POWER

AND

POWER TRANSMISSION.

BY

E. W. KERR, M.E.,

Assistant Professor of Mechanical Engineering,
Agricultural and Mechanical College of Texas.

FIRST EDITION.

FIRST THOUSAND.

NEW YORK:
JOHN WILEY & SONS.
LONDON: CHAPMAN & HALL, LIMITED.
1902.

ROBERT DRUMMOND PRINTER, NEW YORK.

PREFACE.

THE matter contained in the following pages is largely the subject-matter of lectures delivered by the author to students of the elementary principles of engineering.

The author does not and could not presume to have presented much that is new, but rather a collection of such principles and information as would direct the beginner along the proper course of study.

In preparing the work free recourse has been had to various works upon the subjects named, from which much subject-matter has been used.

It is not intended that the subjects treated should be regarded by the student as exhausted, but rather as containing guiding principles for more thorough investigation, either in the classroom or in practice.

The problems at the end of the different chapters are given more for the purpose of fixing in the mind of the student the more important principles contained, than as examples of practice, though in fact many of them are such.

To Prof. R. H. Whitlock, whose assistance and encouragement are largely responsible for the undertaking of the preparation of this work, and under whom he has studied for several years, the author wishes to render thanks.

To Prof. Chas. Puryear thanks are due for assistance in reading proof.

The author is under obligations to the various manufacturers who have loaned electrotypes.

Figures 34, 35, 36, 37, 41, 42, 73, 74, 76, 78, 89, 90, 91, 92, 96, 97, from Robinson's "Principles of Mechanism," Figures 123, 138, 139, 150, from Kinealy's "Steam Engines and Boilers," Figures 158, 179, 180, 181, from Whitham's "Constructive Steam Engineering," and Figures 121, 132, 171, 173, 175, 182, 183, 191, 192, 197, 198, from Hutton's "Mechanical Engineering of Power Plants," and the electros for the same, were used by the permission of the authors of those works, for which the writer is indebted.

E. W. KERR.

COLLEGE STATION, TEXAS,
NOV. 29, 1901.

CONTENTS.

PART I. MACHINERY AND MECHANICS.

CHAPTER I.

CHAPTER II.

CHAPTER III.

CHAPTER IV.

CHAPTER V.

CHAPTER VI.

CHAPTER VII.

CHAPTER VIII.

CHAPTER IX.

CHAPTER X.

CHAPTER XI.

CHAPTER XII.

CHAPTER XIII.

PART II. STEAM-POWER.

CHAPTER XIV.

CHAPTER XVI.

CHAPTER XVII.

CHAPTER XVIII.

CHAPTER XIX.

PART III. PUMP, GAS-ENGINES, WATER-POWER, COMPRESSED AIR, ETC.

CHAPTER XXIII.

CHAPTER XXIV.

CHAPTER XXV.

PART I.

MACHINERY AND MECHANICS.

CHAPTER I.

INTRODUCTION.

Engineering is the art of constructing and using machinery; or the art of executing civil or military works which require a special knowledge of machinery or of the principles of mechanics.

Mechanical Engineering is that branch of engineering which has to do with machinery such as machine-tools, engines, etc.

Civil Engineering is that branch of engineering which relates to the making and care of roads, bridges, railroads, harbors, etc.

Engineer.—This is the term applied to the person who is skilled in the principles and practice of any branch of engineering. Engineers may be classed according to their occupations, as Mechanical, Civil, Military, Naval, Electrical, Mining Engineers, etc.

Machinist.—This name is given to one who is familiar with and is able to operate machine-tools.

Before a piece of machinery can be in readiness for use it must be worked upon by the designer, the draughtsman, the pattern-maker, the moulder, and finally the machinist. The engineer should understand the principles involved in the work of each of the above men and should have had enough practice in them to make himself reasonably familiar with them.

Mechanics is that science which treats of the action of forces on bodies.

A Force is anything that tends to produce or change motion in a body. A body at rest is put in motion by a force; a body in motion is stopped or retarded or accelerated by a force. Or, if the direction of motion of a body be changed, that change is produced by a force.

The unit of force used by engineers in English-speaking countries is the pound avoirdupois. For some scientific purposes physicists have adopted a so-called "absolute unit." This unit is that force which, acting on a unit mass during a unit of time, will produce a unit of velocity. In the English system it is the force which, acting for one second on a mass whose weight is one pound at London, will produce a velocity of one foot per second. It is equal to $\frac{1}{32.187}$ lbs., or roughly half an ounce. As a unit of force it is useless to engineers.

A Machine is a combination of fixed and movable parts so disposed and connected as to transmit force and motion, in order to secure some useful result. The fixed parts constitute the frame or support for the moving parts. The moving parts constitute a train or trains of mechanism. All moving parts of machines may be classified as follows:

1. Revolving shafts. *Examples:* Line-shafts, spindles, etc.

2. Revolving wheels or cams, with or without teeth. *Examples:* Spur-gears, pulleys, etc.

3. Rods or bars with reciprocating or vibratory motions or both. *Examples:* The piston-rod of an engine; the connecting-rod of an engine; links of all kinds.

4. Flexible connectors depending on friction. *Examples:* Belts, ropes, etc.

5. Flexible connectors not depending on friction. *Example:* Link-belt.

6. A column of fluid in a pipe. *Examples:* Steam, compressed air.

Simple Machines.—The simple machines* are: (1) the Lever, (2) the Cord, and (3) the Inclined Plane. The first includes every body that may be revolved on an axis; the second includes all machines in which force is transmitted by means of flexible connectors; the third includes all machines in which a surface inclined to the direction of motion is introduced.

A lever is a rigid bar, movable about a fixed point called a fulcrum. The bar may be straight, bent, or curved. Levers are divided into three classes, according to the relative position of the applied force, the weight, and the fulcrum.

In a lever of the first class the fulcrum F is between the applied force P and the weight W, as in Fig. 1.

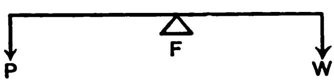


FIG. 1.—Lever of the First Class.

In a lever of the second class the weight is between the applied force and the fulcrum, as in Fig. 2.

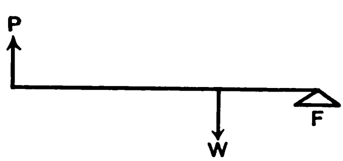


FIG. 2.—Lever of the Second Class.

In a lever of the third class, the applied force is between the weight and the fulcrum, as in Fig. 3.

The law of the lever is the same in all three cases, viz.: The applied force multiplied by its distance from the fulcrum

* These simple machines are also called by earlier writers on Mechanics the "Mechanical Powers." This term is now becoming obsolete. The use of the word "power" is not in accordance with modern usage, in which power is given the meaning "rate of doing work."

is equal to the weight multiplied by its distance from the fulcrum.

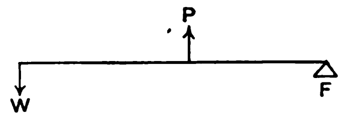


FIG. 3.—Lever of the Third Class.

If the direction of the applied force or of the resistance is not perpendicular to the line of the lever, the "lever-arm" is the perpendicular distance from the fulcrum to the line of action of the applied force or of the load.

If the applied force P, acting through a distance D, moves the load W through a distance d, then $PD = Wd$. This equation may be stated as a law,* applicable to all machines, viz.: *The weight multiplied by the distance through which it is moved is equal to the applied force multiplied by the distance through which it acts.*

The weight of the lever itself is sometimes neglected, but it may be considered as an additional force acting at the centre of gravity of the lever.

An Inclined Plane is usually supposed in calculations of machines to be a perfectly hard and smooth surface. In some cases, however, friction is taken into account. The weight of a body on an inclined plane is partly supported by the reaction of the plane. But as this reaction is normal or perpendicular to the plane, a body on it will slide down, unless restrained

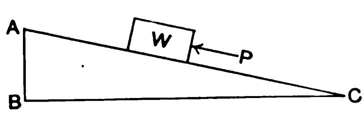


FIG. 4.—Inclined Plane.

by some externally applied force. If this force be applied in a direction parallel to the plane, the force P, Fig. 4, will be to the weight as the height of the plane is to the length of the

* Known as the "Law of Machines."

plane measured on the incline. Expressed as an equation,

$$P = \frac{W \times AB^*}{AC}.$$

If the force is applied in a direction parallel to the base of the plane, then will the applied force be to the weight as the height is to the base; or

$$P = \frac{W \times AB}{BC}.$$

These equations are in accordance with the general law stated above. For if the weight was moved the entire height of the plane the weight would be moved through the height AB. To move the weight through this height the force would be required to act through the whole distance AC or BC, according to the direction in which the force is applied.

If the force be applied to the weight in any other direction than the two just stated, the relation of the force to the weight will be as follows: Let α (Fig. 5) be the angle the plane

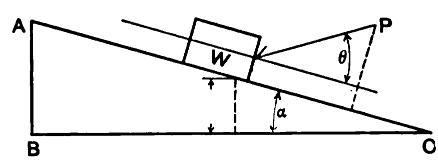


FIG. 5.—Inclined Plane.

makes with the horizontal, and θ the angle the applied force makes with the surface of the plane. Then the force will be to the weight as the sine of α is to the cosine of θ; or

$$P = \frac{W \times \sin \alpha}{\cos \theta}.$$

* The equations on this page apply only to a body in equilibrium.

To solve problems of the inclined plane, use may be made of the triangle of forces. Thus in Fig. 6, let *cd* represent the

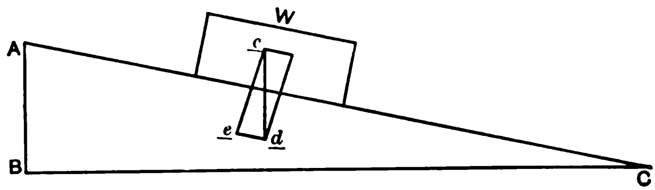


Fig. 6.— Inclined Plane.

weight of the body at rest on the inclined plane, *c* being the centre of gravity of the body. From *c* draw a line *ce* perpendicular to the surface of the plane. From *d* draw a line *ed* parallel to the surface AC, until it intersects *ec* in *e*. Then in the triangle *ced* the force due to gravity, acting on the body, is represented by *cd*, the reaction of the plane by *ec*, and the force which holds the body at rest on the plane, by *de*. If no external force is being applied to hold the body at rest, *de* represents the friction force acting between the plane and body. Since the triangle ABC, representing the plane, is similar to the triangle of forces *ced*, the sides of the plane may be used to determine the relative magnitude of the forces acting on the body. Thus

$$P : W :: ed : cd :: AB : AC.$$

A Wedge is formed by two inclined planes united at their bases. Force is applied to a wedge at its head, the end directly opposite the point. The work of a wedge and of an inclined plane differ in that an inclined plane is generally used to assist in raising a weight, while a wedge is generally used to penetrate a resisting body. *Example:* The use of a wedge to split a block of wood. If friction be neglected, the force required to penetrate a resisting body will be to the resistance as the thickness of the wedge is to the length of the wedge.

Thus, letting $t =$ thickness, $l =$ length, P the force applied, and R the resistance, $P : R :: t : l$,

$$P = \frac{Rt}{l}; \quad R = \frac{Pl}{t}.$$

A Screw is formed by wrapping an inclined plane around a cylinder, so that the height of the plane is parallel to the axis of the cylinder. A nut is formed by wrapping an inclined plane on the internal surface of a hollow cylinder. If it is desired to raise a weight by means of a screw and nut, force is usually applied to the end of a wrench attached to the screw, or to the circumference of a wheel whose axis is that of the screw. Either the screw or nut may remain fixed, the other being rotated in order to raise the weight.

If r be taken as the lever-arm, or as the radius of the wheel to which the force P is applied, p the pitch, or distance between threads, or height of the inclined plane for one revolution of the screw, and W the weight to be raised, and neglecting friction, $P : W :: p : 2\pi r$;

$$P = \frac{Wp}{6.2832r}; \quad W = \frac{6.2832r}{p}P.$$

In practical work, however, friction cannot be neglected, as a large part of the applied force is used up by it, making the screw a very inefficient machine. The practical applications of the screw will be treated more fully in Chapter IX.

The Cam is a revolving inclined plane, which may be wrapped around a cylinder; or it may be curved edgewise and made to rotate in a plane parallel to the base. Its mathematical treatment is the same as that of the screw. The various forms of cams as well as some of their applications will be discussed in Chapter X.

The Pulley or Block consists of a wheel which is able to rotate freely about an axis, together with a flexible cord wrapped around a portion of its circumference. The axis of the wheel may or may not be stationary; hence, pulleys may be classified as fixed or movable. A fixed pulley is shown in Fig. 7. If the system be at rest the tension on the two cords s equal and the applied force F equals the weight P.

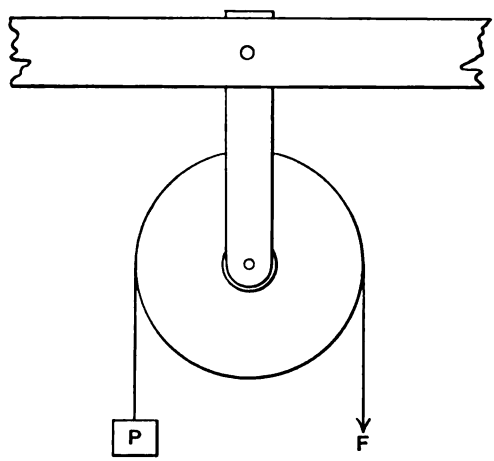


FIG. 7.—Fixed Pulley.

A movable pulley is shown in Fig. 8. Here one end of the cord is attached to a fixed support at A. The weight W is suspended from the axis C of the pulley. The sum of the tensions at A and at P is equal to the tension at B, due to the weight W. As in this case the tension at A is equal to the tension at P, $P = \frac{1}{2}W$.

Movable and fixed pulleys may be combined as shown in Figs. 9 and 10.

Fig. 9 shows one movable and one fixed pulley. The fixed pulley revolving about C merely serves to change the direction of the force P. The relation of P to W is the same in this case as in the case of the single movable pulley, viz., $P = \frac{1}{2}W$.

In Fig. 10 there are three fixed and three movable pulleys. The weight is suspended from the axis B of the movable pulleys. Each movable pulley has two plies of the rope engaging it, or six in all. These six plies will each be shortened by the amount the weight is lifted, and the relation of the applied force to the weight is $P = \frac{1}{6}W$. In general the ratio of W to P is equal to the number of plies that engage the lower block, and also to the number of plies that are shortened by the raising of the weight. In practice the pulleys are seldom arranged as shown in the figure. They are usually side by side and of the same diameter. If the upper block be

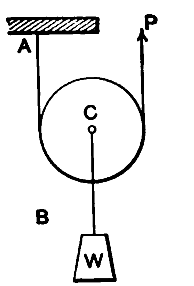


FIG. 8.—Movable Pulley.

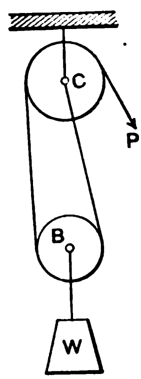


FIG. 9.

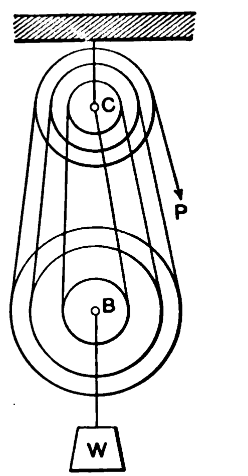


FIG. 10.

provided with three sheaves or pulleys and the lower one with two, the end of the rope will be fastened to a hook at the top of the lower block. In this case five plies will be shortened instead of six and $P = \frac{1}{5}W$. If the end of the rope to which force is applied pass over a sheave in a fixed block, the force may be applied in any direction whatever. If, however, the end passes over a sheave in the movable block, then will it be necessary to apply force in a direction parallel to a line joining the centres of the pulleys. If the force be applied in any other direction, the pulley will be drawn out of the line joining the weight and the fixed pulley, and the maximum effect will not be obtained. The ratio of the effective pull to the actual pull will be equal to the cosine of the angle made by the rope with the vertical.

The **Wheel and Axle** is a modification of the lever. The radius of the wheel may be regarded as one arm of the lever and the radius of the axle as the other. Or a simple arm or wrench may be fastened to the end of the axle instead of the wheel, and the length of the arm be taken as the length of one arm of the lever. Another form of the wheel and axle is shown in Fig. 11. Two cylinders or pulleys of different diameters fastened rigidly together, revolving on the same axis, are used to raise a weight. A rope is wound around the larger cylinder, one end being attached to it. The weight to be raised is attached to a rope fastened to the smaller cylinder. On unwinding the rope from the larger cylinder A, the rope attached to the weight will be wound on the smaller cylinder B. If D represents the diameter of the larger cylinder plus the diameter of the rope, d the diameter of the smaller cylinder plus the diameter of the rope, P the applied force, and W the weight,

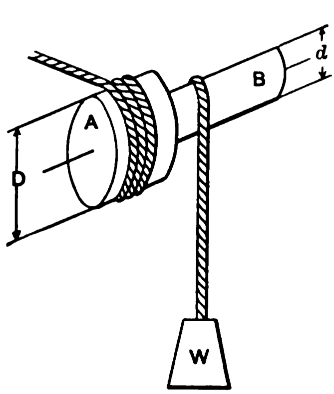


FIG. 11.—Wheel and Axle.

$$P : W :: d : D, \qquad P = \frac{WD}{d}.$$

All moving parts of machines can be resolved into these simple "Elements of Machines." Applications of them will be discussed in later chapters.

ENERGY, WORK, POWER, ETC.

A Motor is any producer of motion.

Energy is the ability to perform work. It is of two kinds, known respectively as *potential* and *kinetic* energy. Potential energy is the ability which a body possesses to do work, owing to its position. Thus a weight raised to a height, or water stored in a reservoir, both possess potential energy. Kinetic energy is the energy possessed by a moving body. Thus if the raised weight be allowed to fall, or the stored water be allowed to flow, both the weight and the water while in motion possess kinetic energy.

Energy manifests itself in various forms, as heat, electricity, mechanical energy, chemical energy, etc.

These different forms of energy may be converted one to the other. Thus heat energy is converted into mechanical energy in the steam-engine. Chemical energy is converted into electric energy in the primary battery, and mechanical energy is converted into electricity in the dynamo.

Conservation of Energy.—The facts stated in the preceding paragraph are comprised in the law known as the law of Conservation of Energy. This law states that "No form of energy can ever be produced except by the expenditure of some other form, nor annihilated except by being reproduced in another form. Consequently the sum total of energy in the universe, like the sum total of matter, always remains the same." (S. Newcomb.)

Potential heat energy exists in coal and other fuels. Potential electric energy exists in a charged storage battery. Potential chemical energy exists in various forms as in gunpowder, etc. The measure of these potential energies is the amount of work that they are able to perform. The actual energy of a moving body is the amount of work it will do against a resistance before that resistance brings it to rest. This energy is equal to the work done on the body in order to bring it to its actual velocity from a state of rest.

Kinetic energy is mathematically equal to $\frac{1}{2}mv^2$, where m, the mass of the body, $= \frac{\text{weight}}{g} = \frac{\text{weight}}{32.2}$, and v is the velocity at the instant of consideration. It is also equal to the weight of the body multiplied by the height from which the body must fall in order to acquire its given velocity. Thus

$$E = \tfrac{1}{2}mv^2 = wh = \frac{wv^2}{2g}.$$

The three principal sources of energy on the earth are the muscular energy of men and animals, the energy of wind or of flowing water, and the energy of fuels. All these sources of energy are due to the heat of the sun.

The first-named group is the least important, as the amount of energy available in either men or animals is limited by the capacity and endurance of the units. Furthermore there is no means of storing energy in them. The second group is more important and has wider application than the first; but still it is not absolutely under the control of man. Energy may be derived from flowing water only in certain locations, as where there is a waterfall or rapidly flowing stream. Then, too, this energy can only be transported to limited distances, as by means of electricity. The longest transmission line for a heavy current of electricity generated by a waterfall exists at present in California. The line is about 150 miles long. The energy due to winds is too uncertain to be depended on to perform any important and continuous work.

Thus it may be easily seen that the third group is the most important source of energy. Fuels are not subject to the limitations of water, wind, and man power.

An enormous capacity for doing work is stored up in very little bulk; it may be liberated from the fuel as gradually as may be desired and the quantity is not limited. It may be obtained in nearly all regions and if not found in certain localities it may be transported there. The most extensive application of this mode of producing energy is that of the Steam-engine and Boiler.

Motive power has different characteristics according to the nature of the source. It may be constant as with a head of water kept at a certain level by a never-failing stream, or it may vary according to fixed laws, like the action of steam in an engine-cylinder; it may be irregular as that of the muscular force of men and animals, or it may be wholly uncertain as in the case of the wind.

These characteristics are not under our control, so that we cannot have power as we want it but must take it as we can get it. In order to make these different sources of power available, some arrangement must be made for controlling them and making them serve our purposes. This is done by means of machinery.

Work is the overcoming of resistance through space. The unit of work is the *foot-pound*, which is the work done in lifting one pound a distance of one foot. Work is done when a body is raised in opposition to the force of gravity. This is the simplest idea of what is meant by work. In general, we say that work is done in moving a body against a resistance and the resistance is overcome by the action of force upon the body moved. From this it is seen that in order that work shall be done, motion must be produced.

Mathematically speaking, work is the product of force in pounds multiplied by the distance through which the force acts in feet, and this product is generally designated as so many *foot-pounds*. Thus, if 10 pounds be lifted through a height of 5 feet, the work done equals $10 \times 5 = 50$ foot-pounds.

Efficiency.—In any machine the work of resistance may be divided into two parts, namely: *useful work* and *lost work*. The former is that which produces desired results and the latter is that which is due to friction and other causes. Of course, the latter is much smaller than the former. For instance, in the arrangement shown in Fig. 7, a force of 20 lbs. applied at F in the direction shown by the arrow should lift an equal weight of 20 lbs. applied at P. But owing to the friction in the axle of the pulley and in the cord, a smaller weight will be lifted; suppose it to be 18 lbs. The work done by the applied force in moving through a distance D is, in this case, $20 \times D = 20D$.

The useful work is $18 \times D$. The *efficiency* then is $\frac{18D}{20D} = \frac{18}{20} = 90$ per cent.

We are now prepared to define efficiency. Efficiency is the ratio of the energy utilized by a machine to the energy supplied to the machine. Or it may be expressed as a fraction, viz.:

$$\text{Efficiency} = \frac{\text{Work put into a machine}}{\text{Work obtained from machine}}.$$

Power is the rate of doing work. Thus the power of a machine may be spoken of as so many foot-pounds, per second, per minute, or per hour as the case may be. The unit of *power* is the *Horse-power*, which is equivalent to 33,000 foot-pounds per minute or 550 foot-pounds per second. An engine of one horse-power will raise one pound 33,000 feet in one minute or 33,000 pounds one foot in one minute.

PROBLEMS.

1. What horse-power will be required to lift a weight of 40,000 lbs. through a height of 100 feet in one minute?

2. What horse-power will be required to lift a weight of 30,000 lbs. through a height of 1000 feet in ten minutes?

3. A pump running at its full capacity lifts 1000 gallons of water into a stand-pipe 50 feet high in one hour. How many horse-power does the pump produce, exclusive of friction?

4. A trip-hammer weighing 2000 lbs., operated by steam, makes 80 drops per minute, the drop being one foot. What is the horse-power of the engine that runs it, supposing that the efficiency is 100 per cent?

5. An elevator rises 200 feet to the top of a building in four minutes. What horse-power is required of an electric motor in

raising it, if the elevator weighs 1000 lbs., supposing the efficiency to be 80 per cent?

6. What is the efficiency of an 18-horse-power engine which will lift a weight of 10,000 lbs. through a height of 100 feet in 2 minutes, when running at its full capacity?

CHAPTER II.

SHAFTING.

SHAFTING is employed in shops for transmitting rotary motion from the motors to the operative machinery.

A **line-shaft** is a continuous run of shafting made up of a number of lengths joined together by couplings, and may or may not be a main line-shaft.

The **main line-shaft** is the line of shafting to which the engine or motor is attached, and which imparts motion to all the other shafts and machines.

Counter-shafts are separate sections, usually short ones, placed between the main shaft and a machine, used to increase or diminish belt-speed, to alter the direction of belt-motion, to carry a *loose* and a *fast* pulley (so that by shifting the belt to the loose pulley it may cease to communicate motion to the machine), or for all these purposes combined.

A **spindle** is a very small shaft, and is usually found on machines.

Hollow shafting is used to a large extent where large quantities of power are to be transmitted; an example of which may be seen in the propeller-shafts of large steamships. It has been found that hollow shafting is stronger than solid shafting for equal quantities of metal.

Flexible shafting is used to transmit rotary motion to any desired distance from the power source through any number of curves, thus allowing the power to be carried to the work instead of the work to the power. It is used extensively for portable work. An example of its use may be seen in the drilling of holes for the rivets in locomotive-boilers where the magnitude of the work prevents taking it to a drill-press of the ordinary type. A more common example is the instrument used by dentists, the power being transmitted from the foot of the operator by means of a small flexible shaft. The construction of this kind of shaft varies with different makes, but it

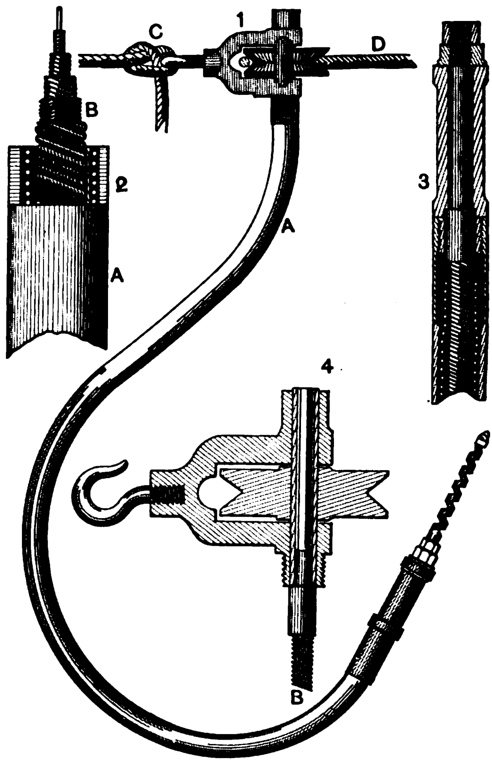


FIG. 12.—Flexible Shaft.

is generally made of a series of steel wires wound upon each other, the alternate layers running in opposite directions. At the ends fittings are attached, one to receive the tools which are to be operated and the other to receive the power to be transmitted.

Shafting is generally made of wrought iron or steel and is made cylindrically true, either by special rolling processes, as in what is known as cold-rolled or hot-rolled shafting, or else it is turned in the lathe. Very large shafting, as in the case of propeller-shafts, is now commonly made up of an ingot forged to approximate shape and then turned in the lathe to a true cylinder. Commercial sizes of solid shafting are made from $\frac{1}{4}$ inch upwards. The sizes are usually given in odd sixteenths of an inch, and advance by eighths. Thus $1\frac{5}{16}$, $1\frac{7}{16}$, $1\frac{9}{16}$ inches, etc. The shafting is rolled or turned accurately to the dimension given, and then the pulley or bearing is bored to a nice fit.

The strains to which a line of shafting is subject are, first, the *torsional* strain due to the twisting effort of the belt on the circumference of the pulley; and, second, the *transverse* strain due to the weight of the shaft and pulleys and to the pull of the belt tending to bend the shaft. In order to keep a shaft from moving out of its place in the direction of its length, it is necessary to use a *collar* or to turn a *shoulder* on the shaft itself. A shoulder is usually inconvenient to manufacture, and

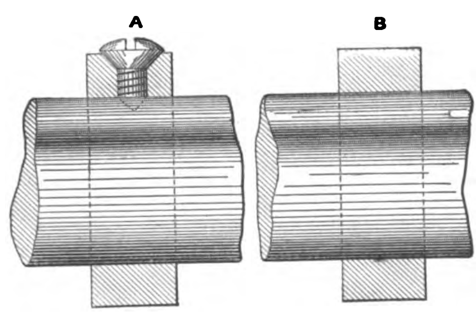


FIG. 13.—Collars. *A*, collar fastened to shaft with a set-screw; *B*, collar fastened to shaft by shrinking.

consequently the collar is generally used for this purpose. The collar is made of wrought iron or steel, and may be fastened to the shaft either by shrinking it on, or by means of set-screws. The collar is placed on the shaft against the bearing, and prevents the shaft from moving in the direction of its length. If the set-screw is used to fasten the collar to the shaft, the head of the screw should be sunk into the collar enough to keep it from catching on belts or clothing.

Shafting which operates wood-working machinery must be run at a higher speed than that used for most metal-working machinery, and taking this as an example it is seen that the speed of shafting, in general, depends on the kind of machinery it is employed to drive. The speed of shafting runs about as follows in practice: For *machine-shops* 120 to 200 revolutions per minute; for *wood-working* 250 to 300 revolutions per minute; and for *cotton-mills* 300 to 400 revolutions per minute.

The cost of a plant may be lessened by running the line-shaft at a high speed, rather than by using large pulleys on it to increase the belt-velocity. The diameter of the shaft should be made as small as considerations of durability will allow. Larger shafts not only increase the weight of the shafting, bearings, couplings, etc., thus increasing the first cost, but also cause a greater amount of friction, due to the larger journals.

A given diameter of shaft will transmit more power in proportion as its speed is increased; that is, a shaft capable of transmitting 10 H.P. when making 100 revolutions per minute will transmit 20 H.P. when making 200 revolutions per minute.

In very large factories long lines of shafting are often used, sometimes as much as 1000 feet in length. In such cases the

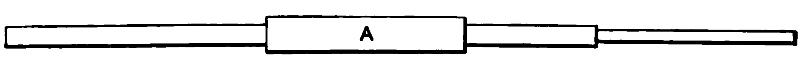


FIG. 14.—Shaft with Different Diameters.

shaft is much larger where it receives the power from the engine than it is farther away, the size of the shaft gradually diminishing as the distance from the motor becomes greater. This consideration suggests another practical rule which should be followed when it is possible; namely, that in arranging a machine-plant, those machines requiring the greatest amount of power should be placed as near as possible to the motor, in order that the diameter and the weight of the shafting, and the friction be reduced as much as possible. For the above reason, in sawmills, the large saws absorbing most power should be driven as directly as possible by the motor, while the spaces farther from the motor should be used for setting up the lighter frame- and circular saws. Economy in the quantity of shafting may thus be practised, as the twisting effort to be resisted by the shaft becomes less and less as the end most distant from the motor is approached, until it becomes almost zero at the end.

The following rule adopted by William Sellers & Co. determines the size of the shaft to be used when the horse-power is given.

RULE.—*Divide the horse-power by the revolutions per minute; multiply the quotient by 125 and extract the cube root of the product. The result is the diameter of shaft required;* that is, $d = \sqrt[3]{\frac{125 \text{ H.P.}}{R}}$.

According to Dr. R. H. Thurston, this coefficient (125) varies with the class of work done by the shaft, and also with the character of the shaft. The coefficient here given is for an iron head-shaft, well supported, and having bearings placed close to the pulleys. For a cold-rolled shaft the formula would read, $d = \sqrt[3]{\frac{100 \text{ H.P.}}{R}}$.

If the shaft under consideration should be a line-shaft, with hangers well spaced, say seven or eight feet apart, the formula would read for iron, $d = \sqrt[3]{\frac{90 \text{ H.P.}}{R}}$; for cold-rolled shafting, $d = \sqrt[3]{\frac{55 \text{ H.P.}}{R}}$. If the shafting is used only to transmit power, and there are no pulleys on it, the formula reads $d = \sqrt[3]{\frac{62.5 \text{ H.P.}}{R}}$ for iron; $d = \sqrt[3]{\frac{35 \text{ H.P.}}{R}}$ for cold-rolled shafting. Here it is noticeable that the size of the shaft decreases as the number of revolutions increases, showing that it is more economical, as far as shafting is concerned, to carry high speeds.

Couplings.—A line of shafting is usually of considerable length, and must therefore be composed of several pieces united, because the difficulties of construction, of transportation, and of setting up forbid its being made in a single piece. The ends of the different pieces are united by means of *couplings*. Couplings for fastening together the ends of the separate sections of shafting are of two kinds, viz.: those which may be used to couple or uncouple at will while the shaft is revolving, and those which require that the rotation of the shaft should cease in order to effect a couple or uncoupling. The former is called a *Clutch*.

Fig. 15 shows a *Box*-coupling, in which the holding power is due to a key. It consists of a box bored out to fit the ends

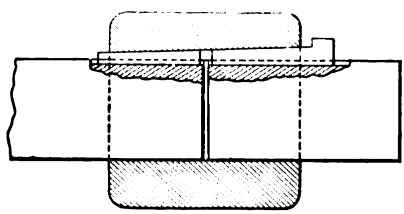


FIG. 15.—Box-coupling.

of the two shafts which are to be connected. It is best to make the key in two pieces, as shown in the cut. The first

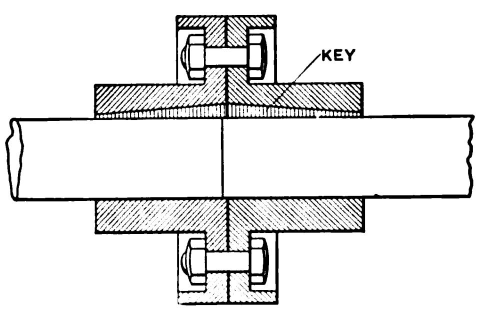


FIG. 16.—Flange-coupling.

half is driven in tight with a drift and afterwards the other part is placed in position. By this method it is not necessary to cut the key and keyway so accurately as when the key is in one piece.

Fig. 16 shows a Cast-iron Flange-coupling. The cast-iron

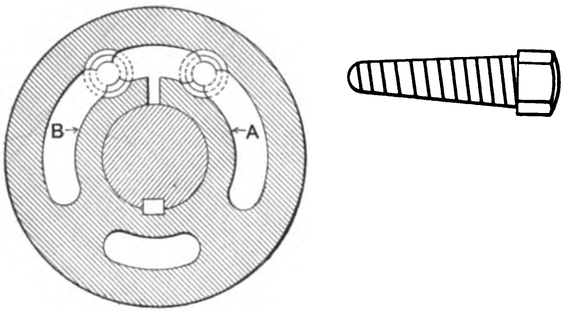


FIG. 17.—Cresson Coupling.

flanges are keyed to the ends of the shafts, and are then bolted together.

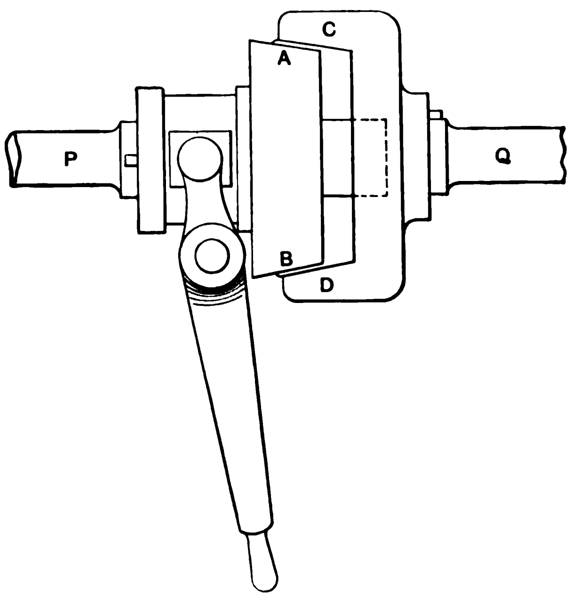


FIG. 18.—Friction-clutch.

In the Cresson compression-coupling, as shown in Fig. 17, the two arms at *A* and *B* are made to clamp the shaft by means of the taper-screws, and the holding effect made still greater by use of the key.

An example of the Clutch is shown in Fig. 18. It is called a friction-clutch. AB is a solid piece of iron of conical shape, which admits of lateral motion on the shaft P from left to right by means of a sliding key. CD is a corresponding piece into which AB may fit and which is fastened to the shaft Q by means of a key. As soon as the one slides into contact with the other, the friction becomes sufficient to engage the two lines of shafting.

PROBLEMS.

1. A cubic foot of wrought iron weighs 480 lbs. Find the weight of a wrought-iron shaft 20 feet in length and 2 inches in diameter.

2. According to the rule adopted by William Sellers & Co., what is the required diameter of a line-shaft which makes 200 revolutions per minute and transmits 125 horse-power? 2

3. What diameter will be required in the above shaft if it is making 600 revolutions per minute?

4. What horse-power will a line-shaft transmit which has a diameter of 2 inches and makes 400 revolutions per minute?

Finding the Work actually transmitted by a Shaft.— The power transmitted by a shaft is measured by means of

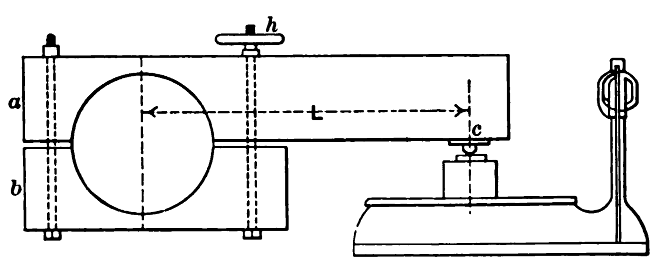


FIG. 19.—Prony Brake.

Dynamometers, of which there are two classes: Absorption Dynamometers and Transmission Dynamometers.

A simple form of the former is shown in the Prony brake, Fig. 19. The shaft whose power is to be measured is clamped by a and b, two blocks of hard wood. At c the block a presses down upon platform-scales, which scales measure the pressure. The load which the shaft may be made to carry is increased by tightening the hand-screw h.

Let $W =$ work of shaft in foot-pounds per minute.
" $P =$ pressure in pounds registered by the scales on the lever-arm of length in feet, L.
" $V =$ velocity in feet per minute of the point c if it were allowed to rotate with the shaft.
" $N =$ revolutions of shaft per minute.

Then

$$W = PV. \qquad (1)$$

$$V = 2\pi LN. \qquad (2)$$

Combining (1) and (2), $W = 2\pi LNP$.

$$\text{H.P.} = \frac{W}{33{,}000}, \quad \text{hence} \quad \text{H.P.} = \frac{2\pi LNP}{33{,}000}.$$

The shaft may be the crank-shaft of a steam-engine, gas-engine, water-motor, or any other shaft.

Dynamometers are used principally in testing motors of different kinds, as will be shown.

Another form of Prony brake is shown in Fig. 20. A rope is wound about a pulley; at one end of this rope known weights are attached; the other end of the rope is fastened to the spring-balance which is securely attached to the floor. The length L in the formula is the radius of the pulley. The factor W is taken as the difference between the known weights and the weight registered on the spring-balance. A Prony brake or any other absorption dynamometer absorbs all the work that may be done by the engine or shaft on which it is used.

A transmission dynamometer measures the power that is being transmitted through a shaft, without absorbing any of it. There are a number of forms of transmission dynamometers,

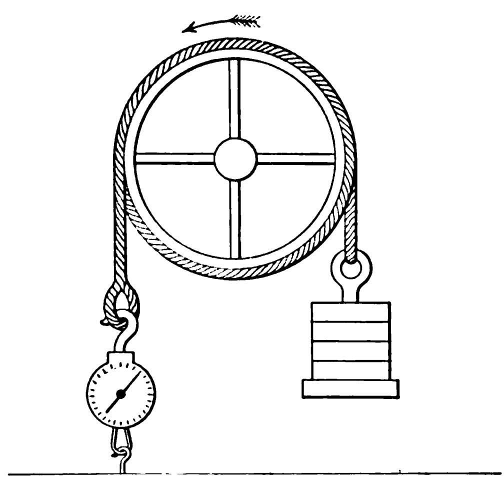


FIG. 20.—Rope Prony Brake.

but they are but very little used at present. For descriptions of them see Flather's book on "Dynamometers" and the Transactions of the American Society of Mechanical Engineers.

CHAPTER III.

BEARINGS.

A BEARING is a support in which a shaft revolves. Generally speaking, the bearing is fixed and the shaft revolves within it; but sometimes the shaft is rigid and the bearing revolves around it; or the bearing and shaft both may be movable as in the case of the connecting-rod end and crank or cross-head pins of an engine.

On account of the great length of line-shafts they must be supported by a greater number of bearings than is necessary for ordinary axles and shafts, for which, as a rule, two bearings are sufficient.

There are three classes to which all bearings may be assigned, viz.: The Journal-bearing, the Pivot-bearing, and

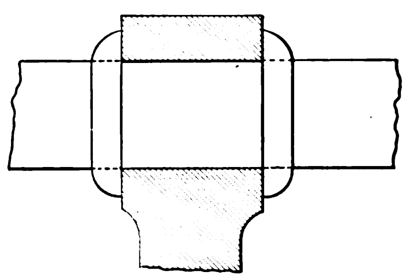


FIG. 21.—Journal-bearing.

the Collar- or Thrust-bearing. If the pressure on the bearing is perpendicular to the axis of the shaft, we have the Journal-bearing.

If the pressure on the bearing is parallel to the axis of the shaft, and the end of the shaft rests on the bearing, it is called a Pivot-bearing. If the pressure on the bearing is parallel to the axis of the shaft and the shaft passes through the bearing we have the Collar-bearing.

The journal-bearings for a line-shaft each support their proportional part of the weight of the shaft, and also the pres-

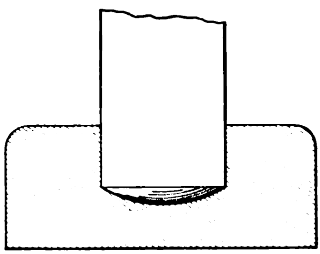


FIG. 22.—Pivot-bearing.

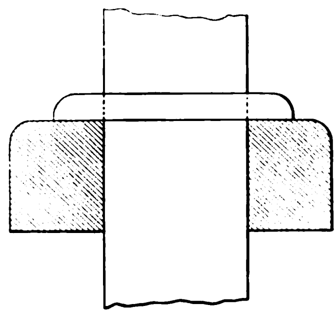


FIG. 23.—Collar-bearing.

sure caused by the pull of the different belts leading off to the machines. In the case of the pivot-bearing the weight of the whole shaft is supported by the one bearing, the shaft being kept in its upright position by other journal-bearings whose

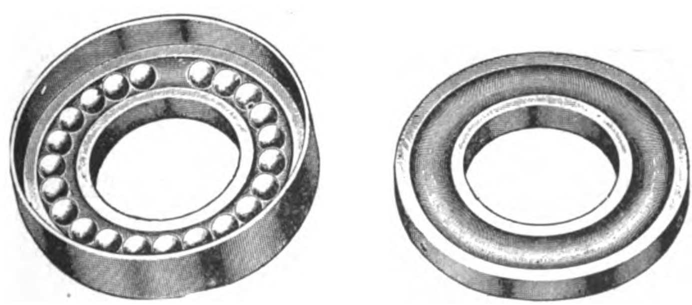


FIG. 24.—Ball-bearing.

axes are vertical. The journal-bearing is the class most generally used and is found in all kinds of machinery.

A notable example of the use of pivot-bearings is found in water-turbines.

The collar-bearing has the advantage that it will not gradually wear and let the shaft drop down, as is the case with the pivot-bearing, because any number of collars may be used on the same shaft, thus increasing the bearing surface and lessening the vertical wear

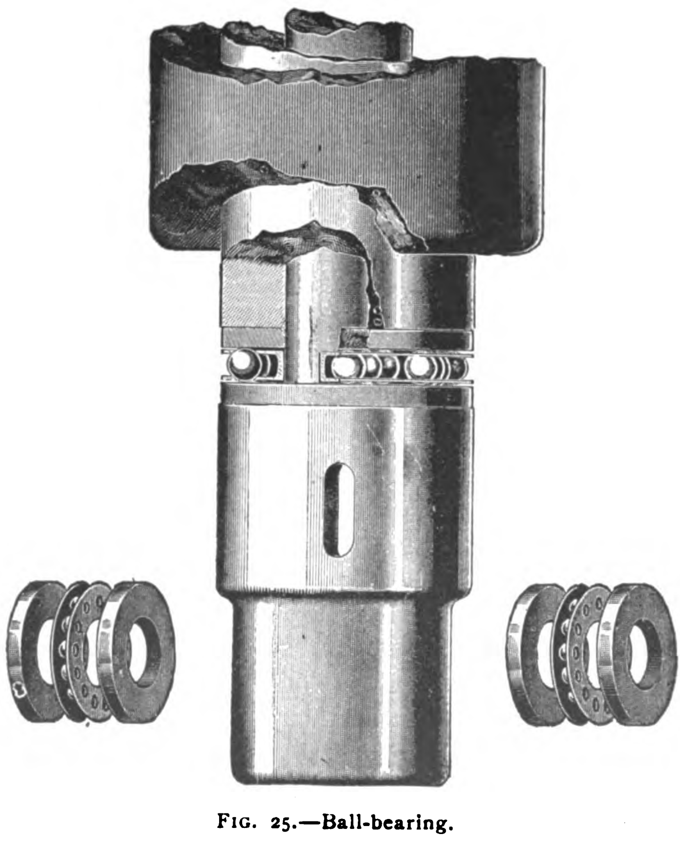


FIG. 25.—Ball-bearing.

