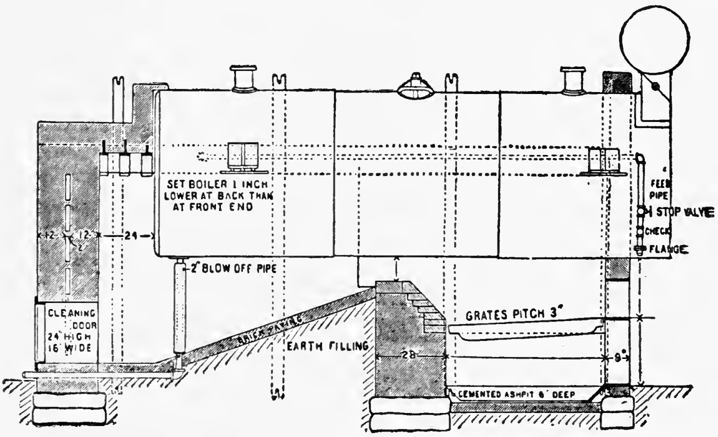


Fig. 95.

the wall and distribute themselves evenly over the whole heating surface of the shell. The removal of soot and ash from the shells is also facilitated, and, moreover, it is found that these deposits do not form so readily when the walls are battered as they do when the walls are straight, and the space between them is correspondingly contracted. The batter also increases the volume of the combustion chamber, and allows of a more thorough mixing of the oxygen and furnace gases, the result being that complete combustion of the fuel is greatly facilitated. The bridge-wall slopes back from about four inches above the grate, at an angle of 40°, in order that the radiant heat from the fire may be diffused over a large portion of the boiler shell. The flame bed back of the bridge-wall slopes down to the level of the boiler-room floor. It is paved for easy cleaning, and the combustion chamber is large enough to make examinations and repairs to the boiler comparatively easy. The cleaning door in the rear wall is placed on a level with the flame bed in order that ashes may be readily removed, and as it is

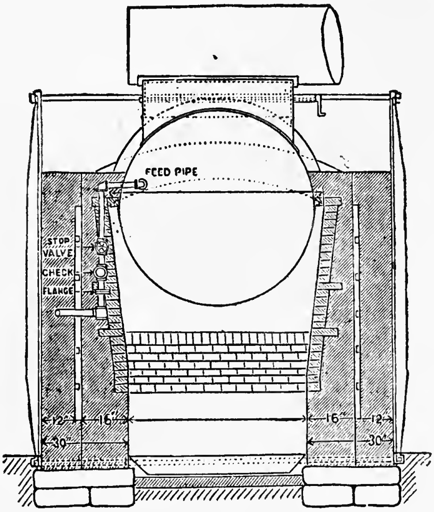


Fig. 96.

below the currents of highly-heated gases loss by radiation through the door is largely prevented. The loss or waste of heat from this cause is often very great and it has not generally received the attention it deserves. Another point that demands more attention than it usually receives, is the liability of *leakage* of cold air through the walls of the setting, with the resulting reduction of furnace temperature. To avoid loss of temperature from this cause heavy double walls are constructed in this company's settings, the outside walls of a battery having a two-inch air space between them. The division walls between two or more boilers should have a half-inch clear space between them, to allow free and independent expansion of the walls. With a solid wall and one or more boilers of the battery stopped, one side of the wall separating a boiler in use from another one out of use would be hot and greatly expanded, while the other side of it would be cool; the result being that the bonded or

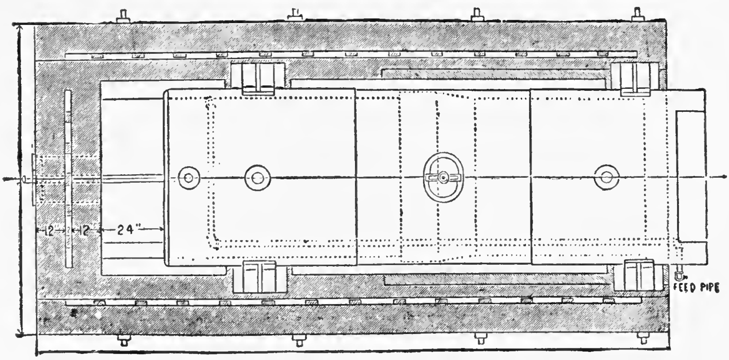


Fig. 97.

solid wall must necessarily be severely strained or injured, and the joints in the masonry quite probably broken by the unequal expansion. Excessive leakage of air is likely to follow. These criticisms apply to all solid-built boiler settings. While the heavy double walls are somewhat more expensive in first cost, the increased economy and capacity of the boilers, as well as the greater durability of the settings, fully warrant their construction. The results obtained in many large plants fully sustain this statement.

The exposed portions of the boiler shells above the settings are covered with plastic non-conducting covering $2\frac{1}{2}''$ thick. This is much lighter than brick, is a better non-conductor, and does not exert a sensible thrust upon the setting walls as a brick arch does. If leaks occur along the joints of the covered part of the boiler, they are quickly noted by the discoloration of the covering, and may be stopped before the injury from corrosion occurs. The illustrations give the general arrangement of the settings above described, in which it is desired to combine durability with simplicity in design and construction, and at the same time to obtain good results from the boilers, both in economy and in capacity."

CHAPTER X.

CHIMNEYS.

85. Chimneys.—Chimneys are to carry the products of combustion away from the boiler, and, by so doing, produce a draft that will cause fresh air to enter the furnace and carry with it the oxygen to be used

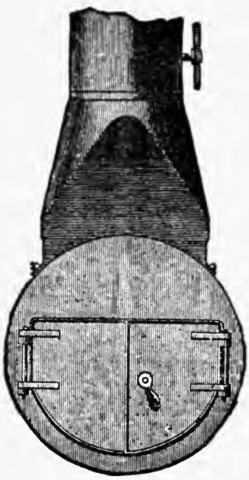


Fig. 98.

in combustion. They are made either of brick or metal; and usually have either an octagonal or circular cross-section. A circular, inside, cross-section is better than either a square or an octagonal cross-section, as it offers less resistance to the flow of the gases. A square, inside, cross-section is, really, equivalent only to a circular cross-section whose diameter is equal to that of a circle inscribed in the square; this is so as the corners of square chimneys become almost dead spaces, on account of the excessive resistance there to the flow of the gases.

The passage through which the gases pass, after leaving the boiler or battery of boilers, on their way to the

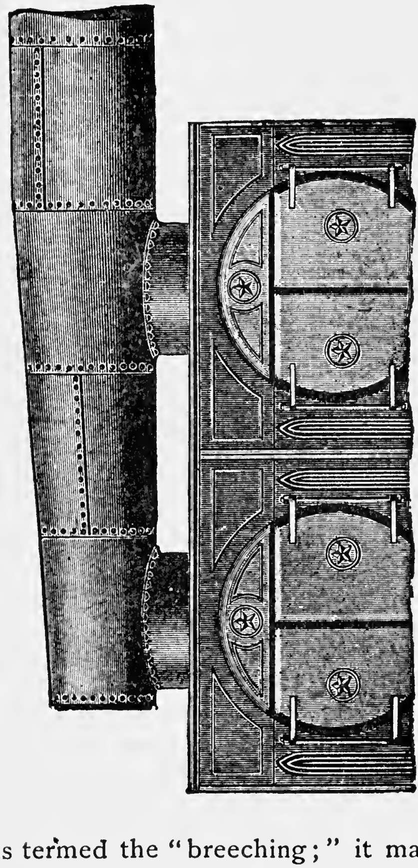


Fig. 99.

chimney, is termed the "breeching;" it may be large or small, long or short, depending upon the number of boilers connected to it and the distance from the boilers to the chimney.

In the case of a single boiler, or a small battery of boilers, the chimney is usually made of No. 16 sheet iron, and is carried directly by the breeching. Fig. 98 shows the breeching of a sheet iron stack for a single boiler with a half-arch front setting; and Fig. 99 shows the breeching for a battery of two boilers, with a full-arch front setting.

In the case of a large battery of boilers, a number of small, sheet iron chimneys, to each of which will be connected two or three boilers, may be used, or all the boilers may be connected to a single large chimney.

Brick chimneys, usually, have two walls with an air space between them. The inner wall may extend up the whole height of the chimney or only a part of the way to the top. The outer wall is for stability, and forms the body of the chimney; while the inner wall is simply a lining to prevent the hot gases from coming in contact with the outer wall, and it should be made entirely of fire brick laid in clay, or, at least, should be lined with fire brick. This lining is necessary, as ordinary brick-work will not stand the heat of the hot gases without deteriorating very much.

Brick chimneys are very much used, although they are expensive, and are apt to open at the joints and let cold air leak into the inside.

In Fig. 100 is shown a section of a large brick chimney

Iron chimneys, of large size, made of thick sheet iron, are becoming more and more extensively used every day. They are usually cheaper than brick chimneys, and are perfectly air tight. They may or may not be lined with fire brick, although it is preferable to have them lined.

In Fig. 101 is shown an elevation and section of a steel plate chimney, such as is made by the Philadelphia Engineering Works, Philadelphia, Pa.

86. Draft of Chimney.— By the draft of a chimney is meant the difference in pressure of the gases in the chim-

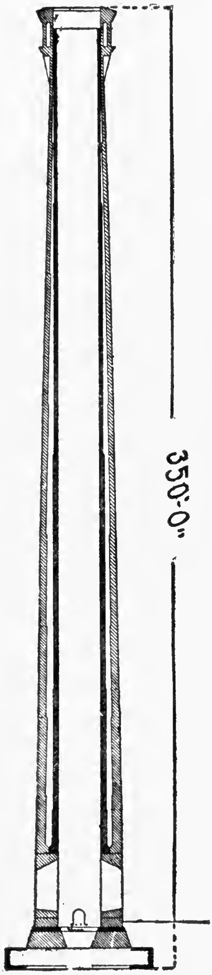


Fig. 100.