Another advantage is that by allowing the shaft to pass on through the bearing, pulleys may be placed beneath it and machinery connected; also the bearing, in this manner, may be kept out of the water, which would be desirable in the case of a water-turbine.

The "ball-bearing" is a contrivance for lessening the frictional resistance by doing away with *sliding* friction and substituting *rolling* friction therefor. It may be used with either of the three classes named above.

The "roller-bearing" is used for purposes similar to those for which the ball-bearing is used, with the difference that the roller-bearing is used for heavier work, as the bearing for a line-shaft. A machine fitted with ball- or roller-bearings will run with a saving of from 25 to 75 per cent of the power required with ordinary bearings, depending upon the nature of the machine. The best form of bearing for any particular use is that which has the necessary strength and at the same time makes the least possible friction.

A "built-up bearing" is one made of two or more pieces, so arranged that it may be adjusted after it is worn so that it will fit the shaft. It is not used much except for large shafts, and generally consists of four pieces, viz.: the bottom, the top or cap, and the two side pieces. The two side pieces are so constructed and arranged that they may be pushed closer to the shaft, after they are worn, by means of wedges as shown in Fig. 26. A bearing which is suspended from a

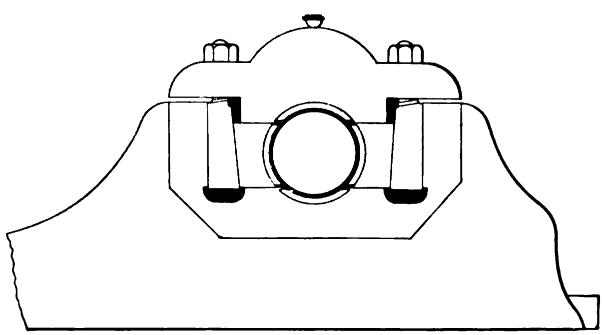


FIG. 26.—Built-up Bearing.

beam is called a "hanger." Likewise a "bracket" is a bearing which supports a shaft along a wall. The bearing is made an easy sliding fit on the shaft in order that the shaft may turn in it. The length of a bearing depends upon the pressure and speed of the shaft which it supports. Suppose the pressure upon a certain bearing, caused by the pull on the shaft, to be 50 lbs. per square inch of bearing surface; then it is not likely to heat because there is not enohug friction. If,

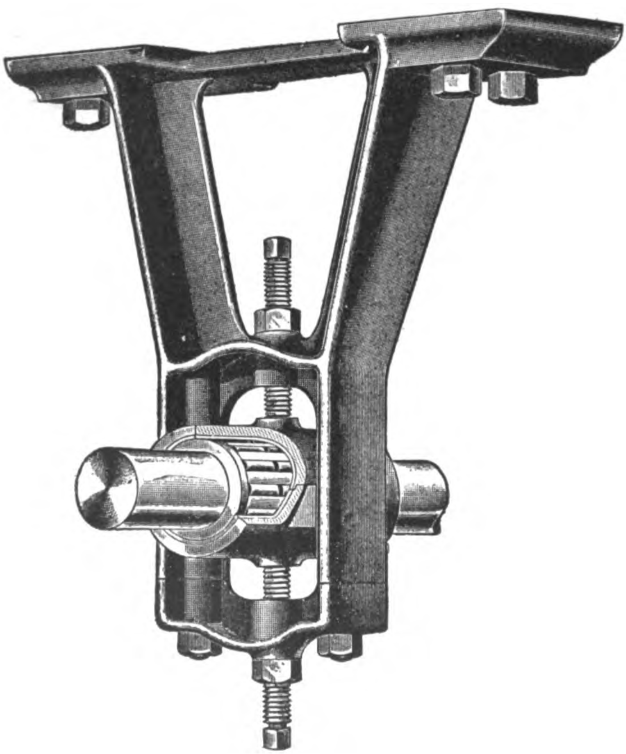


FIG. 27.—Hanger with Roller-bearing.

however, the pressure is increased to 100 lbs. per square inch there may be a heating of the bearing; in which case the bearing should be made twice as long, thus doubling the bearing surface and making the pressure 50 lbs. per square inch again.

Thus it will be seen that the length of the bearing is increased in order to lessen the pressure per square inch of bearing surface by increasing the number of square inches. If there are 10 square inches in a bearing surface and the total pressure is 1000 lbs., then the pressure per square inch is $\frac{1000}{10} = 100$ lbs. Suppose, however, that the bearing is made twice as long, then there will be 10×2 square inches of bearing surface, and the pressure per square inch will be $\frac{1000}{20} = 50$.

By the area of a bearing is meant the area of its projection on a plane perpendicular to the direction of the pressure. It is sometimes called the projected area or bearing surface.

For a journal of diameter D and length L, the bearing surface would be DL.

For a pivot-bearing, the bearing surface would be the area of the cross-section of the shaft, $.7854D^2$.

For a collar-bearing the bearing surface would be the area of the collar, $.7854(D_1^2 - D^2)$, D being the diameter of the collar.

Let R be the total load on a journal-bearing and p the pressure per square inch of bearing surface. Then $p = \frac{R}{DL}$; that is, p equals the total load divided by the number of square inches. Likewise for a pivot-bearing $p = \frac{R}{.7854D^2}$. The allowable pressure per square inch p varies with different speeds, becoming less with increasing speeds; but for ordinary journal-bearings is not more than 200 lbs.; for railway-axles 160 to 300 and for collar-bearings 50 to 70 lbs.

Methods of applying Oil to Bearings.—In order that a bearing may not heat, it is important to keep it well lubricated. Generally it is best to feed the oil to the bearing by the use of some arrangement which will deliver it automatically, and at regular intervals.

Fig. 28 shows what is called a *Sight-feed oiler*. This is an arrangement by which the oil is fed to the bearing drop by drop, and is made adjustable so that the oil may be fed as fast or slow as may be desired. By this means a thin oil may be used on large bearings where heavy oil or grease would otherwise have to be used, it being so arranged that the oil is resupplied to the bearing as fast as it is forced out by the heavy pressure.

Fig. 29 shows what is called the *Compression grease-cup*, which is used to feed tallow or other heavy greases to bearings.

FIG. 28.—Sight feed Oiler.

FIG. 29.—Compression Grease-cup.

The tension in the spring keeps the grease pressed down into the hole in the cap of the bearing. Nearly all bearings have grooves cut in them to serve as reservoirs for surplus oil which is put in, and which would run out if the groove were not there.

PROBLEMS.

1. What is the bearing surface in square inches of a journal-bearing the length of which is 6 inches and diameter 4 inches?

2. What is the bearing surface of a pivot-bearing, the diameter of the shaft being 6 inches?

3. What is the bearing surface of a collar-bearing having four collars, the diameter of the shaft being 6 inches and the diameter of the collars being 12 inches?

4. What is the pressure per square inch upon the bearing surface of a journal, the total load being 2400 pounds, the diameter of the bearing 4 inches, and the length 6 inches?

5. What load may be carried by a journal-bearing 2 inches in diameter and 8 inches in length, allowing a pressure p per square inch on the bearing of 200 pounds?

6. A shaft is 3 inches in diameter and as it runs at a high speed can run safely with a load of 400 pounds per square inch. What should be the length of the bearing if the total load is 6000 pounds?

7. Suppose that the tension on the tight side of a belt is 3200 pounds and the tension on the slack side 1600 pounds; what will be the total transverse force exerted upon the shaft to which it transmits motion?

CHAPTER IV.

FRICTION AND LUBRICATION OF BEARINGS.

Friction is a force which acts between two bodies at their surface of contact so as to resist their sliding on each other, and which depends on the force with which they are pressed together (Rankine). Friction is of three kinds: sliding and rolling friction, which act with solids, and fluid friction, which acts with liquids and gases.

On every surface there exists microscopic irregularities which offer a resistance to the passage of one surface over another. A lubricant introduced between the two surfaces tends to fill the irregular spaces, and causes the surfaces to approach more nearly to being perfectly smooth. The nearer a surface approaches to being smooth, the less frictional resistance will it offer to the passage of a body over it.

The Coefficient of Friction of a body is the ratio of the force required to slide the body along a horizontal plane sur-

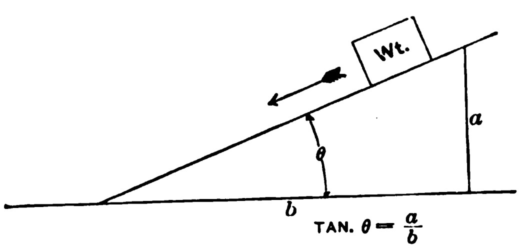


FIG. 30.

face, to the weight of the body. It is usually designated by the letter f.

The Angle of Repose of a body is the angle of inclination to the horizontal of a plane, on which the body will just overcome its tendency to slide. The angle of repose is usually denoted by the Greek letter θ. The coefficient of friction is equal to the tangent of the angle of repose; that is $f = \tan \theta$.

Morin's Laws of Friction.—In 1831 Gen. Morin started a series of experiments which extended over about three years. His results were embodied in the following laws:

1. The friction between two bodies is directly proportional to the pressure; that is, the coefficient of friction is constant for all pressures.
2. The coefficient and amount of friction, pressure being the same, is independent of the areas in contact.
3. The coefficient of friction is independent of velocity.

For about fifty years these laws were accepted without question by engineers. Since about 1880, however, experiments by Thurston, Tower, and others have shown that they are in error; in fact, not even approximately true. The later experimenters have found that with ordinary machinery, friction is not directly proportional to the pressure, is not independent of velocity; and that the coefficients determined by Morin were about ten times too high for modern machinery.

Prof. J. E. Denton, in defence of the laws, claims that Morin made no such special preparations for his tests as are made to-day; that he did not have his running surfaces so thoroughly lubricated as in modern tests, by running the bearing in oil, or by means of an oil-pad. He states that the conditions under which Morin worked were about the same as exist in a journal lubricated with an ordinary restricted-feed oil-cup. He also states that there is an additional resistance in bearings, due to the viscosity of the oil; hence, Morin's laws will not apply to very light pressures.

General Morin himself states that the laws did not pretend to be mathematically exact, but only close approximations to the truth. It is probable that the laws may be safely used for work in connection with ordinary journals, not specially well lubricated, and running under moderately heavy pressure.

The experiments of Prof. Thurston showed that the coefficient of friction varied with different conditions, such as the nature of the surfaces, the quality of the lubricant, etc. He has determined the coefficients for a number of different lubricants, surfaces of contact, velocities, etc.

The following table of Prof. Thurston gives the coefficients of friction for different oils, under varying pressures, at a constant velocity of 720 feet per minute, the journal being of cast iron and the bearing of bronze:

Pressures, lbs. per square inch......	8	16	32	48
Oils.	Coefficients of Friction.			
Sperm, lard, neat's-foot, etc..	.159 to .250	.138 to .192	.086 to .141	.077 to .144
Olive, cottonseed, rape, etc..	.160 " .283	.107 " .245	.101 " .168	.079 " .131
Cod and menhaden	.248 " .278	.124 " .167	.097 " .102	.081 " .122
Mineral oils	.154 " .261	.145 " .233	.086 " .178	.094 " .222

The oil was fed intermittently through an oil-hole and at a temperature of 70° F.

The following laws are derived by Mr. John Goodman, from the experiments of Thurston, Tower, and others.

1. The coefficient of friction, with the surfaces well lubricated, is from one sixth to one tenth that for dry or scantily lubricated surfaces.

2. The coefficient of friction for moderate pressures and speeds varies approximately *inversely as the normal pressure;* the frictional resistance varies as the *area in contact*, *the normal pressure remaining constant.*

3. At very low journal-speeds the coefficient of friction is abnormally high, but as the speed increases from about 10 to 100 feet per minute, the friction diminishes, and again rises

when that speed is exceeded, varying approximately *as the square root of the speed.*

4. The coefficient of friction varies approximately, *inversely as the temperature* within certain limits; viz., just before abrasion takes place.

It has been found by experiment that metals of the same kind running upon each other sometimes cause more friction than do metals of different kinds; the probable reason being that there is a difference in microscopic surface structure; the minute friction points do not so accurately correspond and engage each other. This is not always true, however. A hard-steel shaft, running in a hard-steel bearing, properly polished, has very low friction.

The two following tables, the first by Rankine, the second by Morin, give the comparative value of different bearing surfaces.

FRICTION OF MOTION.

No.	Surfaces.	θ.	f.	$1 \div f$.
1	Wood on wood, dry.......	14° to 26½°	.25 to .5	4 to 2
2	" " " soaped....	11½° to 2°	.2 to .04	5 to 25
3	Metals on oak, dry........	26½° to 31°	.5 to .6	2 to 1.67
4	" " " wet........	13½° to 14°	.24 to .26	4.17 to 3.85
5	" " " soapy.....	11½°	.2	5
6	" " elm, dry........	11½° to 14°	.2 to .25	5 to 4
7	Hemp on oak, dry.......	28°	.53	1.89
8	" " " wet........	18½°	.33	3
9	Leather on oak...........	15° to 19½°	.27 to .38	3.7 to 2.86
10	" " metals, dry....	29½°	.56	1.79
11	" " " wet....	20°	.36	2.78
12	" " " greasy.	13°	.23	4.35
13	" " " oily....	8½°	.15	6.67
14	Metals on metals, dry.....	8½° to 11°	.15 to .2	6.67 to 5
15	" " " wet.....	16½°	.3	3.33
16	Smooth surfaces, occasionally greased............	4° to 4½°	.07 to .08	14.3 to 12.5
17	Smooth surfaces, continuously greased..........	3°	.05	20
18	Smooth surfaces, best results....................	1¾° to 2°	.03 to .036	
19	Bronze on lignum vitæ, constantly wet..........	3°	.05 ?	

COEFFICIENTS OF FRICTION OF JOURNALS (MORIN).

Material.	Unguent.	Lubrication. Intermittent.	Lubrication. Continuous.
Cast iron on cast iron	Oil, lard, tallow.	.07 to .08	.03 to .054
	Unctuous and wet.	.14	
Cast iron on bronze	Oil, lard, tallow.	.07 to .08	.03 to .054
	Unctuous and wet.	.16	
Cast iron on lignum vitæ	Oil, lard.	....	.09
Wrought iron on cast iron. / " " " bronze	Oil, lard, tallow.	.07 to .08	.03 to .054
Iron on lignum vitæ	Oil, lard.	.11	
	Unctuous.	.19	
Bronze on bronze	Olive-oil.	.10	
	Lard.	.09	

Cast-iron Bearings are found to work very smoothly under light duty provided the lubrication is perfect and the surfaces can be kept practically free from dust and grit. The reason for this is that cast iron forms a hard surface-skin when rubbed under light pressure, and so long as the pressure is not enough to cut into this skin, it will make a very bright and smooth wearing surface. Another point in favor of the cast-iron bearing is that it will hold oil better and longer than steel, brass, or wrought iron. This may be proven by trying to clean the oil from bearings made of these metals, when it will be seen that it is almost impossible to clean the cast-iron bearings, while it is comparatively easy to clean the others.

Good examples of the superiority of cast-iron are found in the use of piston-rings and slide-valves. It has been found that cast-iron piston-rings work better in a cast-iron cylinder than those of any other metal. Where the seat of a slide-valve is of cast iron, a cast-iron valve will cause less wear, either to itself or the seat, than one of wrought iron, steel, or brass.

Babbitt Metal.—This is the name generally given to certain soft compounds used in bearings. It is an alloy made of tin, antimony, and copper, mixed in different proportions, depending on the kind of bearing surface desired, whether hard or soft.

Babbitt is used in bearings, because such a bearing is less liable to "overheat" than a bearing of brass or bronze. A bearing of Babbitt will also permit of abrasion or crushing without excessive increase of friction. The various kinds of Babbitt have about the same friction. If the wearing surfaces are kept in good order, the friction will depend not so much on the metal as on the lubricant.

In order to use Babbitt metal the body of the bearing must be made considerably larger than the shaft, and then a bearing which fits the shaft perfectly is made by pouring the melted Babbitt around the shaft while it is in the proper position within the iron part of the bearing.

Qualities of Good Lubricants.—Good lubricants should have the following qualities: 1. Sufficient body (viscosity) to keep the surfaces free from contact under the greatest pressure. 2. The greatest fluidity consistent with the foregoing condition. 3. Power to resist oxidation or the action of the atmosphere. 4. Freedom from corrosive action upon the metals with which they come in contact.

Thus it will be seen that several conditions must be considered in the selection of the proper kind of lubricant for a bearing. The main consideration is the amount of pressure.

For a great pressure a heavy viscous oil, and for light pressure a more fluid oil should be used.

Oil which is suitable for heavy shafting is not suitable for small spindles such as are used in clocks, watches, etc.

Also light sperm-oil is equally unsuited for heavy pressures like that in a car-journal. For very heavy bearings such as those of rolling-mills for rolling iron and steel, tallow and other solid lubricants are used. It is said that in the Waltham Watch Company nineteen different kinds of oil are used, so varied is their machinery.

There are three kinds of lubricating oils, viz.: *mineral*, *vegetable*, and *animal* oils; and besides these, combinations of two or more of them. Soap is a constituent of railway-grease; graphite and steatite, or soapstone, are sometimes used for heavy machinery.

The following list shows the best purpose to which the various lubricants may be applied:

For steam-cylinders: Heavy mineral oils.

For ordinary machinery: Lard-oil, tallow-oil, and heavy mineral or vegetable oils.

For very great pressures with low speed: Graphite, soapstone, etc.

For heavy pressures with low speed: Tallow, lard, grease, etc.

For heavy pressures with high speed: Sperm-oil, castor-oil, mineral oils.

For light pressures with high speed: Sperm, refined petroleum, cottonseed, rape, and olive oils.

For watches, clocks, etc.: Light mineral oils, clarified sperm, neat's-foot, olive, and porpoise.

Sperm, lard, olive, and cottonseed oils may be mixed with minerals oils. Sperm makes the best mixture. The value of the others in mixtures is in the order given.

It should be stated that in all cases, where possible, a mineral oil of suitable body should be selected. If fatty vegetable oils are used in connection with high temperatures, such as exist in the cylinder of a steam-engine, the oil will be decomposed, forming fatty acids, which in the presence of the metal will form metallic soaps, and may cause great damage to the machinery.

Graphite is a solid lubricant; it is most used in the form of a powder. It will work well either used alone or in connection with various oils. It is principally used in connection with heavy pressures, but Thurston states that it may be used to advantage either for light or heavy pressures, especially when mixed with certain oils. It is rather difficult to introduce into bearings, being a solid. Mixing it with water or oil will facilitate its use, however.

CHAPTER V.

FRICTION-WHEELS.

THE use of friction-wheels gives the simplest method of transmitting motion from one shaft to another by means of wheels, the belt or chain being made unnecessary in this case. The transmission of power is often effected by pressing the two wheels together at their circumferences, but sometimes the circumference of one wheel presses on the disk of the other. The transmitting power is due to the friction of the wheels upon each other. The materials used in their construction must be such that the coefficients of friction will be as great as possible, so that the pressure between the two wheels will not have to be abnormally great. For the above reasons wood is often made to work with wood, or wood with cast iron. Sometimes the perimeters of the wheels are covered with leather. Small friction-wheels are sometimes made of solid disks of leather, and sometimes of similar disks made of coarse paper and compressed into the proper form and stability by the use of hydraulic pressure.

In other cases grooves are cut in the circumference of the wheels. The projections on one wheel are forced into the grooves in the other. In this manner a greater bearing-surface is obtained than ordinarily. A pair of such wheels is shown in Fig. 31.

The great objection to the use of friction-wheels is that in order to produce an adequate transmitting friction at the surface of the wheels an excessive quantity of friction and wear is produced in the bearings. Another objection to the use of friction-wheels is that the bearings of the two wheels cannot both be fixed, because it is by moving the two wheels closer together that the pressure can be increased between the two,

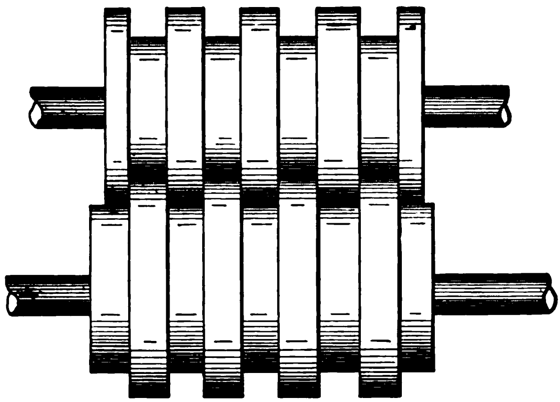


FIG. 31.—Friction-wheels.

and this can only be done by moving the bearings nearer to each other. On this account the friction-wheels are used comparatively little.

It is thought best, however, to discuss them here for the reàson that the principles of mechanism which are shown in

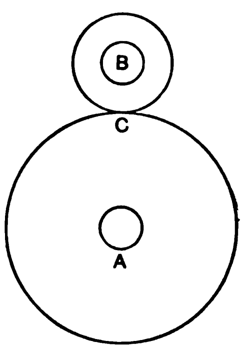


FIG. 32.

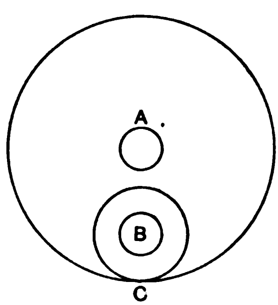


FIG. 33.

their discussion are applicable also to other circular gearings, especially to toothed gears. In Fig. 32 we have two friction-wheels in which the contact is between the axes, and in Fig. 33 two friction-wheels in which the contact is outside of the axes, the axes being parallel in each case. Again in Fig. 34

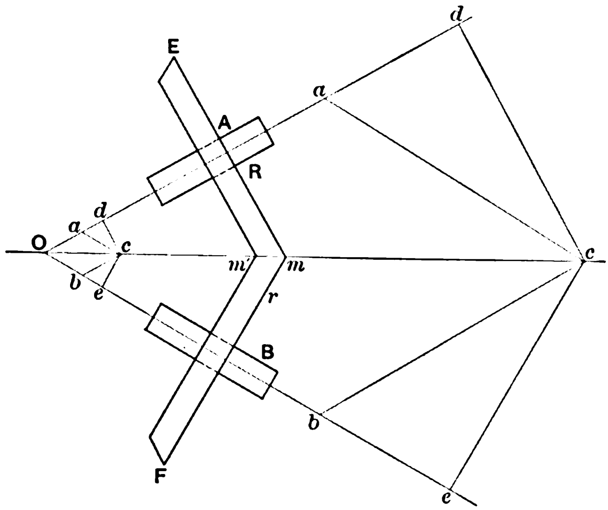


Fig. 34.

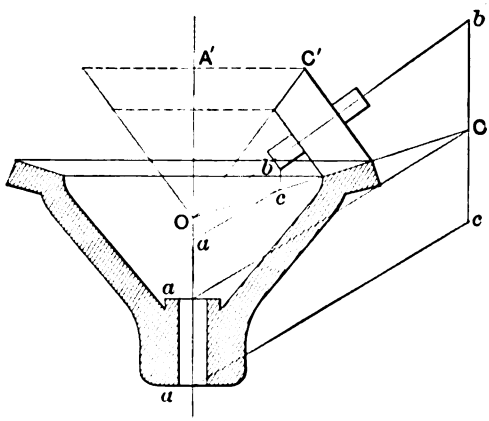


Fig. 35.

we have a case in which the axes meet, the contact being between the axes. Fig. 35 shows intersecting axes with the contact outside the axes.

The ratio of the rotative speeds to the radii of the wheels in any of the above cases may be obtained graphically,* and an example is here given of each. In Fig. 36 let R represent the radius of the large wheel and r the radius of the small wheel, V the velocity of the large wheel, and v the velocity of the small wheel.

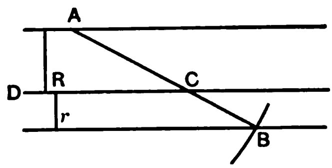


FIG. 36.

The velocities of the two wheels geared with each other are to each other inversely as their radii; hence $\frac{V}{v} = \frac{r}{R}$. Suppose it is desired that the two fixed axes A and B be connected by wheels of such radii that their velocities, or number of revolutions, shall be to each other as 2 to 5. As the sum of the radii equals the distance between the axes, by addition, we have $5 + 2 = 7$. Draw any line AB, seven units in length, which length should be greater than the distance between the axes. Take the distance AC equal to five units and CB equal to two units. Through C draw CD parallel to the axes. Perpendiculars R and r from any point of the axes to the line of contact CD will be the required radii.

Contact outside of the Axes.—In this case the distance between the two axes must be equal to the difference $R - r$ of the two radii. In Fig. 37 take A and B again for the axes. It is desired to find the point C outside the axes A and B. Taking the same velocity ratio, 2 to 5, we have $5 - 2 = 3$. Draw a line AB, in length three units, the difference $(R - r)$, and greater than the distance between the axes. Produce this line to C, making BC equal to two units. Through C, draw CD parallel to the axes. Perpendiculars from CD to the axes will be the required radii.

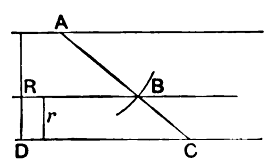


FIG. 37.

These two constructions may be dispensed with, if the dis-

* The constructions given on pages 44 and 45 are taken from Robinson's "Principles of Mechanism."

tance between the axes is divisible into a convenient dimension by the sum or difference of the velocity ratio as the case may be. If the distance is so divisible, the divisions may be made on the perpendiculars to the axes themselves.

Axes Meeting—Contact between Axes.—In Fig. 34 take AO and BO, intersecting at O as the axes, which are to be connected by friction-wheels so that their velocities of rotation shall be as 2 to 5, as in the previous examples. Lay off on OA from O, the distance $Oa = 2$, the relative velocity of A, and on OB the relative velocity of B, which is $Ob = 5$.

Complete the parallelogram $Oacb$, thus finding the point c. Through O and c draw the line cO. This line will be the line of contact of the two wheels. Any number of wheels, varying in size but with the same velocity ratio, may be constructed upon it as a line of contact. Thus the diameter FBm may be drawn and its mate will be mRE, r and R being the radii.

The same construction may be applied to a case in which the axes meet, but with contact outside of them. The construction is shown in Fig. 35, the lettering being the same as for the preceding case. No further explanation is necessary.

It is noticeable that friction-wheels, as well as all other gear-wheels, always work in pairs. The one which imparts motion is called the Driver, and the one which receives motion is called the Driven wheel, or Follower. As has already been stated, the ratio of the revolutions of two friction-wheels in gear is inversely as the ratio of their radii. Let $N =$ number of revolutions of the driver, $n =$ revolutions of driven, $R =$ radius of driver, and $r =$ radius of driven; then the equation, $N \times R = n \times r$, shows the relation of velocity and radii.

PROBLEMS.

1. Two parallel shafts are 18 inches between centres. Find graphically the radii of the two friction-wheels such that one shaft will make 4 revolutions to 5 of the other, the contact coming between the axes.

2. Two parallel shafts are 8 inches between centres. Find

graphically the radii of two friction-wheels such that one may make 5 revolutions to 6 of the other, the contact coming outside the axes.

3. The axes of two shafts meet each other at an angle of 45 degrees. Find graphically the size of the conical friction-wheels such that one may make 2 revolutions to 3 of the other, the contact being between the axes.

4. In Fig. 32 let the radius of the wheel $A = 12$ inches and the radius of $B = 3$ inches. If A makes 100 revolutions per minute, how many revolutions will B make per minute?

5. In Fig. 33 let the radius of the wheel $A = 16$ inches and the radius of $B = 2$ inches. If B makes 400 revolutions per minute, how many will A make?

6. In Fig. 32 B and A make 1200 and 200 revolutions per minute respectively, and A is 18 inches in diameter. What must be the diameter of B?

7. In Fig. 33 B and A make 900 and 100 revolutions per minute respectively, and B is 2 inches in diameter. Find the diameter of A.

CHAPTER VI.

PULLEYS.

PULLEYS for the transmission of power by belts are divided into two classes: the *solid* and the *split* pulley. The solid pulley may be cast solid, or the hub and arms may be cast and a rim of wrought iron or steel riveted on. The latter makes a strong and light pulley.

The split pulley may be made of iron or wood. In either case the two halves are bolted together tight enough to clamp the shaft. When the wood pulley is used a bushing, made of two or more pieces of wood, is put around the shaft and into the eye or hub of the pulley in order to make a tight fit. Pulleys are sometimes made without arms, but with a solid web instead.

Owing to the fact that very large castings sometimes cool unequally, and consequently cause shrinkage, rendering breakage liable, it is customary to cast pulleys of larger diameter than 6 feet in two or more parts. This lessens the liability to damage by shrinkage, and at the same time makes the pulley easier to handle.

Pulleys of small diameter, that is, up to 3 feet, are usually fastened on the shaft by means of set-screws. Pulleys of larger diameters than 3 feet are usually fastened by keys, and sometimes by both keys and set-screws. In either of these cases, if the bore in the hub of the pulley is larger than the shaft, which is generally the case, in order that the pulley may be slipped on with ease, the pulley will be out of balance when the pressure of the set-screw is placed against it.

Especially in the case of pulleys running at high speeds it is often necessary to balance the rim by some means, in order to counteract this objectionable feature. This may be done by attaching a small iron weight to the inner side of the rim opposite the heavy side, as in Fig. 38. When the pulley is unbalanced it causes the shaft to vibrate when rapidly revolv-

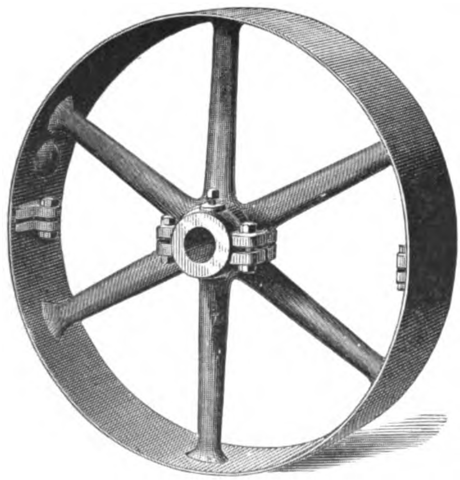


FIG. 38.—Split Pulley.

ing, and this causes unequal strains in the belt at different parts of the revolution, not only injuring the shaft and bearing, but stretching and wearing the belt.

A pulley which transmits motion to a belt is called a driver. A pulley which receives motion from a belt is called a driven pulley. A pulley is said to be *crowned* when the face of the rim is curved, the largest diameter being at the middle. This is done in order that the belt may be kept from running off, because the belt tends to run where it is kept tightest. The amount of convexity, in practice, varies from $\frac{1}{16}$ to $\frac{3}{16}$ inch per foot of width of pulley-face.

A Fast pulley is one which is fastened to the shaft and transmits motion from one pulley to another by means of belts and ropes.

A Loose pulley runs free on the shaft in order to receive the belt, and at the same time transmit no motion to the shaft to which it is attached. It is generally used on countershafts for throwing machines in or out of gear, and has no convexity, or crown, on the face, in order that the belt may be moved aside easily.

A Cone or *Stepped pulley* has a number of faces or grooves of different diameters whereby the speed of a machine may be changed; examples of cone pulleys may be seen on wood- or

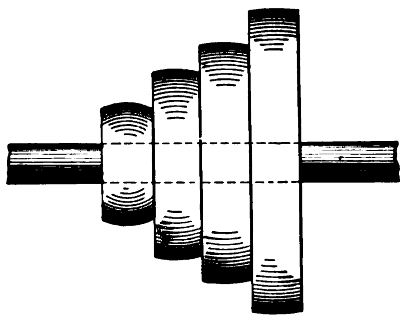


FIG. 39.—Cone Pulley.

engine-lathes. These cone pulleys always work in pairs, one on the machine which is operated by the belt and one on the countershaft. A method of designing cone pulleys is given at the end of this chapter.

Conical pulleys are used where it is desired that a uniform speed be changed to a variable speed, or for changing a variable speed to a uniform speed. They work in pairs like the cone or stepped pulleys. An example may be found in cotton-factories where it is necessary to give the bobbins a gradually increasing speed on account of the unwinding of the thread from the bobbin.

A modification of belted conical pulleys is shown in Fig. 40, which operates upon the principle of friction-gearing. Motion being desired, the ring of leather is moved endwise by a suitable shifting device, and by reason of the ring filling up the space between the pulleys, motion is imparted from the driving pulley to the driven pulley, and so to the machinery.

In all the cases described above, the pulleys have been circular in perimeter, but non-circular or cam-shaped pulleys

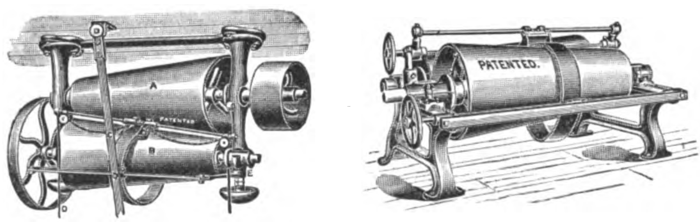


FIG. 40.—Evans' Friction Cone Pulley.

are used occasionally for producing special movements in which it is required that the velocity of the driven should be variable at different parts of the revolution. See Fig. 41.

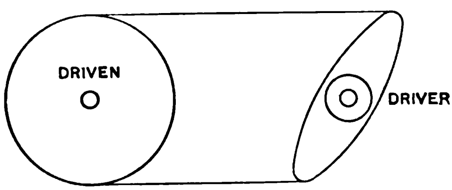


FIG. 41.—Variable Velocity Pulley.

An example of a pulley in which the axis of the pulley does not coincide with the axis of rotation is found in the foot-power lathe-treadle, shown in Fig. 42, which is self-explanatory.

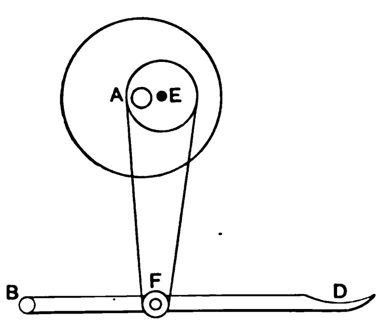


FIG. 42.—Lathe-treadle.

The driving power of a belt and pulley is increased by making the face of the pulley smoother. The holding power of a belt does not depend upon the friction between the belt and the pulley-face but upon the adhesive force between the two. The adhesion depends on the intimacy of contact, and as smoothness of the two surfaces produces a contact between a larger number of particles, it is plain that the above statement is true. A very smooth contact also produces a partial vacuum between the belt and the pulley which increases the tractive force of the belt.

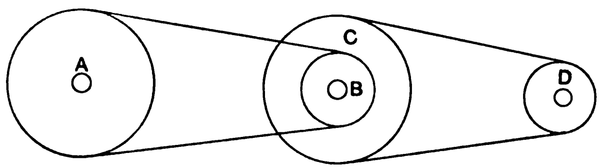


FIG. 43.—Countershaft.

If there is no slip in the belt-connector, the revolutions of two connected pulleys will vary inversely as their diameters, and the relation of velocities to the diameters of the driver and driven is the same as that for friction-wheels.

Let D = diameter of driver;
d = diameter of driven;
R = revolutions of driver;
r = revolutions of driven.

Then the following equations express the relations of the two pulleys with regard to the number of revolutions and diameters:

$$D \times R = d \times r.$$

$$R = \frac{d \times r}{D},$$

$$D = \frac{d \times r}{R},$$

$$r = \frac{D \times R}{d},$$

$$d = \frac{D \times R}{r}.$$

Where a system of pulleys is used, the following rule shortens the calculation:

RULE.—*The revolutions of the first driver multiplied by the continued product of the diameters of the drivers is equal to the revolutions of the last driven multiplied by the continued product of the diameters of the driven pulleys.*

Design of Cone Pulleys.

The following method of designing cone pulleys is taken from Kent's "Mechanical Engineers' Pocketbook." Let EF, Fig. 44, be the distance between the centres of the

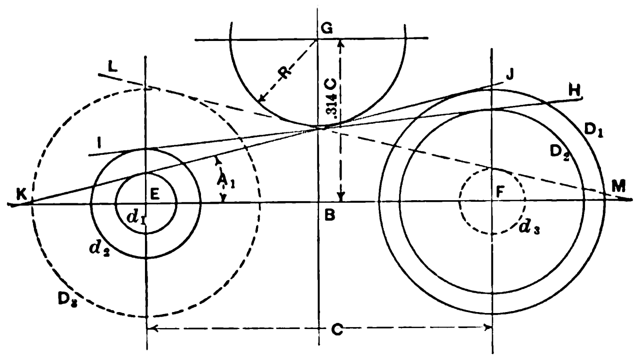


FIG. 44.—Design of Cone Pulleys.

pulleys. Draw the circles D_1 and d_1, representing the first pair of pulleys. The diameters of this first pair can be determined by given conditions. Draw JK tangent to the circles D_1 and d_1. At B, the middle point of EF, erect a perpendicular BG. The length of BG should be .314 EF. With G as a centre draw a tangent circle to JK. The belt-line of any other pair of pulleys must be tangent to this circle. Take any line as HI or LM. The circles about E and F drawn tangent to it will be one of the required pairs.

PROBLEMS.

1. In Fig. 43 A makes 100 revolutions per minute, and the diameters of A, B, C, and D are 16, 8, 20, and 12 inches respectively. Find the number of revolutions made by D.

2. In the same figure let A represent the drive-wheel of an engine

geared by belts to a dynamo-pulley D, and B and C pulleys on a counter-shaft. If the dynamo makes 2000 revolutions per minute and the diameters of A, B, C, and D are 10 feet, 30 inches, 5 feet, 12 inches, respectively, how many revolutions per minute should the engine-pulley make?

3. In the same figure suppose that A makes 100 revolutions per minute, and that D is to make 3000. The diameters of A, B, and C are 10 feet, 30 inches, and 5 feet respectively. What is the diameter of the pulley to be put on the dynamo?

* The circumferential speed of any revolving wheel is the distance, in feet, passed through by a point in its circumference per minute, and is equal to the number of revolutions per minute multiplied by the circumference of the wheel in feet.

4. What is the circumferential speed of D in Problem 1?

5. What is the circumferential speed of A in Problem 1?

6. Suppose that D in Fig. 43 be made the drive-wheel of an engine making 150 revolutions per minute, and that A, B, and D are 20, 4, and 20 inches in diameter respectively. What diameter should be given C in order to give A a circumferential speed of 1000?

---

* The circumferential speed of the pulley equals the velocity of its belt, there being no slip.

CHAPTER VII.

BELT- AND ROPE-GEARING.

Belt-gearing includes all appliances concerned in transmitting motion in the manner of a belt and pulley; such as belts, bands, or chains running on pulleys or sprocket-wheels with continuous motion; or with limited motion, as where a rope, strap, or chain passes partly, wholly, or several times around wheels to which the ends are made fast, as in the case of the windlass. Belt-gearing is very desirable especially where a uniform velocity of driver and driven wheel is not required, because of its noiseless running, lightness, and simplicity of construction as compared with tooth-gearing, link-gearing, etc. In cases where a mathematical relation must be preserved between the speed of the driving pulley and the speed of the driven pulley, the sprocket-wheel, which permits no slipping, is used. Belts or ropes are apt to slip more or less. The driving power of a belt depends mainly upon the tightness with which it is stretched around the pulley; also upon the arc of contact, and upon the condition of the belt and pulley-face with regard to smoothness of surface.

Leather and rubber are the two kinds of belting most generally used, and of the two leather will usually last longer.

Leather belts may be single or of any number of layers cemented together. The object in increasing the thickness is to increase the strength without increasing the width. The best leather belts are made of oak-tanned leather curried with the use of tallow and cod-oil. Such belts have been known to continue in use thirty to forty years, when used as simple driving-belts, transmitting a proper amount of power and being given suitable care. The hair side of the belt should be run next to the pulley. It has been found by experiment that the hair or grain side is the weaker. The hair side will also crack much easier than the flesh side. If the grain side is shaved thin and stretched a little a large number of holes can be seen, showing that the weakness is probably due to the hair having had root on that side. Again the hair side is smoother, and will hug the face of the pulley better, and this is a condition which promotes the tractive force of a belt. When a belt is bent around a pulley, the side of the belt farthest away from the pulley is stretched, while the side next to the pulley is compressed. It is plain, then, that the flesh or stronger side should be on the outer side, which is stretched, while the hair or weaker side should be placed against the pulley, where there is not so much strain.

The safe working tension of a laced belt is 250 to 350 lbs. per square inch of cross-section.

When the driving and driven pulleys, Fig. 45, are at rest

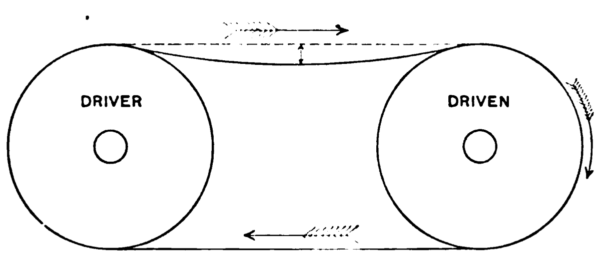


FIG. 45.—Sag of Belt.

the tensions in the two halves of the belt will be the same; but when the driver rotates in the direction of the arrow, the tension on the tight side will be increased, and the tension on the slack side will be diminished. Let T_1 be the tension on the tight side, and T_2 the tension on the slack side, and P the effective force on the circumference of the pulley; then, $P = T_1 - T_2$. Let V be the velocity of the belt in feet per minute; then $VP =$ the number of foot-pounds of work done per minute and H.P. $= \frac{VP}{33,000}$. If the belt passes over a pulley D feet in diameter, that makes N revolutions per minute, then $V = 3.1416 \times D \times N$. Substituting this value of V in the above equation, we have,

$$\text{H.P.} = \frac{3.1416 \times D \times N \times P}{33000}.$$

A short and simple rule for determining the horse-power that will be transmitted by a belt is, that a single leather belt, one inch wide, travelling 1000 feet per minute, will transmit one horse-power. A double belt one inch wide travelling 600 feet per minute will transmit one horse-power. The working strain in this case is 33 lbs. per inch of width.

Different writers give other figures for the speed of belts necessary to transmit one horse-power. The rule above given, however, is a very safe one with which to work.

Mr. F. W. Taylor describes in the Transactions of the American Society of Mechanical Engineers a series of experiments on belting, extending over nine years. His results give rise to principles which, if adopted, would entail heavier expense than is usual in installations of belting. The rules are on this account not much used. Among other things he recommends the splicing and cementing of belts in preference to lacings; the use of narrow, thick belts, even on small pulleys, in preference to wide, thin belts; and that the thickness of belts should be increased as they are made wider.

A belt running at a very high speed will have its effective driving tension diminished by the tension due to centrifugal force. If we let W be the weight of belt per foot of length, V the velocity of the belt in feet per second, and g the acceleration due to gravity $= 32.2$, then the centrifugal tension Tc may be found as follows:

$$Tc = \frac{WV^2}{g}.$$

The weight of leather per cubic foot being 56 lbs., the weight of one square foot one inch thick will be .388 lbs. Then

$$Tc = \frac{.388\,V^2}{32.2} = .012\,V^2.$$

Subtracting this centrifugal tension from the tension on the tight side will give the effective driving tension.

Rubber belting is a combination of rubber and cotton-duck. There are many qualities manufactured, the difference in strength depending upon the quality of the cotton-duck used, as the tensile strength of the belt is in this fabric. The belt is made up of plies of cotton cemented together with rubber, and the entire surface covered with rubber.

The advantages claimed for rubber belting are: perfect uniformity in width and thickness; it is not affected seriously by excessive degrees of heat or cold; it is especially adapted for use in wet or damp places, or where it is exposed to the action of steam; and it is less liable to slip on the pulley.

A comparatively new kind of belt is now made, called the leather link-belt, which is considerably used for heavy work. It consists of many small leather links fastened together with iron rods, as shown in the illustration, Fig. 46. The rods run through the holes in the links and are as long as the width of the belt. It is devoid of the usual great stiffness which is found in ordinary belts and it easily adapts itself to the contour of the pulley, no matter how heavy and thick it may be. Its first cost, however, is an objection which keeps it from coming into more general use.

As a large belt runs on the pulley, a cushion of air is made between the belt and pulley, which lessens the holding power to some extent. Some belt manufacturers diminish this cushioning somewhat by perforating the belt with small holes or slits, so that the air may pass through and allow the belt to stick close to the pulley.

Belts will hold better when the pulleys are at long distances apart than when at short distances. Belts should never, if avoidable, connect two shafts one of which is directly over the other, and, in general, the two pulleys should have a position such that there may be a sag of the belt. It is desirable that

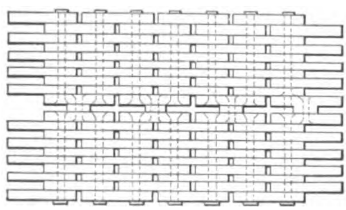


FIG. 46.—Leather Link-belt.

the angle which the belt makes should be not more than 45 degrees with the horizontal.

The tensile strength of the solid leather belting is from 2000 to 5000 lbs. per square inch; but only about 1000 to 1500 lbs. at the lacings. The working strain is taken at not over one third of the strength of the lacing.

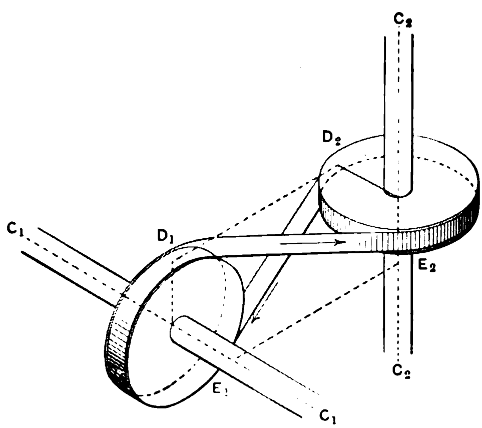


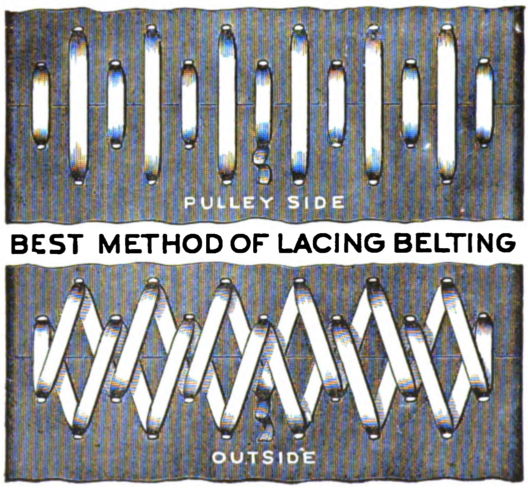
FIG. 47.—Arrangement for Quarter-twist Belt.

Commonly the belt connects pulleys that are on parallel shafts, but this is not necessarily the case. Fig. 47 shows the relative position of the pulleys on two perpendicular shafts which gives the belt a quarter twist. In this case the belt must run squarely onto the pulleys. It may run off the pulleys at any angle. In setting pulleys to give a quarter twist to a belt, the point where the belt leaves the driven pulley must be placed exactly over the corresponding point of the driving pulley. In this arrangement the belt can run in only one direction. If an attempt is made to run the belt in the reverse direction it will be thrown from the pulleys. Other twists may be given by the use of Guide-pulleys.

Methods of Lacing.—The effective strength of a belt, as well as the smoothness and uniformity of transmission, depends on the manner of connecting the ends. When possible, the belt should be endless; that is, it should be joined together in such a manner that the strength of the joint shall be equal to the strength of the belt itself, or as nearly so as possible; also, so that there shall be no extra weight caused by heavy lacing-leather. The heavy joint causes a vibratory movement of the belt when running; this causes variations in the arc of contact and this, in turn, may cause the belt to slip. Where a uniform motion is required, as for a dynamo, this would not be admissible.

The two methods most commonly used in fastening belting together at the ends are the *Butt-joint* and the *Lap-joint.* With the butt-joint and especially with heavy belts, rawhide lacing is used. With smaller belts, wire lacing made of some pliable composition is used considerably. This makes a much less clumsy joint and less waste of strength by the punching of holes than is necessary when the rawhide lacing is used. The lap-joint is made by beveling or scarfing the two ends and then gluing them together, under pressure; by gluing and riveting; and also by interlapping the different plies, when the belt is not single, and then gluing. The lap-joint is best because it makes practically an endless belt.

The joints in rope belting are made similarly by interlapping the strands or fibres and then wrapping them with



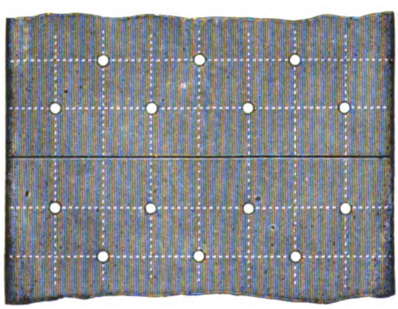


FIG. 48.—Butt-joint.

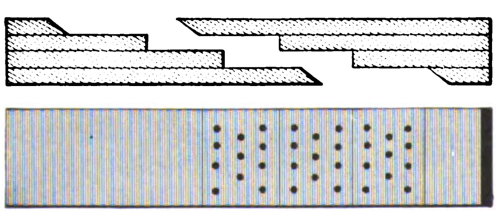


FIG. 49.—Scarf-splice.

cords, thus making a strong endless rope of uniform size.

When the distance between the centres of two shafts and the diameters of the pulleys are given, the length of belting required may be found approximately as follows:

RULE.—*Add the diameters in feet of the two pulleys together, divide the result by 2 and multiply the result by 3.1416. Then add this product to twice the distance between the centres of the two shafts.*

ROPE-DRIVING.

When rope is used the pulley contains a groove or grooves in its face in which the rope runs. *Rope belting* is commonly made of cotton, hemp, or manila, but rawhide, flax, and

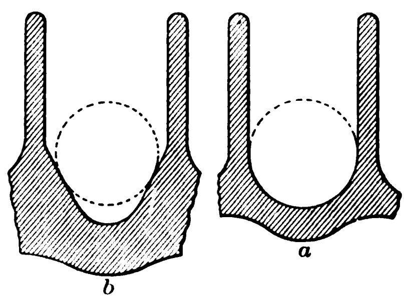


FIG. 50.—Section of Pulley for Rope-drive.

leather are sometimes used. With small pulleys cotton is best as it is softer and more pliable, and there is, therefore, less danger of breaking the fibres.

The principal advantages of rope-driving are quiet running and convenience in application. One of the greatest fields of usefulness for rope-driving is in the transmission of power to a moderate distance, under conditions which are not favorable to the use of leather belting or shafting. With rope-driving one is enabled at a comparatively small cost to transmit power in any direction to a building remotely situated from the power, which would otherwise require a long and expensive line-shaft, or an independent engine or motor. The facility with which a rope-drive may be carried in any direction, across rivers, canals, and streets, above or under ground, up or down hill, over houses and into buildings, is a feature which recommends rope as a means of driving.

The ropes most commonly used are from 1 to 2 inches in diameter. The size of a rope is very often given by its circumference or girth. The tensile strength of ropes for rope-driving varies greatly, though it is generally from 7000 to 12,000 lbs. per square inch.

The weight of the ropes when dry is given approximately by the formula $W = .3D^2 = .032C^2$, where W is the weight per foot of length and D the diameter and C the circumference of the rope. The speed of driving ropes varies from 1500 to 7000 feet per minute.

The accepted authority on rope-driving in this country is Mr. C. W. Hunt. He allows a working-strain on a 1-inch rope of 200 lbs. This makes the working-strain about $\frac{1}{36}$ of the breaking strength of the rope. This strain is about $\frac{1}{25}$ the strength of the splice. In practice, however, this limit is often greatly exceeded, on account of vibrations and imperfect tension-adjusting devices.

In his derivation of a formula for horse-power, he used a constant driving-strain on a 1-inch rope of 200 lbs., and velocities varying from 10 to 140 feet per second. The driving-force will be diminished by the tension due to centrifugal force of the rope passing over the pulley. Where Tc is the tension due to centrifugal force, W the weight of rope in pounds per foot, v the velocity of rope in feet per second, and g gravity,

$$Tc = \frac{Wv^2}{g}.$$

The difference between Tc and the maximum tension gives the force available for power transmission. As a certain amount of tension is necessary on the slack side of the rope to give it adhesion to the pulley, all of this force cannot be used. If the tension on the slack side of the rope is assumed at one half the driving force, the force available for doing work, P, is found as follows:

$$P = \tfrac{2}{3}(T_1 - Tc),$$

where T_1 is the tension on the driving side of the rope and Tc is the tension due to centrifugal force.

Also, the tension, T_2, on the slack side will be

$$T_2 = \frac{T_1 - Tc}{3} + Tc.$$

The tension T_2 will increase as the speed is raised, since Tc increases as the square of the velocity.

With the foregoing assumptions, the formula for horse-power may now be stated

$$\text{H.P.} = \frac{2v(T_1 - Tc)}{3 \times 550},$$

v being the velocity of the rope in feet per second.

The following table gives the horse-power of various ropes at different speeds.

HORSE-POWER OF TRANSMISSION ROPE AT VARIOUS SPEEDS.

(Computed from formula given above.)

Diam. of Ropes.	Speed of the Rope in feet per minute.											Smallest Diam. of Pulleys in inches.
	1500	2000	2500	3000	3500	4000	4500	5000	6000	7000	8000	
$\frac{1}{2}$	1.45	1.9	2.3	2.7	3	3 2	3.4	3.4	3.1	2.2	0	20
$\frac{5}{8}$	2.3	3.2	3.6	4.2	4.6	5.0	5.3	5.3	4.9	3.4	0	24
$\frac{3}{4}$	3.3	4.3	5.2	5.8	6.7	7.2	7.7	7.7	7.1	4.9	0	30
$\frac{7}{8}$	4.5	5.9	7.0	8.2	9.1	9 8	10.8	10.8	9.3	6.9	0	36
1	5.8	7.7	9.2	10.7	11.9	12.8	13.6	13.7	12.5	8.8	0	42
$1\frac{1}{4}$	9.2	12.1	14.3	16.8	18.6	20.0	21.2	21.4	19.5	13.8	0	54
$1\frac{1}{2}$	13.1	17.4	20.7	23.1	26.8	28.8	30.6	30.8	28.2	19.8	0	60
$1\frac{3}{4}$	18	23.7	28.2	32.8	36.4	39.2	41.5	41.8	37.4	27.6	0	72
2	23.2	30.8	36.8	42.8	47.6	51.2	54.4	54.8	50	35.2	0	84

For very light work and for guide-pulleys the rope rests on the bottom of the groove in the pulley (Fig. 50, *a*), but for heavy work the rope works in a groove which is V-shaped (Fig. 50, *b*), whereby the holding power is much increased. Since the power of one rope is limited, and as it is not convenient to use very large ropes, it is necessary, in most cases, to use several ropes. The pulleys have parallel grooves in which the ropes are placed, sometimes as many as 20 or 25.

PROBLEMS.

1. A leather belt is $\frac{1}{8}$ inch thick and 16 inches wide. What tensile force will be required to break it if the tensile strength of leather is 3000 lbs. per square inch?

2. A leather belt running at a velocity of 4000 feet per minute transmits 40 horse-power. Find the driving tension P on the circumference of the pulley.

3. If the tension on the tight side of a belt is twice that on the slack side, find T_1 and T_2 in Problem 2.

4. If the safe tension per inch of width is 90 lbs., find the width of belt required in Problem 2.

5. A rope-pulley is 20 feet in diameter and makes 500 revolutions per minute; find the velocity of the rope.

6. A rope is 1 inch in diameter. What force will be required to break it if the tensile strength is 8000 lbs. per square inch?

7. Find the diameter of each of the 13 ropes which drive 400 horse-power, the velocity of the ropes being 4000 feet per minute.

8. A certain drive has 21 ropes on a pulley 4 feet in diameter making 500 revolutions per minute. What horse-power may be transmitted if the girth of the ropes is 3.14 inches?

9. What is the weight of a rope which is 2 inches in diameter and 10 feet long?

10. A dynamo runs at 1020 revolutions per minute and requires 20 horse-power to operate it. The power is furnished by an engine running at 150 revolutions per minute. The engine drives a countershaft, which in turn drives the dynamo. If the pulley on the dynamo is 12 inches diameter and the fly-wheel of the engine 54 inches diameter and double belts are used, find the size of the pulleys on the counter-shaft, and the width of belts necessary.

11. A Corliss engine runs at 85 revolutions per minute, and develops 186 horse-power. If the main line-shaft which it drives runs at 235 revolutions per minute, and the pulley which receives the power is 5 feet in diameter, find the width of double belt used, and the diameter of the engine fly-wheel.

12. If the tension on the slack side is one half that on the driving side of the belt in Problem 11, find T_1 and T_2. Also determine the tension due to centrifugal force, and the effective driving tension.

CHAPTER VIII.

TOOTHED WHEELS.

TRANSMISSION of power between two shafts, by means of friction-wheels and belt-pulleys, is possible only so long as the resistance to be overcome does not exceed the friction which arises at the circumference of the wheels. When the resistance exceeds the friction a slippage will occur. To prevent this the friction must be made greater, in one case by pressing the friction-wheels closer together, and in the other by making the belt tighter. This excessive amount of pressure causes a corresponding amount of friction of the shafts in their bearings, so that friction-wheels cannot be employed to advantage where the resistance is very great, especially in the case of slow-running shafts.

Neither can they be used where it is necessary that the speeds of the two shafts have an exact ratio at every instant, as in screw-cutting machines, clocks, etc., for experience shows that, even with the greatest pressure, friction-wheels and belts will sometimes slip. To overcome these difficulties the toothed wheel is used.

Suppose two friction-wheels running together have spaces cut in their circumference at regular intervals, and if the material from these spaces be placed on the top of the remaining solid portion, so that the projections of one will fit into the depressions in the other, an approximate form of gear-wheel is produced.

The original diameter of the disk is the "pitch diameter", and the circumference of the disk itself the pitch-circle. The portion of the tooth above the pitch-circle is known as the "face" or addendum of the tooth, the portion below as the

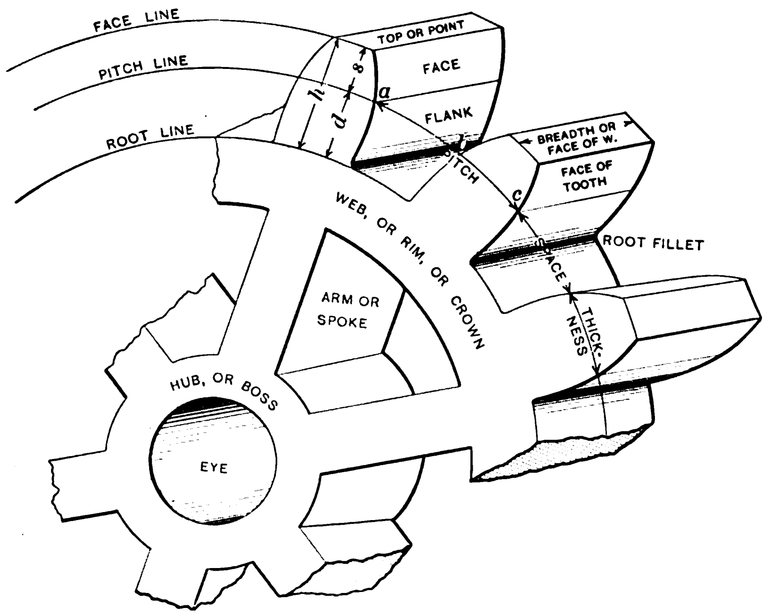


FIG. 51.—Section of Gear-wheel.

flank or dedendum. The distance from the front of one tooth to the front of the next, measured on the pitch-circle, is known as the "circular pitch," or simply as the "pitch." Sometimes the pitch is given as the number of teeth per inch of pitch diameter. This is called diametral pitch. Thus, in a wheel of 36 teeth, pitch diameter 12 inches, the diametral pitch is 3.

Let P_d be the diametral pitch of a gear-wheel of pitch diameter D; let the circular pitch be P_c and the number of teeth be N. Then

$$P_d = \frac{N}{D} = \frac{3.1416}{P_c}; \quad P_c = \frac{D \times 3.1416}{N} = \frac{3.1416}{P_d}.$$

In a spur-wheel (Fig. 51) the teeth are cut in the surface of a cylinder known as a "blank" (Fig. 53). Spur-wheels transmit motion between two shafts with parallel axes.

Bevel-wheels (Fig. 52) are formed by cutting teeth on the surface of a cone or frustum of a cone. Bevel-wheels transmit motion between shafts whose axes intersect.

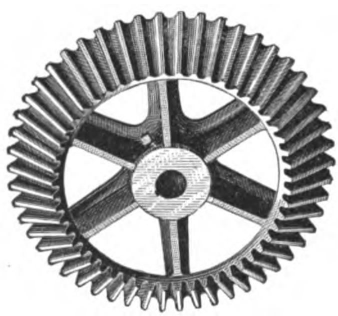


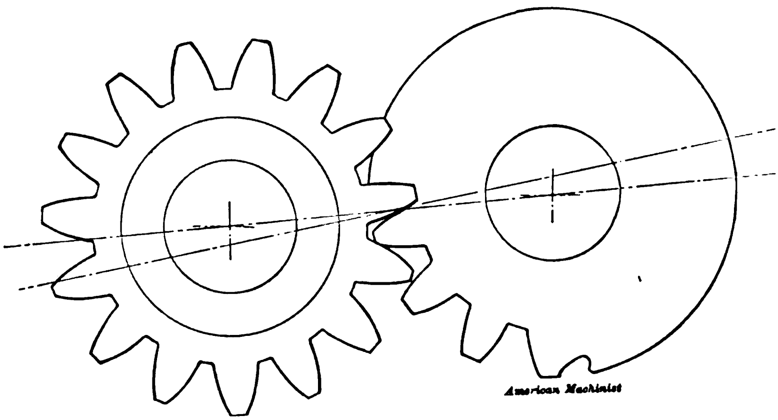
FIG. 52.—Bevel-gear.

Skew-wheels are formed by cutting teeth on the surface of hyperboloids of revolution. They transmit motion between shafts which do not intersect and which are not in the same plane.

If in the elementary gear-wheel, considered earlier in this chapter, the teeth were made in the form of rectangular prisms, they would, in running, wear themselves to approximate forms of either one of two curves, the epicycloid or the involute of a circle. In the practical construction of gear-wheels the teeth are cut or cast in the shape of one of these curves, depending on the use to which the wheel is to be put. The teeth of spur-gears can be easily cut to the proper form in the milling-machine. A bevel-gear, however, cannot be perfectly formed in a milling-machine, as the thickness of the teeth constantly diminishes toward the point of the cone. It requires a special machine, called a gear-shaper. These machines plane the teeth. There are two classes: one generates the tooth itself as it planes it, such as the Bilgram planer; the second class uses a former or template which guides the planing-tool, such as the Fellows gear-shaper.

For rough, heavy work the gears are cast in iron. These are known as *Cast Gears*. Gears made in a milling-machine or gear-shaper are known as *Cut Gears*. Sometimes in very heavy work, as in transmitting power from turbine-wheels, where the noise is intense and disagreeable, one of the gears is provided with wooden teeth which are locked in place by a simple device. These diminish the noise to a great extent. Spur-gears are also made of rawhide or leather, where it is desired to diminish noise. The leather or rawhide is compressed between two steel or brass plates and then is cut as an ordinary iron gear.

To draw all the teeth on a spur-gear would be a very tedious task in drafting. To save time, therefore, gears are



CUTTER BLANK

FIG. 53.—Fellows Gear-cutter and Partially Developed Gear.

usually represented by their pitch-circles (see Fig. 54). They are also shown as blanks, in which a few teeth have been cut, the teeth on each blank being in mesh.

In making gear-wheels, the space is made slightly greater than the thickness of the tooth. This is necessary in order that the teeth shall not bind on each other when running. It is the aim of all designers to make this "backlash" as small as possible. The depth of the space is also made a little greater than is absolutely necessary. The extra distance in this case is termed the "clearance."

The width of the gear is termed the "face. It is generally made from two to three times the circular pitch. Grant's "Gear Book," in a list of stock gears, gives a face of 3 to 4 inches for a gear of 3 diametral pitch, = 1.047 inches circular pitch, and $\frac{1}{2}$ to $\frac{5}{8}$ inch for a gear of 20 diametral pitch, $= 0.157$ inch circular pitch. Another manufacturer gives the face as $1\frac{1}{4}$ inches for a circular pitch of $\frac{1}{2}$ inch, and

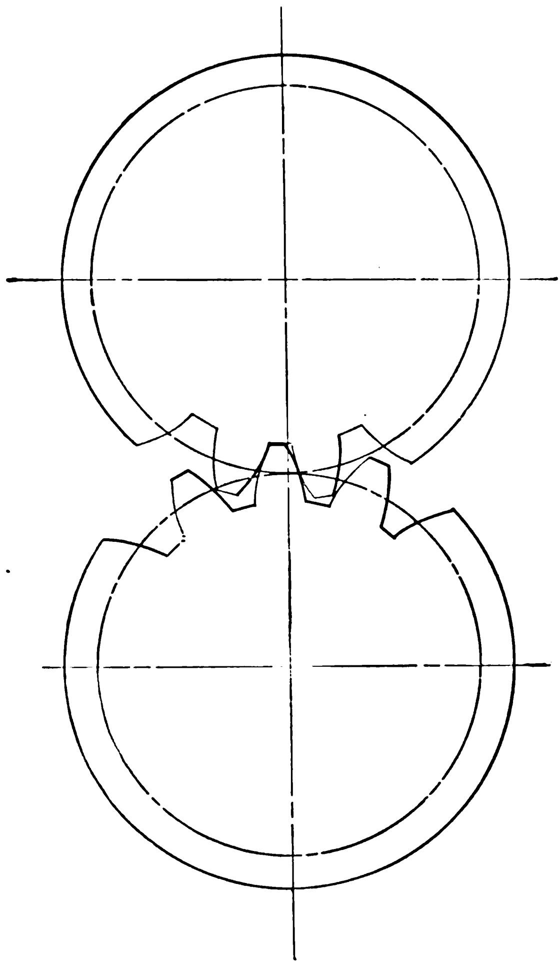


FIG. 54.

various figures up to a circular pitch of 6 inches, where the face is 20 inches.

In the section of a gear-wheel shown in Fig. 51 the following are the dimensions:

Pitch (circular) $= p =$ arc abc; face $= 2.5p$; thickness of tooth = arc $bc = .47p$; space = arc $ab \doteq .53p$; total height of tooth $= h = .7p$; addendum $= s = .3p$; dedendum $= d = .4p$.

In terms of diametral pitch, P_d:

$$s = P_d; \quad d = 1\tfrac{1}{4}P_d; \quad bc = 1.57P_d.$$

An Inside or Annular Gear is a wheel with gear-teeth cut on the inside of the rim as shown in Fig. 55. It works in

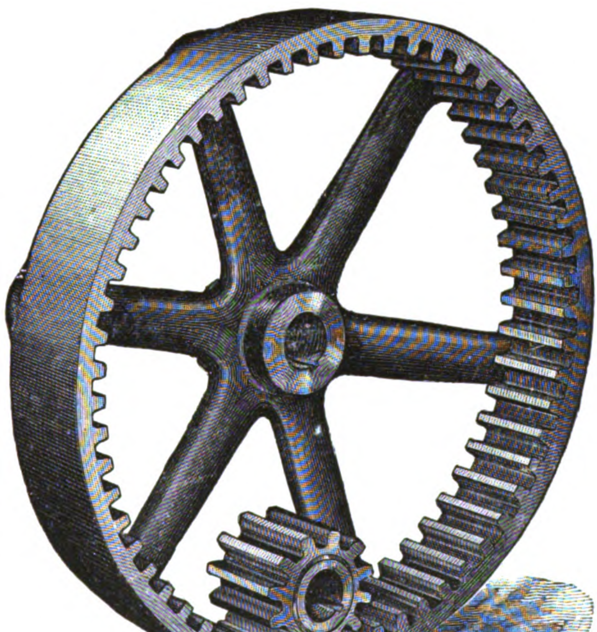


FIG. 55.—Annular Gear.

connection with a pinion (a small spur-gear), and the same considerations as to form and dimensions apply as in spur-wheels.

A Rack is a straight rectangular piece of metal in which teeth have been cut. It is used to convert rotary into reciprocating motion or *vice versa* by means of a pinion. The teeth may be either of the involute or epicycloidal system. An illustration of a rack and pinion is given in Fig. 79.

The same relations as regards diameters and revolutions exist in gear-wheels as in pulleys. But for the diameter of the wheels the number of teeth may be substituted.

Let N and n be the number of teeth on wheels of diameter D and d, making R and r revolutions respectively; then

$$R \times N = r \times n; \quad R = \frac{rn}{N}; \quad N = \frac{rn}{R}; \quad r = \frac{RN}{n}; \quad n = \frac{RN}{r}.$$

If a system of gear-wheels is used to transmit motion from a driver to a follower, the intermediate gears may be neglected in calculating the relative velocities of the driver and followers; the driver and follower may be considered as if they meshed directly into one another.

A train of gears and pinions (Fig. 56) is a train in which

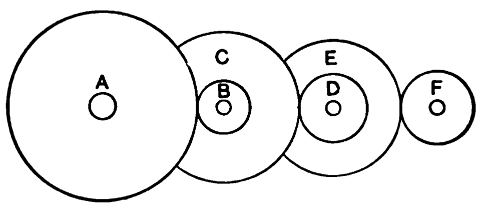


FIG. 56.—Train of Gears and Pinions.

a gear drives a pinion rigidly fastened to a gear on the same axis, which gear in turn drives another pinion, which may or may not be attached to another gear. If the number of teeth in each of the gears and each of the pinions is given, together with the number of revolutions of the first driver, the number of revolutions of the last follower may be obtained as follows:

Multiply the diameters, or the circumferences, or the number of teeth of all the drivers together, and this product by the number of revolutions of the first wheel; divide this product by the continued product of the diameters, or the circumferences, or the number of teeth of all the followers. The quotient is the number of revolutions of the last follower.

In regard to power transmitted by gear-teeth, authorities vary greatly as to the formulæ which should be used. All formulæ for determining the horse-power transmitted by gearing may be reduced to one of three forms:

$$\text{H.P.} = CVpf \quad \text{or} \quad CVp^2 \quad \text{or} \quad CVp^2f,$$

in which C is a coefficient, p the pitch in inches, V the velocity of the pitch-line in feet per second, and f the face of the tooth in inches.

The following is a formula of the first style:

Let $P =$ driving force at pitch-line in pounds;
$D =$ diameter of pitch-circle in inches;
$V =$ velocity of pitch-line in feet per minute;
$N =$ number of revolutions per minute;
$H =$ horse-power transmitted by wheel;
$f =$ face of tooth in inches;
$p =$ pitch of teeth in inches.

Then

$$H = \frac{PV}{33{,}000}, \quad \ldots \ldots \quad (1)$$

and

$$V = \frac{3.1416 \times D \times N}{12}.$$

Substituting this value of V in (1), we have

$$H = \frac{P \times 3.1416 \times D \times N}{33{,}000 \times 12}. \quad \ldots \quad (2)$$

An average value of P from different authorities is $280pf$. Substituting this value of P in (2), we have

$$H = \frac{280pf \times 3.1416 \times D \times N}{33{,}000 \times 12} = .0022\,pfDN.$$

Prof. Harkness gives $\text{H.P.} = \dfrac{0.910\,Vpf}{\sqrt{1 + 0.65\,V}}$, where V is velocity in feet per second.

PROBLEMS.

1. In Fig. 56 A has 80 teeth, B has 20, C has 60, D has 30, E has 40, and F has 10. If A makes 50 revolutions per minute, how many does F make per minute? *Ans.* 1600.

2. In the same figure suppose that A and F make 200 and 4000 revolutions respectively per minute, and that B, E, and F have 50, 40, and 30 teeth respectively, what may be the number of teeth on each of the other wheels?

3. Required the diameter of a spur-wheel which has 100 teeth and a pitch of 1.57 inches. *Ans.* 50 inches.

4. How many teeth in a wheel 10 inches in diameter, the pitch being .2618 inches? *Ans.* 120.

5. Required the pitch of a wheel of 100 teeth, the diameter being 12 inches. *Ans.* .3770 inches.

6. A certain spur-wheel has 40 teeth and its diameter is 10 inches. Find the pitch, thickness of teeth, width of space, total height of tooth, height above pitch-line, and depth below the pitch-line.

7. What is the diameter of the face circle in the above problem?

8. What is the diametral pitch of a spur-wheel of 100 teeth having a diameter of 10 inches?

9. What is the diameter of the pitch-circle in the above problem, also the thickness of the teeth?

10. What would be the diameter of a blank to be used in making a cut gear whose pitch-circle is to be 6 inches in diameter and the pitch $\frac{1}{4}$ inch?

11. What horse-power will be transmitted by a spur-wheel 3 feet in diameter making 200 revolutions per minute, the pitch of the teeth being 2 inches?

12. It is desired that a spur-wheel with a diameter of 24 inches shall transmit 10 horse-power while making 100 revolutions per minute. What should be the pitch of the teeth?

CHAPTER IX.

THE SCREW.

THE screw is a combination of the lever and the inclined plane, and the mechanical advantage depends both on the arm of the working lever and the inclination of the thead or inclined plane which supports the weight.

The efficiency of the screw is very low, from 15 to 45 per cent. A large amount of the force applied is lost in friction in the nut. If the faces of the threads are inclined, as in a V thread, the friction is greater than for a square thread. The efficiency increases if the pitch, or distance between two consecutive threads, is increased.

Applications of the screw may be seen in the jack-screw, the vise, bolts, nuts, etc. The jack-screw, Fig. 57, is a machine for raising heavy weights. It consists of a screw to which is attached a lever for applying force, and a heavy base A, having screw-threads on the inside. When the handle is turned, the screw moves up or down, according to the direction of rotation of the handle. The weight is placed on the head B, which does not turn with the screw, thus allowing the weight to move up without rotation. The equation

$$^{*}F \times 2 \times 3.1416 \times R = W \times \text{pitch} \times (f + 1)$$

gives the relation of the weight to the applied force, in which F = force applied, W = weight, R = radius of handle, and f = coefficient of friction of the screw. This relation is derived by the application of the Law of Machines, Chapter I. With one turn of the handle the applied force moves around the

* The "Efficiency" $= \dfrac{W \times \text{Pitch}}{F \times 2 \times 3.1416 \times R}$.

circumference of a circle, the radius of which is the length of the handle. To find the circumference when the radius is given, multiply the radius by 2×3.1416. If we denote radius by R, we have $2 \times 3.1416 \times R$ as the distance moved through by the applied force during one turn of the handle. During this one turn of the handle the weight is lifted through

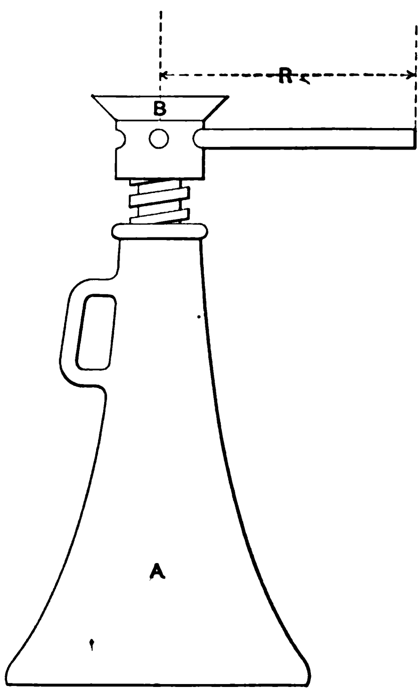


FIG. 57.—Jack-screw.

a distance equal to the length in the direction of the axis of a thread plus a space; this length is called pitch.

The *endless screw*, or worm, is a combination of the screw with a worm-wheel. The worm is secured in bearings so that it cannot move in the direction of its length. The threads of the screw mesh with the teeth of the worm-wheel, and this in turn may impart motion to a train of wheel-work.

In Fig. 58 the force is applied by means of a crank, though a pulley could be used, instead of a crank, and belted to an engine or shaft.

$$F \times 2 \times 3.1416 \times R = W_1 \times \text{pitch}$$

is the equation of work for the screw part of the machine alone. By solving for W_1 we find that the screw will raise a weight

$$W_1 = \frac{F \times 2 \times 3.1416 \times R}{\text{Pitch}}.$$

This weight W_1, in its turn, acts as a turning force against the circumference of the worm-wheel at A, so that we may now

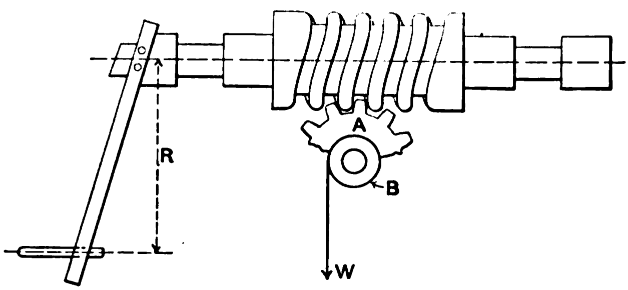


FIG. 58.—Endless Screw.

regard it as force to be used in turning the wheel against the resistance offered by the weight which hangs on the smaller wheel B. Let R_1 be the radius of the large wheel A, and R_2 the radius of the small wheel B. Then, multiplying the force W_1 by its lever-arm R_1, and the weight by its lever-arm R_2, we have

$$\frac{F \times 2 \times 3.1416 \times R}{\text{Pitch}} \times R_1 = W \times R$$

or

$$W = \frac{F \times 2 \times 3.1416 \times R \times R_1}{\text{Pitch} \times R_2}.$$

The relation of the angular velocities of the crank R and the worm-wheel A is shown in the equation $R = N \times r$, in which R and r represent the number of revolutions of the crank

R and the wheel A, respectively, and N the number of teeth on the worm-wheel.

Screw-threads.—Screw-threads are employed for two purposes, one of which is holding or securing, and the other transmitting motion. Examples of the former are bolts, nuts, screws, etc.; of the latter, endless screws and the screw on the engine-lathe for moving the tool-carriage.

Fig. 59 shows the Sellers or United States standard

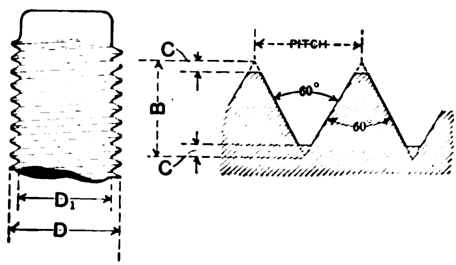


FIG. 59.—United States Standard Thread.

thread, which is used principally in the United States. It will be noticed that the V-shaped threads are flattened a little at the top and bottom. The amount of flat is given by the equation $f = \frac{.125}{n}$, in which f is the width of flat and n is the number of threads to the inch. This makes a solid sound thread avoiding the broken edges which are often the result if the sharp edge is permitted. The sides of the thread make an angle of 60 degrees with the axis of the thread, as shown in the figure.

SCREW-THREADS, UNITED STATES STANDARD.

Diam.	Pitch.	Diam.	Pitch.	Diam.	Pitch.	Diam.	Pitch.	Diam.	Pitch.
$\frac{1}{4}$	20	$\frac{3}{4}$	10	$1\frac{1}{4}$	7	$1\frac{15}{16}$	5	$2\frac{13}{16}$	$3\frac{1}{2}$
$\frac{5}{16}$	18	$\frac{13}{16}$	10	$1\frac{5}{16}$	6	2	$4\frac{1}{2}$	3	$3\frac{1}{2}$
$\frac{3}{8}$	16	$\frac{7}{8}$	9	$1\frac{3}{8}$	6	$2\frac{1}{4}$	$4\frac{1}{2}$	$3\frac{1}{4}$	$3\frac{1}{8}$
$\frac{7}{16}$	14	$\frac{15}{16}$	9	$1\frac{1}{2}$	6	$2\frac{5}{16}$	$4\frac{1}{2}$	$3\frac{5}{16}$	$3\frac{1}{4}$
$\frac{1}{2}$	13	1	8	$1\frac{5}{8}$	$5\frac{1}{2}$	$2\frac{3}{8}$	4	$3\frac{1}{2}$	$3\frac{1}{4}$
$\frac{9}{16}$	12	$1\frac{1}{16}$	7	$1\frac{3}{4}$	5	$2\frac{1}{2}$	4	$3\frac{3}{4}$	3
$\frac{5}{8}$	11	$1\frac{1}{8}$	7	$1\frac{7}{8}$	5	$2\frac{3}{4}$	4	4	3
$\frac{11}{16}$	11								

The relation between the pitch P and the diameter D of the U. S. standard thread is given approximately by the formula

$$P = .24\sqrt{D + .625} - .175.$$

The Whitworth or English standard thread is shown in Fig. 60. The sides of the threads make an angle of 55 degrees

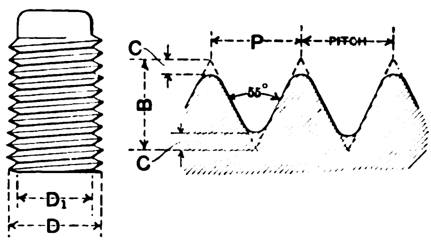


FIG. 60.—Whitworth Screw-thread.

with each other, and the bottom and top of the thread is rounded. Fig. 61 shows a square thread, a part of which is right-handed and part left-handed.

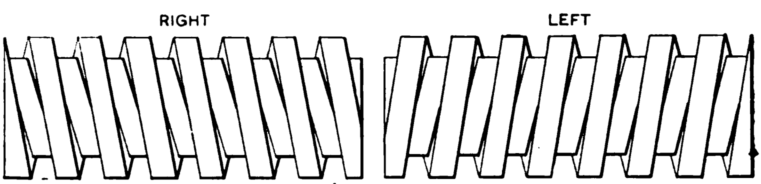


FIG. 61.—Square Thread.

The diameter of a thread is the largest diameter measured perpendicular to the axis of the screw. The depth of thread is the vertical height of the tooth above the bottom. The pitch of the thread is the distance taken up by one thread on the axis. For a square thread the pitch would consist of a thread and a space. The size of thread is generally designated as so many threads to the inch. Screw-threads on the outside of a cylinder, as threads of a bolt, are called male threads, and the threads on the inside, as the threads in a nut, are called female threads.

Screw-threads are made with *taps*, *dies*, *lathes*, and special *screw-cutting machines*. The tap is used for making inside or female threads, and the die for making the outside or male threads, while either kind of thread may be made in the lathe. In threading bolts and piping, different sets of taps and dies must be used, the pipe-thread always being smaller and of greater number to the inch than those on the bolt or screw. They also taper toward the end of the pipe. For this reason, we have what is known as the bolt or standard dies and taps, and pipe dies and taps. For example, the number of threads per inch on a 1-inch bolt, U. S. standard, is 8, while the number of threads per inch on a 1-inch pipe is 11.5. The reason for making more threads on a pipe is that by making more of them the depth of each thread is made less, hence there is less danger of cutting through the thin pipe.

Bolts.—Bolts are made of wrought iron or steel and are forged out by bolt-making machines, and the threads put on with dies or with special thread-cutting machines. The *machine-bolt*, Fig. 62, may have either a square or a hexagonal head

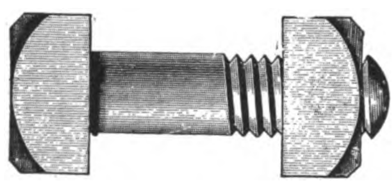


FIG. 62.—Machine-bolt.

with nut to match. The *carriage-bolt*, Fig. 63, differs from the machine-bolt mainly in having a round or oval head and

FIG. 63.—Carriage-bolt.

being square in cross-section for a short distance under the head, and generally has a square nut. The *stove-bolt*, Fig.

64, has a countersunk head with a slot sawed in it for the use of a screw-driver. A *cotter-bolt* or split pin is split along its

FIG. 64.—Stove-bolt.

axis; the split portions are bent at right angles to the axis, thus doing away with screw-threads and a nut. The stud-bolt

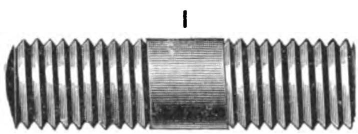


FIG. 65.—Stud.

has no head, but screws directly into the piece which it is to hold, a nut being used on the end; an example may be seen on the cylinder-head of an engine. An *eye-bolt* is one having an eye instead of a head.

Screws.—Fig. 66 represents a cap-screw. It takes the

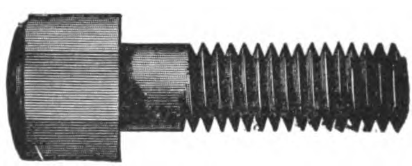


FIG. 66.—Cap-screw.

place of a bolt, and screws into one of the pieces to be held, the shoulder or cap on the end of the screw giving it the holding power. The *lag-screw*, Fig. 67, is used in wood only, usually for hanging shafting. Fig. 68 represents a drive-screw,

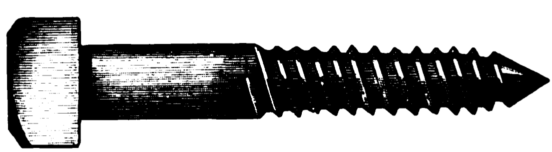


FIG. 67.—Lag-screw.

and Fig. 69 the common wood-screw. The wood-screw may have a round or a flat head. The set-screw is one in which

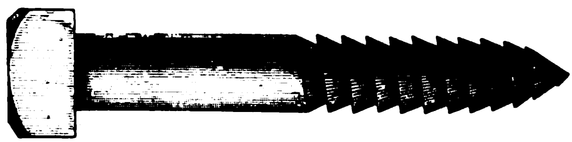


FIG. 68.—Drive-screw.

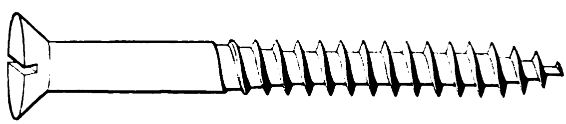


FIG. 69.—Wood-screw.

the holding power is produced by the pressure of the end against a piece of work. The end is hardened.

PROBLEMS.

1. What weight will be raised by a jack-screw, if its handle is 20 inches long, the pitch of the threads $\frac{1}{4}$ inch, and the force applied 130 lbs., neglecting friction? *Ans.* 75,398.4 lbs.

2. Design a jack-screw that will raise a weight of 20,000 lbs. with an applied force of 100 lbs., the length of the handle being 10 inches, neglecting friction. *Ans.* Pitch = .31416 inches.

3. What force must be applied in order to raise a weight of 12,000 lbs. if the handle is 16 inches in length, and the pitch of the thread $\frac{1}{2}$ inch, neglecting friction? *Ans.* 60 lbs. about.

4. Design a jack-screw that will raise a weight of 1600 lbs. with an applied force of 100 lbs., taking the efficiency to be 30 per cent on account of friction.

5. What weight can be raised with an endless screw, arranged as in Fig. 58, if the length of the handle is 20 inches, the pitch of the threads $\frac{1}{2}$ inch, the radius of the worm-wheel A 10 inches, and the radius of the small wheel B 5 inches, the force applied at the end of the handle being 100 lbs., the efficiency being 50 per cent?

6. What must be the length of the handle of an endless screw in order that a weight of 4000 lbs. may be raised by a force of 100 lbs., the other dimensions and efficiency being the same as for problem 5?

7. Design an endless screw similar to the one shown in Fig. 58 which will raise a weight of 8000 lbs. with a force of 100 lbs., neglecting friction.

8. In Fig. 58, how many revolutions will *A* make if *A* has 20 teeth and *R* makes 200 revolutions per minute?

9. How many revolutions of the crank in the above arrangement will be required in causing *A* to make 3 revolutions?

10. What should be the pitch of the threads for a bolt 2 inches in diameter?

CHAPTER X.

CAMS.

THE Cam is a revolving inclined plane. It may be either an inclined plane wrapped around a cylinder, as in Fig. 70, or

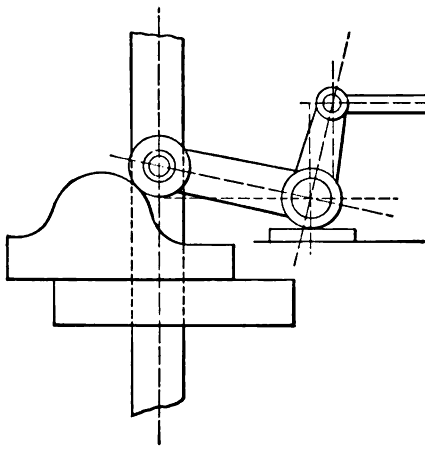


FIG. 70.—Cam.

it may be an inclined plane curved edgewise, as in Fig. 71. This mechanism is generally used for the purpose of producing a reciprocating motion in rods and levers by giving the cam a rotary motion. In Fig. 72, *BCD* represents the cam turning on the axis *A*, and giving a reciprocating rectilinear motion to the heavy rod *EF*, which is constrained to move in its rectilinear path by the guide-rollers. The rotation of the axis being in the direction of the arrow, the rod *EF* has an upward motion until the extreme point *B* of the cam comes in line with the rod, when the portion *BG* of the cam allows the rod to fall by its own weight or by the action of a spring until the point *G* comes in line with the rod, and so on; thus one revolution of the cam here presented will cause the rod to make three upward and three downward strokes.

Within certain limits the use of cams admits of the certain transmission, from a uniformly revolving shaft, of widely varying velocities and in an easily determined manner. For this reason they are

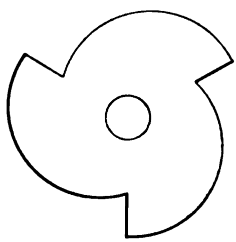


Fig. 71.—Cam.