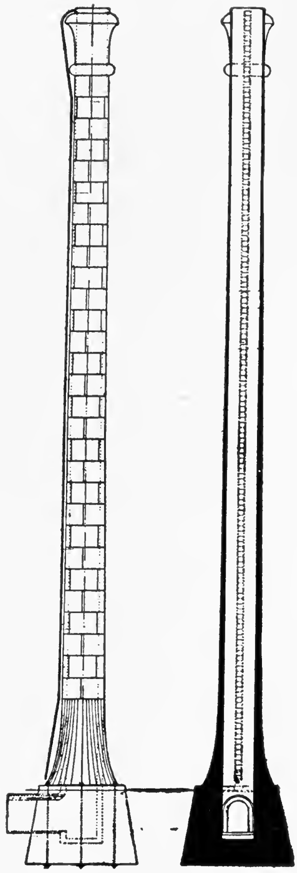


Fig. 101.

ney and that of the air on the outside, measured at, or near, the base of the chimney. It is this difference in pressure, or draft, that makes the air flow into the furnace and force the gases out through the top of the chimney. When the draft is due to the difference in temperatures of the gases in the chimney and the air outside, and to the height of the chimney, it is termed a natural draft; but when the pressure forcing the air into the furnace is that due to a fan or blower, the draft is termed a forced draft, since it is usually much greater than the ordinary, natural, draft. Of course, there is no sharp line of demarkation between natural and forced drafts; as a natural draft may be very high, and a forced draft may be very low.

The draft is usually spoken of as being of so many "inches of water." This method of expressing the draft gives the head of water, in inches, that is equivalent to the difference between the pressure of the air and that of the gases inside of the chimney. The number of inches of draft is measured by means of a U-tube, shown in Fig. 102. The legs of the tube are first filled about half full with water; then, one end of the tube is inserted through a hole in a piece of cork that fits tightly into an opening in either the breeching, near the chimney, or the base of the chimney itself; the other end of the tube is left open to the air. The water will stand higher in the leg in communication with the hot gases than in the one in communication with the air; and the distance, in inches, that the surface of the water in the one leg is above the surface of the water in the other leg is the draft in inches of water.

Except in the case of very high chimneys, the draft of furnaces having natural draft will seldom exceed three-fourths of an inch, and is ordinarily about one-half an inch. The draft in furnaces using forced draft is only limited by the ability of the fan or blower to create it.

87. Velocity of the Gases Passing Through the Chimney.— The velocity of the flow of the gases through the smallest cross-section of the chimney is determined by the law of the flow of gases under a small pressure. We know, from physics, that for small pressures, the velocity, in feet per second, with which a gas will flow

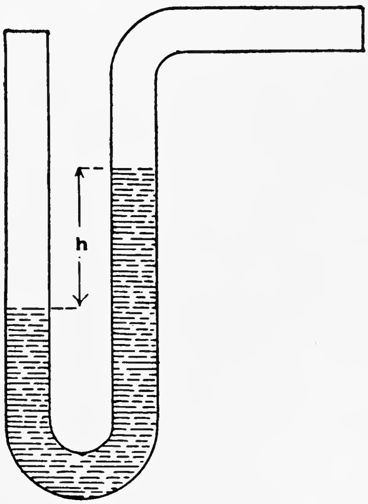


Fig. 102.

from a vessel in which the unbalanced pressure per square foot, p, is that equivalent to a head, h, of the gas, is

(86) $$v = \sqrt{2gh}$$

v is the velocity of flow, in feet per second, of the gas. g is equal to the constant 32. h is the head, in feet of gas, equivalent to the pressure, p, per square foot, that causes the gas to flow.

If the gas has a density, or weight per cubic foot, of D, then $hD = p$, and $h = \frac{p}{D}$.

In the case of a chimney, the pressure causing the gas to flow is equal to the difference between the pressures inside and outside of the chimney.

Let Fig. 103 represent a chimney, whose height in feet is H, with an opening at the bottom. Also, let P_1 be the pressure, per square foot, of the gases inside the chimney; P_0, the pressure, per square foot, of the air outside the chimney; D_1, the density of the gases inside the chimney; D_0, the density of the air outside; and P, the pressure, per square foot, of the air at the top of the chimney.

Then, evidently, $P_0 = P + HD_0$; $P_1 = P + HD_1$; and the pressure that forces air into the opening, and the gases out of the chimney, is

$$P_0 - P_1 = H\,(D_0 - D_1).$$

The head, h, in feet of hot gas equivalent to the pressure $P_0 - P_1$ is, from what has been said before, equal to the pressure divided by the density of the hot gas. Therefore,

$$(87) \qquad h = \frac{P_0 - P_1}{D_1} = H\left(\frac{D_0}{D_1} - 1\right).$$

From (86) we know that the velocity with which the hot gas will tend to flow, when under a pressure equivalent to a head h, is

$$(88) \qquad V = \sqrt{2gh} = 8\sqrt{H\left(\frac{D_0}{D_1} - 1\right)}.$$

Experience has shown, however, that the velocity of the gases in a chimney is reduced by friction, until it is only from one-third to one-half what it would be if there were no friction. Therefore, if we let k represent a factor, varying between one-third and one-half, by which the theoretical expression in (88) must be multiplied in order to obtain the actual value, u, of the velocity of the flow of the gases, we have

$$(89) \qquad u = kv = 8k\sqrt{H\left(\frac{D_0}{D_1} - 1\right)}.$$

The density of air at 32° F., or 493° absolute, is 0.08, and it is sufficiently accurate for us to assume, as is almost true, that the density of the gases in the chimney is, also, 0.08 at 32° F.

Now, if the absolute temperature of the air outside of the chimney is T_0, and that of the gases inside is T_1 we

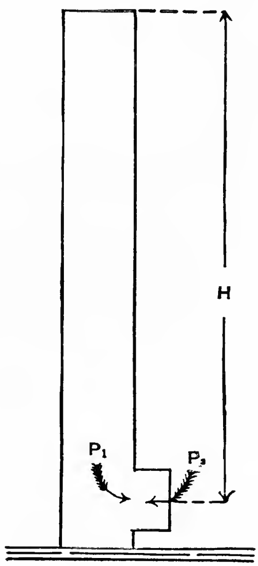


Fig. 103.

know, from what has been said in Chapter I, that, since the density of a gas is inversely as its volume,

$$D_0 = \frac{1}{V_0} = \frac{0.08 \times 493}{T_0}, \text{ and}$$

$$D_1 = \frac{1}{V_1} = \frac{0.08 \times 493}{T_1}.$$

Therefore,

$$\frac{D_0}{D_1} = \frac{T_1}{T_0};$$

and the expression for u, as given in (89), becomes

$$(90) \qquad u = 8k \sqrt{H \left(\frac{T_1}{T_0} - 1 \right)}.$$

The temperature of the gases in chimneys is, ordinarily, between 400° F. and 550° F.; so that the value of T_1 will be between 861 and 1011. The temperature of the outside air varies with the locality and the seasons of the year, but it may be assumed as 60° F., or 521° absolute. Therefore, the value of $\frac{T_1}{T_0}$ may be taken as varying from 1.6 to 2.

As the density of the air varies greatly from time to time, depending upon the amount of moisture in the air, and as the density of the gases inside of the chimney also varies greatly, the value of u can never be very accurately obtained by an equation. The result obtained by the use of (90) is apt to differ more or less from the true value of u, because in (90) it has been assumed that the temperature of the gases is the same at all parts of the chimney, whereas it really becomes less the nearer we approach the top.

APPENDIX.

CARE OF BOILERS.

As it is very important that everbody having anything to do with the operation of a boiler plant should know how to care for the boilers, there is inserted, here, the rules to be observed in order to prevent accidents, to economize fuel, and to preserve the boiler, that are given by the Fidelity and Casualty Company in its little book, The Engineer's Manual.

How to Prevent Accidents.

1. Safety Valves.— These should be of ample size and kept in working order. The valve should be tried daily; this is best done by allowing the pressure to rise gradually until the valve just "simmers," noting the pressure by the steam gauge at the moment. Freedom of action may of course be ascertained by hand, but it cannot be known by this means that the valve will blow off when the proper pressure is attained. Neglect and overloading of this most important adjunct are prolific causes of boiler explosions. Each boiler should have its own safety valve, and no stop valve should be permitted between it and the boiler. See cut "A" (not given here). This illustrates the worst combination of safety and stop valves that could well be contrived.

2. Pressure Gauge.— It is absolutely necessary that the pressure gauge should be trustworthy, and if there is any reason to question its readings, it should be compared with one known to be accurate. The gauge should be

fitted to a "loop" filled with water, which transmits the pressure and prevents contact of steam with the gauge spring. Attach the gauge directly to the boiler and not to the steam pipe, to prevent fluctuations of pressure readings.

3. Water Level. — Before starting, make sure that there is plenty of water in the boiler by trying the gauge cocks. While running do not depend on the gauge glass, but try the gauge cocks often. The water line should be kept at a regular height, and should never be less than three or four inches above the "fire line." The gauge glass should be blown out frequently to see that it is not choked; it is an excellent plan to try the gauge cocks every fifteen minutes. Both gauge and cocks must be kept clean.

4. Damper.— Do not close the damper entirely while there is fire on the grates, as gas may collect in the tubes and cause an explosion.

5. Feed Pump or Injector.— These should be kept in order, and should be of ample size for all requirements. The feed pump, however, ought not to be so large as to render it difficult to feed the boiler continuously at a slow rate of speed. It is always safer to have two means of feeding. An injector should be used when no feed-water heater is provided, as it prevents the contraction of tubes and plates where the feed water comes in contact with them.

6. Low Water.— The blow-out apparatus should be kept tight, as any leakage here may give rise to low water, with the result of overheating the plates. In case of low water, fresh coal, or better still, wetted ashes, must be thrown on the fire at once. Do not turn on the feed,

though if already in motion, allow it to continue, nor start or stop the engine, or lift the safety-valve until the boiler has cooled down. After a case of low water the tube ends in the upper rows should be examined for leaks.

7. Incrustation, Corrosion.— Boilers should be kept free from scale, as its presence increases the liability of burning or cracking the plates and predisposes to explosion. The surest method for preventing internal corrosion is to abandon the use of the water which causes it, but if this is impracticable, a sharp lookout should be kept for defects. Leaks of seams and fittings, drippings from pipes, exposure to the weather, contact of the boiler with brick-work, etc., are causes of external corrosion, and should be at once remedied.

8. Blisters, Cracks, and Burnt Plates.— When these occur they should receive attention at once. Burnt places and blisters should be cut out and a patch put on inside the boiler to avoid making a pocket for the collection of sediment.

9. Fusible Plugs.— These are required by law in some States. To keep them in an efficient condition their surfaces, both on the fire and water sides, must be often scraped clean, but notwithstanding all precautions, they are unreliable.

10. Starting the Engine.— The engine should be started slowly, in order not to make a violent change in the condition of the water and steam, and when possible, the engine should be stopped gradually. The sudden opening or closing of a large stop-valve may produce a violent rush of steam and water against that part of the boiler whence the steam is drawn, the percussion of which may be sufficient to rupture the boiler.

How to Save Fuel.

1. Firing.— The fire should be kept level and of somewhat greater thickness at the bridge wall. This promotes a uniform consumption of fuel, as the air passes more freely through the fire near the bridge and the greater thickness retards its passage. Fuel supplied regularly in small quantities, combined with an even distribution, produces the best results. When anthracite coal is used, the average thickness of the fire should be 6 to 8 inches; with bituminous coal, it should be 8 to 10 inches; with coke, 10 to 12 inches. If the draft is poor, however, a thin fire must be used. Do not fire with large lumps. No fragment ought to be larger than a man's fist.

Complete combustion is only attained when the fuel is burning with a bright flame all over the grate. Blue flames, dark spots and smoke are evidences of the lack of the necessary air which ought to be supplied above the grate. Fires should be "cleaned" no oftener than necessary. In using a caking coal, it is advantageous to make use of a "coking fire," *i. e.*, firing in front and breaking up with a slice bar, and shoving back when coked. The practice of wetting coal before throwing it on the fire is a bad one, as it wastes heat and produces corrosion.

2. Feed-Water.— Heating the feed-water, either by means of exhaust steam or the waste gases in the chimney, adds to the economy of a steam plant. Each increase in the temperature of the feed-water of 11° F. means a saving of fuel of one per cent. No saving in fuel is effected by the use of an injector, but the employment of one promotes the longevity of a boiler by introducing the feed-water at a temperature so high that no injurious contractions are caused in any part of the boiler.

3. Cleaning.— The heating surfaces of a boiler, both inside and out, should be kept clean, in order to prevent a serious waste of fuel. The thickness of the soot or scale which is allowed to accumulate ought never to exceed $\frac{1}{16}$ of an inch.

4. Leaks in Brick-work.— Cracks or openings in the brick-work should be carefully stopped. The admission of air, except at the places provided for it, impairs the draft, cools the gases on their way to the tubes, and sometimes causes jets of flame to impinge so strongly on the shell as to injure the plates.

5. Covering.— Radiation from the dome and the top of the boiler is a source of waste. A covering of asbestos or other suitable non-conducting material should be provided as a protection.

6. Blowing Out.— The bottom blow-out cock should be kept tight to prevent loss by leakage. A plug cock is the simplest, surest and most durable valve for this purpose. When the feed water is of a hard or muddy nature, the boiler should be blown out frequently. A boiler should be emptied every week or two, and filled afresh. The proper manner to use a surface blow-off is to open it for about fifteen seconds every hour rather than for a longer time at greater intervals.

How to Lengthen the Life of the Boiler.

1. Banking Fires.— Contraction and expansion, caused by change of temperature, shorten the life of a boiler. For this reason it is better to bank the fires at night instead of drawing them.

2. Leaks.— Leaks, whether in boiler or fittings, should be repaired at once. Leaks often give rise to corrosion.

3. Filling Up. — Wear and tear of a boiler, arising from unequal expansion and contraction, is increased by allowing the feed-water to enter at too low a temperature. If the use of cold water is unavoidable, the feed-pipe should always be extended into the interior of the boiler. It should enter horizontally through the front head, near one side, and a few inches below the water-line, thence extending back to within a few feet of the back head, crossing over and discharging downward between the tubes and shell. By this means the feed-water is heated nearly to the temperature of water in the boiler, and is discharged at the coolest part of the boiler. The use of an injector or feed-water heater renders this extension of the feed-pipe unnecessary.

4. Blowing Out.— A boiler should never be emptied while the brick-work is hot. When this is done the sediment is baked on the plates, making it difficult to remove.

5. Rapid Firing.— Steam should be raised slowly in a boiler having thick plates or seams exposed to the fire, else overheating or burning results. The greatest effect of a fire on a boiler bottom takes place immediately behind the bridge, and if a seam is located here there is liability of burning the lap. It is best in such cases to change the position of the bridge, so that the seam comes over the bridge, or better still, over the furnace.

6. Moisture. — The exterior of a boiler should be protected from moisture, as it brings about corrosion and consequent weakening of the boiler.

7. Galvanic Action.— Sometimes boilers may be protected from the action of corrosive agents present in the water by means of zinc. As a rule one square inch of surface of zinc to every fifty pounds in the boiler is

sufficient. The plates should be placed in perfect metallic contact with the iron and renewed as they are wasted by oxidation.

8. Disuse of Boiler.— If it is intended not to use the boiler for some time, the boiler should be emptied of its water, dried thoroughly by pans of charcoal, and after placing pans of lime in the interior, closed to prevent oxidation. If this is impracticable, the boiler should be filled with water in which common soda is dissolved.

TABLE I.

PROPERTIES OF STEAM.

Pressure by the Gauge.	Temperature.	Total Heat above 32°.	Latent Heat.	Vol. of one lb. of Steam.
—13	119.	1118.	1031.	223.
—12	137.	1124.	1019.	135.
—11	150.	1128.	1010.	98.9
—10	160.	1131.	1003.	78.3
— 9	168.	1133.	997.	65.0
— 8	175.	1135.	992.	55.9
— 7	181.	1137.	988.	48 9
— 6	187.	1139.	984.	43.6
— 5	191.8	1140.4	980.1	39.31
— 4	197.	1142.	977.	35 8
— 3	201.	1143.	974.	33.3
— 2	205.	1144.	971.	30.6
— 1	208.	1146.	968.	28.4
0	212.0	1146.6	965.7	26.56
1	215.	1148.	964.	25.0
2	219.	1149.	961.	23.6
3	222.	1150.	959.	22.3
4	224.	1150.	957.	21.2
5	227.1	1151.2	955.1	20.16
6	230.	1152.	953.	19.3
7	232.	1153.	952.	18.4
8	235.	1154.	950.	17.7
9	237.	1154.	948.	17.0
10	239.4	1154.9	946.4	16.30
11	242.	1156.	944.	15.7
12	244.	1156.	944.	15.2
13	246.	1157.	942.	14.6
14	248.	1158.	941.	14.2
15	249.7	1158.1	939.3	13.71

Pressure by the Gauge.	Temperature.	Total Heat above 32°.	Latent Heat.	Vol. of one lb. of Steam.
16	252.	1159.	938.	13.3
17	253.	1159.	937.	12.9
18	255.	1160.	935.	12.5
19	257.	1160.	934.	12.2
20	258.7	1160.9	932.7	11.85
21	260.	1161.	932.	11.6
22	262.	1162.	931.	11.3
23	264.	1162.	929.	11.0
24	265.	1163.	928.	10.7
25	266.7	1163.3	927.1	10.36
26	268.	1164.	926.	10.2
27	270.	1164.	925.	9.95
28	271.	1165.	924.	9.75
29	273.	1165.	923.	9.54
30	273.9	1165.5	922.0	9.34
31	275.	1166.	921.	9.16
32	277.	1166.	920.	8.98
33	278.	1167.	919.	8.81
34	279.	1167.	918.	8.63
35	280.5	1167.5	917.3	8.45
36	282.	1168.	917.	8.31
37	283.	1168.	916.	8.16
38	284.	1169.	915.	8.02
39	285.	1169.	914.	7.87
40	286.5	1169.3	913.0	7.73
41	288.	1170.	912.	7.61
42	289.	1170.	911.	7.48
43	290.	1170.	911	7.36
44	291.	1171.	911.	7.23
45	292.2	1171.1	909.0	7.11
46	293.	1171.	908.	7.01
47	294.	1172.	907.	6.91
48	295.	1172.	907.	6.81
49	296.	1172.	906.	6.71
50	297.5	1172.7	905.2	6.61
51	299.	1173.	904.	6.52
52	300.	1173.	904.	6.43

Pressure by the Gauge.	Temperature.	Total Heat above 32°.	Latent Heat.	Vol. of one lb. of Steam.
53	301.	1174.	903.	6.34
54	302.	1174.	902.	6.25
55	302.4	1174.2	901.6	6.16
56	303.	1174.	901.	6.08
57	304.	1175.	900.	6.00
58	305.	1175.	900.	5.93
59	306.	1175.	899.	5.85
60	307.1	1175.6	898.4	5.77
61	308.	1176.	898.	5.70
62	309.	1176.	897.	5.63
63	310.	1176.	897.	5.57
64	311.	1177.	896.	5.50
65	311.5	1176.9	895.1	5.43
66	312.	1177.	895.	5.37
67	313.	1178.	894.	5.31
68	314.	1178.	893.	5.25
69	315.	1178.	893.	5.19
70	315.8	1178.2	892.1	5.13
71	317.	1179.	892.	5.08
72	317.	1179.	891.	5.02
73	318.	1179.	890.	4.97
74	319.	1179.	890.	4.91
75	319.8	1179.4	889.1	4.86
76	321.	1180.	889.	4.81
77	321.	1180.	888.	4.77
78	322.	1180.	887.	4.72
79	323.	1180.	887.	4.68
80	323.7	1180.6	886.3	4 63
81	324.	1181.	886.	4.59
82	325.	1181.	885.	4.54
83	326.	1181.	885.	4.50
84	327.	1182.	884.	4.45
85	327.4	1181.7	883.6	4.41
86	328.	1182.	883.	4.37
87	329.	1182.	883.	4.33
88	330.	1182.	882.	4.28
89	330.	1183.	881.	4.24