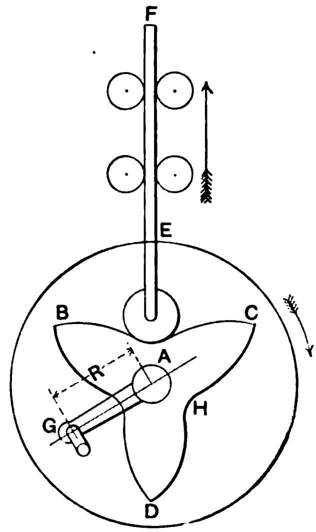


Fig. 72.—Cam and Follower.

often convenient. By varying the curve of the cam any law of motion may be given to the rod. The rod *F*, Fig. 72, is called the *follower*, and is generally provided with a roller as shown in the cut, by means of which the contact between the cam and the follower is changed from a sliding contact to a rolling contact, thus lessening, to some extent, the friction and wear. The use of a cam is accompanied by a very large amount of friction due to the contact of the follower and the driver, especially where it is not possible to use a roller on the follower. This causes a wear of the parts, which in time makes a backlash and, with high speeds, much noise. The cam is the mechanical movement that the designer usually calls to his aid as the last resort, after having failed to obtain the necessary motion of a piece by other means which would have made lighter and quieter running parts. The cam is, however, a very useful movement, and in certain cases must be accepted, though it is to be avoided where possible.

The path of the follower may be a straight line, a circle or any other curve. Fig. 73 illustrates a cam with a *swinging* follower. In this case the path of the point D of the follower will be the arc of the circle EF.

In some cam movements the follower has a flat bearing-piece, Fig. 74, instead of a point, which for the same cam changes the law of motion of the follower, but gives a more extended bearing surface to the cam. It is called a *flat-footed* follower. Cams often have grooves in their perimeters for the purpose of confining the follower to its proper path.

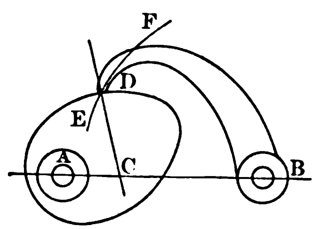


FIG. 73.—Cam with Swinging Follower.

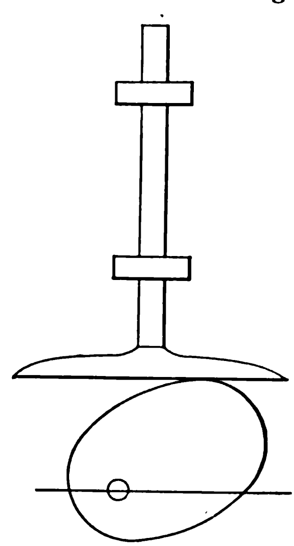


FIG. 74.—Cam and Flat-footed Follower.

To find the curve forming the edge of the cam so that the velocity ratio of the rod and the axis of the cam may be constant:

In Fig. 75 let A be the centre of the cam. From A as a centre with any convenient distance AC as a radius describe the circle $CEDBN$. On BA take Ba equal to the length of the stroke of the rod; divide it into any number of equal parts, say five, in the points b, c, d, e, and divide the semicircle $BDEFG$ into the same number of equal parts by the radial lines AD, AE, AF, and AG. From A as a centre with Ab, Ac, Ad, and Ae as radii describe the dotted arcs cutting AD, AE, etc., at the points s, k, l, m; then through these points draw the curve *asklmpn*. This curve is the *Spiral of Archimedes*. The peculiarity of this curve is that a point following it will move outward radially equal distances when passing through

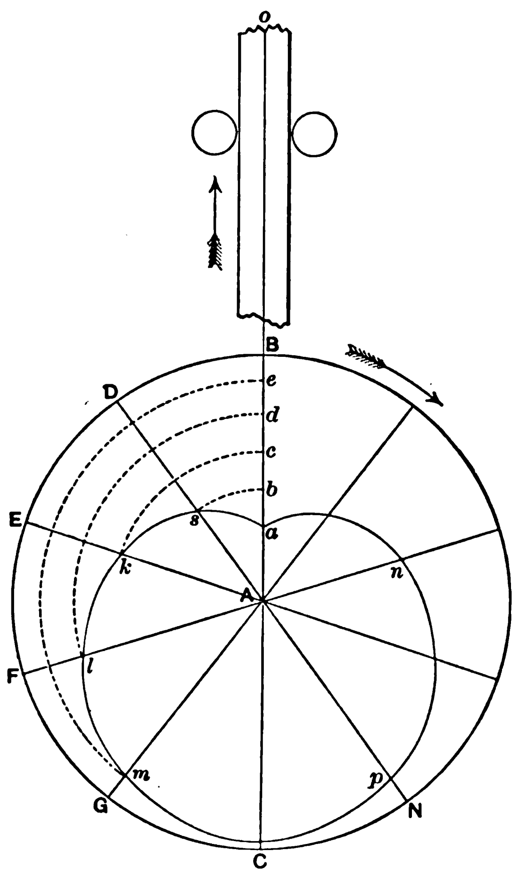


FIG. 75.

equal angles. All lines drawn through the centre *A* of this curve are equal: thus *aC* equals *ln* = *sp*. Hence if the rod had two pins placed at *a* and *C*, the cam would revolve between them, and would cause the rod to make a downward as well as an upward stroke.

The following example * gives a solution of the cam and follower in which the cam has a variable velocity of revolution about its axis, and the follower moves through a given desired curve. In Fig. 76 let *D* be the follower, a part of which is left out of the figure. *A* is to be the centre of rotation of the cam, and it is desired that the path of the follower-point be a curve 1 2 3 4 5. To find the curve of a cam necessary to fill these conditions, proceed as follows: For convenience cut a templet to the follower-path 1 2 3 4 5 with the centre point *A* marked. Then with the angles 1*A*2, 2*A*3, etc., laid off according to the velocities during their part of the stroke, the several curves may be struck by the templet. Now drawing in the arcs from the points 1, 2, 3, etc., of the follower-path, we obtain intersections and can draw the curve 1 2 3 4, etc. This is called the method of intersections and is the one usually employed in practice.

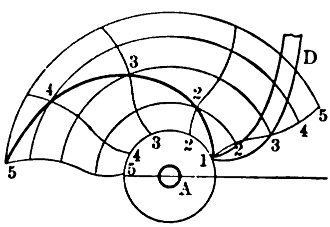


FIG. 76.

Cams are often used on engines for giving the proper movements to valves. Fig. 77 shows a cam movement sometimes used in operating shearing- and punching-machines. The cam revolves about its axis *O*, presses against the under side of the lever *A*, thus causing an upward and downward

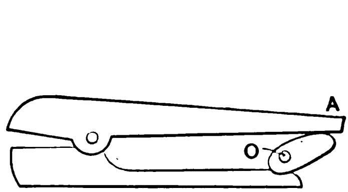


FIG. 77.—Shear operated by a Cam.

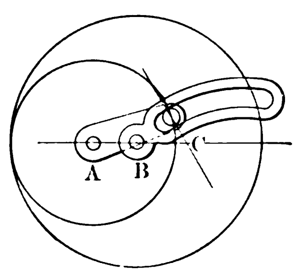


FIG. 78.

movement of the shear. The *inverse cam* is a movement in which are the elements of the grooved cam and follower, but

* Robinson's "Principles of Mechanism."

where the driver has the pin or roller and where the follower has a groove. It is sometimes called the pin and slit. An example is shown in Fig. 78, in which the slotted piece B is the follower and A the driver.

The relation of the applied force to the resistance is calculated for cams in the same manner as in the case of the screw; that is, by the application of the Law of Machines. In Fig. 72 let the length of the handle be R. Let $AC - AH = d =$ the distance through which the follower is lifted. The follower will make three movements for each revolution of the cam; hence, applying the Law of Machines, we have

$$F \times 2 \times 3.1416 \times R = W \times 3 \times d$$

as the equation of work, F being the force applied and W the weight of the follower. This of course applies only where friction is neglected.

CHAPTER XI.

THE LEVER AND SOME OF ITS MODIFICATIONS.

As has been stated before, the lever and its modifications make up a large part of the mechanism of much of the machinery now used, different methods of using it being employed as may be convenient. A *crow-bar* is an excellent example of the lever. Examples are also seen in the *rack*

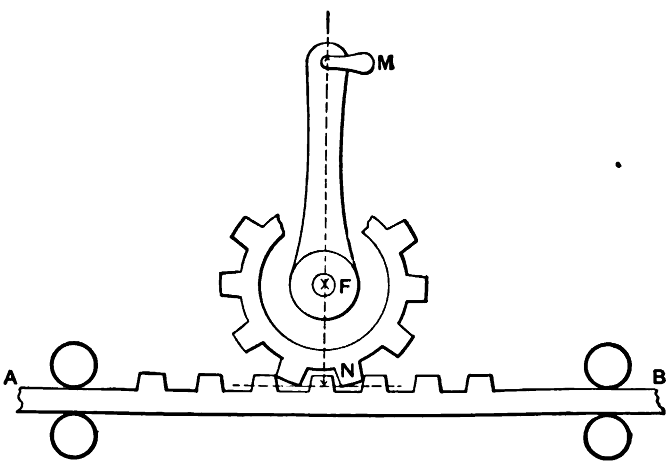


FIG. 79.—Rack and Pinion.

and pinion, the *pulley*, the *wheel and axle*, *cranes*, etc., the action of the lever in these last-named cases being continuous.

Law of the Lever.—*The applied force multiplied by its distance from the fulcrum equals the weight multiplied by its distance from the fulcrum.*

Fig. 79 shows the *rack and pinion*. In this arrangement the lever-arm of the applied force is the length of the crank FM, and the lever-arm of the resistance is the radius FN of the pitch-circle of the pinion.

The **Moving Strut** and the **Toggle-joint** are two applications of the lever. The moving strut consists of a bar, Fig. 80, which rests against some projection. The weight W to be moved rests on a plane, and the extremity of the bar is placed against it. If a force P be applied in the direction of the arrow, the strut will be forced down and the weight will move away from the fixed point. If the angle between the strut and the plane on which the weight rests is small, a comparatively small force will move a very heavy weight. If the angle be denoted by α, the applied force by P, and the resistance of the weight to being moved by R, then

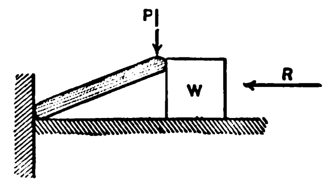


FIG. 80.—Moving Strut.

$$P \times \cos \alpha = R \times \sin \alpha.$$

If α be 5°, cos α = .99619, sin α = .08716, and R = 11.44F.

The resistance is not always the weight of the body to be moved; it may be principally the friction of the body on the plane.

The toggle-joint is a combination of two moving struts. It is shown in Figs. 81 and 82. It is used where a large resistance is to be overcome through a short distance. The two struts have force applied at their junction; one end of one strut rests against a fixed point, and the other against the body to be moved. The force is applied in a direction perpendicular to the direction of motion of the body.

Let α be the angle each strut makes with the line joining the points about which the outer ends of the strut rotate, P the applied force, and R the resistance. Then

$$2R \sin \alpha = P \cos \alpha.$$

An example of the toggle-joint is found in stone-crushers.

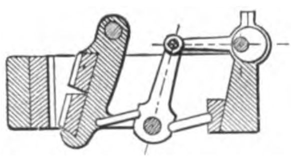


FIG. 81.

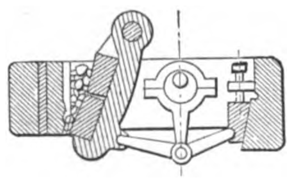


FIG. 82.

WHEEL-WORK, CRANES, ETC.

Fig. 83, two wheels of unequal diameters keyed to the same shaft, illustrates the application of the lever to wheel-work. That is, the radius of one wheel is the lever-arm of the *applied force*, and the radius of the other the lever-arm of the *weight*, the pivot on which the two wheels turn being the fulcrum.

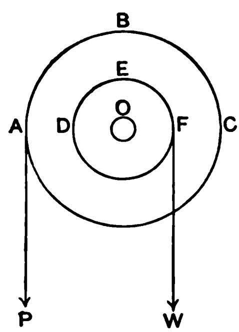


FIG. 83.

This is the principle of the wheel-work of cranes for raising heavy weights, as used for loading and unloading ships, handling ore, etc., in which several pairs of wheels are used, in connection with each other, for the purpose of making a still larger mechanical advantage, but which gives a corresponding loss of speed. In calculating such an arrangement, the wheels may be considered by pairs, applying the law of the lever in each case; if so desired, however, the work may be abridged considerably by the use of the following equation, which is deduced from the law of the lever and which is known as the Law of Wheel-work: *The continued product of the weight and the radii of the wheels is equal to the* *continued product of the applied force and the radii of the pinions.*

If, in Fig. 84, we let R, R_1, R_2, R_3, R_4, R_5, R_6, R_7, be the radii of the wheels A, B, C, D, E, F, G, H, respectively, then $P \times R \times R_2 \times R_4 \times R_6 = W \times R_1 \times R_3 \times R_5 \times R_7$. These

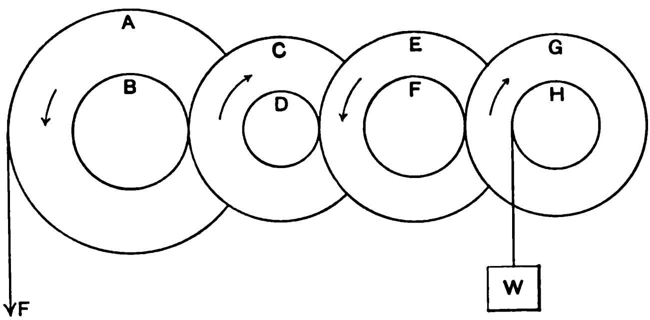


FIG. 84.—Wheel-work, Ten Wheels.

are spur-wheels which mesh into each other. The force may be applied at the first wheel A by means of a crank, or it may be turned by an engine, as in the case of the hoisting-engine. Sometimes the various pairs of wheels in wheel-work are connected by means of belts or sprocket-chains instead of teeth. In the figure here shown the circles which are tangent are the pitch-circles of the spur-wheels which mesh with each other and are so drawn to save the trouble of drawing in the teeth.

The Block or Pulley.—A *pulley-block* in a simple form consists of two metal plates carrying a grooved cylindrical disk or sheave. The number of sheaves may be increased and the pulley is then described as a double, triple, etc., block. The drawing shows a triple block in perspective. The mathematical discussion of the pulley will be found in Chapter I, page 9.

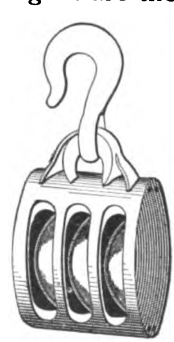


FIG. 85.
Triple Pulley-block.

The Differential Windlass.—This machine is shown in Fig. 86*a* and consists of two drums of unequal diameter, D and

d, upon the opposite sides of which two ropes are fastened and wound. By this arrangement the rope winds upon one of the drums, while it winds off the other. A movable pulley hangs in the loop of the rope, and to this pulley is attached the weight. The equation of motion is derived from the Law of Machines. The distance passed through by the applied

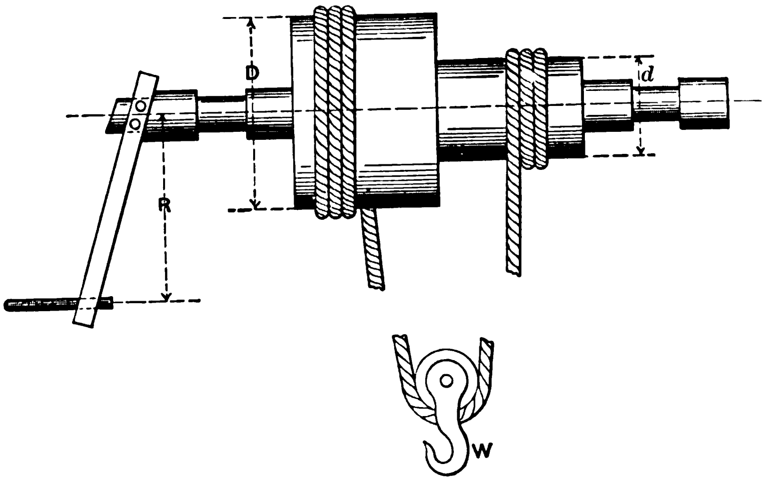


FIG. 86a.—Differential Windlass.

force in one turn of the crank is the circumference of the circle of which R is the radius, that is, $2 \times 3.1416 \times R$. The distance passed through by the weight is

$$\frac{3.1416 \times D - 3.1416 \times d}{2},$$

since the rope is winding on the large drum and off the small one, and because the movable pulley will divide the distance moved through by the weight by 2. Then, multiplying the applied force and the weight by the distances through which they move, respectively, we have the following equation of work:

$$F \times 2 \times 3.1416 \times R = W\frac{(3.1416 \times D - 3.1416 \times d)}{2}.$$

Dividing through by 3.1416, we have

$$F \times 2 \times R = W \times \frac{D - d}{2}$$

for the desired equation.

The Differential Pulley, or differential hoist, operates on the same principle as the differential windlass. It consists of two pulleys A and B, Fig. 86*b*, keyed to the same shaft, and which have pockets in their circumferences, into which fit the links of a chain passing around the pulleys. The chain is endless and, in one of its loops, supports a single movable pulley, C. If R be the radius of the large pulley A, and r the radius of the small pulley B, F the applied force, and W the weight to be raised, then

$$F \times R = W \times \tfrac{1}{2}(R - r).$$

There are other differential hoists, which depend for their action on trains of gear-wheels, which usually have a sun-and-planet motion. They are, however, too complicated to be discussed here.

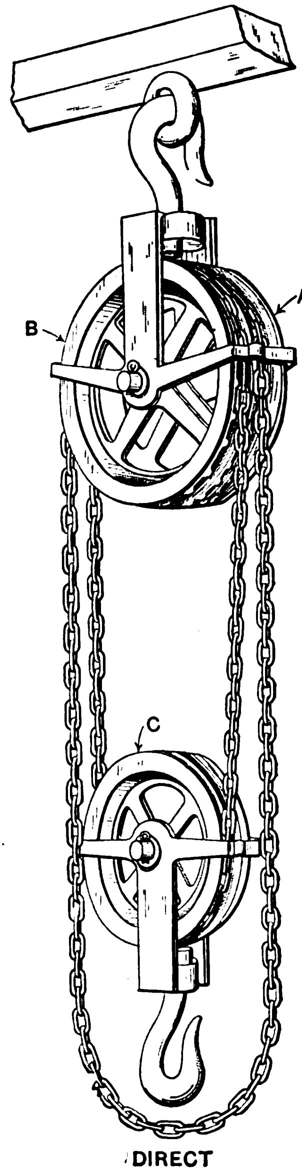


FIG. 86*b*.

PROBLEMS.

1. In Fig. 87 let $FM = 4$ feet, $FN = 12$ inches, and the weight 4000 lbs. Find the applied force in pounds, by the application of the law of the lever. *Ans.* 1000 lbs.

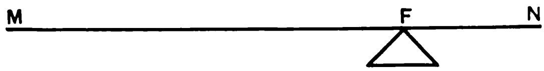


FIG. 87.—Lever.

2. In the same figure, let the weight be 2000 lbs., the applied force 100 lbs., and FN 4 inches. How long should FM be? *Ans.* 6 feet 8 inches.

3. In Fig. 79 let the length of handle be 20 inches and the radius of pinion 4 inches, and the force applied at the end of the handle 200 lbs. What resistance will it overcome, neglecting friction? *Ans.* 1000 lbs.

4. In Fig. 79 let the radius of the pinion be 5 inches, the resistance 600 lbs., and the applied force 100 lbs. Find the length of the handle.* *Ans.* 2 feet 6 inches.

5. If the resistance in Fig. 79 is 6000 lbs., the length of the handle 30 inches, and the radius of the pinion 2 inches, what force at the end of the handle is required to overcome it? *Ans.* 400 lbs.

6. Find the radius of the pinion and the length of the handle of a rack and pinion that will raise a weight of 2000 lbs. with an applied force of 100 lbs.

7. In Fig. 83 let the radius of the larger wheel be 20 inches, and the radius of the small wheel 4 inches. What weight can be raised with an applied force of 100 lbs.? *Ans.* 500 lbs.

8. Find the diameters of two wheels, similar to those above, which will raise a weight of 2000 lbs. with an applied force of 50 lbs., the efficiency being 60 per cent.

9. In Fig. 84 let the radii of the wheels A, C, E, and G be 24 inches, 20 inches, 16 inches and 12 inches, respectively, and the radii of the pinions B, D, F, and H be 5 inches, 4 inches, 6 inches, and 3 inches, respectively. Find what weight can be raised with an applied force of 100 lbs. *Ans.* 25,600 lbs.

---

* Friction is not considered in Problems 4, 5, 6, 7, 8, 9, and 10.

10. Design a set of 4 wheels and 4 pinions which will raise a weight of 8000 lbs. with an applied force of 100 lbs.

Here we must have $100 \times$ the product of the radii of the four wheels $= 8000 \times$ the product of the radii of the four pinions; or, letting P and S represent the above products, respectively, we have the equation, $100 \times P = 8000 \times S$. We may assume S to be $3 \times 2 \times 4 \times 2 = 48$, that is, we may assume that the radii of the pinions are 3 inches, 4 inches, 2 inches, and 2 inches, respectively. Substituting this in the above equation, we have $100\ P = 8000 \times 48 = 384,000$ and $P = 3840$. The product P of the radii of the four wheels must then be 3840; that is, we may make the radii of the four wheels of any length so their product is 3840. These radii may be found by trial. $10 \times 8 \times 6 \times 8 = 3840$. Therefore the radii of the wheels may be 10 inches, 8 inches, 6 inches, and 8 inches, respectively. Where the number of teeth on each wheel is known, number of teeth may be substituted for radius in the above equation.

11. Find the diameters of the three wheels and three pinions in a crane which will raise a weight of 20,000 lbs. with an applied force of 200 lbs., the efficiency of the machine being 50 per cent.

12. What weight can be raised with an applied force of 100 lbs. with a system of pulleys in which the block contains three pulleys as in Fig. 85, the efficiency being 50 per cent?

13. A weight of 1800 lbs. is to be raised with a system of pulleys in which the block contains 3 pulleys. Find the force necessary, if friction is neglected, to lift it.

14. A weight of 2 tons is to be raised with an applied force of 500 lbs. How many pulleys should be put in the block, friction being neglected as unimportant?

15. What weight can be raised with a differential windlass, the large and small drums being 12 and 8 inches in diameter, respectively; the force applied 100 lbs. and the length of the handle 20 inches?

16. What force will be required to raise a weight of 8000 lbs. with a differential windlass, the large and the small drums being 20 and 12 inches in diameter, respectively, and the length of the handle 16 inches?

17. Design a differential windlass that will raise a weight of 4000 lbs. with an applied force of 400 lbs., the length of the handle being 20 inches.

CHAPTER XII.

LINK-WORK.

THE term link-work is applied to such machinery as consists of rods, cranks, levers, bars, etc., either with parallel axes, intersecting axes, or axes not in the same plane. As an instance, take A and B, Fig. 88, as fixed centres of motion

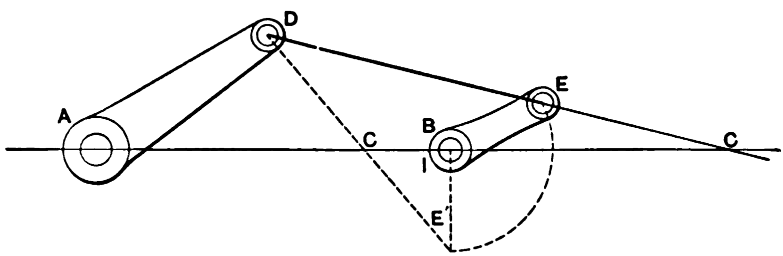


FIG. 88.

AD as the driving crank, BE as the driven crank, and DE as the connecting rod, bar, or link. As AD turns about its centre A, the rod DE compels BE to turn also. Here DE is the link. When the point C, where the axis of the link intersects the line of centres, is outside either centre, A or B, the cranks will turn in the same direction, and in opposite directions when C is between the centres. The connecting-rod of an engine, the pitman for giving motion to the sickle of a mower, and the valve-rod of a Corliss engine are examples of link-work.

Link-work is the lightest running mechanism known, the only resistance being due to the slight friction made by comparatively small pins in well-oiled bearings. A pin rarely makes more, and usually much less, than one complete turn in its bearing in a complete movement; while in the corresponding movement with a cam the roller (in the best arrangement) makes from 6 to 12 or more turns on its pin, and even this is not so prejudicial as regards resistance as the movement of a roller along the surface of the cam-groove. For the above reason, link-work is much more durable than other forms of mechanism; hence it should be adopted, wherever possible, in preference to toothed gearing, cam-work, belted gearing, etc. Most link-work belongs to the class in which the axes are parallel, and some examples will be given in order to explain their action and use.

Fig. 89 is a diagram of the Corliss valve-gear. The

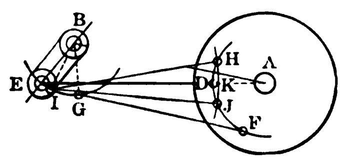


FIG. 89.

wrist-plate makes a partial rotation in order to carry the pin D back and forth from H to F. The valve and the stem at B are moved by the link BE as it swings from BE to BG.

As another example * suppose that a point is required to move from E to F, Fig. 90, and return within the sixth part of the revolution of the main shaft A, and then allowed to remain quiet at E for the remaining five sixths of the turn. By the use of a cam this movement can be easily made; but if there is no particular objection to the additional movement from E to G and return, link-work may be employed, as follows: A is the main shaft, AD the crank, De the pitman, and $eBG = HBF$ a bell crank-lever, the arrangement being such that while the crank-pin moves from c to b the required sixth part of the turn of A, a moves to H and back, and E moves to F and back, thus meeting the essential conditions of the movement.

* Robinson's "Principles of Mechanism."

Paths of Various Points.*—In the study of link-work it is often desirable to determine the different simultaneous positions of each of the joints, beginning with the driver. These positions

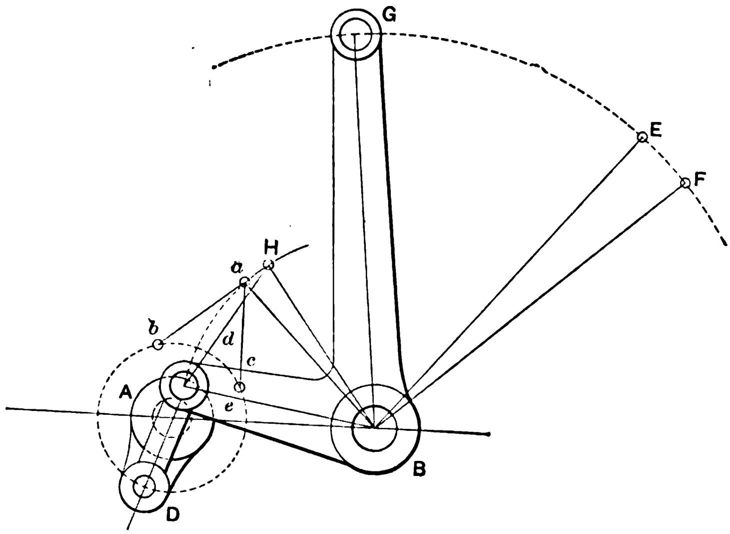


FIG. 90.

for uniform motion should be equidistant, as shown in Fig. 91. The crank-pin D moves around the circle 1 2 3 4 5 6 7 8

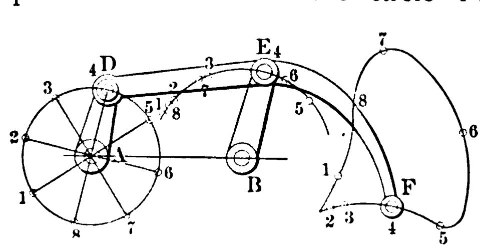


FIG. 91.

with A as a centre. The link BE moves about B as a fixed centre and about E as a movable centre, E describing an arc 1 8 2 7 3 4 6 5.

It is required to find the path described by the point F.

By an examination of the figure it will be seen that the shortest distance between D and F, and the shortest distance

* Robinson's "Principles of Mechanism."

between E and F, will not change for the different simultaneous positions.

Suppose the crank AD to be at a point 1 on the circle described by the crank; with a radius equal to DE, and with 1, 2, 3, etc., as centres, describe arcs. Their intersections with the arc 1 8 2 7 3 4 6 5 will be the various positions of E. Again, with DF as a radius, and 1, 2, 3, etc., of the circle 1 2 3 4 5 6 7 8, as centres, describe arcs. Also with 1, 2, 3, etc., of the arc 1 8 2 7 3 4 6 5, as centres, and with radius EF, describe other arcs. The corresponding arcs intersect at 1, 2, 3, etc., which are required points of the path of F. Other points may be found in the same manner and the path more fully determined.

Equivalents for Link-work.—For every elementary combination in link-work the equivalent motion can be obtained by wheels in rolling contact, these wheels being non-circular in form (see Fig. 92).

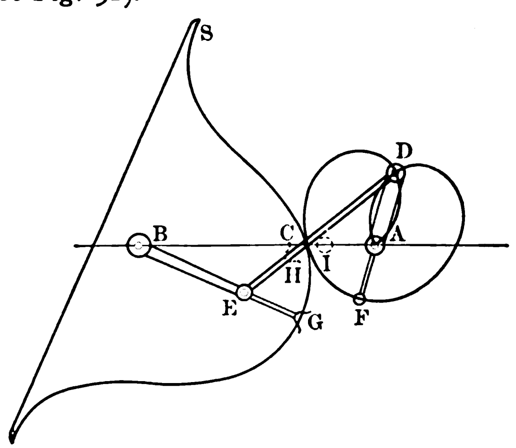


FIG. 92.

Dead-points.—One of the objections to the use of link-work, which has to be provided against, is what is known as the dead point.

A dead-point or a dead-centre is a point or set of points or positions of the links at which, if a certain one of the links in combination be made driver, the linkage will be found locked.

An example often seen is that of the crank and connecting-rod of an engine. When the crank and connecting-rod are in line the crank cannot be started by any amount of force applied to the cross-head. It is evident that in the combination shown in Fig. 93 the shorter lever is capable of turning completely

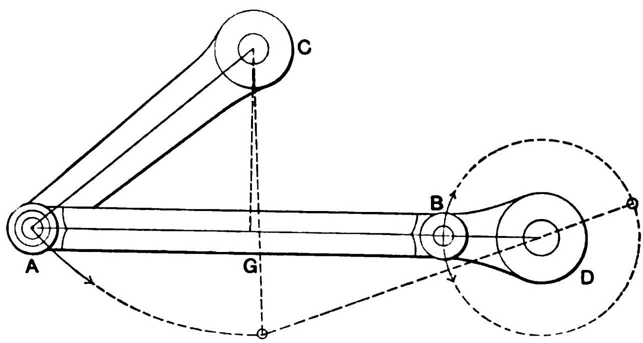


FIG. 93.—Dead-point.

around, hence it is called a crank. It is obviously possible for the system to come into either of the positions shown in Figs. 93 and 94 in which *AB* and *BD* coincide. This position in which the links coincide is called the dead-point.

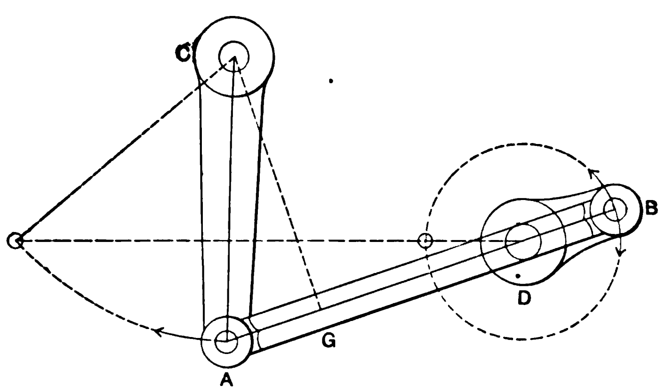


FIG. 94.—Dead-point.

Dead-points are provided against by special attachments for the purpose. In the steam-engine the fly-wheel serves the purpose, the momentum of the wheel carrying the crank over the dead-centre. Sometimes springs are used. In the single-acting engine the crank must not stop on the dead-centre. In locomotives, two sets of cranks are used, being placed nearly at right angles to each other, so that while one crank is on the dead-centre the other is acting at its best advantage, thus entirely obviating the liability of a dead-point. Sometimes an extra link is added, as in Fig. 95, in order to destroy the dead-point.

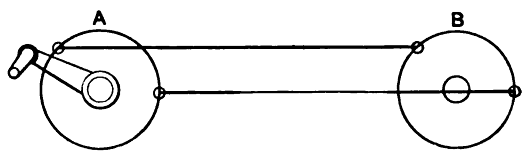


FIG. 95.

The examples and explanations thus far given of link-work have been of those in which the axes of the driver and follower were parallel. Its use may also be extended to work in which the axes intersect. This is sometimes called conical work or solid link-work, the principal essential being the bringing of all the axial lines of shafts and pins to a common point *O*, as in Fig. 96. The use of the conical link-work is similar to that

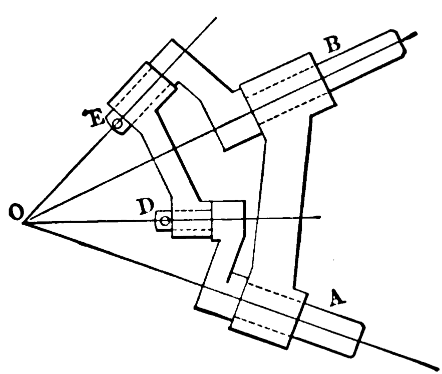


FIG. 96.—Conical Link-work.

of beveled and skew-bevel wheels in that it makes connections for shafts which intersect. Any of the examples under axes parallel may be carried into conical link-work, even to the extent of continued trains. As with parallel axes, every elementary conical link combination may have its equivalent in non-circular wheels in rolling contact, and will also be subject to the same conditions in regard to dead-points.

A simple example of this class of link-work may be seen in the Hooke's universal joint, which is employed as a shaft-coupling. The joint is shown in Fig. 97, where *A* and *B* are

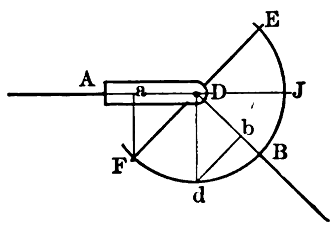


FIG. 97.—Hooke's Universal Joint

the shafts to be connected, *AD* and *FBE* half hoops between which is a cross with one branch at *EF*, parallel to the paper, and the other at *D* perpendicular to the paper. The branches of the cross are pivoted at *E* and *F*, and the two points *D* in *AD*.* Sometimes sliding parts are introduced into link-work, generally for simplifying the mechanism, a notable example being that of the cross-head of an engine.

* Robinson's "Principles of Mechanism."

CHAPTER XIII.

PIPE-FITTINGS.

Valves.—The term "pipe-fittings" is used to designate all the pieces necessary for the control of liquids and gases such as water, steam, oil, ammonia, etc., water and steam being the fluids most generally dealt with. These fittings are exclusive of the piping itself and may consist of valves and those parts of which the elbow and the plug are examples.

Valves may be divided into the following classes:

(1) *Lifting Valves;* as globe-, gate-, ball-, conical-, and some safety-valves.

(2) *Rotary Valves;* as cocks, faucets, throttles, etc.

(3) *Hinging Valves;* as clack or butterfly check-valves.

(4) *Spring-valves;* in which the valve is held on its seat by means of a strong spring, an example of which is the pop safety-valve.

(5) *Sliding Valves;* as the slide-valve on a locomotive.

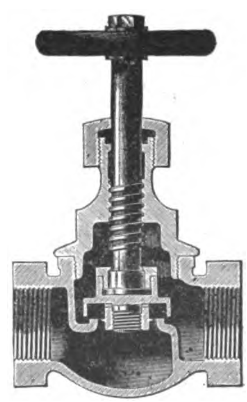


FIG. 98.
Jenkins Globe-valve.

The *Globe-valve* is the most generally used valve in pipe-work, and is used to control the passage of fluids through a straight pipe. It generally consists of a conical-shaped disk which fits in a similar conical-shaped opening, the raising or lowering of which causes the passage to be opened or closed. It derives its name from the external appearance, which is somewhat globular in form. The Jenkins Globe-valve has a flat disk and seat instead of a conical one. The valve-disk contains a vulcanized-rubber ring which rests upon the seat when the valve is closed. This makes a good valve because the rubber makes a water- or steam-tight joint. There are many valves which are similar to the Globe-valve, the difference being that they change the direction of the passing fluid; as the Angle-valve, the Cross- or Tee-valve, and the Y-valve.

The Gate- or Straight-way Valve.

By an inspection of the sectional cut which is shown, of the globe-valve, it will be noted that the fluid in passing through the valve does not move in a straight line but makes a turn of almost a right angle. This produces friction and retards the passage of the fluid to some extent. The gate-valve shown in section in Fig. 99 allows a straight passage. For the control of water, this form is especially desirable.

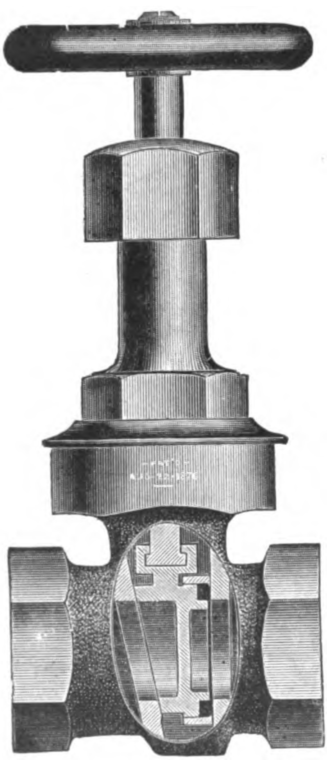


FIG. 99.—Jenkins Gate-valve.

A *Check-valve* is used where it is desired that a fluid may pass in one direction only and be prevented from going back by the action of the fluid itself. The valve is automatic, the backward pressure of the fluid pushing the valve against the seat and the forward pressure pushing it off the seat. The check-valve may be a lifting-valve, a butterfly-valve, or ball-valve.

The *Throttle-valve* is used on engines for turning the steam on or off. The globe-valve is often used for this purpose, but the throttle-valve has the advantage that it is much quicker in its action. This is very desirable especially in case of accident.

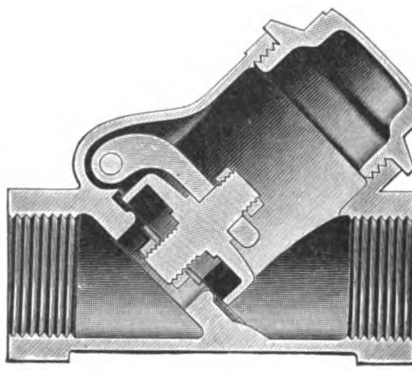


FIG. 100.—Check-valve.

The throttle is found on all locomotives. It may be of the rotary or sliding type of valves, or a double poppet-valve, shown at *C*, Fig. 150.

An example of the Clack-valve may be seen on the plunger of a suction-pump, Fig. 202.

When a globe-valve leaks it is generally caused by the wear of the valve-disk or the seat. This may be remedied by taking the valve apart and scraping the disk and seat until a good surface is obtained. With the Jenkins valve the leak can generally be stopped by putting in a new rubber disk. The *Valve-regrinder* is a small machine which is used for reseating valves which are worn. It is really a small lathe, which fits into the bonnet of the valve, with cutters that may be adjusted to a valve of any size. It may be used on the globe-valve with either the conical or flat seat. This is an invention which makes a great saving, because it makes possible the use of old valves which would otherwise have to be thrown away, besides the saving of time, it being used on the valve without taking it from its position on the line of piping.

The *Slide-valve* as used on steam-engines is shown in Fig. 177.

The *Poppet-valve*, which is also used in engines, air-compressors, etc., for the control of operating-fluid is shown in Fig. 232.

For different gases and acids it is necessary to use valves made of different materials; for instance, a valve used for the control of ammonia in ice-machinery is generally made of iron, the common brass valve being subject to destructive chemical action. Small valves are generally made of brass, but where large piping is used or where large pressures are to be resisted *iron-body valves* are used.

The *Back-pressure valve* is similar to the check-valve in that it allows a fluid to pass in only one direction. It differs from the check-valve, however, in having its valve held upon its seat by means of a lever and ball as shown in Fig. 101. With this arrangement, the fluid cannot pass except at

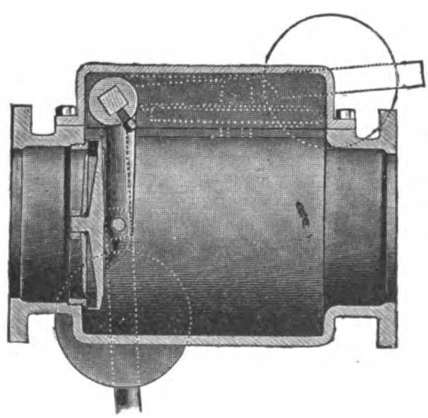


FIG. 101.—Jenkins Back-pressure Valve.

or above a pressure which can be fixed at will by moving the weight further away from or closer to the pivoted end of the lever.

A section of a *reducing-* or *regulating-valve* is shown in Fig. 102. This valve is designed to reduce and maintain even steam-pressure, regardless of the initial pressure. It will automatically reduce boiler or air-receiver pressure in all places when it is desirable to use lower pressure than that of the boiler or receiver. An example of its use may be noted with the steam-accumulator, Fig. 210.

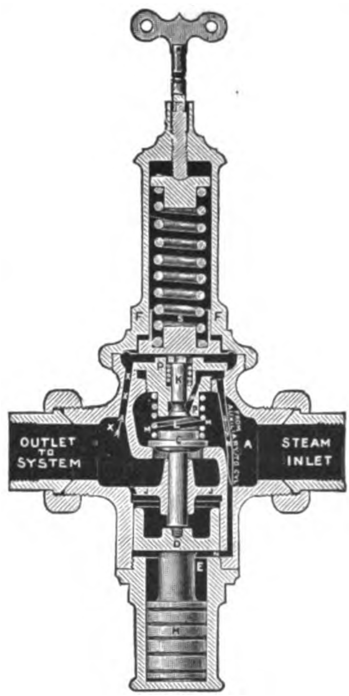


FIG. 102.—Mason Reducing-valve.

The following is its mode of operation: Steam from the boiler enters at side "steam-inlet" and passing through the auxiliary valve *K*, which is held open by the tension of the spring *S*, passes down the port marked "from auxiliary to cylinder" underneath the differential piston *D*. By raising this piston *D*, the valve *C* is opened against the initial pressure, since the area of *C* is only one half of that of *D*. Steam is thus admitted to the low-pressure side, and also passes up the port *XX* underneath the diaphragm below *S*. When the low pressure in the system has risen to a required point, which is determined by the tension of the spring *S*, the diaphragm is forced upward by the steam in the chamber, the valve *K* closes, and no more steam is admitted under the piston *D*. The valve *C* is forced to its seat by the initial pressure, thus shutting off steam from the low-pressure side. This action is repeated as long as the low pressure drops below the required amount. The piston *D* is fitted with a dash-pot *E*, which prevents chattering or pounding. From the description it is seen at once that this is both a reducing- and a regulating-valve.

Piping is generally made of wrought iron, sometimes cast iron. In all steam-piping wrought iron is used. An example of the use of cast iron is that of the piping used in water-mains, the different sections of pipe being put together by what is known as the "*socket-and-spigot*" *joint*, or more commonly called the *bell-joint*. One end of each section is enlarged to a bell shape into which fits the end of the next section, a joint being made by pouring in molten lead and filling up the space between the two pieces. With wrought-iron pipe the joint is made by the use of screw-threads or flanges and bolts.

There are two kinds of pipe-fittings in use, those which join to the pipe by means of screw-threads and those which join by means of flanges. For small work the screw-thread joints may be used, but for heavy work the flange-joints are better.

Fig. 103 is an *Elbow* which is used for making a right angle in a line of piping.

Fig. 104 represents a *Tee*. Fig. 105 is a *Nipple*. Fig. 106 represents a *Coupling* which is used to join the ends of two

FIG. 103.—Elbow.

FIG. 104.—Tee.

FIG. 105.—Nipple.

FIG. 106.—Coupling.

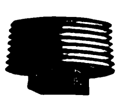


FIG. 107.—Plug.

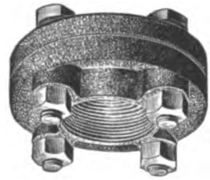


FIG. 108.—Flange-coupling.*

pipes. Fig. 107 shows the *plug* which is used for closing the end of a pipe.

* The above cuts are furnished by the Lunkenheimer Company.

Fig. 108 shows the *Flange-coupling*. A flange of cast iron is screwed to the end of each of the two pieces of pipe and the flanges are then fastened together by means of bolts. In order to make a tight joint a *gasket* made of rubber or copper is generally used. When the screw-thread connection is made, a tight joint may be obtained by putting *red lead* on the threads. It should be remembered that the examples given above are not meant to include all the fittings used in practice but are mentioned only to give a general idea of what they are like.