Pressure by the Gauge.	Temperature.	Total Heat above 32°.	Latent Heat.	Vol. of one lb. of Steam.
90	330.9	1182.8	881.0	4.20
91	332.	1183.	881.	4.16
92	332.	1183.	880.	4.13
93	333.	1184.	880.	4.09
94	334.	1184.	879.	4.06
95	334.4	1183.9	878.5	4.02
96	335.	1184.	878.	4.00
97	336.	1184.	878.	3.97
98	336.	1185.	877.	3.93
99	337.	1185.	877.	3.90
100	337.6	1184.9	876.0	3.86
101	338.	1185.	876.	3.83
102	339.	1185.	875.	3.80
103	340.	1186.	875.	3.77
104	340.	1186.	874.	3.74
105	340.9	1185.9	873.8	3.71
106	342.	1186.	873.	3.68
107	342.	1186.	873.	3.65
108	343.	1186.	872.	3.63
109	343.	1187.	872.	3.60
110	343.9	1186.8	871.4	3.57
111	345.	1187.	871.	3.55
112	345.	1187.	871.	3.52
113	346.	1187.	870.	3.50
114	346.	1188.	870.	3.47
115	346.9	1187.7	869.3	3.45
116	348.	1188.	869.	3.43
117	348.	1188.	868.	3.40
118	349.	1188.	868.	3.38
119	349.	1189.	868.	3.35
120	349.8	1188.6	867.1	3.33
121	350.	1189.	867.	3.31
122	351.	1189.	866.	3.28
123	352.	1189.	866.	3.26
124	352.	1189.	865.	3.23
125	352.6	1189.5	864.9	3.21
126	353.	1190.	865.	3.19
127	354.	1190.	864.	3.17

Pressure by the Gauge.	Temperature.	Total Heat above 32°.	Latent Heat.	Vol. of one lb. of Steam.
128	354.	1190.	864.	3.14
129	355.	1190.	863.	3.12
130	355.4	1190.3	863.0	3.10
131	356.	1191.	863.	3.08
132	357.	1191.	862.	3.06
133	357.	1191.	862.	3.05
134	358.	1191.	861.	3.03
135	358.0	1191.1	861.0	3.01
136	359.	1191.	861.	2.99
137	359.	1192.	860.	2.97
138	360.	1192.	860.	2.96
139	360.	1192.	860.	2.94
140	360.7	1191.9	859.1	2.92
141	361.	1192.	859.	2.90
142	362.	1192.	858.	2.88
143	362.	1192.	858.	2.87
144	363.	1193.	858.	2.85
145	363.2	1192.7	857.2	2.83
146	364.	1193.	857.	2.81
147	364.	1193.	857.	2.80
148	365.	1193.	856.	2.78
149	365.	1193.	856.	2.77
150	365.7	1193.4	855.4	2.75

NOTE.—Although the quantities in the table are not carried out to as many significant figures as in many tables, they are sufficiently exact for practical purposes. The volumes have been calculated upon the assumption that the mechanical equivalent is 778, instead of 772. All the volumes have been calculated up to 20 lbs. pressure; above that they have been calculated only every five pounds, and the intermediate values interpolated.

TABLE II.

HYPERBOLIC LOGARITHMS.

Number.	Hyperbolic Logarithm.	Number.	Hyperbolic Logarithm.	Number.	Hyperbolic Logarithm.	Number.	Hyperbolic Logarithm.
1.0	0.00	3.5	1.25	6.0	1.79	8.5	2.14
1.1	0.10	3.6	1.28	6.1	1.81	8.6	2.15
1.2	0.18	3.7	1.31	6.2	1.82	8.7	2.16
1.3	0.26	3.8	1.34	6.3	1.84	8.8	2.17
1.4	0.34	3.9	1.36	6.4	1.86	8.9	2.19
1.5	0.41			6.5	1.87		
1.6	0.47	4.0	1.39	6.6	1.89	9.0	2.20
1.7	0.53	4.1	1.41	6.7	1.90	9.1	2.21
1.8	0.59	4.2	1.44	6.8	1.92	9.2	2.22
1.9	0.64	4.3	1.46	6.9	1.93	9.3	2.23
		4.4	1.48			9.4	2.24
2.0	0.69	4.5	1.50	7.0	1.95	9.5	2.25
2.1	0 74	4.6	1.53	7.1	1.96	9.6	2.26
2.2	0.79	4.7	1.55	7.2	1.97	9.7	2.27
2.3	0.83	4.8	1.57	7.3	1.99	9.8	2.28
2.4	0.88	4.9	1.59	7.4	2.00	9.9	2.29
2.5	0.92			7.5	2.01		
2.6	0.96	5.0	1.61	7.6	2.03	10.0	2.30
2.7	0.99	5.1	1.63	7.7	2.04	10.1	2.31
2.8	1.03	5.2	1.65	7.8	2.05	10.2	2.32
2.9	1.06	5.3	1.67	7.9	2.07	10.3	2.33
		5.4	1.69			10.4	2.34
3.0	1.10	5 5	1.70	8.0	2.08	10.5	2.35
3.1	1.13	5.6	1.72	8.1	2.09	10.6	2.36
3.2	1.16	5.7	1.74	8.2	2.10	10.7	2.37
3.3	1.19	5.8	1.76	8.3	2.12	10.8	2.38
3.4	1.22	5.9	1.78	8.4	2.13	10.9	2.39

TABLE III.

FACTORS OF EVAPORATION.

Temperature of Feed-water	Pressure of Steam, by Gauge, in Pounds per Square Inch.												
	0	40	50	60	70	80	90	100	110	120	130	140	150
40	1·179	1.203	1.206	1.209	1.212	1.214	1.217	1.219	1.221	1.222	1.224	1.226	1.228
50	1.168	1.192	1.195	1.198	1.201	1.203	1.206	1.208	1.210	1.211	1.213	1.215	1.217
60	1.158	1.182	1.185	1.188	1.191	1.193	1.196	1.198	1.200	1.201	1.203	1.205	1.207
70	1.148	1.172	1.175	1.178	1.181	1.183	1.186	1.188	1.190	1.191	1.193	1.195	1.197
80	1.137	1.161	1.164	1.167	1.170	1.172	1.175	1.177	1.179	1.180	1.182	1.184	1.186
90	1.127	1.151	1.154	1.157	1.160	1.162	1.165	1.167	1.169	1.170	1.172	1.174	1.176
100	1.117	1.141	1.144	1.147	1.150	1.152	1.155	1.157	1.159	1.160	1.162	1.164	1.166
110	1.106	1.130	1.133	1.136	1.139	1.141	1.144	1.146	1.148	1.149	1.151	1.153	1.155
120	1.096	1.120	1.123	1.126	1.129	1.131	1.134	1.136	1.138	1.139	1.141	1.143	1.145
130	1.085	1.109	1.112	1.115	1.118	1.120	1.123	1.125	1.127	1.128	1.130	1.132	1.134
140	1.075	1.099	1.102	1.105	1.108	1.110	1.113	1.115	1.117	1.118	1.120	1.122	1.124
150	1.065	1.089	1.092	1.095	1.098	1.100	1.103	1.105	1.107	1.108	1.110	1.112	1.114
160	1.054	1.078	1.081	1.084	1.087	1.089	1.092	1.094	1.096	1.097	1.099	1.101	1.103
170	1.044	1.068	1.071	1.074	1.077	1.079	1.082	1.084	1.086	1.087	1.089	1.091	1.093
180	1.033	1.057	1.060	1.063	1.066	1.068	1.071	1.073	1.075	1.076	1.078	1.080	1.082
190	1.023	1.047	1.050	1.053	1.056	1.058	1.061	1.063	1.065	1.066	1.068	1.070	1.072
200	1.013	1.037	1.040	1.043	1.046	1.048	1.051	1.053	1.055	1.056	1.058	1.060	1.062
212	1.000	1.024	1.027	1.030	1.033	1.035	1.038	1.040	1.042	1.043	1.045	1.047	1.049

TABLE IV.

HEATING POWER OF FUELS.

COMBUSTIBLE.	C	H	O	N	S	Ash.	Heat Units per Pound.
Wood, air dried........	40.4	4.90	32.70	0.90	......	1.20	6400
Peat....................	40.8	3.30	26.30	1.00	......	7.70	6800
" air dried..	46.1	4.60	23.60	1.00	......	1.50	7600
Petroleum, crude, from Baker, Russia	86.5	12.00	1.50	......	......	......	19800
Petroleum, heavy crude, from Pennsylvania	84.9	13.70	1.40	......	......	......	19200
Petroleum, common, from Virginia.......	85.3	13.90	0.80	......	......	......	18100
Lignite, American.....	50.1	3.90	13.70	0.90	1.50	13.20	10300
" Australian...	64 3	4.20	10.00	1.00	0.60	10.00	11000
Coal, Welsh	83.78	4.79	4.15	0 98	1.43	4.91	15100
" Newcastle.......	82.12	5.31	5.69	1.35	1.24	3.77	15200
" Lancashire	77.90	5.32	9.53	1.30	1.44	4.88	14600
" Scotch	78.53	5.61	9.69	1.00	1.11	4.03	14900
" Big Muddy, " Jackson Co , Ill	69.80	5.26	8.35	1.33	2.02	6.90	12600
" Johnson Co., Arkansas......	83.74	4.52	0.54	1.50	1.60	6.63	14400
" Block, Id........	82.70	4.77	8.81	1.74	0.98	1.00	14000
" Hocking Valley, Ohio...........	72 29	6.53	8.28	1.50	0.43	2.72	13400
" Coking, Pittsburgh, Pa.....	79.81	5.98	4.80	1.50	1.35	6.48	14400
" Anthracite	91.50	3.50	2.60	......	......	......	15200
" " Pennsylvania, Buckwheat............	81.32	......	......	......	0.67	10.96	12200

C means per cent of carbon contained in the combustible; H, the per cent of hydrogen; O, the per cent of oxygen; N, the per cent of nitrogen, and S, the per cent of sulphur.

PROBLEMS.

1. How much work is done in lifting a weight of 20 lbs. through a height of 20 ft.? Ans., 400 ft.-lbs

2. How much work is done in moving a weight of 100 lbs. along a horizontal plane surface against a resistance of 10 lbs. through a distance of 6 ft.? Ans., 60 ft.-lbs.

3. If the resistance to be overcome on a railroad is 10 lbs. for each ton of weight of the cars, what horse-power will be required to move a train of cars weighing 100 tons at a speed of 40 miles per hour? Ans., 107 horse-power.

4. How many units of heat per minute are equivalent to one horse-power? Ans., 42.4.

5. A piece of iron weighing 5 lbs. is heated to 212 degrees and then dropped into a vessel containing 16.5 lbs. of water at 60 degrees. If the temperature of the water is increased 5 degrees by the heat from the iron, what is the specific heat of the iron? Ans., 0.112.

6. The specific heat, c_p, of air at constant pressure, expressed in heat units, is 0.24. What is the specific heat expressed in ft.-lbs. at constant pressure, K_p, and at constant volume, K_v?

Ans., $K_p = 186.7$ ft.-lbs., $K_v = 134.4$ ft.-lbs.

7. A quantity of air at a temperature of 60 degrees under a pressure of 14.7 lbs. per square inch, has a volume of 5 cubic feet. What is the volume of the same air

when its temperature is changed to 120 degrees at constant pressure? Ans., 5.57 cub. ft.

8. The volume of a quantity of air at a temperature of 60 degrees under a pressure of 14.7 lbs. per square inch is 10 cub. ft. What is the volume of the same air when the pressure is changed at constant temperature to 60 lbs. per square inch? Ans., 2.45 cub. ft.

9. Assume that the initial pressure, volume, and absolute temperature of a gas are P_1, V_1, and T_1; and that after a change the final pressure, volume, and absolute temperature are P_2, V_2 and T_2. Prove that

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}.$$

Let the pressure remain constant at P_1 while the temperature is changed from T_1 to T_2. The volume will change from V_1 to some volume that we may call V'. From (7) we have

(a) $$\frac{V_1}{T_1} = \frac{V'}{T_2}$$

Now let the absolute temperature remain constant at T_2 while the pressure is changed from P_1 to P_2. The volume will change during this change of pressure from V' to V_2. From (8) we have

(b) $$P_1 V' = P_2 V_2.$$

Multiply (a) by (b) and we have

$$\frac{P_1 V_1 V'}{T_1} = \frac{P_2 V_2 V'}{T_2}, \text{ or } \frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

10. The volume of a quantity of air at 70 degrees under a pressure of 16 lbs. per square inch is 20 cubic feet. What is the temperature of this air when the volume is 4 cubic feet and the pressure is 70 lbs. per square inch? Ans., $T = 464.6$, and $t = 3.6$.

11. What is the weight of the quantity of air which occupies a volume of 10 cubic feet at a temperature of 100 degrees under a pressure of 50 lbs. per square inch?

When the pressure is in pounds per *square* foot, we know that $\frac{P\ V}{T} = w\ 53.15$, where w is the weight of the air in pounds. Hence $w = \frac{P\ V}{53.15T} = \frac{50 \times 144 \times 10}{(461+100)\ 53.15} = 2.4$ lbs.

12. How much work is done by a quantity of air while expanding under a constant pressure of 80 lbs. per square inch from a volume of 2 cubic feet to a volume of 6 cubic feet?

For expansion at constant pressure, the work is equal to the pressure per square foot multiplied by the change of volume, or

$$W = P\ (V_2 - V_1).$$

$$P = 80 \times 144,\ V_2 = 6, \text{ and } V_1 = 2.$$

$$\text{Work} = 80 \times 144\ (6-2) = 46080 \text{ ft.-lbs.}$$

13. How much heat, expressed in foot-pounds, must be given to the air during the expansion in Problem 12?

We know that $H = S + L + W$. For a perfect gas whose weight is w we know that $S = w K_v\ (T_2 - T_1)$, and $L = O$. In this case $W = P_1\ (V_2 - V_1)$. Therefore,

$$H = wK_v\ (T_2 - T_1) + P_1\ (V_2 - V_1).$$

But

$P_1\ V_2 = R\ w\ T_2$ and $P_1\ V_1 = R\ w\ T_1$. Hence $w\ T_2 = \frac{P_1 V_2}{R}$ and $w\ T_1 = \frac{P_1 V_1}{R}$. Put for $w\ T_2$ and $w\ T_1$ their values and get

$$H = \frac{K_v}{R}\ (V_2 - V_1)\ P_1 + P_1\ (V_2 - V_1)$$

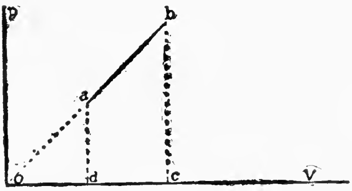
$$= P_1\ (V_2 - V_1) \left(\frac{K_v}{R} + 1\right).$$

$$\frac{K_v}{R}=\frac{K_v}{K_p-K_v}=\frac{1}{\gamma-1}, \text{ and } \frac{K_v}{R}+1=\frac{1}{\gamma-1}+1=\frac{\gamma}{\gamma-1}$$

Hence,

$$H=P_1(V_2-V_1)\frac{\gamma}{\gamma-1}=80\times 144\ (6-2)\ \frac{1.41}{0.41}$$
$$=158{,}500 \text{ ft.-lbs.}$$

14. How much heat is given to a quantity of air while it changes in such a manner that $\frac{P_1}{V_1}=\frac{P}{V}$, from an initial volume of 9.23 cubic feet under a pressure of 100 lbs. per square inch, to a volume of 18.46 cubic feet under a pressure of 200 lbs. per square inch?



As before, $H=S+W$; and $S=w\ K_v\ (T_2—T_1)$; where w is the weight of the air; T_2 the final absolute temperature; and T_1 the initial absolute temperature.

$$W=\text{area } abcd, \text{ in the figure},=\frac{P_1+P_2}{2}(V_2-V_1)$$

$$T_2=\frac{P_2\,V_2}{w\,R}, \text{ and } T_1=\frac{P_1\,V_1}{w\,R}. \text{ Therefore,}$$

$$w\,K_v\,(T_2-T_1)=\frac{K_v\,(P_2\,V_2-P_1\,V_1)}{R}=\frac{K_v\,(P_2\,V_2-P_1\,V_1)}{K_p-K_v}$$
$$=\frac{P_2\,V_2-P_1\,V_1}{\gamma-1}$$

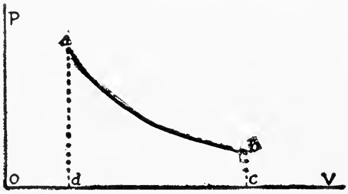
$$H=S+W=\frac{P_2\,V_2-P_1\,V_1}{\gamma-1}+\frac{P_1+P_2}{2}(V_2-V_1)$$
$$=\frac{200\times 144\times 18.46-100\times 144\times 9.23}{1.41-1}+$$
$$\frac{300\times 144\ (18.46-9.23)}{2}$$

$$=1{,}172{,}000 \text{ ft.-lbs.}$$

15. What fraction of the heat in Problem 14 is transformed into work? Ans., 0.17.

16. How much heat must be given to a quantity of air which expands isothermally, at a temperature of 60 degrees, from a valume of 0.83 cubic feet under a pressure of 60 lbs. per square inch, to a volume of 3 cubic feet?

Here, no heat is required to change the temperature of the air; all is used in doing external work.



$H = 0 + W.$

$W =$ area $abcd$, in the figure,

$$= P_1 V_1 \ hyp. \ log. \ \frac{V_2}{V_1}.$$

Ans., 9180 ft.-lbs.

17. How much heat in ft.-lbs. must be given to 1.3 cubic feet of air which is heated at constant volume from an absolute temperature of 520 degrees under a pressure of 2 lbs. per square inch, to an absolute temperature of 1000 degrees?

Since the volume is kept constant, the external work is zero and

$$H = w K_v (T_2 - T_1)$$

$$w = \frac{P_1 V_1}{R T_1}. \quad \text{Hence,}$$

$$H = \frac{P_1 V_1 K_v}{T_1 R} (T_2 - T_1).$$

$$\frac{K_v}{R} = \frac{K_v}{K_p - K_v} = \frac{1}{\gamma - 1} = \frac{1}{1.41 - 1} = \frac{1}{0.41}, \text{ and}$$

$$H = \frac{P_1 V_1}{T_1 0.41} (T_2 - T_1)$$

$$= \frac{2 \times 144 \times 1.3 \ (1000 - 520)}{520 \times 0.41} = 842.9 \text{ ft.-lbs.}$$

18. How many heat units are given to the air in Problem 17? Ans., 1.08.

19. In which is there the greater amount of energy: 1 lb. of air at 60 degrees under a pressure of 100 lbs. per square inch, or 1 lb. of air at 60 degrees under a pressure of 15 lbs. per square inch? Give the reasons for your answer.

20. Given a quantity of air whose volume is 3 cubic feet at 60 degrees under a pressure of 45 lbs. per square inch. What is the volume and temperature of this air after it is expanded adibatically until its pressure is 15 lbs. per square inch?

$$\text{Ans.} \begin{cases} V = 6.54 \text{ cub. ft.} \\ T = 378.6; \text{ and } t = -82.4. \end{cases}$$

21. (a) What is the work done during the expansion in Problem 20? (b) What is the heat, in heat units, converted into work?

$$\text{Ans.} \begin{cases} \text{(a) } 13{,}170 \text{ ft.-lbs.} \\ \text{(b) } 16.9 \text{ heat units.} \end{cases}$$

22. Given a quantity of air whose volume is 2 cubic feet at a temperature of 60 degrees under a pressure of 80 lbs. per square inch. (a) What is the weight of the air? (b) What will be the temperature and pressure if the air be expanded adibatically until its volume is 8 cubic feet? (c) How much work will be done during this expansion? (d) How much work will be done if the air be expanded isothermally until its volume is 8 cubic feet?