It should also be remembered that any of the above may be made with either the screw-thread or flange-joint. For instance, the elbow may be fastened to the pipes by either screw-threads or flanges. In the latter case the elbow is cast with flanges on it. Other fittings are the *Cross*, the *Union*, the *45° Elbow*, the *Cap*, the *Right* and *Left Coupling*, the *Reducing Coupling*, the *Return Bend*, and others too numerous to mention. When the diameter of a pipe is spoken of the inside diameter is meant.

PART II.

STEAM-POWER.

CHAPTER XIV.

STEAM-BOILERS.

STEAM for use in steam-engines is made in a closed vessel, called a boiler. It is necessary that the vessel should be closed in order that pressure higher than that of the atmosphere may be obtained. Steam may be generated in an open vessel, but it is evident in this case that the pressure of steam is equal to that of the atmosphere. The apparatus for manufacturing steam, including the boiler, the furnace, the chimney, the feed-pump, etc., is called a steam-boiler plant. The manner of producing steam is as follows: Fuel is burned in a furnace which is so situated with respect to the boiler that the heat from the burning fuel comes in contact with the surface of the boiler, part of the heat being taken up by the water in the boiler, thereby converting the water into steam. The remaining heat is either taken up the chimney by the force of the draft or is lost by radiation from the furnace, boiler, and pipes, or by leakage. The office of the furnace is to generate heat, that of the chimney to carry off the products of combustion and create a draft, and that of the boiler to transfer heat into the water, producing steam, and to confine the steam under pressure.

Classes of Boilers.

Steam-boilers may be classed as either *Fire-tube boilers* or *Water-tube boilers*, according to whether the heated gases pass through the tubes which are surrounded by water or around the tubes which contain water. These two classes may be further classed as Upright or Horizontal, Marine or Land, Internally-fired or Externally-fired.

Fire-tube Boilers.

Fig. 109 shows in elevation a fire-tube boiler, known as the return-tubular boiler, and Fig. 110 shows the same in section. The fuel is burned on the grate in the furnace. The heated gases rise and pass over the bridge-wall to the back end of the boiler, then return through tubes or flues to the front end; thence up the chimney which is over the front end of the furnace. Part of the heat of combustion is given to the under side of the shell of the boiler, the gases being somewhat cooled thereby. A further amount of heat is taken up by the water surrounding the tubes, as the hot gases pass through them. Thus the heated gases are made to remain in contact with the surfaces which are in contact with water as long as possible so as to utilize a large part of the heat. The volume of the boiler-shell consists of two parts, the *water-space* and the *steam-space*. The water-space consists of that part in which the water is contained. All above this is steam-space. The steam-space should be sufficiently high to separate the particles of water carried up into the steam by the disturbances caused by boiling. For the purpose of providing for dry steam, the steam-dome is often used, but it is now often dispensed with, as being unnecessary. From the top of the steam-space a pipe takes the steam off to the engine, thus drawing off the dryest steam.

Heating Surface.—That surface of the boiler which comes in direct contact with the heat from the furnace is called the

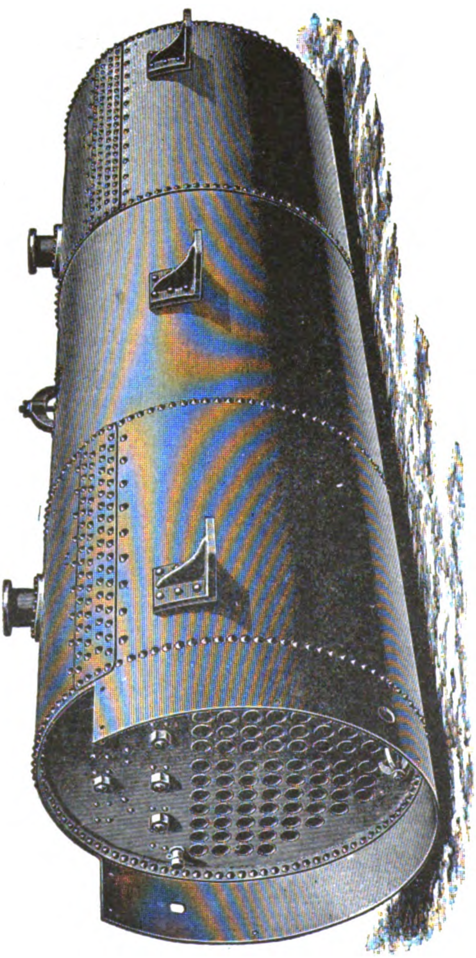


FIG. 109.—Elevation of Return-tubular Boiler.

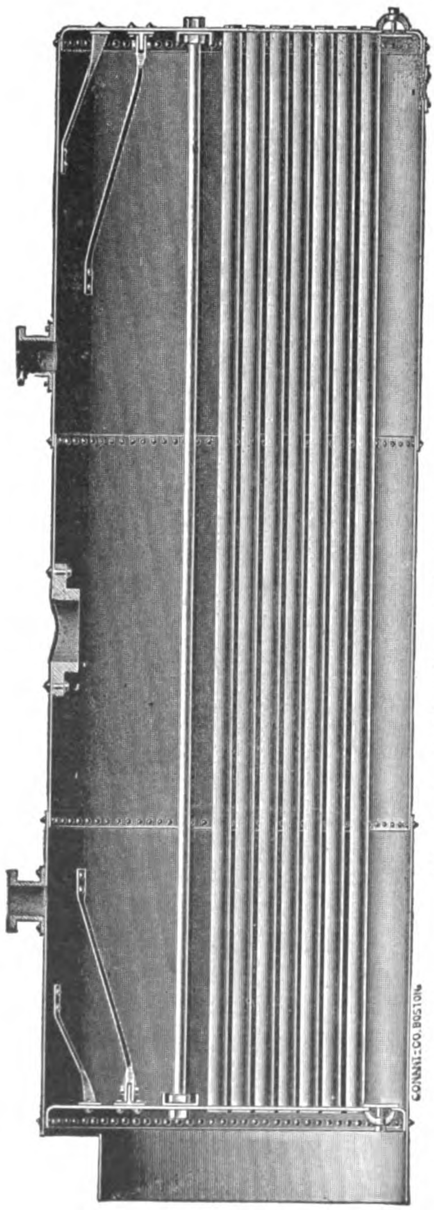


FIG. 110.—Section through Return-tubular Boiler.

"heating surface." In the return-tubular boiler it consists of the surface of the shell in contact with the hot gases (generally about two thirds of the boiler-shell) plus the inner surface of the tubes. Let $D =$ diameter of the shell in feet, L the length in feet, n the number of tubes, l their length in feet, and d their diameter in inches. Then the heating surface of shell in square feet $= 3.1416D \times L \times \frac{2}{3}$, the heating surface of tubes in square feet $= \frac{3.1416d \times l \times n}{12}$. Total heating surface $= \frac{2}{3} \times 3.1416D \times L + \frac{3.1416d \times l \times n}{12}$. The shell of this type of boiler is made of steel or wrought iron. Its thickness is usually from $\frac{1}{4}$ to $\frac{1}{2}$ inch. The plate is put together with rivets. One of the greatest causes of trouble with this type of boiler is due to the expansion and contraction of the tubes, which is apt to cause leakage where the tube joins the head-plate. Tubes with diameters less than 6 inches are called *tubes*, while those with diameters larger than 6 inches are generally called *flues*. The former are fastened to the head-plates by means of tube-expanders, the latter are riveted in. The ends of the boiler above the tubes are strengthened by tying them to the side of the boiler by means of *stays* (see Fig. 110). This type of boiler gives good results when correctly proportioned and well managed. One objection to it is that it sometimes is the cause of great loss of life by explosion, by reason of having the whole steam-pressure in one large vessel.

The fire-tube boiler may be classed as either externally-fired or internally-fired. The externally-fired boiler has already been described.

Internally-fired Boilers.—The internally-fired boiler has its furnace within the shell in a flue which is made large enough to contain a grate. The gases may pass to the back in this flue and then return to the front in small tubes above the fire-flue as in the case of the return-tubular boiler, or they may pass to the back and then to the front through openings in the brick setting and then to the back again and out the chimney by another passage under the shell. The latter plan is illustrated by Fig. 111, showing the Cornish boiler, one of the earlier forms of boiler, of English origin.

An example of the former is shown in the Cylindrical Marine, or Scotch, boiler, Fig. 112, in which the combustion takes place in two large flues, passes to the "back connection" and then to the front through small tubes above the

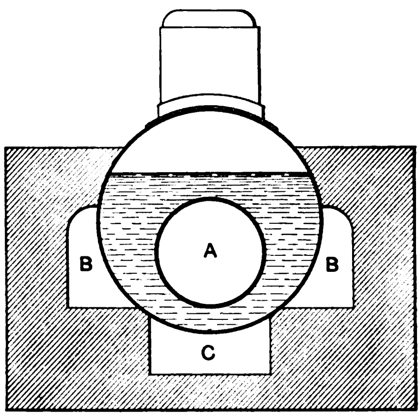


FIG. 111.—Cornish Boiler.

flues and up the stack. The Locomotive boiler, Fig. 113,* is an internally-fired boiler, in which the heated gases of combustion pass directly from the front to the rear of the boiler through fire-tubes and then out through the back connection and the smoke-stack. The fire-box is of a rectangular form as shown in cut.

Water-tube Boilers.—As before stated, the water-tube boiler has its tubes filled with water, and surrounded by hot gases. Fig. 114 shows a side view of the Babcock & Wilcox boiler, which is an example of this type. The boiler is made up of a number of vertical sections, standing side by side. Each section consists of several tubes connected one above the other, at each end to a *header*, the headers being connected at the top to a large steam- and water-drum. Fig. 115 shows one

* Figs. 109, 110, 112, 113 are furnished by Messrs. E. Hodge & Co.

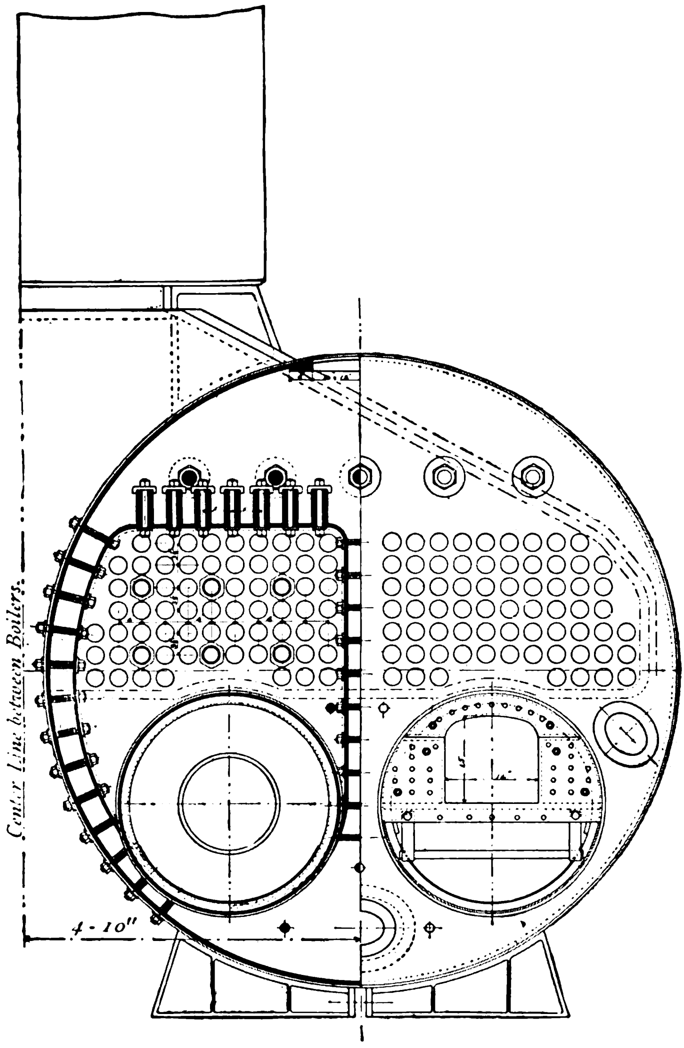


FIG. 112.—Scotch Boiler.

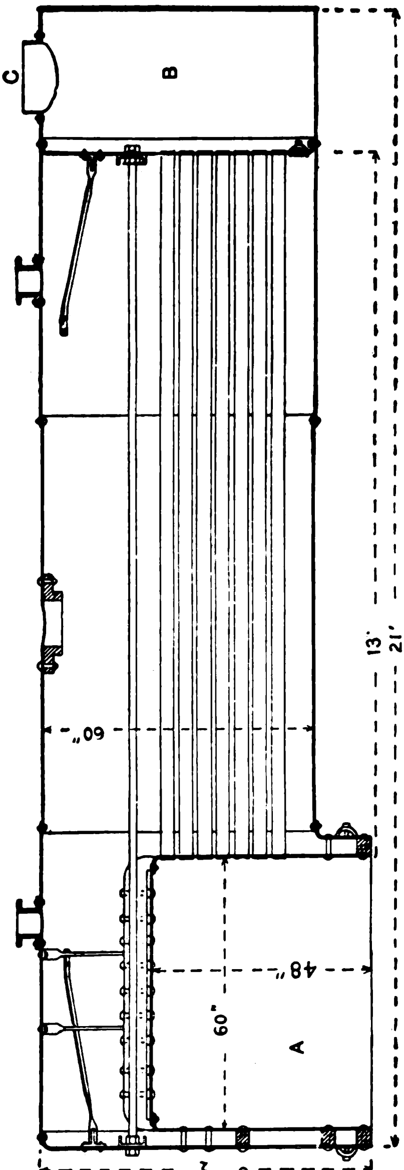


FIG. 113.—Locomotive Boiler.

of these headers. It is made of cast iron or wrought steel. The tubes are placed over each other in zigzag fashion in order to intercept as much as possible the gaseous currents. The tubes are made of wrought iron or steel, and are fastened to

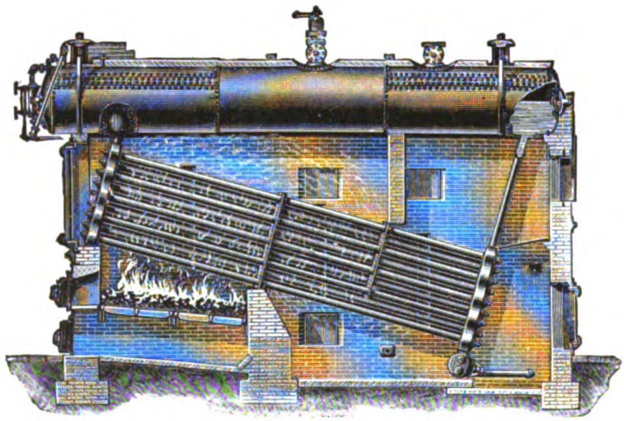


FIG. 114.—Babcock and Wilcox Boiler.*

FIG 115.—Babcock and Wilcox Header.*

the header by means of a tube-expander. Fig. 116 shows a portion of the end of a tube and a header in section showing the arrangement for getting at the end of the tubes for the purpose of inspection and cleaning. By taking off the hand-hole nut the hand-hole cover may be taken off and an opening made in the header-wall opposite the wall entered by the tube. The same figure shows how the feed-water enters near the bottom of the drum. The tubes slant downwards toward the back in order to facilitate the circulation of the water. The heated water tends to rise and the water in the drum above flows down through the rear headers to take its place. A mud-drum is

* From "Steam," by permission of the Babcock and Wilcox Co.

connected to the lower end of the rear headers, as shown in the side elevation, Fig. 114. Sometimes the tubes are all fastened to one header at each end as in the Heine boiler,

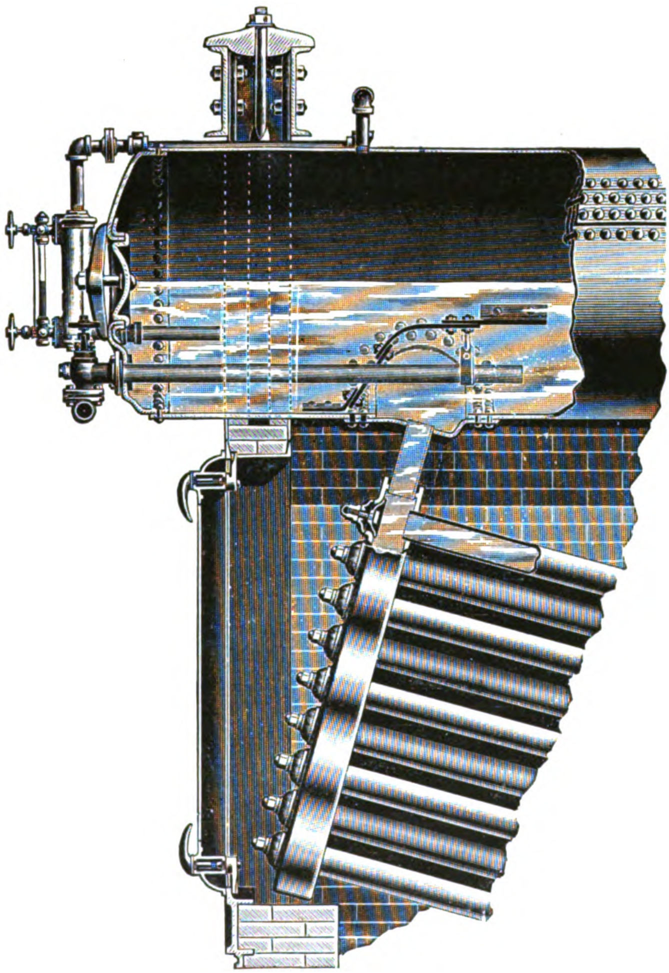


FIG. 116.—Babcock and Wilcox Header and Drum.

Fig. 117. The boiler, however, made up of sections, has the advantage of being more easily handled.

Another type of the water-tube boiler is shown in the Stirling boiler, Fig. 118, which consists of three large drums connected to a fourth drum below them by means of nearly vertical small tubes. The hot gases come in contact with these small tubes and part of the drum-surface, being made to pass in contact at the proper time and in the proper direction by means of *deflecting-plates*, as shown in the figure.

Water-tube boilers are "quick steamers" because of the small volume of water in each tube, each tube also being surrounded almost entirely by hot gases. They have an advantage also in that the pressure is confined mainly in small tubes.

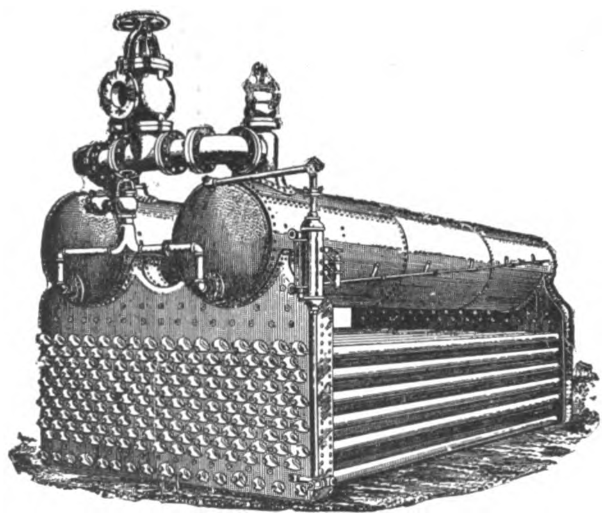


FIG. 117.—Heine Boiler.

This makes the resistance to rupture much more perfect, and for this reason many makers of water-tube boilers call their boilers "safety-boilers." The heating surface of water-tube boilers consists of the outside surface of the tubes, headers, etc., in contact with hot gases.

BOILER-SETTING.

By the term *setting* is meant the general arrangement of the boiler, furnace, and chimney with regard to each other and the manner of closing the boiler in. By an inspection of the

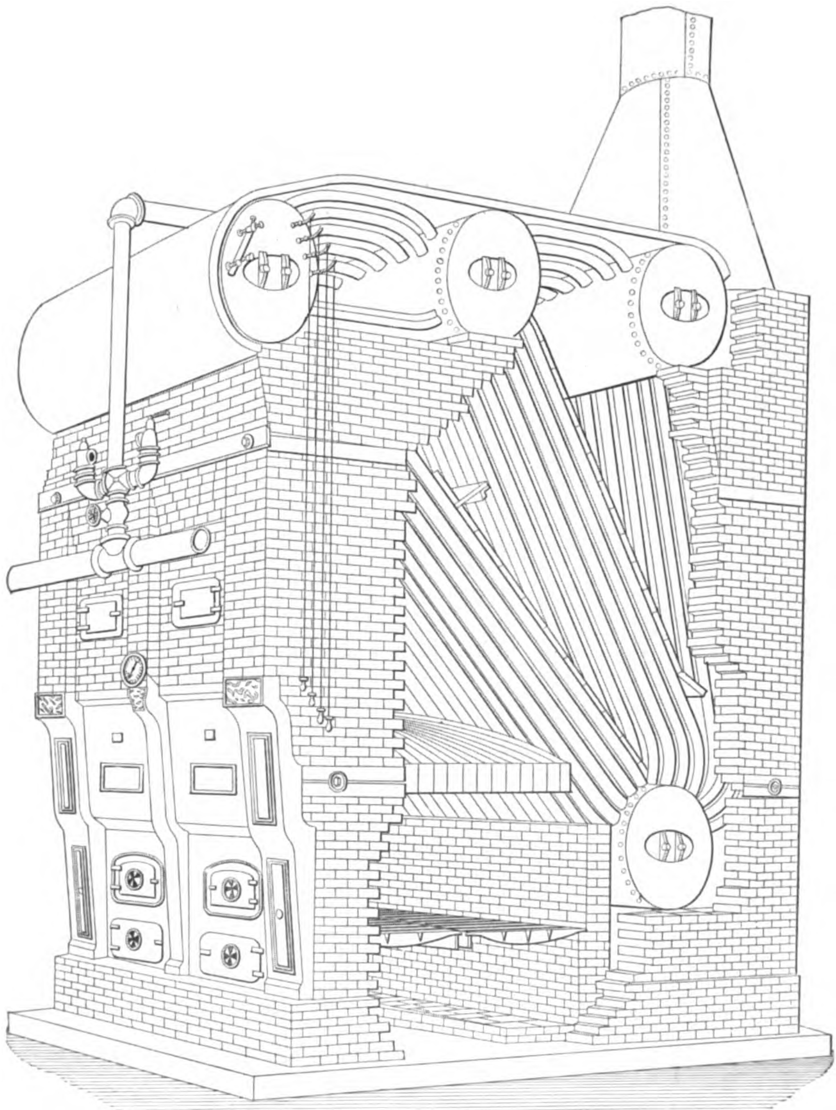


FIG. 118.—Stirling Boiler.

different forms and shapes of the boilers already described it is readily seen that the setting will depend largely upon the make of the boiler. Fig. 119 and Fig. 120 show two views of the

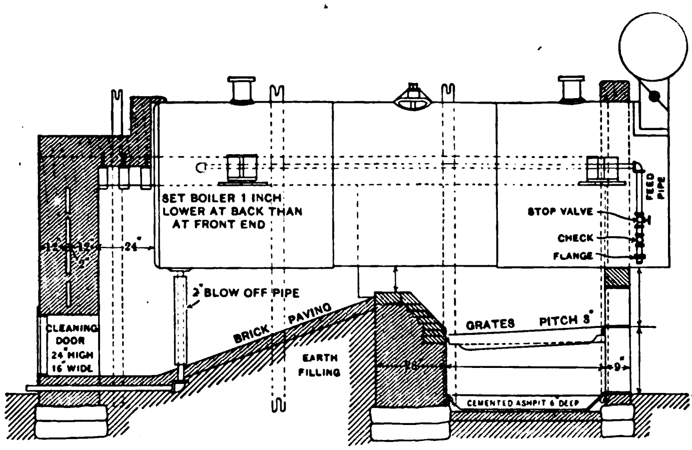


FIG. 119.—Boiler-setting.

"setting" for a return-tubular boiler. The walls of the furnace are sometimes "battered," that is, the inner surface of the walls is inclined outward from the vertical. This enlarges the combustion-chamber.

The line along the side of the boiler at which the brick-work joins the shell is called the *fire-line*. This determines the amount of shell-surface in direct contact with the heat. It is generally about two thirds of the boiler-surface. The walls should be thick enough to prevent radiation. It is best to have a double wall with an air-space between. This prevents cracking and consequent leakage of heat of the walls due to unequal expansion. This double wall is equally important where there are several boilers side by side (called a *battery*).

The grate-surface is made up of a number of grate-bars placed side by side. The grate-bar is seldom longer than 4 feet. The total length of grate is seldom more than 7 feet. For this case it is necessary to have two lengths of grate-bars placed end to end and supported at the middle. The grate rests on the dead-plate at the front and on the bridge-wall at the back end. The grate is inclined slightly from the front toward the back end, usually about $\frac{3}{4}$ inch fall for every foot of length of grate.

The bridge-wall rises from the back end of the grate, making an angle of about 45° with the horizontal. The object

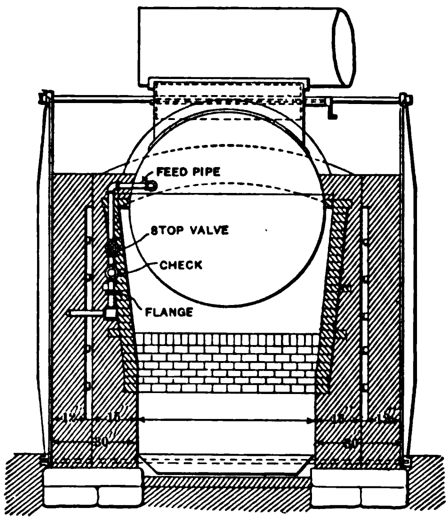


FIG. 120.—Boiler-setting.

of this is to deflect the furnace flames in such a manner as to spread them and cause them to follow along the under surface of the boiler. On this account the hottest part of the boiler-furnace is just behind the bridge-wall. Sometimes the top of the bridge-wall is a straight horizontal line as in Fig. 120. In other cases it is curved to conform to the shape of the boiler.

The ash-pit is the space between the grate and the bottom of the furnace. It is formed by the sides of the furnace and paved at the bottom, usually in order that water may lie in it.

The flame-bed is that part of the furnace just back of the bridge-wall and extending to the back end of the furnace. It inclines downward from near the top of the bridge-wall to the level of the boiler-room floor. This inclination causes the soot and ashes drawn over the bridge-wall by the force of the draft to fall towards the back. In this way the cleaning is facilitated. A cleaning-door is made in the back wall through which ashes, etc., are taken out. The boiler is set a trifle lower at the back end than at the front end. This is done in order to drain the boiler toward the blow-off pipe.

Hanging the Boiler.—Supporting the heavy weight of the boiler itself and its volume of water is effected in two ways: The boiler is supported at its sides by the furnace-walls, or it is hung from beams above by means of links and eyes attached to its upper part. The first plan is carried out by riveting "brackets" of steel or cast iron, at least two to each side, to the boiler as in Fig. 109. These brackets rest on a place made for them on the walls. The walls of the furnace are apt to be spread apart by reason of the expansion of the boiler when heated, if the bracket is fixed movably on the wall.

Rollers are sometimes put between the bracket and its seat for the purpose of avoiding this trouble. Fig. 121 shows an

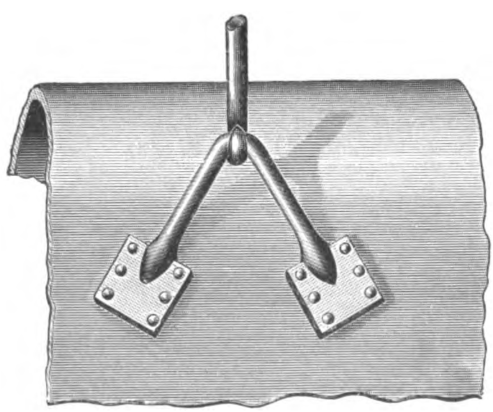


FIG. 121.—Eye Riveted to Boiler-shell.

eye riveted to the boiler-shell, into which is fitted a hooked link for supporting the boiler from above. Owing to the different forms and dimensions of water-tube boilers, no established rule can be adopted for setting them. Each make has its own special setting, which is usually specified by the maker. Fig. 114 shows the setting of the Babcock & Wilcox boiler, and Fig. 118 the setting of the Stirling boiler.

CHIMNEYS.

The connection of the chimney to the setting will depend upon the type of the boiler. For the return-tubular boiler shown in Figs. 109 and 110, the stack connects with the front end. For the locomotive boiler it is on the end of the boiler opposite the furnace. With most water-tube boilers the stack is at the back end.

The material used in building stacks is usually either steel, wrought iron, or brick. The brick chimney is of much greater first cost and it is apt to leak by reason of cracks. Its great weight makes the construction of the foundation a serious problem. The brick chimney is usually made with double walls with an air-space between. The outer wall is thick, and of ordinary brick, giving stability to the structure, while the inner wall is thin and made of fire-brick (see Fig. 122).

Steel or wrought-iron chimneys are made of sections riveted together. They may be made practically air-tight. The higher the chimney the greater the force of the draft. The air on the inside of the chimney, being heated, expands and becomes lighter and therefore lower in pressure than the air

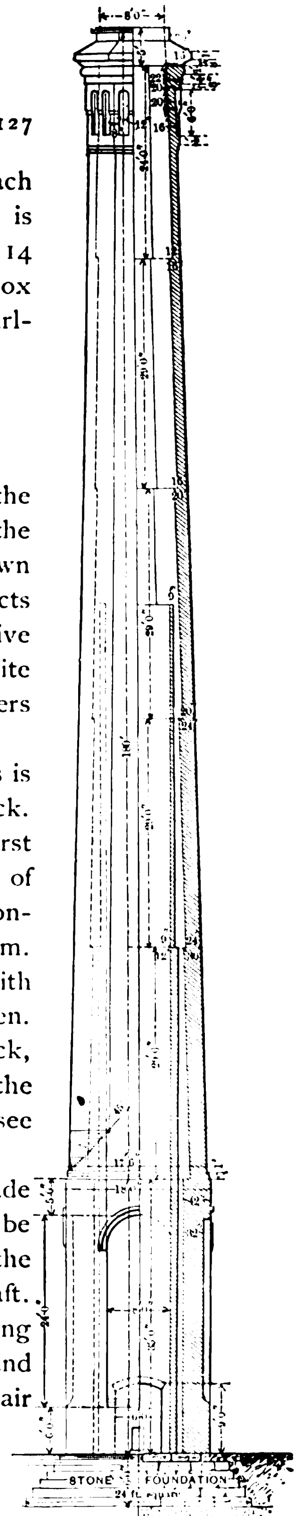


FIG. 122.—Chimney.

on the outside, hence the temperature of the gases entering the bottom of the chimney affect the draft force. The force of the draft is equivalent to the difference in weight of the column of heated gases inside of the chimney and that of an equal column of the outside air. The draft is usually spoken of as "so many inches of water." It is determined by means of the U-tube gauge shown in Fig. 123.

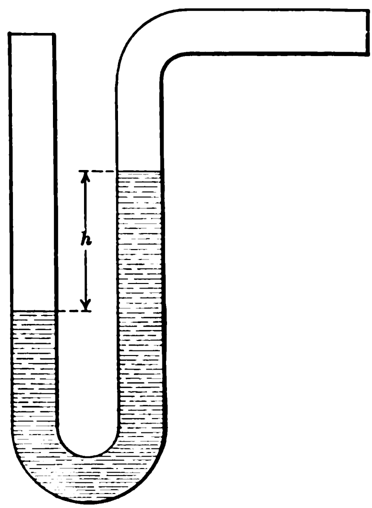


FIG. 123.—U-tube Draft-gauge.

One end is connected by a rubber connection or otherwise to the inside of the stack near the base, the other end being open to the atmosphere. The difference in level in the two legs measures the draft in inches of water $= h$ in the figure. It usually ranges from $\frac{1}{4}$ to $\frac{3}{4}$ inch.

The following formulas for dimensions of chimneys are given by Kent:

$$\text{H.P.} = 3.33E\sqrt{H} = 3.33(A - 0.6\sqrt{A})\sqrt{H}.$$

$$E = A - 0.6\sqrt{A} = \frac{0.3\ \text{H.P.}}{\sqrt{H}}.$$

$$H = \left(\frac{0.3\ \text{H.P.}}{E}\right)^2.$$

For round chimneys, diameter of chimney = diameter of $E + 4$ inches.

For square chimneys side of chimney in feet $= \sqrt{E} + 4$ inches.

In these formulas H.P. = commercial horse-power of boiler, 1 H.P. taken as equivalent to 5 lbs. of coal burned per hour; A = area of chimney and E = effective area in square feet. H = height in feet.

It is usual in chimney design to assume the height of the chimney such that the smoke, etc., is carried above the surrounding buildings and then find the corresponding area by means of the formula.

The passage which connects the different boilers of a battery to the main chimney is called the "breeching." It is generally made of sheet iron.

Forced Draft is the term used in designating all means of draft production other than that of the chimney. In the locomotive forced draft is produced by passing the exhaust steam through the stack in such a manner that a current of air is induced through the furnace. This method is also frequently adopted in stationary practice, but it has the objectionable features of making a disagreeable noise, especially in an iron stack, and of rusting the metal parts due to the moisture in the steam. It is also wasteful of steam, on account of its causing back-pressure on the engine.

The most common method of producing a forced draft is to produce a current through the furnace and stack by means of a fan or blower, which is usually run by a small independent engine. Here the force of the draft may be controlled by the speed of the engine. Fig. 124 shows an arrangement for producing induced draft. The fan is placed at the back end of the boiler. The fan consists of a number of vanes on a revolving shaft. It revolves within an iron casing. The small engine which runs it is shown at the right of the cut. Fig. 125 shows another method of producing draft in which the grate is made up of hollow pipes.

The pipes are perforated and air is forced through them and the fuel by means of a blower. By having forced draft

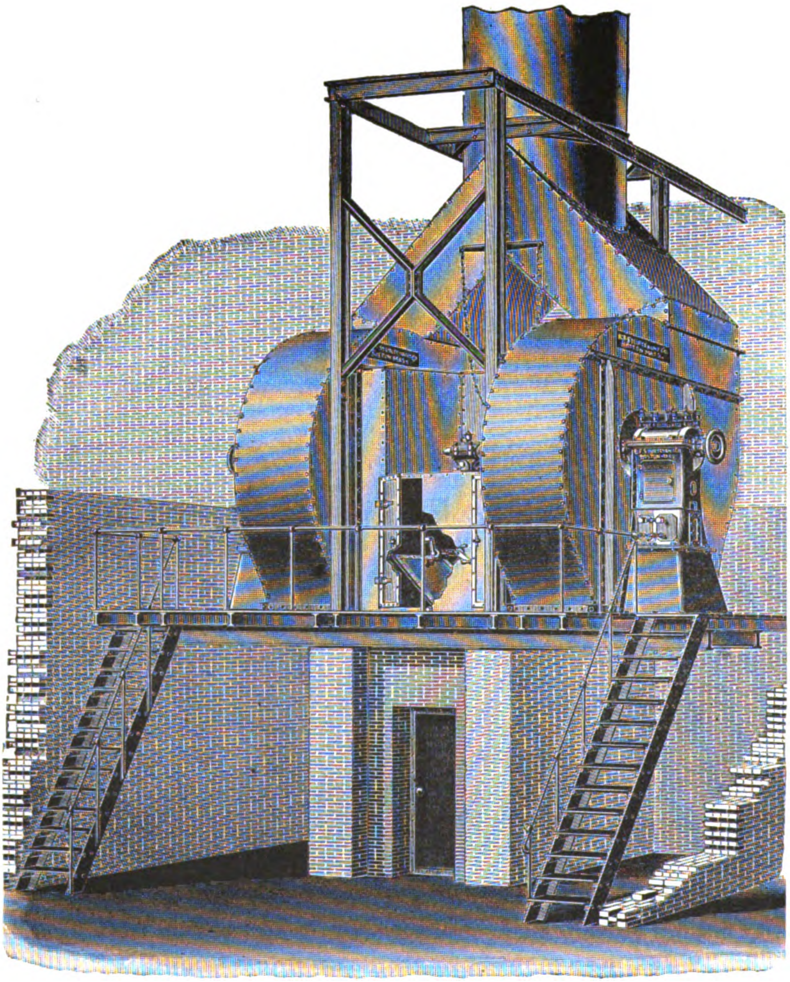


FIG. 124.—Sturtevant Blower Plant.

the chimney or stack needs only to be high enough to carry the smoke above the surrounding buildings.

FUELS AND COMBUSTION.

Combustion is rapid oxidation accompanied with heat and light.

The Rate of Combustion is measured by the number of pounds of fuel burned on one square foot of grate-surface in one hour. It depends mainly upon the nature of the fuel and the force of draft. A maximum rate is generally produced when forced

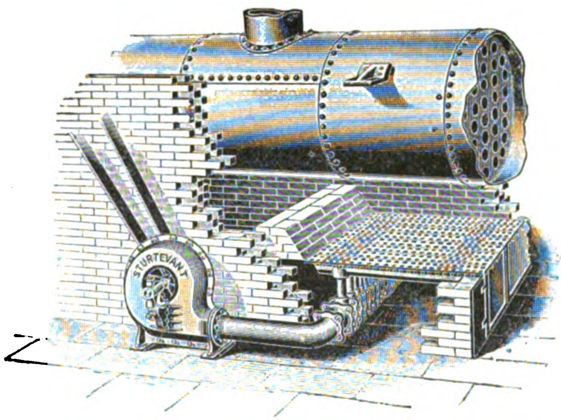


FIG. 125.—Forced-draft Fan.

draft is used with an inferior fuel, while the minimum rate is produced with natural draft and the best grades of fuel.

In locomotive practice a rate of combustion as high as 200 lbs. of coal per square foot of grate per hour is sometimes used. The ordinary rate for stationary boilers generally ranges between 10 and 25 lbs., and as low as 3 or 4 lbs. is sometimes found in small boilers used for steam-heating.

The principal elements in fuels are Hydrogen, Carbon, and Oxygen. Carbon may combine with oxygen and form two different chemical compounds, carbon monoxide, CO, and carbon dioxide, CO_2; that is, one atom of carbon and one of oxygen or one atom of carbon with two of oxygen. The former is produced by limiting the supply of air and the latter by an abundant air-supply. When it burns to CO_2 or carbon dioxide, 14,600 heat-units are produced per pound of carbon. When it burns to CO or carbon monoxide, 4450 heat-units per pound of carbon are produced. Hence 10,150 heat-units per pound of fuel are lost by improperly ventilating the fuel during combustion.

Hydrogen combines with oxygen in the proportion of 1 lb. of hydrogen to 8 lbs. of oxygen, or 2 atoms of hydrogen to 1 of oxygen. In so doing a pound of hydrogen gives up 62,000 heat-units. The oxygen in the fuel does not burn, but will neutralize one eighth of its weight of hydrogen. Then if C, H, and O, represent the proportions of carbon, hydrogen, and oxygen in the pound of fuel, the number of heat-units,

$$h = 14{,}600\mathrm{C} + 62{,}000\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right). \quad . \quad . \quad . \quad (1)$$

The proportion of different elements in a pound of fuel is determined by chemical analysis. Owing to the different wastes of heat by radiation, by dropping of fuel through the grate, by incomplete combustion, and by the hot gases escaping through the chimney, much of this heat is not taken up in producing steam. The amount utilized may range from 50 to 75 per cent of the total heat, depending upon the excellence of design, and upon the conditions of operation.

The *total heat required* to be generated in the furnace is calculated by first finding the weight of steam required per hour to run the engines or other apparatus. By the use of the steam-table the number of heat-units required to produce this weight of steam is found as follows: Let $H =$ the total heat above 32° of 1 lb. of steam, as taken from the steam-table for the given gauge-pressure, and let h be the difference between the temperature of the feed-water and 32°; then $H - h$ is the heat required to evaporate 1 lb. of water from the given feed-water temperature into steam of the given pressure.

If $K =$ the number of heat-units that may be generated by the complete combustion of 1 lb. of the coal which is to be used, and E the efficiency of the boiler which may be expected under running conditions (say from 0.5 to 0.7) then $\frac{H-h}{EK}=$ the quantity of coal required to generate each pound of steam.

Air Required for Combustion.

The quantity of air supplied to the furnace should be more than the quantity actually necessary for the combustion of the fuel. It is necessary to supply not less than about 18 lbs. of air to the furnace for every pound of carbon in the coal in order to obtain complete combustion.

Fuels.

The principal fuels used in boiler-furnaces are wood, coal, and petroleum. Wood and coal are burned on the ordinary grate. Petroleum requires a special apparatus for feeding which will be described later. Coal is used more than any other fuel. Wood is mostly used where local conditions call for it, as in a sawmill plant, or in a wooded locality. Wood burns with a bright flame and rapidly. Its percentage of carbon is comparatively small and that of the volatile gases large.

Coal may be divided into two great classes: Anthracite or hard coal and Bituminous or soft coal. Anthracite burns slowly with little flame. Its percentage of carbon is large and hence it is a great producer of heat. The percentage of volatile gases is small. Bituminous coal breaks easily, burns more rapidly, and with more flame than the anthracite. Its percentage of volatile matter is from 20 to 50.

A better classification is the following, taken from Kent's "Steam-boiler Economy."

	Fixed Carbon. Per cent. (In the Combustible.)	Volatile matter. Per cent. (In the Combustible.)	Heating Value per lb. Combustible.	Relative Value of the Combustible. Semi-bt. = 100.
Anthracite	97 to 92.5	3 to 7.5	14,600 to 14,800	93
Semi-anthracite	92.5 " 87.5	7.5 " 12.5	14,700 " 15,000	94
Semi-bituminous	87.5 " 75	12.5 " 25	15,500 " 16,000	100
Bituminous, Eastern	75 " 60	25 " 40	14,800 " 15,200	95
" Western	65 " 50	35 " 50	13,500 " 14,800	90
Lignite	under 50	over 50	11,000 " 13,500	77

The anthracite and semi-anthracite coals are found in eastern Pennsylvania; the semi-bituminous in a very narrow stretch of territory from central Pennsylvania to the southern boundary of Virginia; the eastern bituminous coals in the remainder of the Appalachian coal-field from northern Pennsylvania and Ohio to Alabama. The western bituminous coals and the lignites are found west of the State of Ohio. They are characterized by being high in moisture as well as in volatile matter.

The figures in the above table refer to the combustible portion of the coal, that is, the carbon and the volatile matter, not including the ash and the moisture. The percentage of ash varies greatly in all the several classes. It may be as low as 5 per cent and as high as 30 per cent.

Coke is made from bituminous coal in a manner corresponding to that of producing charcoal from wood. It produces little smoke and is an efficient heat-giving fuel.

Petroleum has a heating value of about 50 per cent greater than average good coal per pound. It is burned with less cost of labor in feeding it.

Firing.—The term firing is used to designate the process of burning the fuel on the grate. It consists of keeping the fire in a clean condition, regulating the draft-supply, etc. There are systems of firing, among which is the spreading system, in which the fuel is spread all over the grate-area in thin layers at frequent intervals of time. Thick layers always choke the draft and consequently cool the furnace.

The alternating system requires a wide grate. A charge is first put on one side and allowed to burn a while, when another charge is placed on the other side. This keeps one side of the furnace hot all the time and the volatile gases from the new charge pass over the hot part of the furnace and are burned.

The coking system consists of placing the charge upon the dead-plate in front of the grate and gradually pushing it back toward the bridge-wall. This makes the hottest part of the furnace toward the back, and the volatile gases, being driven off soon after the charge is placed in the front of the furnace, pass over this heated part at the back and are burned. This process of firing is usually carried out by means of a mechanical arrangement called a Mechanical Stoker. The fuel is placed in a hopper and is taken thence to a moving grate, either endless or reciprocating, which causes the fuel to move gradually to the back end, burning in the meantime. Fig. 126 shows the Roney stoker. The crank-disk which receives its motion direct from the engine gives motion to the whole stoker. The coal is placed in the hopper. It is pushed over the dead-plate to the grate-bars by means of the pusher and the feed-plate. The grate consists of bars arranged in steps which take an inclined and then a stepped position, which causes a movement of the fuel toward the back end. Motion is transmitted to the grate-bars through the connecting-rod and rocker-bar. The quantity of fuel fed out of the hopper is regulated by the feed-wheel.

Petroleum Fuel.—The usual method is to supply the oil to the furnace in the shape of finely divided particles, which are forced into the furnace by a steam- or air-injector. Steam or compressed air is made to pass through annular openings drawing the oil up. This makes a finely divided mixture of air and oil, or steam and oil, which is sprayed into the ordinary furnace and burned. Using steam for this purpose has the advantage of being the cheaper arrangement, but it has the objectionable feature of introducing moisture from the steam into the furnace. When compressed air is used an air-compressor supplies air to a large reservoir, which holds the air necessary to start again after the steam-pressure in the boiler is down. Fig. 127 shows the arrangement of a plant for burning petroleum by means of compressed air. Fig. 128 is a sectional view of an oil-feeder.