Ans., (a) 0.83 lbs. (b) -166 degrees and 11.3 lbs. per sq. in. (c) 24,450 ft.-lbs. (d) 32,000 ft.-lbs.

23. (a) What is the temperature of the steam in a boiler

whose gauge pressure is 90 lbs.? (b) What is the weight of one cubic foot of the steam? Ans. $\begin{cases} \text{(a) 330.9 degrees.} \\ \text{(b) 0.238 lbs.} \end{cases}$

24. How many heat untis are required to heat 16 lbs. of water from an initial temperature of 60 degrees and evaporate it under a pressure of 30 lbs. by the gauge?

Ans., 18,200.

25. The temperature of the water entering a boiler, in which the gauge pressure is 60 lbs. per square inch, is the same as the temperature of the steam in the boiler. (a) What is the external work done in evaporating one pound of water? (b) What is the internal work done in evaporating one pound of water? Ans. $\begin{cases} \text{(a) 62,300 ft.-lbs.} \\ \text{(b) 636,600 ft.-lbs.} \end{cases}$

26. Given a quantity of air whose temperature is 80 degrees; whose pressure is 100 lbs. per square inch; and whose volume is 2.2 cubic feet. It is made to pass through the following Carnot cycle: It is expanded isothermally until its volume is 4.0 cubic feet; then expanded adiabatically until its temperature is 30 degrees; then compressed isothermally; and finally it is compressed adibatically until its volume, pressure, and absolute temperature are the same as at the beginning of the cycle. (a) What is the total quantity of heat, H, given to the air? (b) What is the heat, U, taken from the air? (c) What is the work, W, done during the cycle? (d) What is the efficiency, E, of the cycle?

$$H = P_1 V_1 \, hyp. \, log. \frac{V_2}{V_1} = 19{,}000 \text{ ft.-lbs.}$$

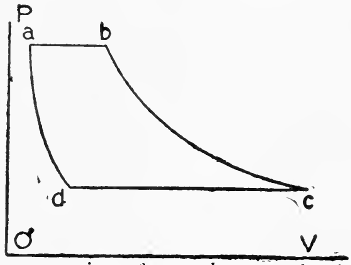
$$U = P_4 V_4 \, hyp. \, log. \frac{V_3}{V_4} = \frac{P_1 V_1 T_2}{T_1} \, hyp. \, log. \frac{V_2}{V_1}$$

$$= 17{,}300 \text{ ft.-lbs.}$$

$$W = H - U = 1{,}700 \text{ ft.-lbs.}$$

$$E = \frac{W}{H} = \frac{T_1 - T_2}{T_1} = 0.093.$$

27. One pound of air is made to pass through the following cycle: It is expanded at constant pressure; then expanded isothermally; then compressed at constant pressure; and then compressed isothermally until the cycle is completed. What are the expressions for H, U, W, and E?



The work diagram is shown in the figure. Let the co-ordinates of a be V_1, P_1, T_1; of b be V_2, P_1, T_2; of c be V_3, P_2, T_2; and of d be V_4, P_2, T_1.

During the expansion from a to b the heat given to the pound of air is $K_v\,(T_2—T_1) + P_1\,(V_2—V_1)$; and during the expansion from b to c the heat given to the air is

$P_1\,V_2\,hyp.\,log.\,\frac{V_3}{V_2}$. Adding these expressions we have that the total heat, H, given to the air is

$$H = K_v\,(T_2-T_1)+P_1\,(V_2-V_1)+P_1\,V_2\,hyp.\,log.\,\frac{V_3}{V_2}$$

$$=\frac{P_1\,(V_2—V_1)}{\gamma—1}+P_1\,(V_2—V_1)+P_1\,V_2\,hyp.\,log.\,\frac{P_1}{P_2}$$

$$=\frac{P_1\,(V_2—V_1)\,\gamma}{\gamma—1}+P_1\,V_2\,hyp.\,log.\,\frac{P_1}{P_2}$$

During the compression from c to d the heat taken from the air must be the same that would be put into it during expansion from d to c, or $K_v\,(T_2—T_1)+P_2\,(V_3—V_4)$; and the heat taken from the air during compression from d to a is

$P_1\,V_1\,hyp.\,log.\,\frac{V_4}{V_1}$. Therefore, the heat, U, taken from the air is

$$U = K_v(T_2—T_1) + P_2(V_3—V_4)+P_1\,V_1\,hyp.\,log.\,\frac{V_4}{V_1}.$$

$$=\frac{P_1\,(V_2—V_1)}{\gamma—1}+P_2\,(V_3—V_4)+P_1\,V_1\,hyp.\,log.\,\frac{P_1}{P_2}.$$

Since $P_2 V_3 = P_1 V_2$, and $P_2 V_4 = P_1 V_1$, we have $P_2 (V_3 - V_4) = P_1 (V_2 - V_1)$; and

$$U = \frac{P_1 (V_2 - V_1)}{\gamma - 1} + P_1 (V_2 - V_1) + P_1 V_1 \textit{ hyp. log.} \frac{P_1}{P_2}$$

$$= \frac{P_1 (V_2 - V_1) \gamma}{\gamma - 1} + P_1 V_1 \textit{ hyp. log.} \frac{P_1}{P_2}$$

$$W = H - U = P_1 (V_2 - V_1) \textit{ hyp. log.} \frac{P_1}{P_2}$$

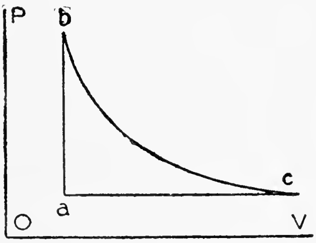
$$E = \frac{W}{H}$$

28. If in Problem 27, the weight of air used is 0.25 lb.; P_1 is 11,520 lbs. per square foot; P_2 is 2,200 lbs. per square foot; V_1 is 0.61 cubic foot; and V_2 is 2 cubic feet; what will H, U, W, and E equal?

Since the expressions derived for H, U, W, and E, involve only the pressures and volumes of the gas, the weight need not be considered. Substitute the values of P_1, V_1, and P_2 in the expressions for H, U, W, and E and get: $H = 93{,}300$; $U = 66{,}700$; $W = 26{,}600$; and $E = 0.285$.

29. Find the expressions for H, U, W, and E, for one pound of air working according to the following cycle: It is heated at constant volume; then expanded adiabatically; then compressed at constant pressure to its initial condition.

The work diagram is shown in the figure. The co-ordinates of a are P_1, V_1, T_1; of b, P_2, V_1, T_2; and of c, P_1, V_2, T_3.



During the change from a to b the air is heated at constant volume, no work is done, and the heat put in during this change is $K_v (T_2 - T_1)$. During the change from b to c no heat is either given to, or emitted by, the air. During the change from c to a the same amount of heat is taken from the body that must be given to it for a change from a to c, or,

$$K_v (T_3 - T_1) + P_1 (V_2 - V_1).$$

Therefore we have

$$H = K_v (T_2 - T_1) = \frac{V_1 (P_2 - P_1)}{\gamma - 1}$$

$$U = K_v (T_3 - T_1) + P_1 (V_2 - V_1),$$

$$= \frac{P_1 (V_2 - V_1)}{\gamma - 1} + P_1 (V_2 - V_1) = \frac{P_1 (V_2 - V_1) \gamma}{\gamma - 1}.$$

$$W = H - U.$$

$$E = \frac{W}{H}.$$

30. In Problem 29 let P_1 be 15 lbs. per square inch; P_2 be 80 lbs. per square inch; V_1 be 1.3 cubic feet; and V_2 be 4.26. Find the values of H, U, W, and E.

$H = 29{,}700$; $U = 22{,}000$; $W = 7{,}700$; and $E = 0.26$.

31. Deduce the expressions for H, U, and W for one pound of air for the following cycle: Air expanded at constant pressure, P_1, from V_1 to V_2; then expanded adiabatically from V_2 to V_3; then compressed at constant pressure, P_2, from V_3 to V_4; then compressed adiabatically to its initial condition.

$$H = \frac{\gamma P_1 (V_2 - V_1)}{\gamma - 1}; \qquad U = \frac{\gamma P_2 (V_3 - V_4)}{\gamma - 1}$$

$$= \frac{\gamma P_1^{\frac{1}{\gamma}} P_2^{\frac{\gamma - 1}{\gamma}} (V_2 - V_1)}{\gamma - 1}$$

$$W = H - U.$$

32. In Problem 31, let P_1 be 12,000 lbs. per square foot; P_2 be 3,000 lbs. per square foot; V_1 be 0.8 cubic foot; and

V_2 be 3.2 cubic feet; and determine the values of H, U, W, and E.

$H = 99,000$; $U = 66,200$; $W = 32,800$; and $E = 0.33$

33 Determine the horse-power of a double-acting engine whose cylinder is 13 inches in diameter and which has an 18-inch stroke, when making 220 revolutions per minute while taking steam at 80 lbs by the gauge and cutting off at $\frac{1}{4}$ stroke. Neglect the clearance and assume that the mean back pressure is 20.5 absolute.

Ans., 96.7 horse-power.

34. An engine has a clearance volume which is 0.08 of the volume swept through by the piston per stroke. If the steam be cut off at $\frac{1}{5}$ stroke, what will be the number of times it is expanded? Ans., 3.86 times.

35. Assume that the mean back pressure is 19 lbs. absolute, and that the clearance volume is 10 per cent. of the volume swept through by the piston per stroke; and determine the horse-power developed by a double-acting engine, whose cylinder is 12 inches in diameter and has a 14-inch stroke, when making 260 revolutions per minute, while taking steam at 70 lbs. by the gauge and cutting off at $\frac{3}{8}$ stroke. Ans., 97 horse-power.