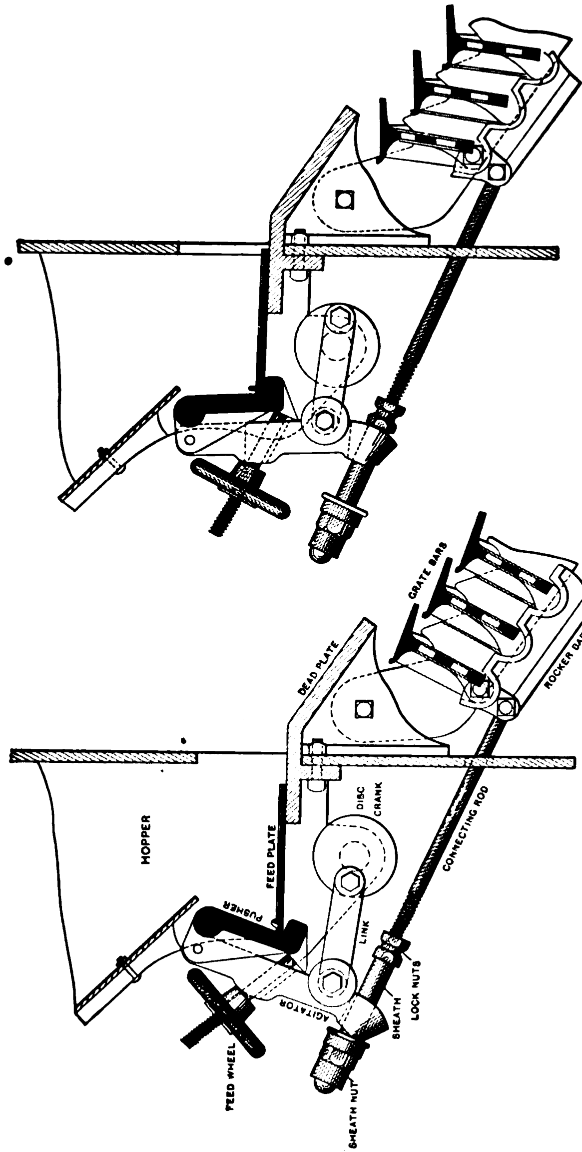
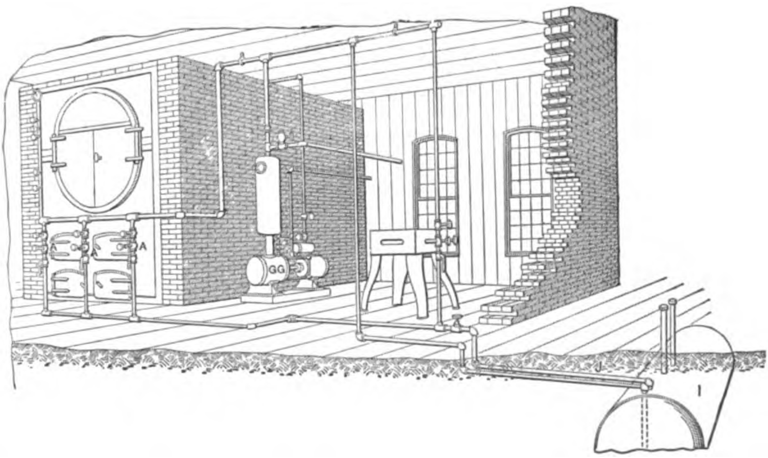
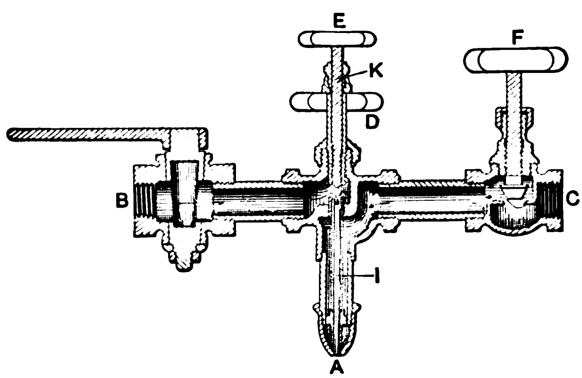


FIG. 126.—Roney Mechanical Stoker.



* FIG. 127.—Plant for Burning Petroleum Fuel. *I* = oil-tank ; *J* = pipe leading from *I* to the burner *A* ; *GG* = air-compressor which keeps a supply of air in the tank just by its side.



* FIG. 128.—Section of Burner. *C* = air-entrance ; *B* = oil-entrance ; *F* = air-valve ; *A* = pipe leading to combustion-chamber under boiler.

* From "A Treatise on Fuel," by Arthur V. Abbott.

BOILER ACCESSORIES.

The articles named and described under this head are usually necessary for any boiler no matter what the make or type.

Steam-gauge.—This is an instrument connected by a small pipe to the steam-space of the boiler for indicating at a glance the condition of the steam-pressure in the boiler. Fig. 129

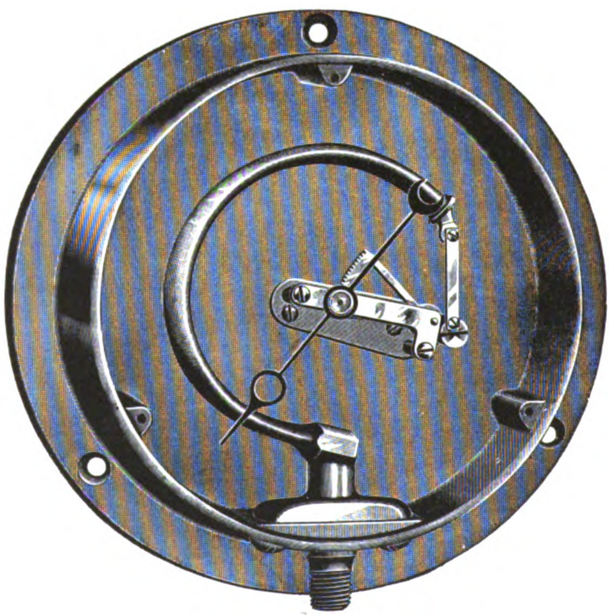


FIG. 129.—Ashcraft Steam-gauge.

shows the internal mechanism of a steam-pressure gauge. The steam-pressure enters it at the bottom and enters the flat curved tube, which is connected at one end to the needle. When the pressure is increased the ends of the spring spread apart and the needle is turned around the dial accordingly.

The Water-gauge.—This consists of a glass tube, generally about 12 inches long, one end of which is on a level with the water-space and the other on a level with the steam-space.

The water always stands at the same level in the glass as in the boiler, and the gauge thus shows at a glance the height of the water in the boiler. For the same purpose gauge-cocks, generally three, are placed in connection with the boiler, one with the steam-space, one with the water-space, and one at about the level of the water-line. By opening these the height of the water may be determined. They are used for the purpose of having a reliable source of information even if the gauge-glass should get out of order.

The Water-column.—The gauge-glass, pressure-gauge, and gauge-cocks are all usually connected to the water-column as shown in Fig. 130. The water-column is usually made of cast iron. In order to keep the tube in the gauge, to which the needle is attached, from becoming too hot and thus making an inaccurate reading, water is kept in it by means of a loop in the pipe-connection of the gauge, which loop is called a siphon.

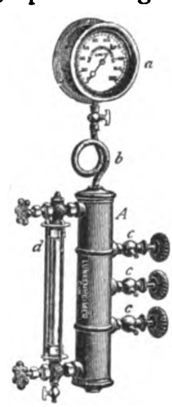


FIG. 130.—Water-column.*

Safety-valve. — All boilers are designed of such strength that they will confine the steam at a given pressure, and if this pressure is exceeded the boiler is liable to burst. Some method must be used for keeping the steam-pressure from rising above this given pressure. The safety-valve is used for the purpose. It consists of a valve so arranged that it will open and let the steam escape when the pressure for which it was set is reached. There are two kinds: the weight-and-lever safety-valve and the pop safety-valve. In the former a lever of the third class is used in connection with a heavy weight for retaining the steam. The pressure at which the steam blows off will depend on the distance of the weight from the fulcrum. Fig. 131 shows this safety-valve in elevation. The lever is pivoted at *C*. The pressure of steam is exerted upon the lever

* *A* = water-column, *b* = siphon; *c* = gauge-cocks; *d* = gauge-glass.

at B. Then applying the law of levers or moments we have, taking moments about B,

$$P \times s = W \times m + w \times n + w_1 \times s$$

or

$$P = \frac{Wm}{s} + \frac{wn}{s} + w_1,$$

in which $P =$ total upward pressure on the valve, W the weight of the ball, w the weight of the lever, w_1 the weight

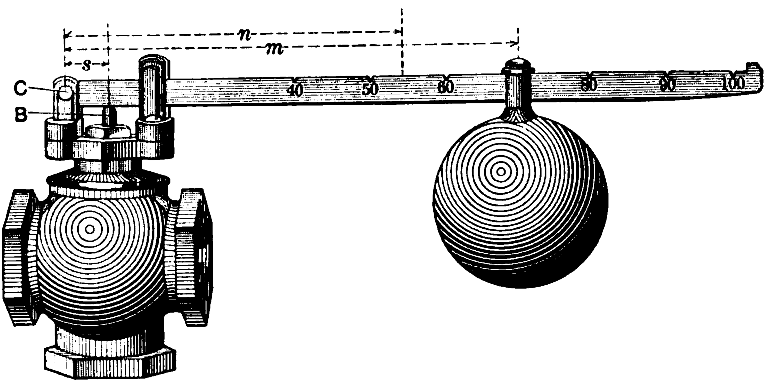


FIG. 131.—Safety-valve.

of the valve and spindle, s the distance from the centre of the fulcrum to the centre of the valve-spindle, m the distance from the fulcrum to the centre of the ball, and n the average lever-arm of the weight of the lever = the total length divided by 2.

The pop safety-valve gets its power to hold steam-pressure by means of a strong spring above the valve, which takes the place of the lever and weight of the other type. Different pressures are held by increasing or decreasing the tension in the spring by means of a nut. Fig. 132 is an example of this type.

Feed-water.—The water out of which steam is made is forced into the boiler against the steam-pressure therein by two distinct means; viz., the feed-pump and the injector. The

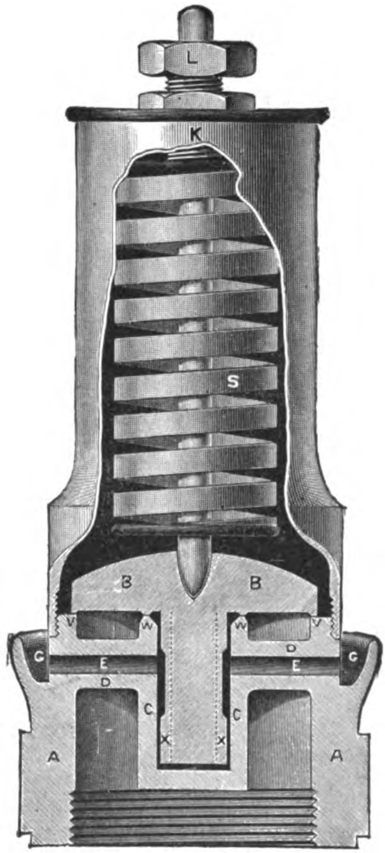


FIG. 132.—Pop Safety-valve.

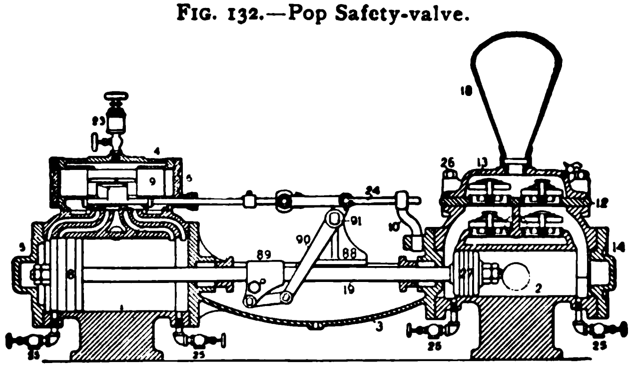


FIG. 133.—Feed-pump.

feed-pump puts water into the boiler cold or at the temperature at which it receives it, while the injector warms it in putting it in. The feed-pump is used principally in stationary practice. The injector may be used on any boiler; but is always found on the locomotive. The feed-pump is generally considered more reliable than the injector, but it takes up more room and is of greater first cost, besides a greater cost for repairs. Fig. 133 shows a sectional cut of the Deane boiler feed-pump. The water-piston is actuated by a steam-piston. The steam is controlled by a common D-valve. The valve-motion is produced by a vibrating arm.

The Injector.—The water is drawn through the injector and fed into the boiler by means of an induced current produced by the flow of steam from the boiler through the injector. Fig. 134 is a very simple example of the injector. Steam from the boiler enters at *V* and passes through *R* into the space surrounding *R* and *S*, where it is partially condensed. The condensation makes a vacuum, and besides this the velocity of the steam and water passing through *R* and *S* causes the water to be drawn toward it and through *S*, *Y*, and *O* to the boiler. The opening marked "overflow" is brought into use only while starting the injector, that is, when the water is to be lifted and put into the boiler, the entering stream will be blown to waste through the overflow during a few seconds necessary to exhaust the air in the water-supply pipe. As soon as the steam which is passing out of the overflow turns to water it shows that the lifting of the water has begun. The above makes plain the fact that for pumping water by this process there must be a condensation as well as a very high

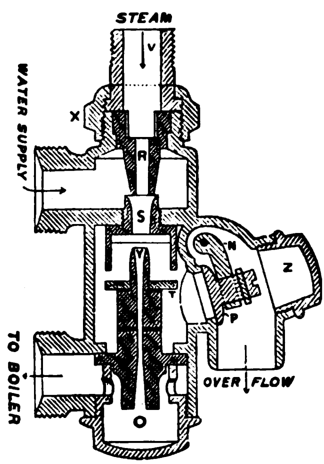


FIG. 134.—Penberthy Injector.

velocity of steam. The first requires that to be injected the water must be cold, otherwise the injector will not work. The high velocity of steam is produced by the rapidity with which it is condensed by the cold water. The injector cannot be used between the feed-water heater and the boiler because of the high temperature of the feed-water taken from the former. Fig. 135 shows another injector of more complicated design,

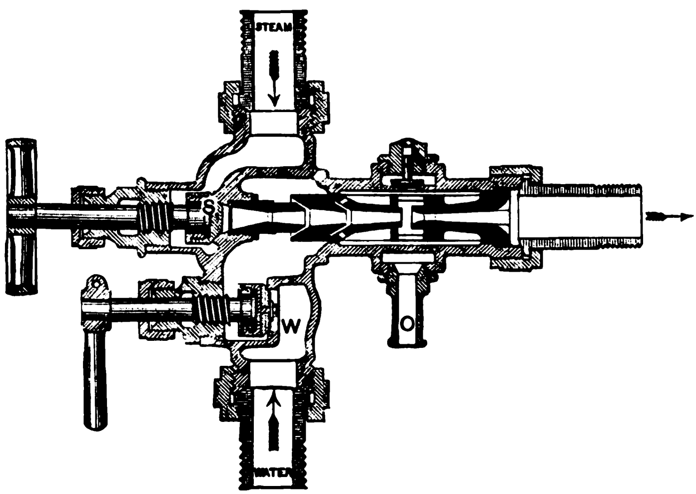


FIG. 135.—Monitor Injector.

the Monitor Injector. *S* is the steam-valve, *W* the water-valve, and *O* the overflow. To start, first open the steam-valve *S* a little to let the condensed water in the steam-pipe out through the overflow, and shut again as soon as clear steam appears. Next open the water-valve *W*, after which open the steam-valve slowly and the injector is at work. The maximum temperature of water at which this injector will cease to feed is about 130° F. The economy of the injector is due to the fact that it returns to the boiler all the heat of the steam required to operate it; nevertheless it is not as economical as a feed-pump used in connection with an exhaust-steam heater, for the latter utilizes heat which would otherwise be wasted.

Feed-water Heater.—The number of heat-units required to evaporate a pound of water is $H - (h - 32)$, in which $H =$ the total heat of evaporation, and h the temperature of the feed-water. By increasing h the heat required per pound of steam is reduced. As has just been shown, the injector does this, but at the expense of steam taken from the boiler. The exhaust steam from non-condensing engines, which would otherwise exhaust into the atmosphere, is generally used for heating the feed-water. The exhaust steam usually surrounds a number of pipes through which the feed-water circulates. Fig. 136 shows a sectional view of a heater of this type. The feed-water enters at the bottom at the right hand and passes through the tubes to the top and down the left-hand tubes and out to the boiler. The heat of the exhaust steam surrounding the tubes heats the water.

The Feed-pipe.—The pipe which takes the water from the injector or the feed-pump to the boiler is called the feed-pipe. It may enter the boiler at the bottom near the back end, in which case the boiler is lower at the back end, so that the feed-pipe may also serve as a blow-off pipe and to drain the boiler. A better method is to introduce the feed-pipe into the front of the boiler just above the top row of tubes and extend it a few feet within the boiler. By this means the contact of the cold water with the hot plate of the boiler is avoided, thus avoiding damage to the shell of the boiler. Another method is to utilize the heat from the chimney-gases for heating the feed-water. Coils of pipe through which the feed-water passes on the way from the feed-pump to the boiler are placed so that the flue-gases pass in contact with them, thus heating the feed-water. This apparatus is called an *economizer*, of which Fig. 137 is an illustration. Here the economizer is placed by the side of the furnace and in the flue from the back end of the setting around to the stack. The feed-pipe usually has two *check-valves* with a *gate-valve* between the boiler and feed-pump.

Blow-off Pipe.—When the boiler contains sediment due to the precipitation from impure water and the high temperatures, an opening is made somewhere in the lowest part of the boiler.

FIG 136.—Feed-water Heater.

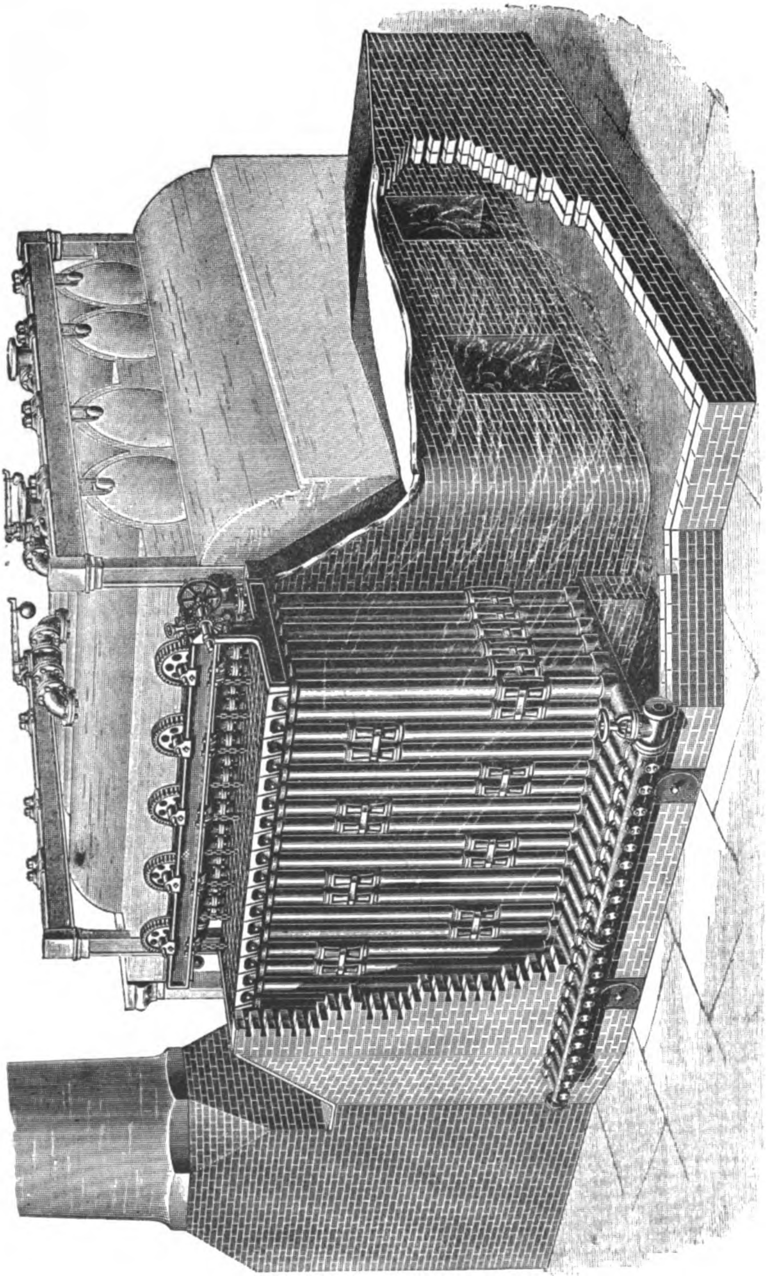


FIG. 137.—Economizer.

and the rush of the steam towards this point carries the impurities with it. For this purpose the blow-off pipe is placed at the lowest point of the boiler. By it the boiler may also be drained of water. The blow-off valve consists usually of a plug-cock. The blow-off pipe should be of large diameter in order to prevent clogging up by the acccumulation of solid matter in the bottom of the boiler. The blow-off pipe in water-tube boilers generally leads off from the mud-drum.

Dampers.—The draft in a boiler-furnace is regulated by means of the damper, or draft-doors. That is, by closing the draft-doors the draft stops and *vice versa*. This is frequently done automatically, by a mechanical device which causes the doors to close or open according to the variations of pressure of the steam in the boiler.

PROBLEMS.

1. A return tubular boiler has 82 tubes. The diameter of the boiler is 5 feet, the length is 18 feet. The length of the tubes is 16 feet and their inside diameter is $2\frac{1}{2}$ inches. Find the heating surface in square feet if the setting allows $\frac{2}{3}$ of the shell in contact with the fire.

2. What is the rated horse-power of the above boiler, allowing 15 square feet of heating surface per H.P.?

3. The grate for the above boiler is 4 feet 6 inches by 4 feet 6 inches. What is the grate surface in square feet?

4. What is the ratio of the heating surface to the grate surface?

5. On the grate given above a ton of coal is burned every 24 hours. Find the rate of combustion.

6. Find the number of lbs. of coal burned per hour per H.P. from the above.

7. A furnace burns 50 lbs. of coal per hour. A chemical analysis shows 60 per cent carbon, 5 per cent hydrogen, and 10 per cent oxygen. Find the number of heat-units generated per hour.

8. If 3000 lbs. of steam per hour is required from a boiler, the steam-pressure being 100 lbs. gauge, and the feed-water temperature

200° F., how many lbs. of coal per hour will be burned, the heating value of the coal being 13,000 heat-units per lb., and the efficiency of the boiler being 60 per cent ?

9. A round chimney is to be 100 feet high for a 200 H.P. boiler. Find the necessary diameter.

CHAPTER XV.

SIMPLE STEAM-ENGINE.

A STEAM-ENGINE is either *reciprocating* or *rotary*. All engines, whether reciprocating or rotary, are either *simple-* or *multiple-expansion* engines, and are also either *condensing* or *non-condensing*. Reciprocating engines also are either single- or double-acting.

In a reciprocating engine, the steam gives to a piston, operating in a closed cylinder, a back-and-forth motion. This back-and-forth or reciprocating motion is usually converted into rotary motion by means of a crank, but sometimes, as in the case of pumps, steam-hammers, etc., it is not so converted.

A rotary engine uses the expansive force of steam to produce rotation directly, without the interposition of cranks, pistons, etc. This style of engine is usually very uneconomical and but little used. Recently, however, the steam-turbine has been developed. It runs at a high rate of speed and is fairly economical in its use of steam. It will be discussed in the section on steam-turbines, Chapter XXII.

In a simple engine the steam does its work in one cylinder, and is then exhausted into the atmosphere or a condenser.

In a multiple-expansion engine the steam is exhausted from the first cylinder into a receiver. From the receiver it is taken into a second and larger cylinder, where it is used a second time. From this second cylinder it may either go to a second receiver to be used still another time, or it may go to the condenser, or exhaust to the atmosphere.

An engine using the steam twice is a compound engine;

one using it three times is a triple-expansion engine; one using it four times is a quadruple-expansion engine, etc.

A non-condensing engine exhausts directly into the atmosphere. A condensing engine exhausts into a condenser, where the exhaust steam is condensed by means of cold water, and is then removed to the atmosphere by means of a pump.

In a single-acting engine the steam acts only on one side of the piston. In a double-acting engine it acts on both sides of the piston alternately.

This chapter will treat only of reciprocating engines. Rotary engines will be discussed later.

The first engine to be considered is the simple, double-acting, non-condensing, D slide-valve engine, shown in Figs. 138 and 139. The various parts are named in the illustrations.

The cylinder is a hollow, cylindrical vessel, closed at both ends. The piston is a disk, flat or otherwise, which moves inside the cylinder with a reciprocating motion. The piston is fitted, so that it is steam-tight within the cylinder, by means of piston-rings. These are cast-iron rings, which are split diagonally at one point of their circumference, and which are sunk into grooves in the piston. When made, they are turned slightly larger than the diameter of the cylinder, and then split. Enough metal is taken out at the split so that the rings can be compressed to a smaller diameter than the cylinder. When placed in the piston the rings spring out against the walls of the cylinder, making a steam-tight yet easy sliding fit. The cylinder is counterbored at each end, to facilitate the introduction of the piston.

Steam from the boiler enters the steam-chest by means of the steam-pipe. The D-valve, so-called on account of its resemblance to the letter D, being in the position shown, the steam from the chest passes into the steam-port, and from thence into the cylinder. The steam forces the piston back. This motion is transmitted to the fly-wheel by means of the piston-rod, cross-head connecting-rod, and crank, and causes the fly-wheel to revolve. On the crank-shaft of the engine is placed the *eccentric*. This is a disk, eccentrically placed on the shaft and fastened to it. Around the eccentric is placed the *eccentric-strap*. This strap completely encircles

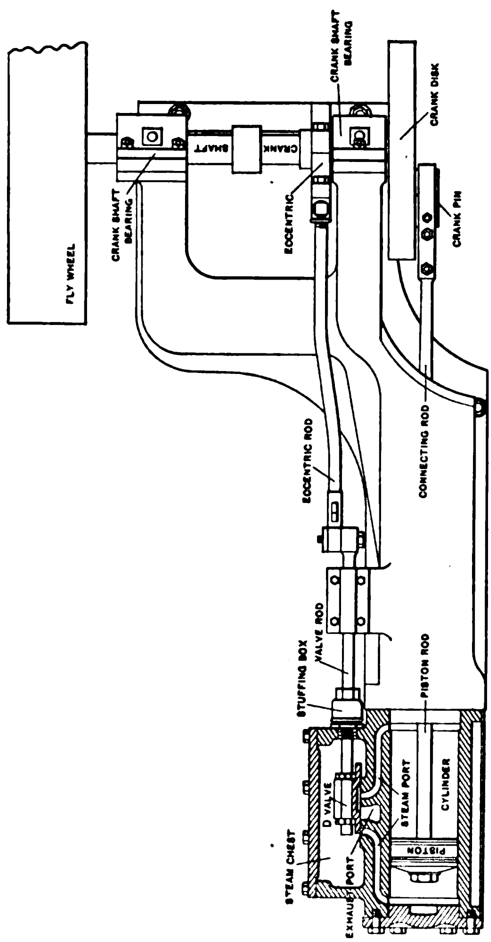


FIG. 138.—Simple Steam-engine.

the eccentric, with a sliding fit, and is attached to the *eccentric-rod*. The D-valve is joined to the eccentric-rod by means of the valve-rod as shown in the illustration. The amount of eccentricity, or distance between the centre of the shaft and the centre of the eccentric, determines the amount of movement of the valve; the travel of the valve is twice the eccentricity.

As the shaft rotates, the D-valve is moved by the eccentric,

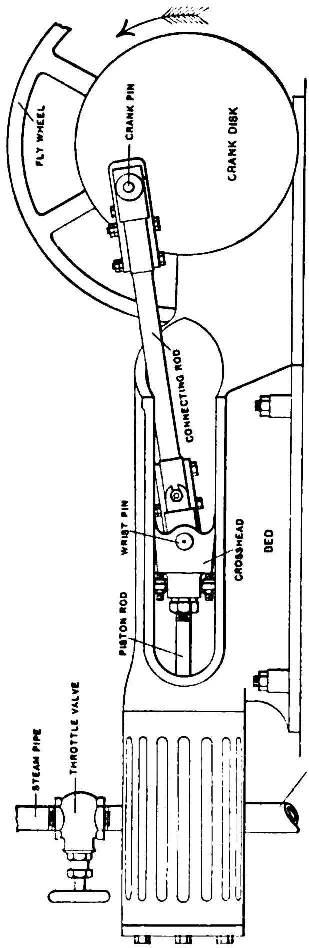


FIG. 139.—Simple Steam-engine.

and at some predetermined point of the stroke of the piston it closes the port. The steam is now prevented from entering the cylinder, or is *cut off*. The point in the stroke where the valve closed is the *point of cut-off*. The steam already admitted to the cylinder expands, nearly according to Mariotte's law, which is: The volume of a gas varies inversely as its pressure, the temperature being uniform; or more concisely, $p \times v = constant$, and by its expansion, the piston is forced to the end of the stroke. By the time the stroke is completed, the eccentric has moved the valve over so that communication is established between the steam-port and the exhaust-port by means of the valve. By this means the steam is now released from the cylinder and flows to the atmosphere by way of the steam port, valve, exhaust-port, and exhaust-pipe. The point in the stroke at which the steam is released is called the *Release point*, or simply the *release*.

At or near the time that the steam was released from the head end of the cylinder,* the valve uncovered the steam-port leading to the crank end, thus admitting steam behind the piston. The point of the stroke where the valve uncovers the port, so as to admit steam to the cylinder, is called the *point of admission*.

The piston is moved in the opposite direction to that in which it started, by the steam which is admitted behind it. It drives the steam from the head end until the valve is moved so that the communication between the exhaust- and steam-ports at the head end is closed. Some steam remains in the cylinder, and is compressed by the motion of the piston. The point of the stroke at which the exhaust-passage is closed is called the *point of compression*. When the piston reaches the end of the stroke, or usually just before, the valve admits steam to the head end of the cylinder again, and the cycle of the engine is complete.

The cylinder of the engine is usually made of cast iron. The walls are made heavy enough to withstand the unbalanced pressure of the steam on the interior. The thickness of the walls may be determined by the following formula, given by Kent:

$$t = 0.0004Dp + 0.3 \text{ inch},$$

* The head end of the cylinder is the end away from the crank of the engine. The crank end is the end next to the crank.

where t is the thickness of the cylinder-wall, and D the diameter of the cylinder, both in inches, and p is the pressure of steam in pounds per square inch.

The cylinder-heads, or covers, are also of cast iron. They are usually made slightly thicker than the walls of the cylinder. Thurston says the excess over the walls should not exceed 25 per cent. Sometimes the head is stiffened by ribs, radiating from the centre. In this case it is not necessary to make the cylinder-head so heavy. The cylinder-heads are fastened to the cylinder by means of machine-bolts or studs. Studs are preferable for a horizontal engine. These studs or bolts are screwed into flanges on the cylinder, which flanges are made slightly heavier than the cylinder-walls. The flange on the head through which they are screwed is usually of the same thickness as the flange on the cylinder.

The piston is also usually built of cast iron. It may be cast in one piece and the grooves for the piston-rings turned in it, or it may be built of a number of pieces, fitted together. The latter kind, shown in Fig. 140, is termed a built-up piston. The piston should be heavy enough to stand the difference of pressure of the live steam and that of the steam which is being exhausted.

The piston-rod is made of steel or wrought iron. It is usually tapered on the piston end and fits into a taper-hole. On the opposite side of the piston it is secured by a nut. It passes through the cylinder-head, through a stuffing-box, shown in Fig. 141, which keeps the rod steam-tight. The end of the rod outside the cylinder is fastened to the cross-head. The method of fastening varies in different cases. Sometimes the rod is screwed into the cross-head. In locomotives it is tapered to fit a tapered hole, and a taper-key is driven through it, from one side of the cross-head to the other, thus holding the rod firmly in place. Unwin gives the following formula for the diameter of the piston-rod:

$$d = bD\sqrt{p},$$

where d is the diameter of the rod, and D the diameter of the cylinder, both in inches; p is the maximum unbalanced pressure in the cylinder in pounds per square inch, and b is a constant. For iron $b = .0167$; for steel $b = .0144$.

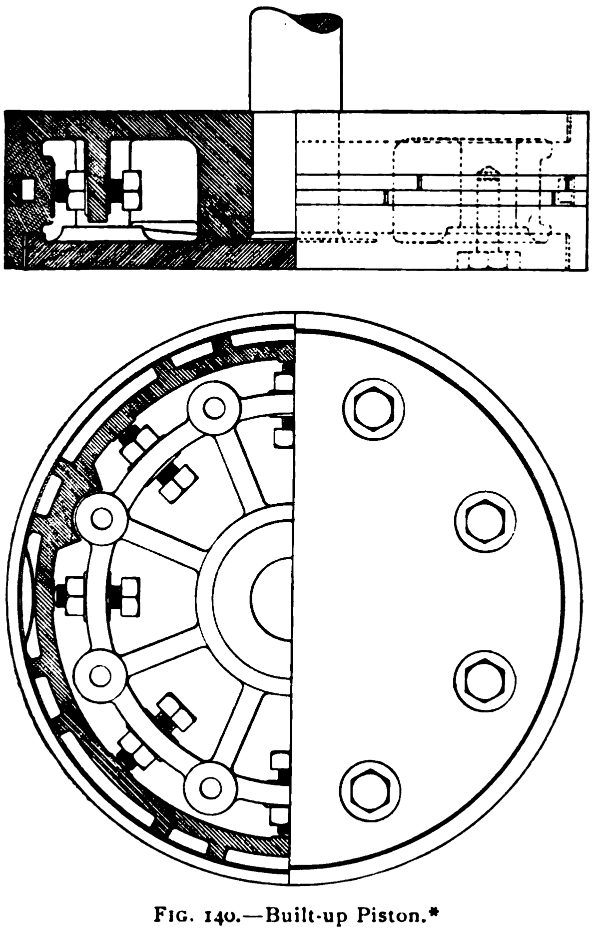


FIG. 140.—Built-up Piston.*

The cross-head, shown in Fig. 142, is often a solid block of wrought iron or steel. It serves to unite the piston- and connecting-rods. It is joined to the connecting-rod by means

* Bass-Corliss engine.

of a wrist or cross-head pin. This is a steel pin of generous proportions which retains the connecting-rod end in a slot in the back of the cross-head. The pin is seated in the sides of the slot, and is usually stationary, the rod moving around it.

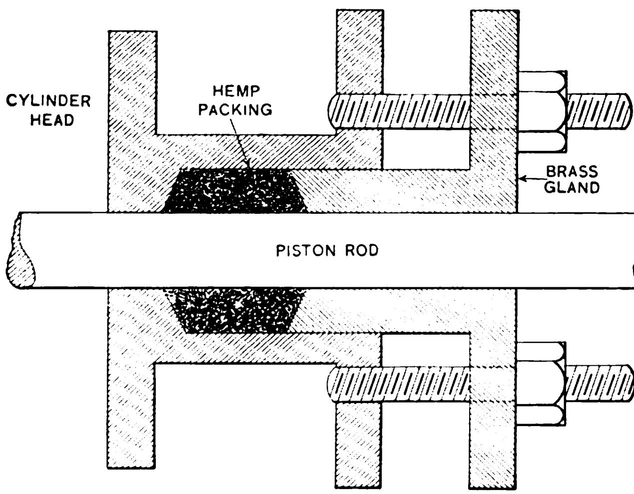


FIG. 141.—Stuffing-box.

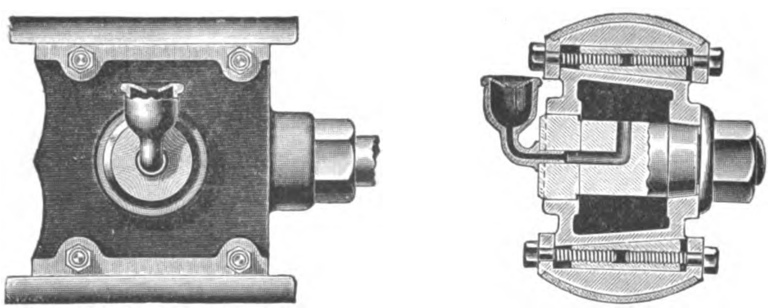


FIG. 142.—Cross-head of the Buffalo Automatic Cut-off Engine.

Sometimes, however, it is made a part of the rod and rotates in bearings in the cross-head. A rough rule for the dimensions of cross-head pins is to make its length from .25 to .3 the diameter of the piston and its diameter .18 to .2 that of the piston. The cross-head slides back and forth between two bars termed guides. The wearing surface of the cross-head may be lined with brass or some anti-friction metal.

The guides are two long, smooth surfaces, placed parallel to the axis of the cylinder at its crank end. They guide the cross-head in its proper path. The area of the cross-head bearing surface, according to Seaton, should be such that the bearing surface on which the thrust of the connecting-rod is taken will admit of a pressure of 400 lbs. to the square inch. That is, the area of that portion of the guide on which the cross-head bears when the engine is standing still. But for good working the surfaces should be made so that this pressure will not exceed 100 lbs. The formula given for the area of the slides by various authorities is

$$A = P \tan \theta \div p_0,$$

where P is the total unbalanced pressure, θ the angle whose sine $= \frac{1}{2}$ stroke of piston $\div$ length of connecting-rod. If the engine rotates in the direction of the arrow, Fig. 139, the wear will come principally on the upper guide. If it rotates in the opposite direction it will come principally on the lower one. If the engine is a reversing one the greatest wear will be on either guide, depending on the direction of rotation.

The connecting-rod is a steel or wrought-iron bar joining the cross-head to the crank. In length it is usually four to five times the length of the crank. In section it may be either circular, rectangular, or of I section. Rods of circular section are sometimes made heavier in the middle than at either end. Whitham gives for the diameter of the connecting-rod at the middle the following formula:

$$d = .0272\sqrt{Dl\sqrt{p}},$$

where D is the diameter of the cylinder in inches, l length of connecting-rod in inches, and p is the maximum steam-pressure in pounds per square inch. The diameter of the rod at its ends may be seven eighths of the diameter at the middle.

The ends of the connecting-rods are secured to the crank and cross-head pins in various ways. A brass bearing is placed around the pins, and then a strap passes over the bearing and end of the rod and is secured in place by a taper-key, passing through both rod and strap as in Fig. 143. In marine engines

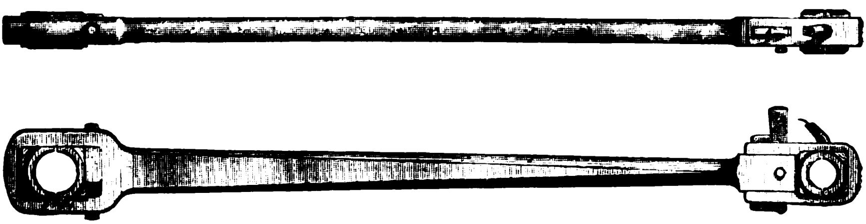


FIG. 143.—Connecting-rod of the Porter-Allen Engine.

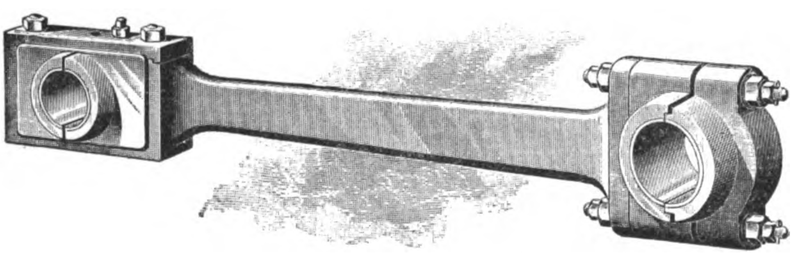


FIG. 144.—Marine Connecting-rod.

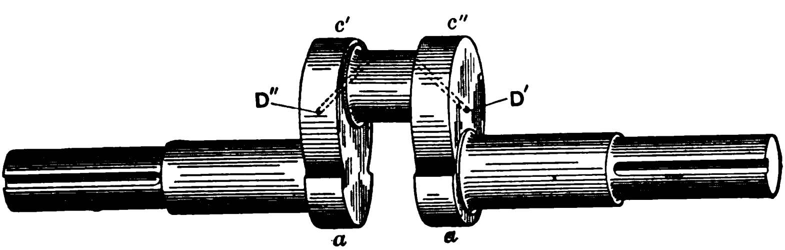


FIG. 145.—Solid Crank and Shaft.

the brasses are generally secured to the rods by bolts passing through the brasses, into an end forged directly on the rod. See Fig. 144.

The crank may be made in various ways. It may be an ordinary crank, Fig. 147, forged of wrought iron or steel, or it may be made in the shape of a disk with a crank pin set in it as in Figs. 138 and 139. The crank may be overhanging as in Fig. 138, or it may have an outboard bearing. In this case another disk or arm is placed parallel to the first, as shown in Figs. 145 and 146, with the crank-pin set between

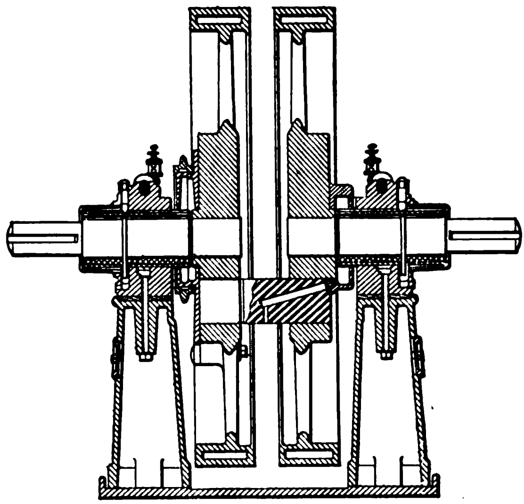


FIG. 146.—Crank of the Straight-line Engine.

them. This disk or arm is forced on a shaft which runs in a bearing outside of the engine. The diameter of the crank-pin is found by the formula given by Unwin:

$$d = \sqrt{\frac{5.1}{t}} \times \sqrt{P\frac{l}{d}},$$

where t is the allowable stress on the metal, being about 9000 lbs. for iron. P is the maximum load on the piston, l and d are the length and diameter of the pin respectively in inches. The ratio $\frac{l}{d}$ is assumed. If either the length or diameter is given the formula will read:

$$d = \sqrt[3]{\frac{5.1}{t}}\sqrt[3]{Pl}.$$

For the length of the crank-pin Whitham gives

$$l = .9075 f \frac{\text{I.H.P.}}{L},$$

in which f is the coefficient of friction of the pin; this coeffi-

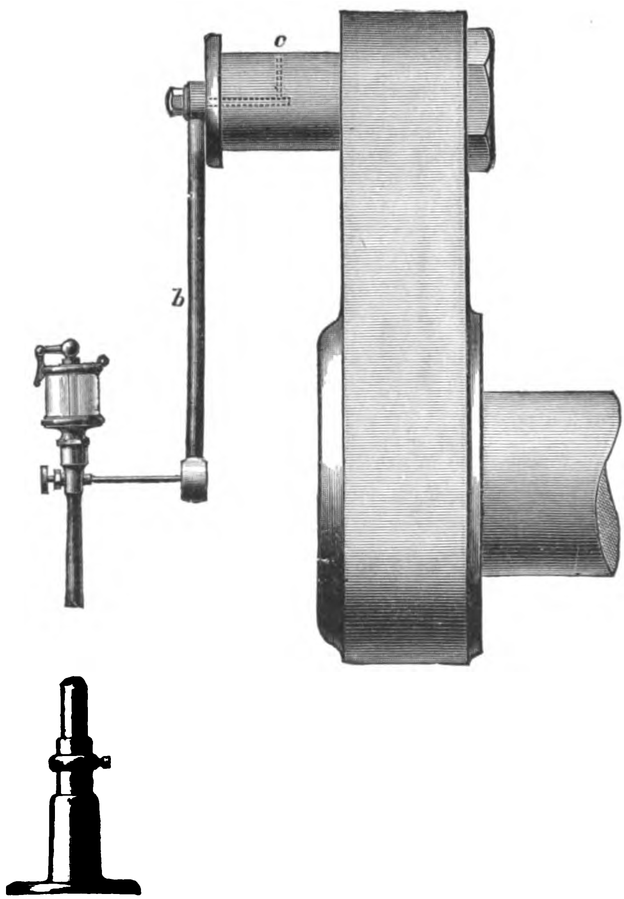


FIG. 147.—Crank.

cient may be taken as .03 to .05 if the pin is perfectly lubricated; if not, take $f = .08$ to .1. L is the length of stroke of the engine in feet, and I.H.P. represents the theoretical indicated horse-power of the engine.

The crank-shaft carries the fly-wheel, and when turned by the crank, it causes the fly-wheel to revolve with it. It should be made of wrought iron or steel. Its diameter should be, according to Unwin,

$$d = a\sqrt[3]{\frac{\text{I.H.P.}}{\text{R.P.M.}}},$$

where R.P.M. represents the revolutions per minute and a is constant, depending on the strength of the material and the factor of safety. He gives $a = 3.294$ for wrought iron and $a = 2.877$ for steel.

If the engine is direct-connected to an electric generator, the armature of the generator is carried by the crank-shaft. The shaft in this case must be made very much larger than the diameter determined by the formula, to decrease the deflection of the shaft due to the weight of the armature as much as possible.

The fly-wheel of the engine serves to preserve a uniform speed of rotation of the engine during a revolution. It stores up energy, or gives it off in accordance with the fluctuations of the load. The fly-wheel is sometimes used as a band-wheel, by means of which the engine transmits power. If the engine runs faster than its usual rate, as may be caused by the governor-belt breaking, the fly-wheel, if not strong enough, may burst from centrifugal force. Fly-wheels should be designed so as to withstand a moderate increase in speed. The maximum velocity at which a cast-iron fly-wheel should be run is 6000 feet per minute. Then the diameter, D, of the wheel will be

$$D = \frac{6000}{\pi R} = \frac{1910}{R},$$

R being the number of revolutions per minute.

The eccentric is a flat disk of cast iron, set eccentric with the shaft. The amount of eccentricity, e, Fig. 148, determines the travel of the valve. It may be either keyed to the shaft or attached by means of set-screws. The eccentric-strap which fits around the eccentric is made in two pieces usually. The two pieces are joined with bolts; the strap is fitted so that the eccentric may rotate freely within it, and yet have no lost motion. The eccentric-rod is fastened to the eccentric-strap usually by means of bolts, passing through the rod and strap. It serves to communicate motion to the slide-valve, by means of the valve-rod.

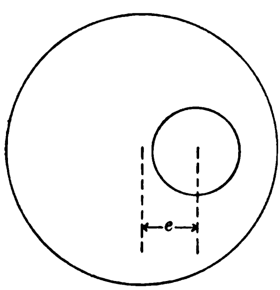


FIG. 148.—Eccentric.

The valve-rod is a steel or wrought-iron rod, which moves the slide-valve. One end is attached to the valve, the rod usually passing through the valve, by a nut on each side of the valve. The rod passes through the steam-chest through a stuffing-box, similar to that used on the piston-rod. The rod usually joins the eccentric-rod by means of a rocker. The valve-rod may be made one third the diameter of the piston-rod.

The steam-chest is a closed box, into which the steam first passes from the boiler. It may be either rectangular or circular. The chest may be cast solid with the cylinder or bolted to it. Its position with regard to the cylinder varies. It is usually placed on the side of stationary-engine cylinders and on top of locomotive cylinders. Sometimes it is placed beneath the cylinder. The same formula applies to the thickness of the steam-chest walls as to the walls of the cylinder, both being subject to the same pressure. The bottom of the chest is a wall of the cylinder. There are three openings in the bottom, which is made a plane surface. The two smaller of these openings are the steam-ports which lead to the cylinder. The third and larger one is the exhaust-port, which communicates with the atmosphere. The length of the steam-ports is usually made 2 inches less than the diameter of the cylinder. The area is found as follows: Let A be the area of the piston in square feet, let L be the stroke in feet. Let the engine make N revolutions per minute. Then the volume of steam in cubic feet required to fill the cylinder in one stroke will be $V = LA$; in N revolutions the volume required will be $V = LAN$. Live steam is allowed a velocity of 6000 feet per minute and exhaust steam a velocity of 4000 feet. The steam-port has to convey exhaust as well as live steam, hence calculations must be made for the exhaust steam. Hence the area of the port to carry the exhaust steam away is $\frac{LAN}{4000}$.

The wall separating the ports is called the bridge. If the bridge is made too narrow, the valve will travel over it, and allow steam to pass into the atmosphere from the steam-chest. Hence the bridge must be made so wide that there is no possibility of such a waste.

The exhaust-port communicates with the atmosphere by means of a pipe. This pipe is termed the exhaust-pipe, and should be laid with as few bends as possible. The area of the exhaust-port should be 33 per cent greater than the area of the steam-port at the valve-seat. It should gradually increase in area until it is 50 per cent greater at the entrance to the exhaust-pipe.

The valve of the engine will be discussed in a later chapter, under the head of "Valves and Valve-gears."

The bed of the engine is a heavy cast-iron frame, which is bolted firmly to foundations of masonry or concrete. It has various forms with different makers. The cylinder of the engine is sometimes cast solid with the bed and sometimes is bolted on. The guides are also sometimes cast on the bed, and sometimes not. The frame usually carries the crank-shaft bearings, Fig. 149.

The *governing*, or *regulation* of the speed of the engine, is accomplished by means of the governor, shown in Fig. 150.

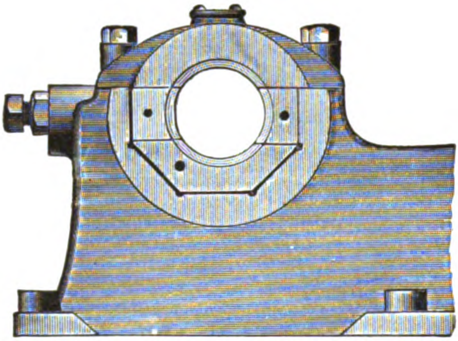


FIG. 149.—Crank-shaft Bearing of the Houston, Stanwood & Gamble Engine.

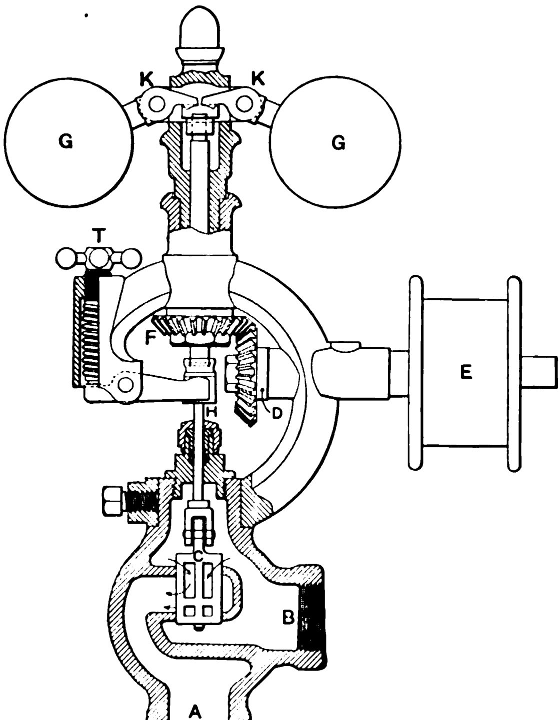


FIG. 150.—Governor.

If the load on the engine was suddenly lightened, and the steam-supply remained the same, the engine would commence to run faster and if not checked might cause its fly-wheel to burst. Its speed is checked by the governor. When the engine speeds up, the balls GG, driven through the pulley E, and bevel-gears F and D, by a belt to the crank-shaft, fly out from centrifugal force. As they fly out they rotate about the pivots KK. The rod H is depressed, partly closing the valve C in the steam-pipe AB and shutting off the steam-supply. The decrease in the steam-supply causes the engine to run slower. If it runs too slowly the balls drop and open the valve C, thus admitting more steam to the engine. The hand-wheel and spring at T serve to regulate the speed at which the governor will act. A governor such as has been described is known as a *throttling* governor.

Clearance is the volume contained between the piston and the end of the cylinder when the piston is at the end of the stroke plus the volume of the steam-passage or port. The clearance is prejudicial to the heat efficiency of an engine because it preserves a volume of steam which is not active and which is subject to condensation. By so doing it increases the amount of steam required to do a given amount of work. It varies according to the make of the engine from 2 per cent to 12 per cent. *Back-pressure* is the resistance on the side of the piston opposite the live steam and is due to the pressure of the atmosphere plus the resistance of the exhaust steam on passing out through the exhaust-pipe which offers friction. An engine should have as little back-pressure as possible.

The *stroke* of an engine is the distance passed through by the piston in one movement in one direction and is approximately equal to the length of the cylinder minus the thickness of the piston. There are two strokes to each revolution of the crank-shaft. For a given velocity of piston a higher speed of rotation may be had by making the stroke short and a slow speed by making the stroke long. The velocity of the piston is equal to the length of the stroke multiplied by the number of strokes per minute.

The foregoing is a discussion of a simple non-condensing engine. All that would be necessary to convert the engine into a condensing engine would be to connect the exhaust-pipe to a condenser having an adequate supply of cold water.

To convert the engine to a multiple-expansion engine, it would be necessary to provide the requisite number of cylinders with volumes of the proper ratio, and the receivers. Compound and multiple-expansion engines will be treated separately.

The formulæ given in this chapter apply in the main to the essential parts of all reciprocating engines. The only difference between the engine just described and a Corliss or automatic cut-off engine is in the valve-gear and governing. Otherwise both in construction and operation they are identical. Such changes in the given formulæ as are necessary to suit these special cases will be made at the proper time.

CHAPTER XVI.

AUTOMATIC CUT-OFF ENGINES.

HIGH-SPEED ENGINES.

AUTOMATIC cut-off engines are of two kinds: 1. Long-stroke engines, having a moderate rotative speed, say about 60 to 120 revolutions per minute. The Corliss, Brown, and Greene engines are in this class. 2. Short-stroke engines having a high rotative speed, 200 revolutions per minute and upwards. The Buckeye, Porter-Allen, and Ball & Wood engines are in this class.

The high-speed automatic cut-off engine is that type in which the regulation of the speed of the engine is effected by changing the travel of the valves in such a manner as to control the time of admission and cut-off of steam from the steam-chest into the cylinder. This is a modern type of engine which is run at a high speed. The high speed does not mean necessarily high piston-speed but high rotative-speed by means of a very short stroke; the diameter being in many cases equal to or greater than the stroke. This means a short engine, if we assume that the length of an engine is always equal to four times the stroke, which is an average value if the length of the connecting-rod is taken as two and one half times the stroke.

The Buckeye engine shown in Fig. 151 is an example of this type. The increasing or decreasing of the number of revolutions for a sudden change of load amounts to about 1 per cent and the change is for a few revolutions only.

The greatest source of trouble with high-speed automatic engines is the heating of the main bearings, caused by the high

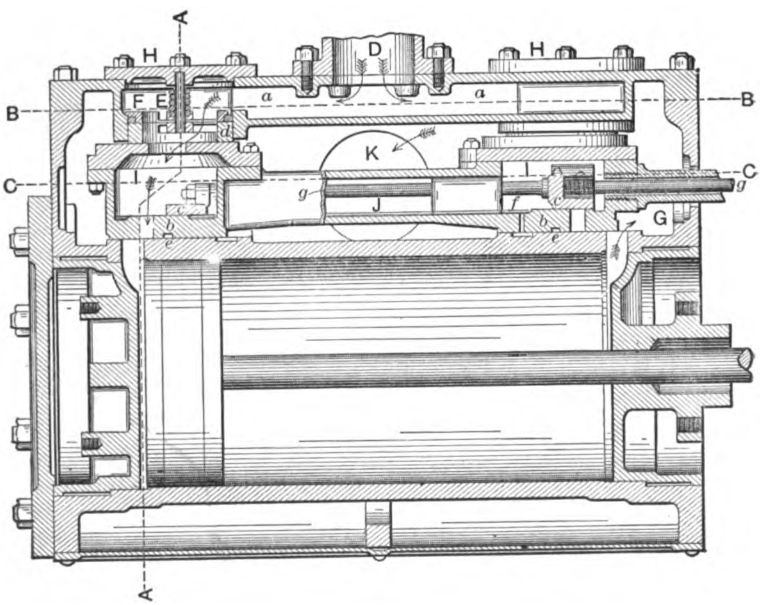


FIG. 151.—Buckeye Engine. Section through Valve and Cylinder.

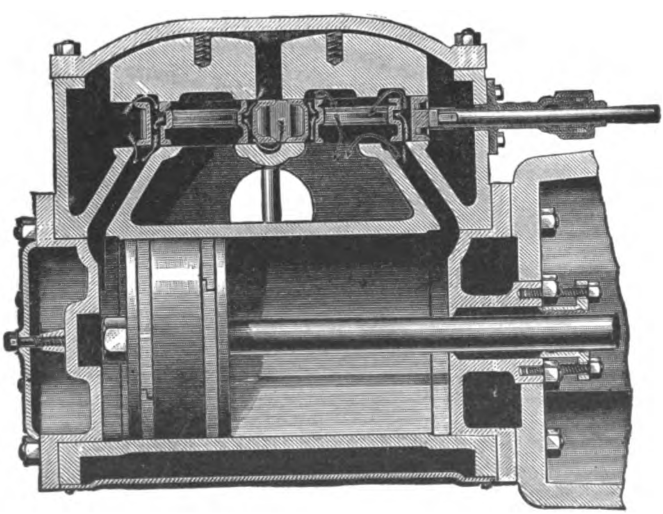


FIG. 152.—Watertown Engine. Section through Valve and Cylinder.

speed of rotation. For this reason large bearings, well oiled, must be provided.

On account of the comparatively large diameter of these engines, the clearance is large, about 5 to 10 per cent. The mean back-pressure is 3 to 4 lbs. above atmospheric pressure.

This type is fairly economical in its use of steam; it occupies small space and regulates well but requires careful attention, especially to bearings. The valve in this engine is generally balanced and multiple-ported as shown in Fig. 152, in order to give a large port-opening with a short travel of the valve. Here it should be remembered that the steam has two openings through the valve into the steam-port, making an ample area for the steam to pass through without being retarded. Also the steam enters above and below the valve, producing a balance and preventing the friction of the valve against the seat caused by steam-pressure upon one side only.

It is seen that when the left edge of the valve moves to the right until it passes the left-hand steam-port there is a double port-opening.

Some engines also use an auxiliary valve, in connection with the main valve, to control the cut-off more fully than is possible with a single valve. The Buckeye engine uses such a valve; it is shown in Fig. 151.

Piston-valve.—Another form of balanced valve used on this type of engine is the piston-valve, of which Fig. 153 is an illustration.* In this case the steam enters in a direction which is the reverse of that with the ordinary valve; that is, it comes into the cylinder through the middle and exhausts through the hollow valve and out at the ends. This is evidently a balanced valve, the pressure being equal on all sides.

The governor of an automatic engine is placed in the fly-wheel as shown in the illustration of the Straight-line Engine,

---

* The piston-valve here spoken of controls the passage of the steam into and out of the small cylinder. The valve of the large cylinder is of the ordinary balanced type.

Fig. 154. Its construction will be discussed more fully in the chapter on "Valve Motions."

However, it may be proper to say here that the mechanism

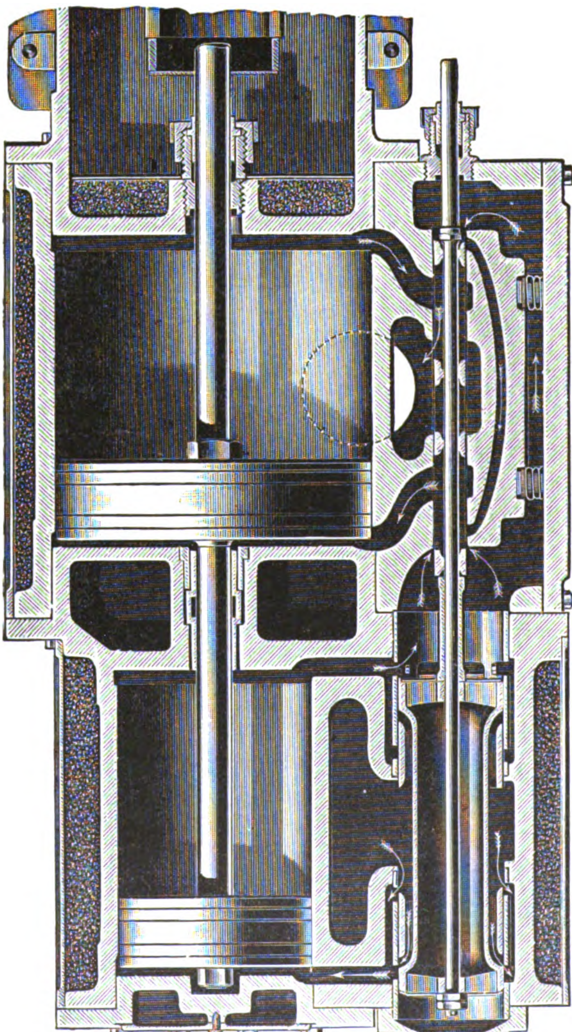


FIG. 153.—Section through Cylinders and Valve-chests of the Ideal Compound Engine, showing Piston-valve.

of the governor and eccentric is such that, for an increase in the speed of the engine, the travel of the valve is made less and the cut-off occurs earlier, causing a diminution of speed;

for a diminution of the speed the cut-off occurs later. This arrangement causes each stroke to use just what steam is necessary for that stroke and no more, the quantity depending upon the load.

CORLISS ENGINES.

By the pure Corliss type is meant an engine having four rotary valves for the control of the steam, two for *admission* and two for *exhaust* (see Fig. 156). Under the Corliss type is generally included all those engines whose valves are rotary

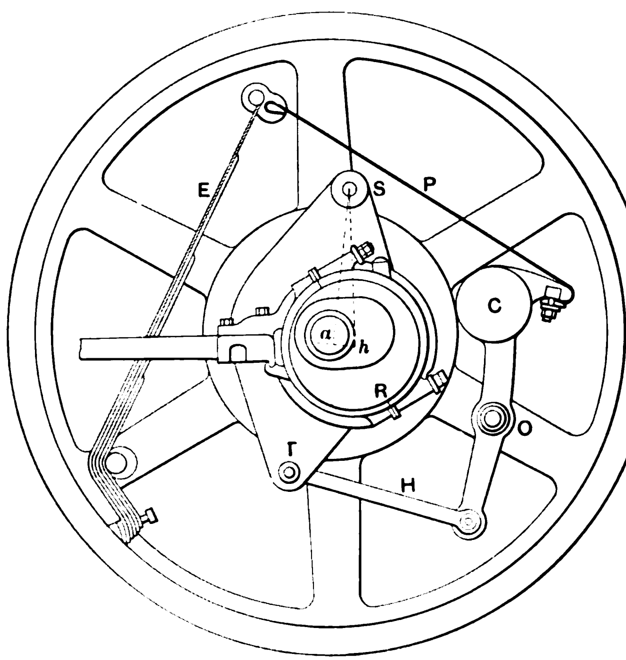


FIG. 154.—Governor of the Straight line Engine.

in their motion or which have more of the pure Corliss characteristics than of the other types. These engines have a high piston-speed but a slow rotary speed. This is effected by a comparatively long stroke. The average ratio of the diameter to the stoke is one half. A small diameter with short ports

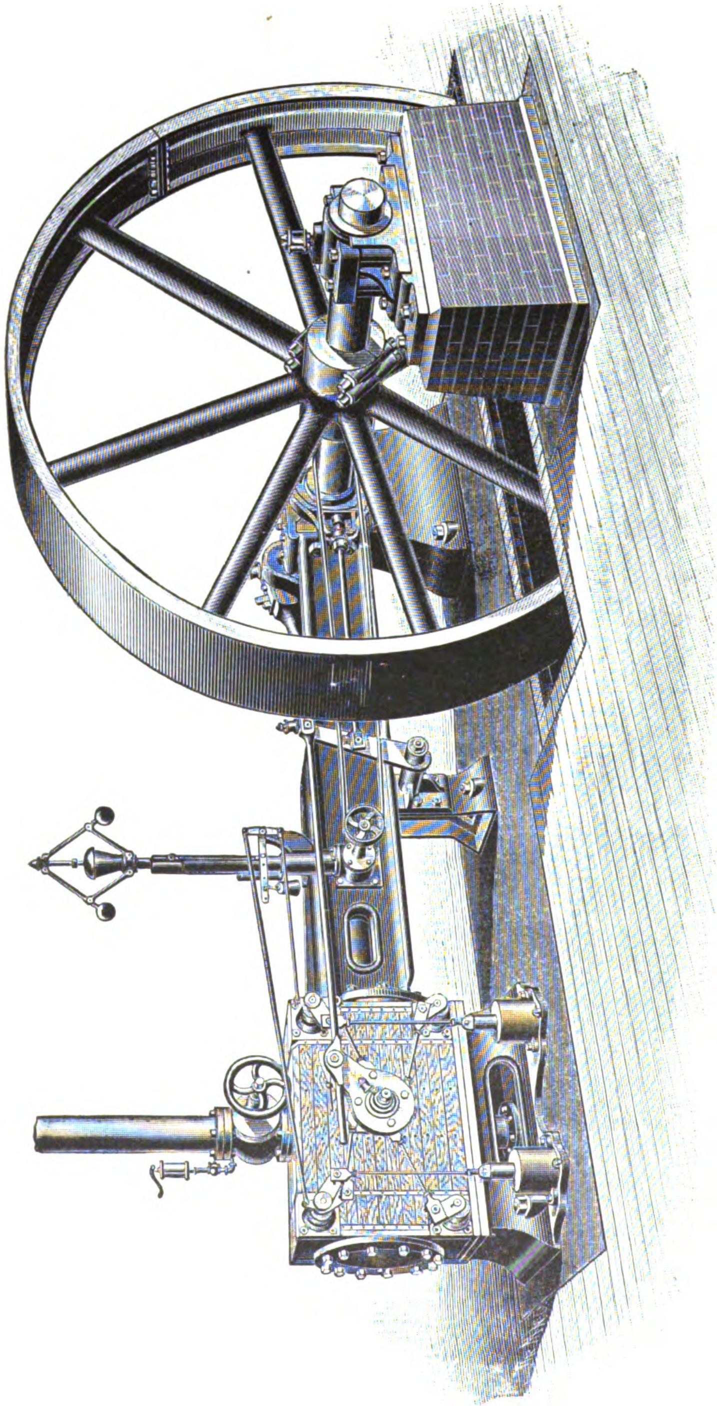


Fig. 155.—Bass-Corliss Engine.

means a small clearance, which is from 1 to 5 per cent. Corliss engines usually make about 100 revolutions per minute or less. The system of valves for each end of the cylinder makes

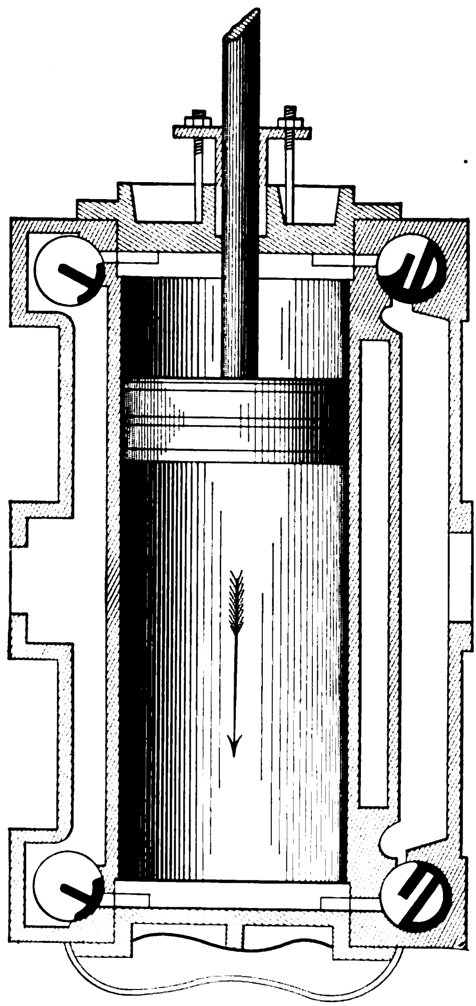


Fig. 156.—Corliss Engine Cylinder.

it possible to cut off the steam to suit the requirements of the load without changing the amount of compression, as is necessary with the single slide-valve.

Under varying loads, the Corliss engine acts precisely the same as the high-speed automatic engine. The governor alters the valves, so that the time of cut-off occurs earlier or later in the stroke, according to whether the load is decreased or increased, thus admitting to the cylinder the amount of steam necessary to do the work.

The mean back-pressure for this type, on account of the slow speed, is small, being from 1 to 3 lbs. per square inch above atmospheric pressure.

This type is the most economical of the three types discussed in the use of steam, but it takes more skill to keep it in order than do the others, for the reason that it has more parts, some of them being very delicate.

The long stroke gives the engine great length and it is of high first cost.

The arrangement of the four valves is shown in elevation in Fig. 155, and in section in Fig. 156. The valves are of the form shown in Fig. 156 and are rotated by the valve-rods. These valve-rods acquire their motion from the *wrist-plate* which is partially rotated about its centre. This rotation is accomplished by means of the ordinary eccentric. The eccentric is assisted in the movement of the valves by the dash-pots, shown in Fig. 155. These are used in order to give a quick shutting of the admission-valves, whereby the port is given a wide opening up to the time of cut-off, and this prevents wire-drawing. The governor is usually of the weighted-ball type and controls the time of cut-off, according to the speed of the engine. It is connected to the valves by means of the rods as shown in the diagram. The action of the valves and governor will be more fully discussed in the chapter on "Valves and Valve-gearing."

CHAPTER XVII.

INDICATORS.

THE indicator is an instrument used for determining the actual amount of work that an engine is performing, besides giving other information as to the conditions of working, such as the operation of the valves, etc.

The diagram obtained by the use of the indicator is called the *indicator-card.*

Fig. 157 shows in section the Tabor indicator, with a reducing attachment. It comprises a paper-drum B on which blank paper is held by means of two clips; a cylinder M in which works a steam-tight piston connected to a piston-rod; this piston-rod connects at the top with a lever which carries a pencil. The paper-drum is connected by a suitable reducing motion to the cross-head of the engine. As the piston makes a stroke the paper-drum is turned on its axis by the pull of the cord. A strong spring on the inside of the drum causes it to make its backward movement. The cylinder M is connected by steam-piping $\frac{1}{2}$ inch in diameter to the end of the engine-cylinder. Between the piston of the indicator and the upper end of the cylinder in the space marked M is placed a spring of known strength, which offers a measured resistance to the upward movement of the pencil. The steam from the engine-cylinder enters the indicator at L, pushes the piston up, and this in turn causes the pencil to rise. At the same time the cord causes the paper-drum to revolve. By means of the system of levers attached to the pencil-lever, the pencil is made to move in a straight line parallel to the axis of the drum. When the pencil is at rest, the line made by pressing the pencil to the paper and revolving the drum should be parallel to the base of the drum. When the pencil is pressed

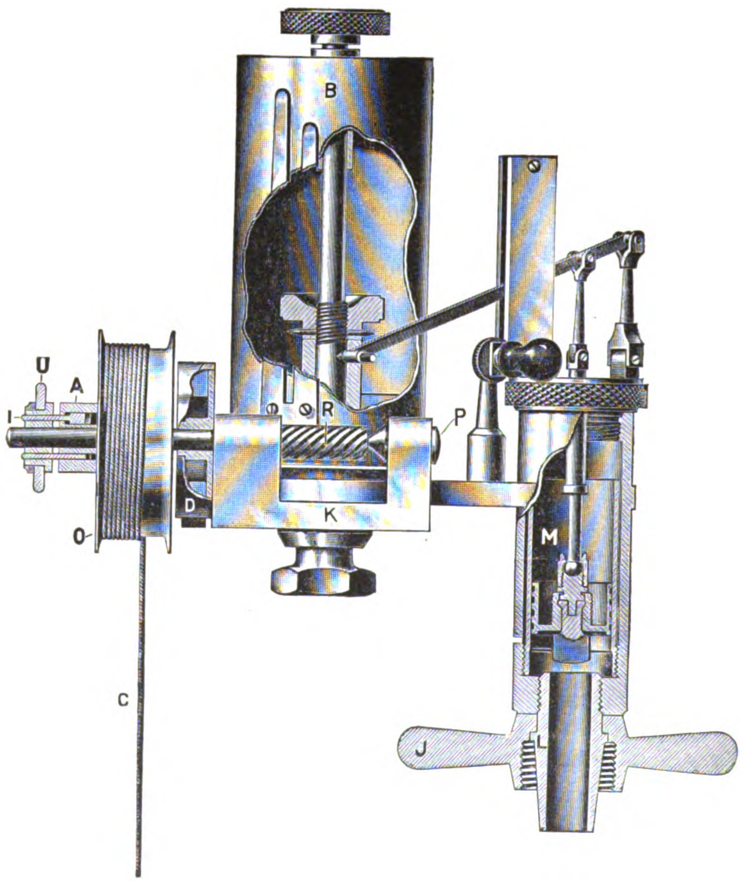


FIG. 157.—Tabor Indicator.

against the paper, it makes a line every point of which represents at once the pressure in the engine-cylinder and the position of the engine-piston within the cylinder. By shutting off the indicator from the steam of the engine, opening connection from the indicator to the atmosphere and pressing the pencil to the paper, a line parallel to the base of the drum is made, called the *atmospheric line*. The pressure of the steam as shown on the indicator-card may be referred to this line. The springs used in the cylinder of the indicator are numbered according to their strength; that is, a No. 40 or 40-scale spring is one which allows a movement of the pencil of 1 inch for a pressure of 40 lbs. per square inch.

The following are the prinicipal conditions required in a good indicator:

The line made by pressing the pencil to the paper and keeping the pencil at rest while the paper-drum revolves should be perpendicular to the line made by keeping the drum at rest and moving the pencil.

For equal amounts of increase in the pressure the pencil should rise equal distances.

The spring should be tested while hot. The tension of the spring to be used in the indicator will depend upon the speed of the engine and the pressure of steam used. Strong springs should be used for high speeds and high pressures and weaker ones for low speeds and low pressures. This difference in strength is made partly on account of the effects of inertia.

The indicator should be as light as is consistent with the proper strength, and no joints should be tight enough to cause the least binding.

The cord should be small and yet not small enough to allow it to stretch

In taking the cards from an engine it is necessary to take the cards from the two ends at precisely the same time or as nearly so as possible in order that the relations of the conditions in the two ends may be shown correctly. This can be done by the use of two indicators, one on each end, but one indicator may be used for one cylinder with the arrangement shown in Fig. 158.

Reducing Motions.

The paper-drums for indicators are usually 1½ or 2 inches in diameter. It is desirable that the length of the card be less

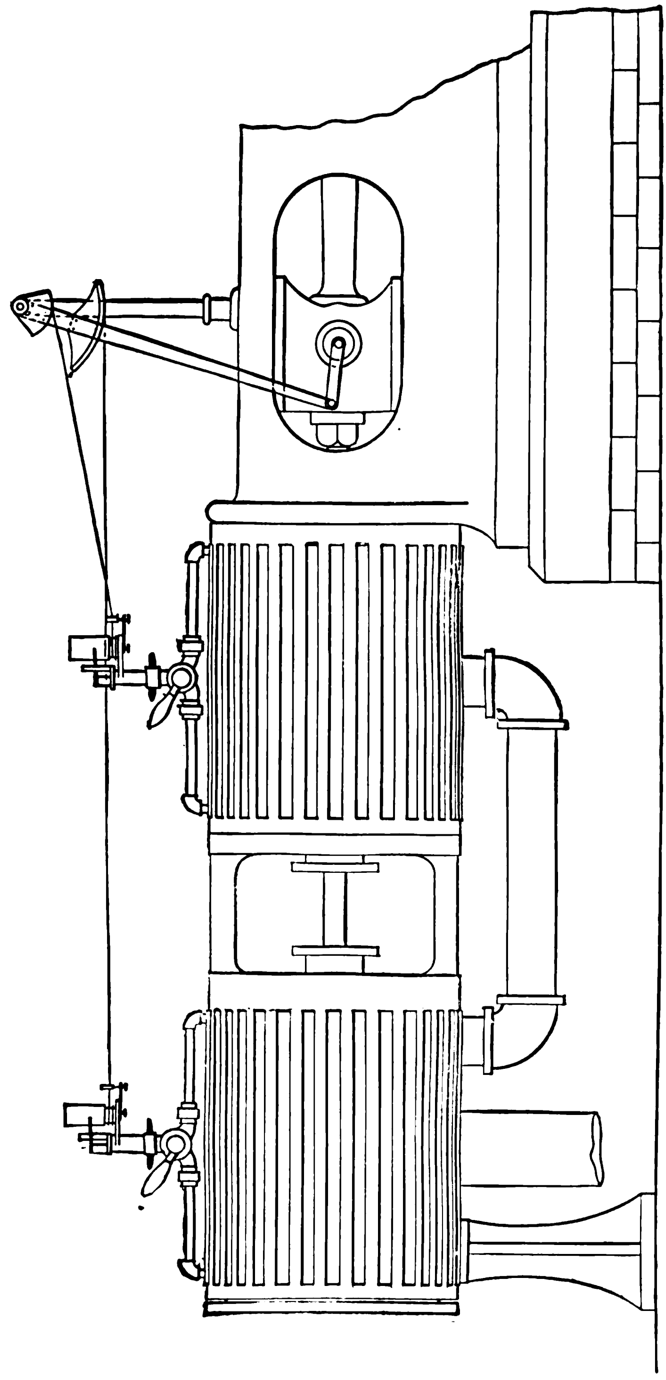


Fig. 158.—Ordinary Form of Indicator Rig for a Compound Engine.

than the circumference, hence the lengths of the cards usually taken with the above two sizes are usually 3 and 4 inches, respectively. As the length of the stroke of the engine is many times the length of either one of these, it is evident that some reducing motion must be used between the cross-head and the indicator. Fig. 158 shows the pendulum-reducing motion in which the length of the cord depends upon the distance from the fixed pivot to the point where the string is tied. The pendulum is caused to swing by being fastened to the cross-head by a link. In an indicator reducing motion the cord should lead off from the reducing motion, for a short distance at least, parallel to the axis of the engine.

The *Pantograph*, or lazy tongs, is shown in Fig. 159. It consists of a number of wooden links pivoted together as shown. The end A is fastened to the cross-head of the engine

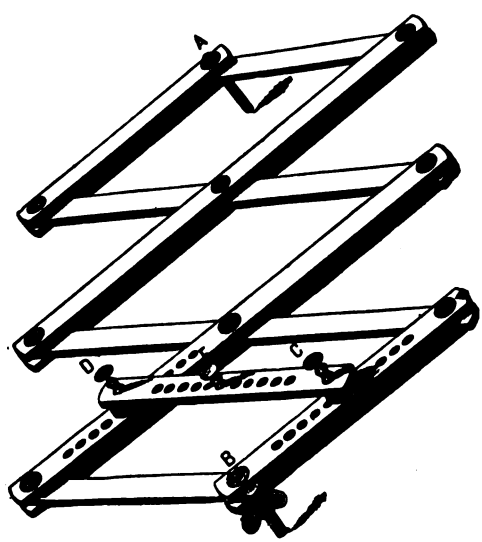


FIG. 159.

and the other end B to any stationary point. The cord leads off from the point E on the link CD. The point E must be on the centre line AB.

NOTE.—The most convenient and most improved reducing motion of those here given is that shown in Fig. 157 which is attached directly to the indicator. The reduction of motion is produced through the worm R.

TAKING THE CARD.

After the reducing motion and indicator have been connected, the cylinder of the indicator should first be warmed by turning on the steam, after which the steam is turned into the indicator-cylinder, first from one end of the engine-cylinder and then from the other end, pressing the pencil to the paper in each case for an instant.

The steam should then be shut off from the indicator, and an atmospheric line taken. The atmospheric line should be taken by disconnecting the string from the reducing motion, and pulling it by hand, so that the drum makes a complete revolution. If this is not done the atmospheric line will be of the same length as the card, which is not desirable.

The following data should be placed on every card: Date, time, revolutions per minute, gauge-pressure, length of stroke, diameter of piston, diameter of piston-rod; and the card should

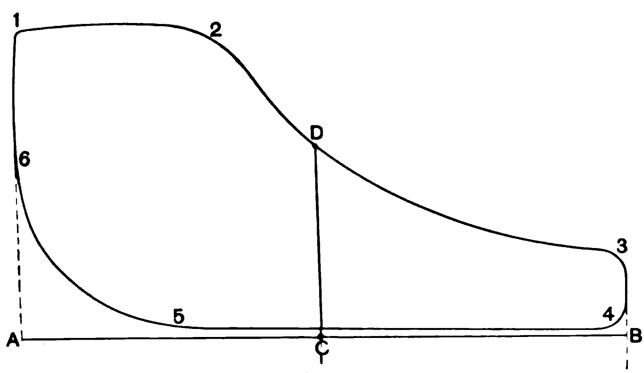


FIG. 160.—Indicator-card. 1–2, steam-line; 2, point of cut-off; 2–3, expansion-line; 3, point of release; 4–5, back-pressure line; 5, point of compression; 5–6, compression-line; 6, point of admission; *AB*, atmospheric line.

also be marked so as to show whether it is from the head or crank-end of the cylinder.

Fig. 160 shows a typical indicator-card in which the names of the points and parts are given.

It should be kept in mind that vertical distances represent pressures and that horizontal distances represent volumes; that is, in Fig. 160 the point D shows that the piston is at the middle of its stroke, passing from left to right and that the pressure in the cylinder at that part of the stroke is represented by the distance CD. The average height may be found by dividing the area of the card as determined by a planimeter by the length, or it may be done by the method shown in Fig. 161, by dividing the card into a number of rectangles, finding

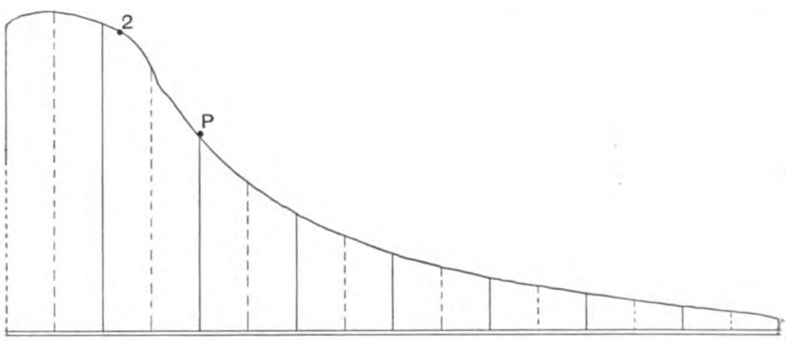


FIG. 161.

the middle height of each rectangle as shown by dotted lines, adding together the heights thus found and dividing by the number of rectangles. This average height multiplied by the scale of the spring gives the average pressure per square inch of the steam in the cylinder during the whole stroke, and is usually called the *Mean Effective Pressure*, or M.E.P.

Horse-power.—The work done in a cylinder in foot-pounds per minute is equal to the average pressure on the piston in pounds multiplied by the distance moved through by the piston in one minute, that is, Work $= \phi V$, in which $\phi =$ the total pressure on the piston and $V =$ velocity of piston in feet per minute.

Let $P =$ the M.E.P., L the length of the stroke in feet, A the area of the piston in square inches, N the number of strokes per minute, and H.P. the horse-power; then

$$V = LN. \quad \phi = PA,$$

and

$$\text{H.P.} = \frac{P \times A \times L \times N}{33,000} = \frac{PLAN}{33,000}.$$

The area of the piston is different in the two ends of the cylinder, owing to the presence of the piston-rod in the crank-end. The area of the rod should be calculated and subtracted from the area of the piston.

To Calculate the Weight of Steam per hour per H.P. used by the Engine.—If we calculate the volume of the cylinder, or the volume of steam used per stroke, and multiply by the number of strokes per minute, and then by 60, we have the volume of steam used per hour. Knowing from the diagram the terminal pressure of this steam (that is, the pressure which would be found at the end of the stroke if the release did not occur before the end), we can find from a steam-table the weight of a cubic foot, and thus the weight of the steam used per hour. This amount will be considerably less than the real quantity used because of condensation in the cylinder. It is useful, however, in that it gives the theoretical minimum or ideal consumption of steam.

Let A = area of piston in square inches;
L = length of stroke in feet;
N = number of strokes per minute;
P = M E.P. indicated by card.

Then

$$\frac{A}{144} = \text{area of piston in square feet;}$$

$$\frac{AL}{144} = \text{contents of cylinder in cubic feet;}$$

and

$$\frac{AL}{144} \times 60 \times N = \text{cubic feet of steam used per hour.}$$

The horse-power is $\frac{PLAN}{33,000}$. Dividing by this value we get the volume of steam used for H.P. per hour, the same letters having identically the same value in each formula; that is,

$$\frac{\frac{AL60N}{144}}{\frac{PLAN}{33,000}} = \frac{AL60N33,000}{144PLAN} = \frac{13,750}{P} \text{ cu. ft.}$$

To find the weight of this volume of steam, let Fig. 162

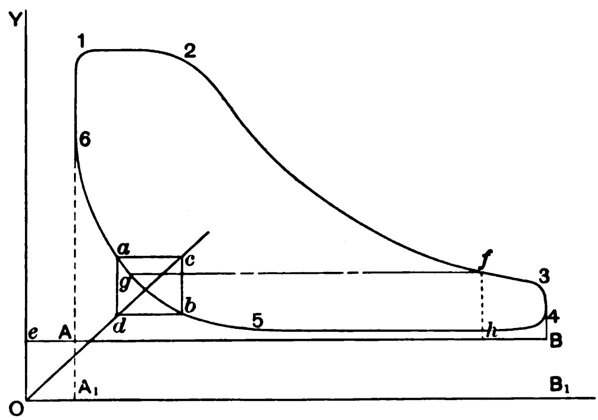


FIG. 162.

represent the card in question. Select some point as f just before release on the expansion-curve, draw fg parallel to the atmospheric line; then $\frac{fg}{AB}$ is the fraction of the volume of steam in the cylinder represented by the point f. The pressure of this volume is hf times the scale of the spring. Hence $\frac{13,750}{\text{M.E.P.}} \times \frac{fg}{AB} =$ volume of steam per hour at pressure hf.

Let w be the weight of steam per cubic foot corresponding to the pressure hf, as found in a steam-table; also let $\frac{fg}{AB} = F$.

Then

$$\frac{13,750}{\text{M.E.P.}} \times F \times w = \frac{13,750Fw}{p} = \text{wt. of steam per H.P. hr.}$$

To Construct the Theoretical Expansion-line.

Expansion of steam in a cylinder is the result of shutting off the steam-supply before the end of the stroke. In some early engines live steam was admitted during the whole stroke. Suppose for instance that steam is passing into a cylinder at full pressure until the middle of the stroke is reached; we then have a half cylinder full of steam which, during the remainder of the stroke, expands according to Mariotte's law, the pressure varying inversely as the volume. The curve representing this variation in the pressure is a hyperbola.

Let Fig. 163 represent an indicator-card. It is desired to find how nearly it comes to ideal conditions of expansion.

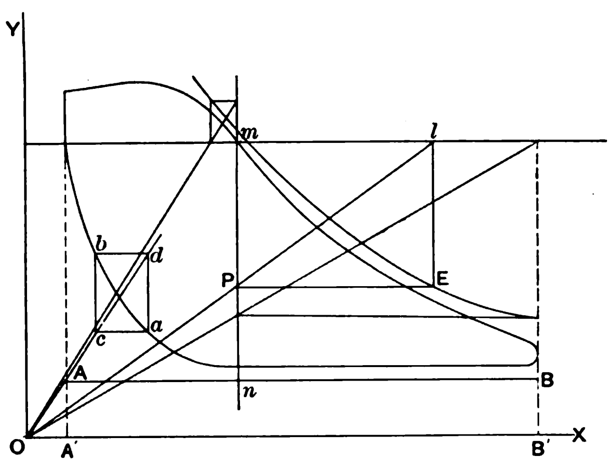


FIG. 163.

First it is necessary to find the ratio of the clearance to the volume swept over by the piston. Take any two points a and b on the compression-line and on them erect the rectangle $acbd$. Draw the vacuum-line. This is $A'B'$, parallel to AB, the atmospheric line, and at a distance below it corresponding to atmospheric pressure; for a 40-lb. spring it would be $\frac{14.7}{40}$ inches. Draw the diagonal *dc* meeting *A'B'* at *O*, which will be the origin of the hyperbola of expansion. To construct the

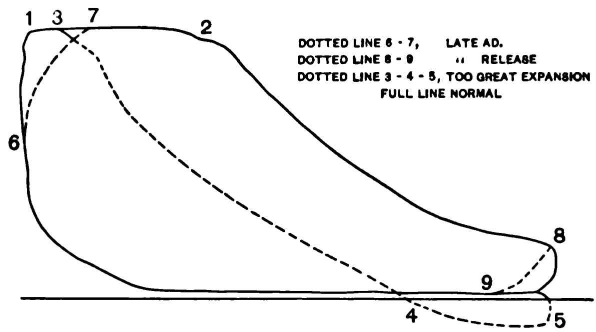


FIG. 164*a*.

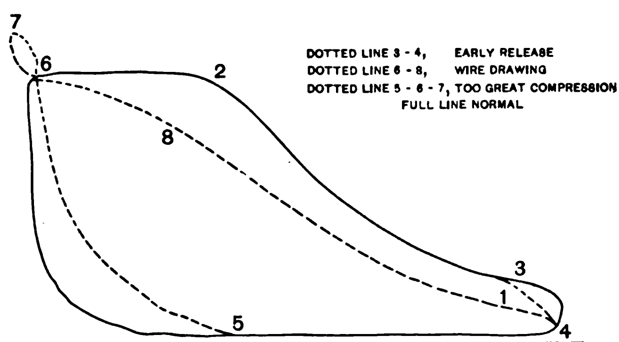


FIG. 164*b*.

theoretical expansion-curve take any point *m* on the expansion-line of the card, after the cut-off, and draw *mn* perpendicular to *AB*.