36 What is the weight of the steam used per stroke by the engine in Problem 33? Ans., 0.075 lbs.

37. Neglect the clearance volume of the engine in Problem 35, and determine the weight of steam used per hour per indicated horse-power Ans., 22.8 lbs.

On account of the loss by condensation and other causes, the weight of steam actually used per hour per indicated horse-power will be from $\frac{1}{5}$ to $\frac{1}{3}$ greater than 22.8 lbs., or between 27 and 31 pounds

38. Assume that the temperature of the water entering the boiler is 150 degrees, and determine the efficiency of the steam in Problem 33. Ans., $E = 0.117$

39. If it were possible to use the steam in a perfect engine, working according to the Carnot cycle, between the same limits of temperatures as in Problem 38, what would the efficiency be? Ans., 0.22.

40. The cylinders of a locomotive are 19 inches in diameter and have a 24-inch stroke; the driving wheels are 7 feet in diameter; and the mean back pressure against which the pistons work is 19 lbs. absolute. Determine the horse-power developed by the locomotive when taking steam at 150 lbs. by the gauge and cutting off at $\frac{5}{8}$ stroke, while traveling at a speed of 40 miles per hour.

The number of revolutions made per minute by each driving wheel is equal to the number of feet in one mile, 5,280, multiplied by the number of miles traveled per hour, and divided by 60 times the circumference of one driving wheel, $\dfrac{5280 \times 40}{60 \times 3.14 \times 7} = 160$. Horse-power $= 1458$.

41. Assume the mean effective pressure to be 40 lbs., the number of revolutions to be 75 per minute, and the length of stroke to be 42 inches; and detemine the diameter of the cylinder of a double-acting engine which will develop 200 horse-power. Diameter $= 20$ inches.

42. If the mean back pressure is 20 lbs. absolute, how many times must steam at an initial pressure of 80 lbs. by the gauge, be expanded in order that the mean effective pressure shall be 40 lbs.?

Here we have

$$40 = \frac{95\,(1 + hyp.\ log.\ r)}{r} - 20$$ From this we get

$hyp.\ log.\ r = 0.632\ r - 1.$

In order to solve this we must assume various values of r and try them in the equation; we shall finally get a value of r that will satisfy it.

If $r = 3$,	*hyp log.* $r = 1.10$;	and we have	$1.10 > 1.89 - 1$
" $r = 4$,	" " $= 1.39$;	" "	$1.39 < 2.52 - 1$
" $r = 3.5$,	" " $= 1.25$;	" "	$1.25 > 2.21 - 1$
" $r = 3.6$,	" " $= 1.28$;	" "	$1.28 > 2.27 - 1$
" $r = 3.7$,	" " $= 1.31$;	" "	$1.31 < 2.34 - 1$

r is equal to 3.6, about.

43 About how many revolutions per minute should be made by an automatic high speed engine whose stroke is 18 inches? Ans., 218.

44. About what should be the diameter of the cylinder of an automatic high-speed engine whose stroke is 16 inches? Ans., 12 inches.

45. About what should be the length of the connecting rod of an automatic high-speed engine whose stroke is 14 inches? Ans., 35 inches.

46. About how many revolutions per minute should be made by a Corliss engine whose stroke is 42 inches? Ans., 67.

47. About what should be the diameter of the cylinder of a Corliss engine whose stroke is 36 inches? Ans., 18 inches.

48. About what should be the length of the connecting rod of a Corliss engine whose stroke is 54 inches? Ans., 162 inches.

49. How does increasing the angle of advance affect

the lead, the point of cut-off, and the point of compression?

50. Through what distance will the valve move, if the eccentric be turned through an angle equal to the angle of advance?

51. What must be done to make the cut-off occur later, on a single-valve engine, and not change the point of release or the point of compression?

52.* Find the steam lap and the lead of a valve, whose travel is $4\frac{1}{8}$ inches, that admits steam when the piston is $\frac{1}{128}$ of the stroke before the beginning of the forward stroke, and that cuts off at $\frac{5}{8}$ of the stroke.

Ans., Lap $= 1\frac{1}{8}$ in.; lead $= \frac{1}{4}$ in.

53. Find the steam lap and the lead of a valve, whose travel is $4\frac{1}{2}$ inches, that admits steam when the piston is $\frac{1}{64}$ of the stroke before the beginning of the forward stroke, and that cuts off at $\frac{3}{8}$ of the stroke.

Ans., Lap $= 1\frac{19}{32}$ in.; lead $= \frac{5}{16}$ in.

54. Steam is admitted when the piston is at the beginning of the stroke and is cut off at $\frac{1}{2}$ of the stroke, by a valve whose steam lap is $2\frac{1}{8}$ inches. Find the lead, the eccentricity, and the angle of advance.

Ans., Lead $= 0$; eccentricity $= 3$ in.; angle of advance $= 45°$.

55. Steam is admitted when the piston is $\frac{1}{128}$ of the stroke before the beginning of the forward stroke, and is cut off at $\frac{1}{4}$ of the stroke, by a valve whose steam lap is $1\frac{7}{16}$ inches. Find the lead, the eccentricity, and the angle of advance.

Ans., Lead $= \frac{5}{32}$ in.; eccentricity $= 1\frac{3}{4}$ in.; angle of advance $= 63°–15'$.

56. Steam is admitted when the piston has made $\frac{1}{128}$ of

*In working the valve diagram problems it will be well to make the crank circle 8 inches in diameter.

the stroke, and is cut off at $\frac{9}{16}$ of the stroke, by a valve whose lead is $-\frac{5}{16}$ of an inch. Find the steam lap and the eccentricity. Ans., Lap $= 1\frac{25}{32}$ in.; eccentricity $= 2\frac{15}{32}$ in.

There are some special cases where the construction shown in Fig. 45 fails, and other constructions must be used. The most common case that occurs is when the line *eh* is so nearly parallel to *ef* that it is impossible to determine with any accuracy their point of intersection, O'. In such cases the construction must be exactly the same as for Fig. 45 until the point *h* is fixed, then instead of drawing the line *eh* draw *hg*, as shown in Fig. 45*a*. Then draw *dk* through *d* parallel to *hg*, and continue it until it cuts *fg* prolonged at *k*. Through *k* draw $O'k$ cutting *Ob* at *b*, and *ef* at O'. $O'b$ is the steam lap. If $O'd$ be drawn, the angle it makes with *Ob* will be the required angle of advance. $O'd$ is the eccentricity.

If the lead be zero, the points *O*, *d*, and *e*, in Figs. 45 and 45*a* will coincide. In this case the method of Fig. 45 fails but the method shown in Fig. 45*a* may be used for the

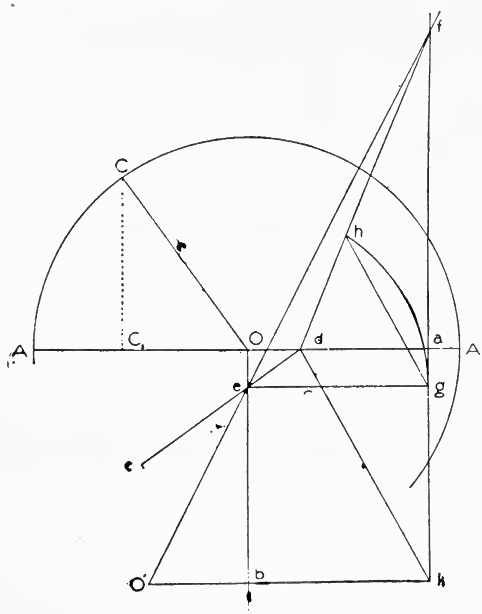


Fig. 45*a*.

solution of the problem. Sometimes it is preferable to use the following method, indicated by Fig. 45*b*: Since he

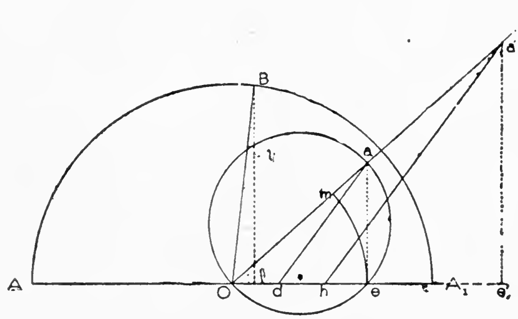


FIG. 45*b*

lead is zero we know that the crank is in the position OA_1 in Fig 45*b* when the steam is admitted. Let OB represent the position of the crank at cut-off. Now we know that the center of the eccentric is somewhere on the line, OD, bisecting the angle BOA_1. Assume any point, a, as a trial center of the eccentric; and draw the valve circle aeO. With O as a center draw the lap circle em, cutting Oa at m. If Oa were the eccentricity and Oe the steam lap, the maximum opening of the valve would be am. But $am = Oa—Om = Oa—Oe = Oa\ (1—cos\ DOA)$. Since the angle DOA, is constant, we see that the maximum opening of the valve is, in this case, directly proportional to the eccentricity. Therefore, make Od equal to ma; and Oh equal to the required maximum opening. Draw da; then draw ha' parallel to da and cutting Oa at a'. Oa' is the required eccentricity. Through a' draw a line perpendicular to OA, and cutting it at e'. Oe' is the required steam lap.

In Problem 57 the student may use the regular construction shown in Fig. 45 or the construction shown in Fig. 45*a*. In Problem 58 the regular construction fails, and the construction shown in Fig. 45*a* or that in Fig. 45*b* must be used. In Problem 59 it will probably be best to use the construction shown in Fig. 45*a*.

57. Steam is cut off at $\frac{5}{8}$ of the stroke by a valve whose maximum opening is $\frac{3}{4}$ of an inch, and whose lead is $\frac{1}{8}$ of an inch. Find the steam lap, the eccentricity, and the angle of advance.

Ans., Lap $= 1$ in.; eccentricity $= 1\frac{3}{4}$ in.; angle of advance $= 40°$.

58. Steam is cut off at $\frac{1}{2}$ of the stroke by a valve whose maximum opening is $\frac{5}{8}$ of an inch, and whose lead is $-\frac{1}{16}$ of an inch. Find the steam lap and the eccentricity

Ans., Lap $= 1\frac{7}{16}$ in.; eccentricity $= 2\frac{1}{16}$ in.

59. Steam is cut off at $\frac{1}{4}$ of the stroke by a valve whose maximum opening is $\frac{1}{2}$ inch, and whose lead is zero. Find the steam lap, and the eccentricity

Ans., Lap $= 3\frac{5}{32}$ in.; eccentricity $= 3\frac{21}{32}$ in.

60. The sine of the angle of advance is $\frac{14}{23}$, the eccentricity is $2\frac{7}{8}$ inches, and the compression begins when the piston has made $\frac{11}{16}$ of the return stroke. Find the exhaust lap and the point of release.

Ans., Lap $= 1\frac{7}{16}$ in.; release begins when the piston is $\frac{1}{256}$ of the stroke from the end of the stroke.

61. The sine of the angle of advance is $\frac{11}{16}$, the eccentricity is 2 inches, and the point of compression is $\frac{19}{32}$ of the stroke from the beginning ot the return stroke. Find the exhaust lap and the point of release.

Lap $= 1\frac{5}{32}$ in.; release begins when the piston is $\frac{1}{128}$ of the stroke from the end of the forward stroke.

62. The point of release is $\frac{3}{128}$ of the stroke before the end of the forward stroke, compression begins at $\frac{9}{16}$ of the return stroke, and the eccentricity is $3\frac{1}{2}$ inches. Find the exhaust lap and the angle of advance.

Lap $= 1\frac{15}{16}$ in.; angle of advance $= 50°$-$45'$.

63. Find the center of suspension of the eccentric of an automatic high-speed engine on which the cut-off changes from $\frac{1}{8}$ to $\frac{5}{8}$ of the stroke, and the maximum opening of the valve when cutting off $\frac{1}{4}$ stroke is $\frac{3}{8}$ of an inch. The lead of the valve shall be $\frac{1}{32}$ of an inch positive, for cut-off at $\frac{1}{8}$ the stroke; zero, for cut-off at $\frac{1}{4}$ the stroke; and $\frac{3}{16}$ of an inch negative, for cut-off at $\frac{5}{8}$ of the stroke.

64. For an indicator pendulum motion, such as is shown in Fig. 52, what should be the shortest lengths of the distances *Bc* and *BD* in order to get a card 3 inches long from an engine whose stroke is 18 inches?

Ans., $Bc = 36$ in.; and $BD = 6$ in.

65. The indicator card taken from an engine whose cylinder is 13 inches in diameter and which has a stroke of 21 inches, is $3\frac{3}{16}$ inches long, and has an area of 1.46 square inches. What was the horse-power developed by the engine if the card were taken with an 80-lb. spring while the engine was making 180 revolutions per minute?

Ans., 93 horse-power.

66. On an indicator card $3\frac{3}{16}$ inches long, taken from an engine whose cylinder is 13 inches in diameter and whose stroke is 21 inches, the length of the line corresponding to the line *fg* in Fig. 58 is $2\frac{3}{8}$ inches. If the pressure corresponding to the point *f*, in Fig. 58, be 13 lbs. per square inch by the gauge, what is the weight of steam used per stroke by the engine? Ans., 0.0825 lbs.

67. If the engine from which the card in Problem 66 is taken, develop 93 indicated horse-power when making 180 revolutions per minute, what is the weight of steam used per hour per indicated horse-power? Ans., 19.1 lbs.

68. Assume c in (63) to be 0.85; E in (64) to be 9; P_1 to be 105 lbs., by the gauge; P_3 to be 8 lbs. absolute; the length of the stroke to be 20 inches; the number of revolutions to be 200 per minute; and the ratio of the volume of the low-pressure cylinder to that of the high-pressure cylinder to be 3; and determine the diameters of the cylinders of a compound-engine to develop 200 horse-power.

Diameter of low pressure cylinder = 20.7 in.
" " high " " = 12.0 in.

69. How many times is the steam expanded in the high pressure cylinder in Problem 68? Ans., 3 times.

70. An engine takes steam at an initial pressure of 80 lbs. by the gauge and expands it 3.7 times against a mean back pressure of 18 lbs. absolute. How much would the horse-power of the engine be increased by the use of a condenser which reduces the mean back pressure to 6 lbs. absolute? Ans., 29 per cent.

71. To what could the number of expansions of the steam in the engine of Problem 70 be changed, and the engine continue to do the same work with the condenser that it did without?

From (68) we have

$$\frac{1 + hyp.\ log.\ r}{r} = \frac{1 + hyp.\ log.\ 3.7}{3.7} - \frac{18-6}{95} = 0.50.$$

Therefore, we have

$$hyp.\ log.\ r = 0.50r - 1$$

If $r = 3$ we have $1.10 > 1.50 - 1$
" $r = 5$ " " $1.61 > 2.50 - 1$
" $r = 6$ " " $1.79 < 3.00 - 1$
" $r = 5.5$ " " $1.70 < 2.75 - 1$
" $r = 5.4$ " " $1.69 < 2.70 - 1$
" $r = 5.3$ " " $1.67 > 2.65 - 1$
$r = 5.4$, about.

72. To what could the boiler pressure of the steam in Problem 70 be reduced, and the engine continue to do the same work with the condenser that it did without?

Ans., about 65 lbs. by the gauge.

73. If the condensing water enters the condenser at 70 degrees and leaves it at 110 degrees, how many pounds of water will be required to condense one pound of steam exhausted from the engine in Problem 70?

Ans., 26.9 lbs.

74. About what is the vacuum, in inches, of Mercury, maintained by the condenser in Problem 70?

Ans. We may say that, roughly, the difference between the pressure against which the steam is exhausted without and with the condenser is equal to the difference between the mean back pressures without and with the condenser, or to $18-6=12$ lbs. That is, the pressure in the condenser is 12 pounds less than atmospheric pressure, or is only 3 lbs. absolute. This corresponds to a vacuum of $12\times 2=24$ inches.

75. Calculate the factor of evaporation for a gauge pressure of 75 lbs. and an initial temperature of the feed water of 135 degrees.

76. A boiler evaporates 5000 lbs. of water per hour from an initial temperature of 145 degrees, and under a pressure of 80 lbs. by the gauge. What is the equivalent water evaporated per hour from and at 212 degrees?

Ans., 5515 lbs.

77. What is the boiler horse-power of a boiler which evaporates 3080 lbs. of water per hour from an initial temperature of 135 degrees and under a pressure of 100 lbs. by the gauge? Ans., 100.

78. A boiler evaporates 3500 lbs. of water per hour from an initial temperature of 120 degrees and under a

pressure of 80 lbs. by the gauge; a second boiler evaporates 4000 lbs. of water from an initial temperature of 110 degrees and under a pressure of 60 lbs. by the gauge. Which of the two boilers utilizes the greater amount of heat per hour?

79. Calculate the number of heat units evolved by the complete combustion of one pound of coal which contains 69.8 per cent. of carbon; 5.26 per cent. of hydrogen; and 8.35 per cent. of oxygen.

Ans., 12,750 heat units.

80. Calculate the number of heat units evolved by the complete combustion of one pound of petroleum which contains 84.9 per cent. of carbon; 13.7 per cent. of hydrogen; and 1.40 per cent. of oxygen.

Ans., 20,700 heat units.

81. How many pounds of water can be evaporated from and at 212 degrees by the heat evolved by the complete combustion of one pound of coal containing 65.2 per cent. of carbon; 4.92 per cent. of hydrogen; and 8.64 per cent. of oxygen? Ans., 12.65 lbs.

82. Assume that one cord of wood weighs 3000 lbs. and that each pound of wood will evolve 5000 heat units when completely burned, and determine when it is cheaper to buy wood than to buy the coal in Problem 81.

Cheaper to buy wood as long as one ton (2000 lbs.) of coal costs more than 1.57 as much as one cord of wood.

83. If 40 per cent. of the heat evolved by the combustion of each pound of the coal in Problem 79 is lost, how many pounds of coal will be required to evaporate 5650 lbs. of water from an initial temperature of 130 degrees and under a pressure of 80 lbs. by the gauge?

Ans., 800 lbs.

84. Suppose that in burning the coal in Problem 81 it is found that one-half only of the carbon is completely burned, and that the other half is burned to carbon monoxide; what is the heat evolved per pound of coal burned?
Ans., 8540 heat units.

85. How many pounds of air are required for the complete combustion of one pound of coal containing 79.65 per cent. of carbon, 5.58 per cent. of hydrogen, and 4.64 per cent. of oxygen? Ans., 11.4 lbs.

86. Assume that 20 lbs. of air at 60 degrees are admitted to a furnace for each pound of the coal in Problem 85 that is burned; that the specific heat of the gases in the chimney is 0.24; and that the temperature of the escaping gas is 430 degrees; and determine the number of heat units carried off by the gases per pound of coal burned.

All the carbon, hydrogen, and oxygen in the coal that is burned is carried up the chimney. Therefore, the weight of the gases carried up the chimney per pound of coal burned is the weight of the air admitted per pound of coal plus the weight of the combustible and volatile matter in the coal, or it is $20+0.7965+0.0558+0.0464$, equal 20.9 lbs. Heat carried off $=1850$ units.

87. Determine the area of the heating surface of a return tubular boiler 66 inches in diameter, 16 ft. long, and containing 98 tubes each 3 inch in diameter, that is set so that $\frac{2}{3}$ of the circumference of its shell is exposed to the hot gases. Ans., 1414 sq. ft.

88. What would be the area of the heating surface of the boiler in Problem 87 if it were set so that but $\frac{1}{2}$ of the shell was exposed to the hot gases? Ans., 1368 sq. ft.

89. Assume $12\frac{1}{2}$ sq. ft. of heating surface per boiler horse-power, and determine the horse-power of the boilers in Problem 87 and 88.

90. Assume k in (90) to be $\frac{1}{3}$, and determine what will be the velocity of the gases in a chimney 120 ft. high, when the temperature of the gases is 450 degrees and that of the air is 65 degrees. Ans., 25 ft. per second.