Through *m* also draw a line *ml* parallel to *AB*. Draw *Ol* cutting *mn* at *P* and *ml* at *l*. On the points *P* and *l* construct a rectangle. The point *E* will be a required point on the curve. Any number of points may be found in the same manner, and the curve filled in as shown. In this example, it is seen that the actual expansion-line falls below the theoretical line, indicating a considerable loss of some kind.

The compression-line may be determined in the same manner.

Indicators are also used in determining the conditions of working and work done in the cylinders of Gas-engines, Air-engines, Air-compressors, Water-engines, etc.

TO DETERMINE H.P. FROM THE TEMPERATURE OF THE STEAM.

Let P_1 be the absolute initial pressure of the steam which enters the cylinder before the cut-off, and r the number of expansions; then for all ordinary purposes the temperature of the exhaust steam will be $\frac{P_1}{r}$. *Absolute temperature* is the temperature given by a Fahrenheit thermometer $+ 461°$. *Absolute pressure* is the pressure-gauge reading plus 14.7.

Work and heat are mutually convertible, and whenever heat is used up in doing work the quantity of heat used is exactly proportional to the amount of work done; conversely, by the expenditure of a certain amount of work the same amount of heat will be produced. The unit of work is the foot-pound. The unit of heat is the *quantity of heat required to raise 1 lb. of water from 62° to 63° Fahrenheit*. This is called a *British Thermal Unit*, or abbreviated, B.T.U. According to Rowland one heat-unit is equivalent to 778 ft.-lbs. of work. Hence, by finding the number of heat-units consumed per minute by an engine and multiplying by 778 the work, W, which the engine should perform per minute under ideal conditions may be found. Then

$$\frac{W}{33,000} = \text{H.P.}$$

The number of heat-units in 1 lb. of steam is found by use of Table I, at the end of the book, as follows: Let T_1 be the temperature of the exhaust steam and P be the gauge-pressure of the steam supplied to the engine; also let T be the temperature at the pressure P, and let l be the latent heat at the same pressure, both being taken from the table. Then the total number of heat-units in a pound of steam is $T + l - T_1$. Hence $(T + l - T_1)\ 778 =$ the work done in foot-pounds.

As an example, suppose that an engine uses 2 lbs. of steam per minute at a gauge-pressure of 80 lbs. per square inch. By referring to Table I we find $T = 323.7$ and $l = 886.3$. T_1, the temperature of the exhaust, differs according to the number of expansions. Suppose that it is found to be 250° by means of a thermometer introduced into the exhaust-steam pipe.

Then $W = 2 \times (323.7 + 886.3 - 250°)778 = 2 \times 960 \times 778 = 1{,}493{,}760$ ft.-lbs.

$$\text{H.P.} = \frac{1{,}493{,}760}{33{,}000} = 45.2.$$

Owing to the fact, however, that part of the heat thus consumed is expended in warming the cylinder-walls, radiation, etc., this method is not reliable.

PROBLEMS.

1. At what parts of the stroke did cut-off, release, compression, and admission occur in the engine from which the card (Fig. 162) was taken?

2. Draw an indicator card similar to Fig. 162, and find the average height.

3. Calculate the H.P. in the above example, a 40-lb. spring being used.

4. In Fig. 161 what was the pressure of steam in the cylinder at that part of the stroke represented by the point P if a No. 40 spring was used?

5. What part of the engine-stroke does the point 2 represent in the same figure (161)?

6. What back-pressure in the engine is shown by this diagram?

7. Reproduce Fig. 160 on a piece of paper and find the M. E. P.

spring No. 40. Find also the clearance percentage and construct the theoretical expansion line.

8. Taking the M. E. P. found in the above, find the H.P. of the engine, the stroke being 24 inches, the diameter of the piston 12 inches, and the number of revolutions per minute 100.

9. Find the steam-consumption per hour per H.P. of the above engine.

10. If cut-off occurs at $\frac{1}{8}$ of the stroke, how many expansions are there during the whole stroke?

11. The initial pressure of steam entering a cylinder is 120 lbs. absolute. What is the final pressure if cut-off occurs at $\frac{1}{6}$ of the stroke? What is the pressure at $\frac{1}{2}$ of the stroke?

CHAPTER XVIII.

COMPOUND ENGINES.

IN the Simple Engine the steam after cut-off expands until the release-point is reached. The degree of expansion depends upon the point of cut-off. The earlier the point of cut-off the greater the expansion and *vice versa*. By making cut-off very early or by connecting a condenser, the steam may be expanded until its pressure becomes less than atmospheric pressure (see Fig. 164*a*). The steam, after expansion has ceased, is exhausted. Here the whole process of expansion has been accomplished in one cylinder. Now this process of expansion is sometimes carried on by means of two cylinders or more instead of in one. That is, the steam enters one cylinder and is expanded a certain amount, after which it is exhausted into another, in which the expansion continues. This is called compounding, and any engine in which the steam passes through two or more cylinders consecutively is called a Multiple-expansion Engine. The term compound is commonly used to designate those engines in which the expansion is accomplished in two cylinders.

In a compound engine the steam at boiler-pressure enters a comparatively small cylinder called the high-pressure cylinder. It is exhausted into a larger one called the low-pressure cylinder. If there are three cylinders, with three stages of expansion, the last entered by the steam is the low-pressure cylinder, and the one between the high pressure and low pressure is called the intermediate cylinder. If there are four

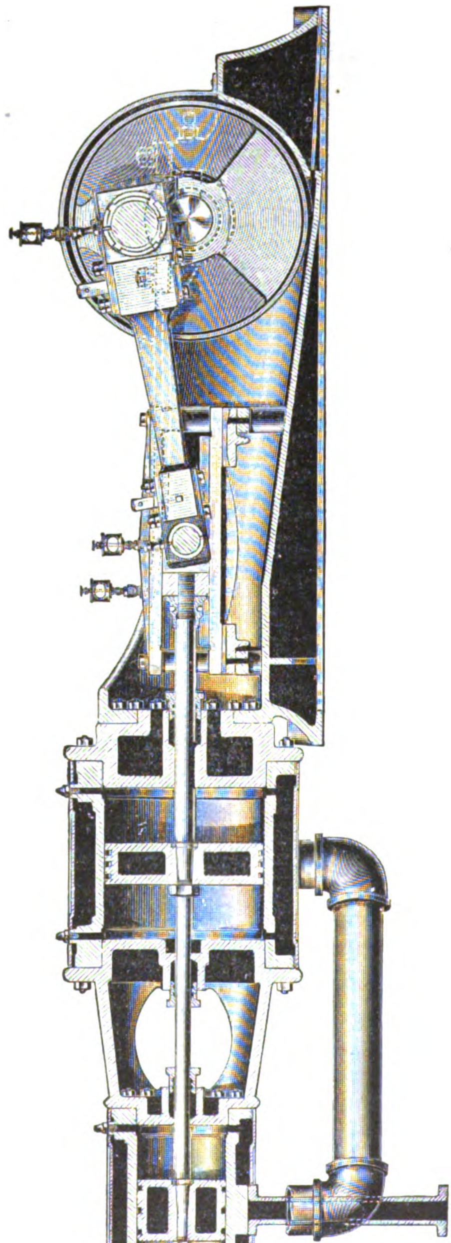


FIG. 165.—Tandem Compound Engine.

cylinders, with four stages of expansion, the one next to the high-pressure cylinder is called the first intermediate and the one next to the low-pressure cylinder is called the second intermediate.

An engine in which the steam exhausted from the first, or high-pressure, cylinder passes into two other cylinders, half into each, is called a three-cylinder compound engine, and an engine with four cylinders, in which the steam exhausting from the second, or intermediate, cylinder passes into two cylinders, is called a four-cylinder triple-expansion engine.

Of the compound engine there are two types, viz.: the Tandem and the Cross-compound. Those in which the cylinders are placed end to end and having only one piston-rod, as in Fig. 165, are Tandem Engines and those having their cylinders side by side and two piston-rods are Cross-compounds, Fig. 166.

In the tandem shown in the figure the steam from the boiler is entering the smaller or high-pressure cylinder to the left of the piston and driving it to the right.

The exhaust is leaving the right-hand end of the h.p. cylinder through the large pipe at the bottom and entering the l.p. cylinder at the left of the piston. It will be noticed here that this pressure of steam against the low-pressure piston is also back-pressure against the h.p. piston, but owing to the greater area of the l.p. piston a working effect is produced. The steam-distribution in a cross-compound engine is practically the same as the process just described for the tandem compound.

The tandem compound has only one piston-rod to which both pistons are attached. This rod may be made in two sizes as shown in cut. The tandem is much simpler than the cross-compound by reason of the fact that it is practically a single engine, having two cylinders and two pistons. This makes its cost small and it requires a small floor-space.

The cross-compound has two piston-rods, and in fact, is practically two separate engines operating one crank-shaft.

The cranks on these shafts are usually put on at an angle of 90° with each other, by this means distributing the strain on the crank-shaft so that all the effort is not exerted on one point. Another advantage of this arrangement is that while

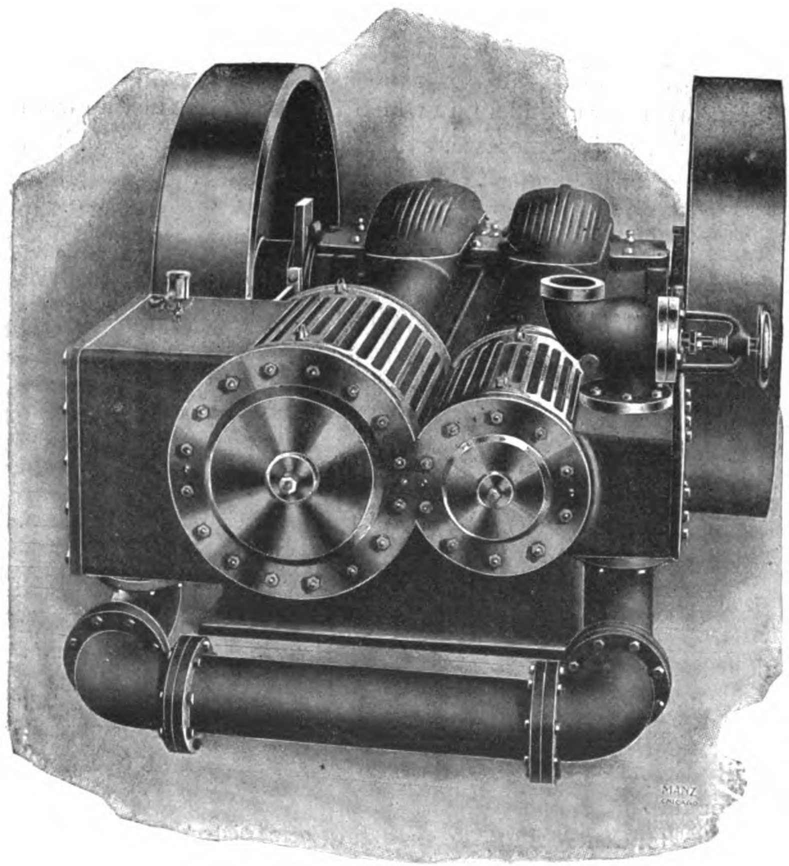


FIG. 166.—Ball Cross-compound Engine.

one crank is exerting its least effect the other is exerting its greatest effect. By this means, the dead-point or dead-centre is always passed. It is evident from the above that the cost and floor-space required for this engine will be great as compared with the tandem.

Simple and Compound Engines Compared.

The object of a compound engine is to economize in the use of steam. The question may be asked: Is it better to make a given number of steam-expansions in two cylinders rather than in one? The question is answered by saying, in some cases yes, in some, no.

It is evident that by making the point of cut-off early and the cylinder large enough, any number of expansions may be made in a single cylinder, and that compounding does not necessarily cause greater expansion.

The advantage of the compound is that in some cases the losses are made less than in the simple engine. The losses to which all engines are subject are, principally: loss of heat by radiation, heating the walls of the cylinder, and the loss due to condensation of the entering steam by the comparatively cool cylinder-walls, besides the losses due to friction (mechanical losses).

The loss due to radiation may be reduced by covering or lagging, and this may be done equally as well with the simple engine as with the compound engine, so that there is no advantage to be claimed for the compound in that respect.

The loss of heat due to heating the walls of the cylinder may be reduced by giving the piston a high speed, thereby shortening the period of contact of each particle of steam. Here again the compound engine has no advantage, since the simple engine is, in fact, more easily run at high speeds.

The losses of condensation, caused by the cooling of the entering steam, however, are in some cases made less by expanding in the two cylinders of a compound engine rather than in the single cylinder of a simple engine. The following is an explanation: When a certain number of expansions, say eight, take place in one large cylinder the steam at a temperature a little less than that of the boiler enters the cylinder and drives the piston forward at full pressure and heats that end of the cylinder up to its own temperature. The cut-off occurs at one eighth stroke, after which the expansion of steam begins and the temperature is correspondingly decreased until the end of the stroke is reached. This action leaves the temperature of the cylinder-metal much lower at the end of the stroke than at the beginning. This difference of temperatures is called the range of temperature.

The steam on entering for the return stroke gives up a large quantity of heat in warming the cool end of the cylinder up to its temperature, and so for every stroke there is an alternate cooling and warming, very detrimental to economy.

By making the eight expansions take place in two cylinders, with cut-off say at one half stroke in the high-pressure cylinder, making two expansions, and then exhausting into a low-pressure cylinder with a volume four times as large as the high-pressure cylinder, making a total of eight expansions, the range of temperature in each cylinder is reduced. This is the principal advantage of the compound engine. Another advantage is that by the use of two cylinders for expansion, and a given pressure for the exhaust, much higher initial pressures of steam may be used. It is known to be economical to use steam at very high pressures and temperatures, because the cost of obtaining very high pressures in a boiler is comparatively very small after a moderately high pressure is once attained. This is accounted for by the fact that the greater part of the heat at low pressures is spent in breaking up the molecular construction of the water, while it is comparatively easy to increase the pressure of the steam after it has once become a perfect gas.

Another advantage of the compound engine is that the effort may be distributed at different angles on the crank-shaft, if a cross-compound is used. The hottest steam is used in the cylinder of the smallest volume. This causes a diminution of the loss due to radiation.

There are objections to the compounding principle, such as increase in first cost, the friction is increased by reason of the increased number of moving parts, especially in the cross-compound, the greater loss of radiation from two cylinders instead of one, and the fact that the engine is wasteful of steam as compared with the simple engine, unless nearly the average load for which it was designed is carried.

Ratio of Cylinders.—It has been shown that the low-pressure cylinder is of much larger volume than the high-pressure cylinder. Since the volume of a cylinder varies as the square of the diameter, if R be the ratio of the volume of the high-pressure cylinder to the volume of the low-pressure cylinder, then $R = \frac{D^2}{d^2}$, D and d being the diameters respectively of the low-pressure and high-pressure cylinders.

The value of R depends upon the make and type of the engine and is usually from $2\frac{1}{4}$ to $4\frac{1}{2}$.

The total number of expansions in a compound engine is equal to the number of expansions in the high-pressure cylinder multiplied by R.

Receiver.—It has been explained that in a tandem compound engine, the steam-pressure acts against the same side of the high-pressure and low-pressure pistons at the same time. This, however, is not the case with the cross-compound, because the cranks are usually placed on the shaft at an angle of 90°. Here the exhaust steam from the first cylinder cannot be used directly in the low-pressure cylinder.

The exhaust from the high-pressure cylinder is therefore passed into an intervening vessel, called a receiver, where it remains during a half stroke or while the crank makes a turn of 90°.

The space between the exhaust-port of the h.p. cylinder and the steam-port of the l.p. cylinder, in either a tandem or a cross-compound, is called *receiver-space*. It includes the exhaust-pipe of the h.p. cylinder and the steam-chest of the l.p. cylinder. This space should be as large as possible in order to make frictional resistance of the steam small.

The expansion of the steam in the h.p. cylinder is attended with a lowering of the temperature. For the purpose of reheating this steam, some manufacturers have introduced a coil of pipe carrying hot steam from the boiler into this receiver-space between the two cylinders.

Indicator-cards.—In order to get an indicator-card from the two cylinders of a compound engine it is necessary to attach an indicator to each cylinder and connect them individually or by the same string to the reducing motion.

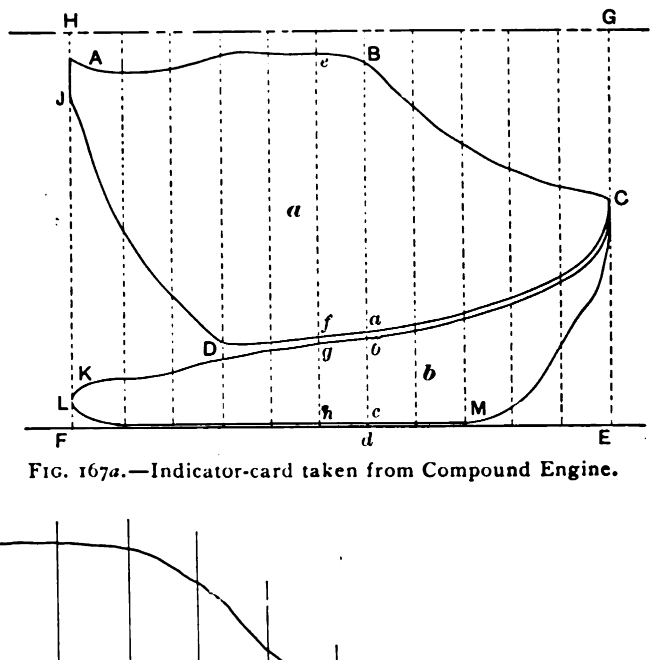


Fig. 167*a*.—Indicator-card taken from Compound Engine.

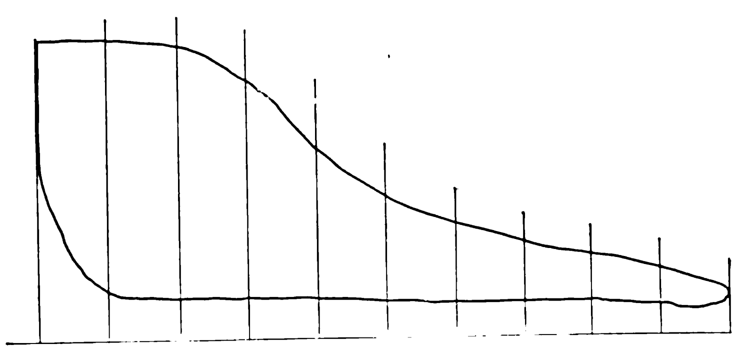


Fig. 167*b*.—Card from High-pressure Cylinder.

In Fig. 167 (*a*) is a card taken from the h.p. cylinder and (*b*) one taken from the low-pressure cylinder of a compound engine. The length of the engine-stroke is represented by

FE, and boiler-pressure by *FH*. *FE* is the atmospheric line. *AB* is the steam-line, and *B* is the point of cut-off for the h.p. cylinder. As the piston advances the volume of steam in the h.p. cylinder increases and the pressure decreases correspondingly, producing *BC*, the high-pressure expansion-curve. The point *C* represents the position of the high-pressure piston when its exhaust-valve is opened and the steam allowed to pass into the low-pressure cylinder.

This point *C* also represents the end of the h.p. piston-stroke, after which its return-stroke begins, the pressure falling to *D* as the steam expands into the l.p. cylinder. The point *D* is the point of compression for the h.p. cylinder, that is, when the valve which opens at *C* closes. *DJ* is the compression-curve, and *J* the point of admission of the h.p. cylinder.

Let us now turn our attention to the low-pressure card. The same edge of the valve which exhausts the steam from the h.p. cylinder admits it to the low-pressure cylinder, hence *C* is the point of admission for the l.p. cylinder. Also, since the h.p. exhaust is the l.p. steam, the line *CD* for the two cylinders should be common. If there is a space between them, it is caused by frictional losses.

PROBLEM.

1. Find the H.P. of the compound engine from which the card in Fig. 167*a* was taken, the stroke of both pistons being 12 inches, the diameter of the high-pressure piston 8 inches, and the diameter of the low-pressure piston 12 inches, piston-rods $1\frac{1}{2}$ inches, and the number of revolutions per minute 150.

To Combine the Indicator-cards of a Compound Engine.—Let Figs. 167*b* and 167*c* represent average cards from a compound engine, obtained by averaging the cards from the head and crank end of each cylinder. Divide each card into, say, ten equal parts, and erect ordinates at the points of division. Now take a line *AB*, Fig. 167*d*, as an atmospheric line, and *CD* as a line of zero volume. Select on *AB* some convenient distance, as *EF*, for the length of the high-pressure card on the combined diagram. Divide this distance into the same number of equal parts as the h.p. card and erect ordinates. The clearance of the high-pressure cylinder being known, lay off the distance GH to represent the clearance. GH should be

FIG. 167c.—Card from Low-pressure Cylinder.

the same percentage of EF as the clearance is of the volume of the cylinder. Then enlarge* the ordinates on the high-pressure card, Fig. 167*b*, to a convenient scale and take the ordinates erected in Fig. 167*d* as equal to the length found by this enlargement. Draw a curve through the extremities of the ordinates. This curve will be the high-pressure card on the combined diagram.

Now let the volumes of the high- and low-pressure cylinders, swept over by the piston, be to each other as $\frac{V}{V_1}$. Then the length JK of the low-pressure card on the combined diagram will be

$$JK = EF\frac{V_1}{V}.$$

Lay off the distance GK as the clearance of the low-pressure cylinder, GK being the same proportion of JK as the clearance of the low-pressure cylinder is of its volume.

Then erect the same number of ordinates on the atmospheric line as were erected on the low-pressure card, Fig.

* Multiply.

167*c*. Determine the length of these ordinates by enlarging or reducing those on the original low-pressure card to the same scale as the high-pressure ordinates on the combined card, and taking their length as the length of the ordinates on the com-

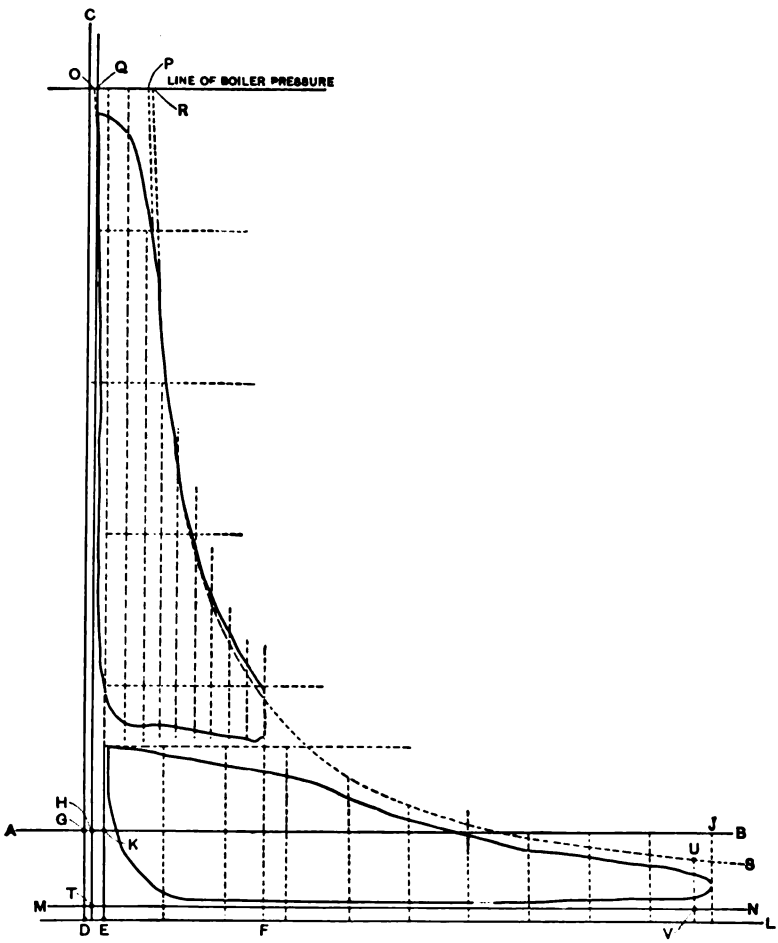


FIG. 167*d*.

bined card. Draw the card through the ext ities of these ordinates; draw *DL*, the line of zero absolute pressure, and *MN*, the line of back pressure.

Now if the actual area of the combined indicator-cards be compared with the theoretical area for the volume of steam at boiler-pressure admitted to the high-pressure cylinder expanding to the volume in the low-pressure cylinder, the back pressure being taken as that in the condenser, the ratio of the area of the cards to the theoretical area is the ratio of the work done in the cylinder to the theoretical work.

The theoretical area is determined by extending the compression-curve according to the law $pv =$ constant until it meets the line of boiler-pressure at O, and also the expansion-line from some point below the cut-off until it meets the boiler-pressure line at P. Make QR equal to OP and draw Mariotte's curve RS, taking T as the origin. Lay off the distance TV, making it equal to JK, the length of the low-pressure card. Erect the ordinate VU. The actual areas and the theoretical area are found by planimeter; the theoretical area is $QRUVTQ$.

It is assumed in this case that the length of stroke and number of revolutions is the same for each cylinder

CHAPTER XIX.

CONDENSERS.

THE condenser is frequently used with the simple engine, but the common practice is to make compound or multiple-expansion engines condensing. The object of the condenser is to economize in the use of steam, or, in other words, to get more nearly the entire force from the steam in an engine than could be obtained without one.

The steam from an engine if exhausted into the atmosphere causes a small back pressure on the side of the piston opposite the live steam. Making the exhaust-pipe discharge vertically, making it very long, or causing it to make sharp turns, all tend to increase this back pressure. For illustration let *A* and *B* in Fig. 168 represent a cylinder and piston. The live steam is entering and pushing the piston towards the right. Suppose that *b* opens to the atmosphere, then the pressure on the side *C* will be 14.7 pounds plus that due to the friction of the steam in the exhaust-pipe and to inertia. This is a real loss because it neutralizes just that much of the pressure of the live steam on the side *A*.

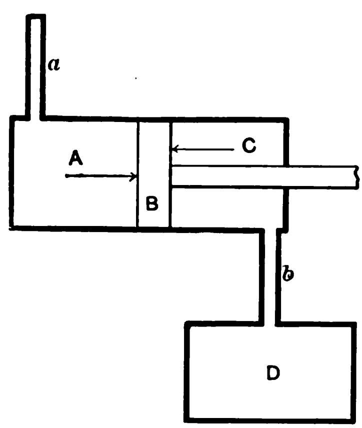


FIG. 168.

Suppose, however, that *b* is made to open into a chamber *D*, and that by a spray of water or otherwise the steam is condensed immediately upon entering. By this means the entire volume of steam is reduced to a few drops of water in the bottom of D. It is evident that this will produce a partial vacuum in D. This changes the back pressure to an effective forward pressure. This partial vacuum is usually made still more perfect by means of an air-pump which draws not only the air from D but keeps it free from water.

The foregoing is merely to illustrate the principle of the condenser. The principal difference in the different makes of condensers is in the manner of cooling the exhaust steam. This is done either by bringing it in direct contact with a jet or spray of water, or by bringing it in contact with tubes which are kept cool by circulating water through them. The former is called the *Jet-condenser*, the latter the *Surface-condenser*.

The Jet-condenser.—In a jet-condenser the exhaust steam from the engine enters a chamber called the condensing-chamber, where it is intercepted by a spray of cold water and condensed; it then falls with the condensing water to the bottom of the condensing-chamber. From here it is drawn off, usually by an air-pump, to a reservoir called the hot-well. From the hot-well it is taken as it is needed to the boiler by the feed-pump, providing it has been previously freed from oil, by means of a separator or filter.

Fig. 169 is an illustration of this type. It is made by the Worthington Hydraulic Works. The exhaust steam from the engine enters at A. F is the condensing-chamber into which the condensing-water is introduced at B and through D which sprays it. The condensed steam falls to the bottom and is forced out through J by the air-piston G. This air-piston is worked by the steam-engine at K. The water is carried from J to the *hot-well*. The amount of condensing-water admitted is controlled by means of the hand-wheel E.

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

Surface-condenser.—In the surface-condenser the exhaust steam from the engine enters a condensing-chamber, which consists of a vessel in which a number of tubes are placed, with water circulating through them. The hot steam is cooled

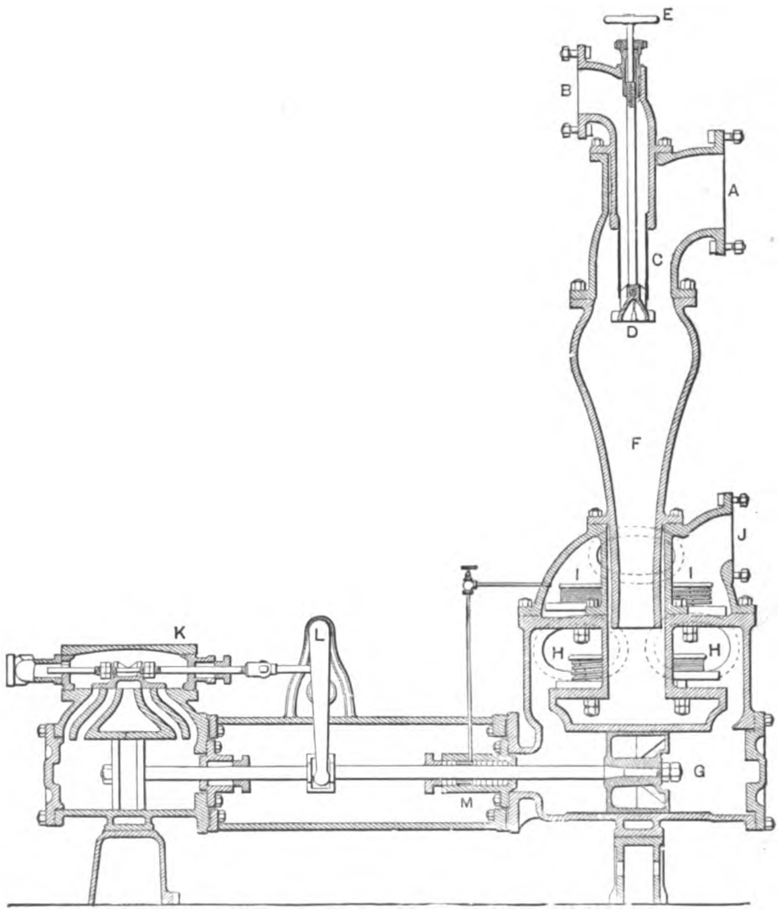


Fig. 169.—Jet-condenser.

and condensed on coming in contact with the cool pipes. It falls to the bottom of the condensing-chamber, as in the jet-condenser, and is drawn off to the hot-well by the air-pump.

Fig. 170 shows in section a condenser of this type made by the Wheeler Condenser and Engineering Co. The exhaust steam enters the top of the condensing-chamber, striking the baffle-plate, which spreads it and protects the tubes

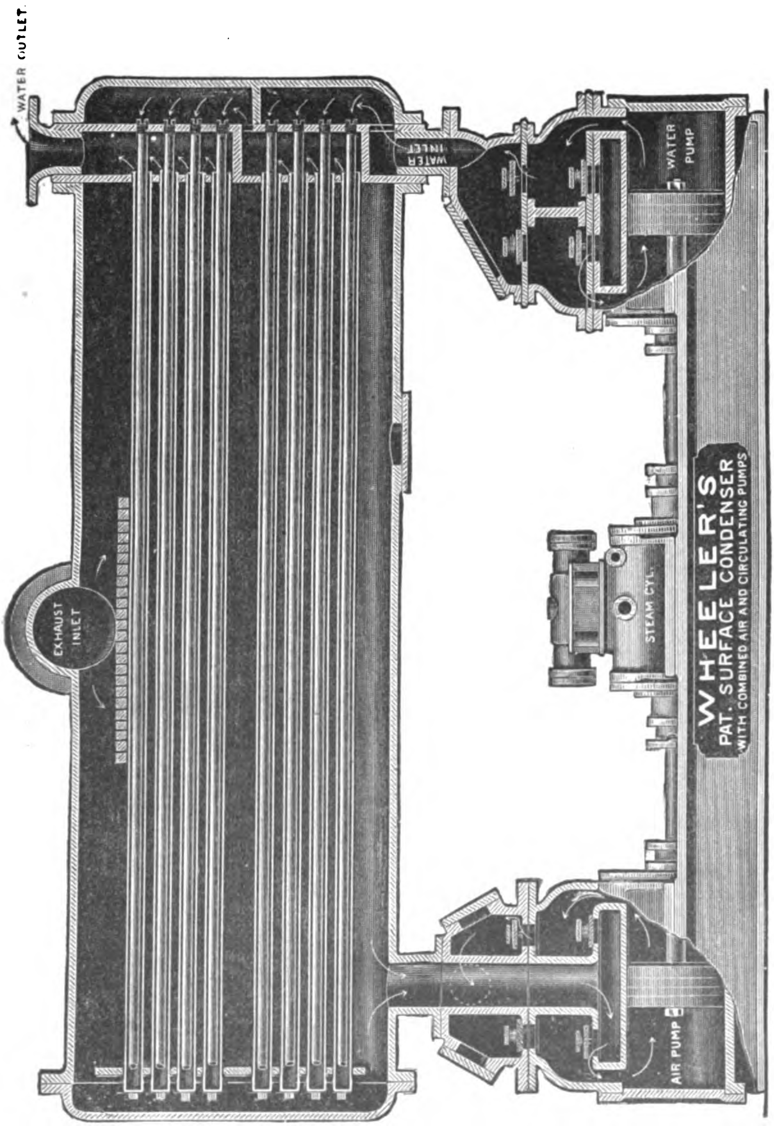


Fig. 170.—Surface-condenser.

nearest the entering steam. The steam is condensed upon striking the tubes, and falls to the bottom of the condenser.

whence it is drawn off by the air-pump and, if the oil in the water has been removed, is taken off to the hot-well or to the boiler through the outlet shown in the figure by the dotted circle.

The principal parts of the surface-condenser are: The condensing-chamber, the air-pump, the hot-well, and the circulating-pump, the same as for the jet-condenser with the addition of the circulating-pump. The condensing water enters at the bottom, thus causing the coolest water to meet the cooler steam and *vice versa*. This reduces the damage done by uneven expansion in the tubes and tube-plates.

Jet- and Surface-condenser Compared.—From the two foregoing examples it may be seen that the jet-condenser is much simpler in construction than the surface-condenser. For this reason the jet-condenser is used largely on land. Since the cooling water comes in direct contact with the steam it is obvious that pure water must be used for cooling purposes if it is used in the boiler again.

The surface-condenser has the advantage that its condensing water does not come in direct contact with the steam. This makes it possible to use any kind of water for cooling purposes. Salt water at sea, muddy water, etc., may be thus used. For this reason the surface-condenser is used on sea-going vessels, in which case the condensing water is salt and it is allowed to run back into the sea. When surface-condensers are used on land the condensing water, after it has accomplished its cooling effect and been heated, is usually allowed to run to waste. Sometimes, however, this water is used over again after cooling it by some method. One of these methods is, to cause it to run along a series of shallow troughs in the open air exposed to the cooling effect of natural winds. The troughs are arranged one over the other with a slight grade so that the water finally reaches the bottom trough cooled, from which it is pumped again through the condenser. Another later and better method is to cause the heated water to descend in a cooling-tower, in a divided state, through trays of wire gauze. A circulating-fan induces a current of air which causes a vaporization and cooling of the water. When it has

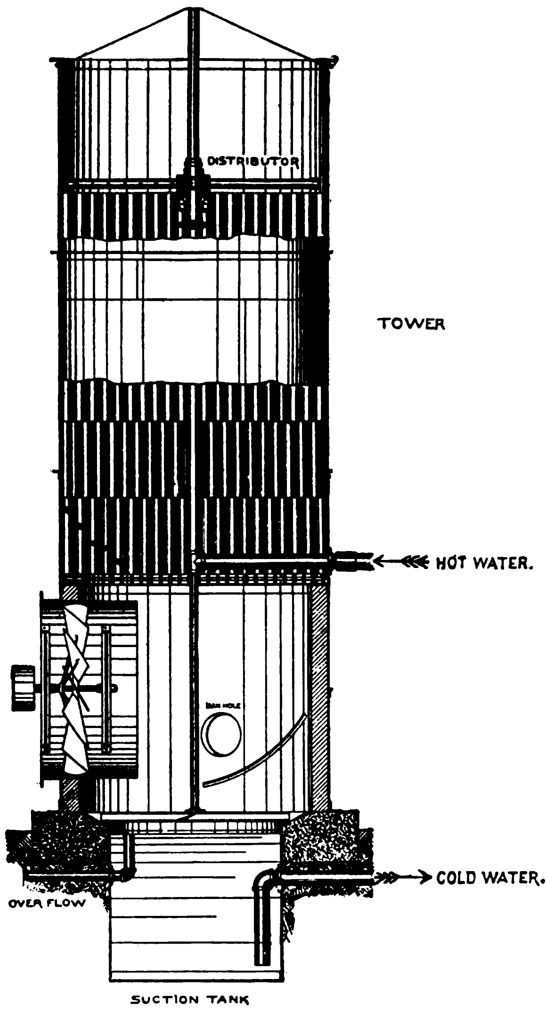


FIG. 171.—Cooling-tower.

fallen to the bottom of the tower and become cool it is again pumped through the condenser.

Fig. 171 shows this arrangement used in connection with a jet-condenser.

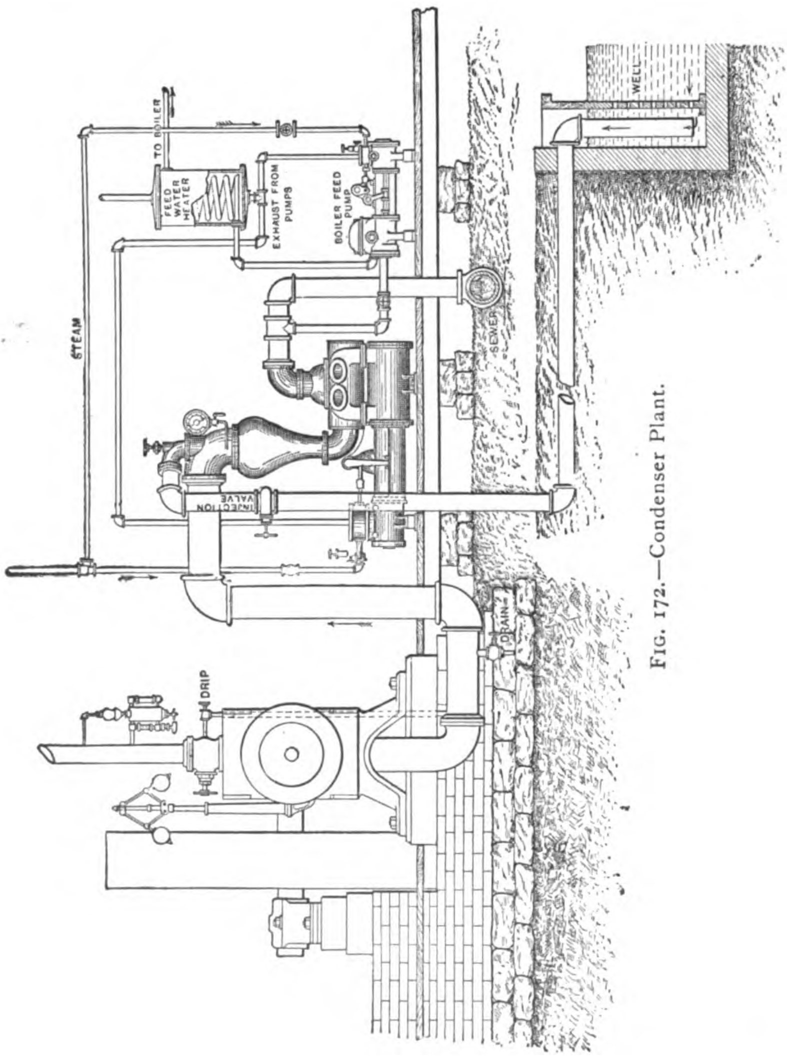


FIG. 172.—Condenser Plant.

One of the principal objections to the surface-condenser is that the expansions and contractions of the tubes caused by their heating and cooling causes much trouble from their leak-

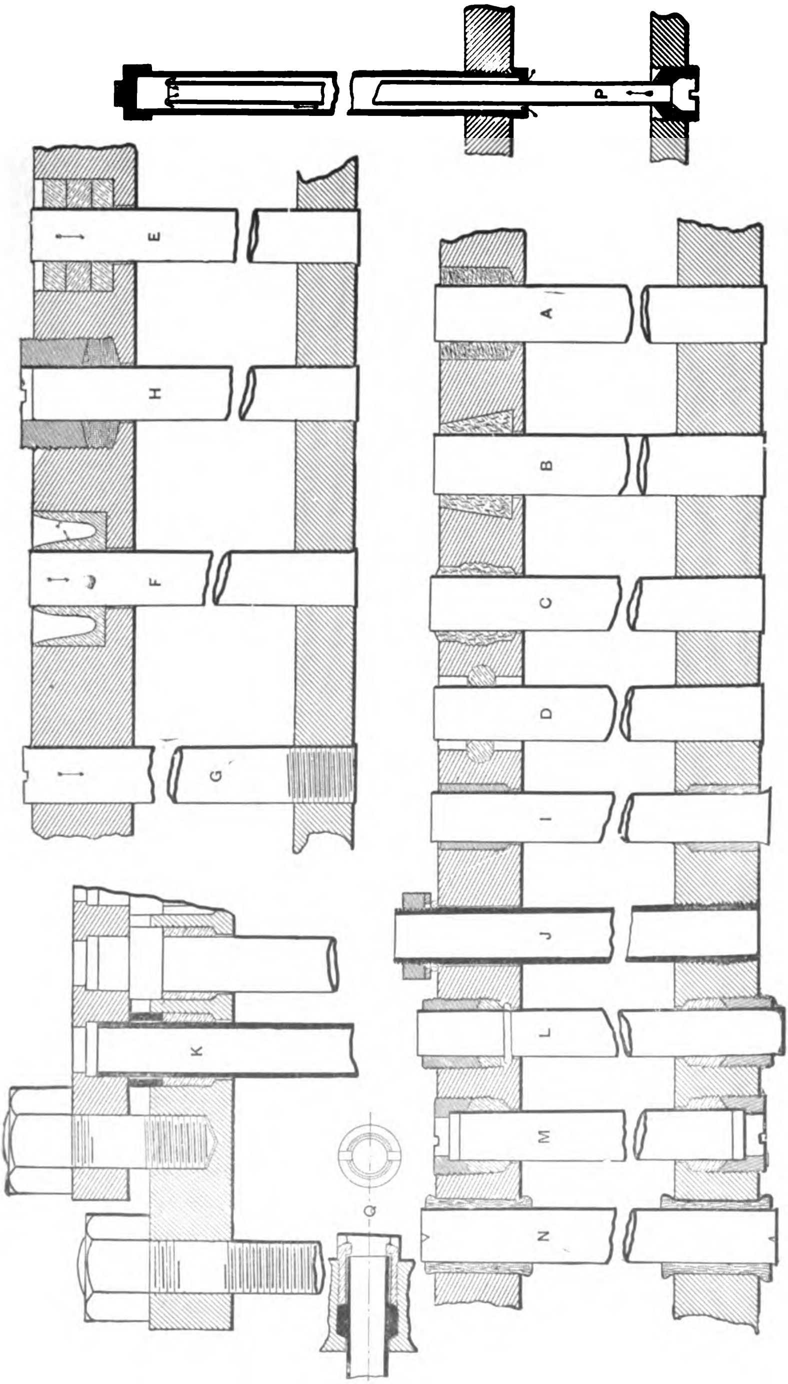


FIG. 173.—Condenser-tube Joints.

ing, similar to those in tubular boilers. This is generally obviated by making the tubes fast to one head-plate and movable with reference to the other, a joint being made in the manner of a stuffing-box. The tubes are also liable to be destroyed by corrosion.

Fig. 173 illustrates several ways of making such a joint. When the tube increases in length by reason of expansion by heating, this joint allows the end to slide. Fig. 170 shows a unique method of avoiding the difficulty stated above. The condensing water passes through a set of double pipes, one within the other, the smaller one fastened to one and the larger one or casing to the other head-plate. The water enters the small pipe at *F* and flows to the end at the left and empties into the larger pipe in which it returns to *G*, from whence it goes to the outlet *D*. By this means the small pipe may slide within the other and prevent the disastrous effects of contraction and expansion.

The Air-pump.

The question is sometimes asked why should the pump which keeps the condenser drained of water be called an air-pump.

In reality, it does assist not only in pumping water but also in maintaining and increasing the vacuum which has been partially made by condensation and because it also removes air which has entered in the steam.

The air-pump is a lifting or suction-pump, and usually has its valve in the piston.

The pump is usually placed below the condensing-chamber in order that the water may enter it by gravity. The air-pump piston may be driven by a belt from the engine or it may be driven by a separate engine, especially for the purpose. This kind is called the independent condenser, of which the Wheeler and Worthington condensers are examples.

The belt-driven condenser-pump has the advantage of small first cost and simple construction, but it is obvious that the speed depends on the speed of the engine itself; and that if belted directly to the engine, it must be in the plane of the engine in order to receive the belt-connection, making it necessary to place it oftentimes where it is in the way.

The independent condenser is of higher first cost and more complicated in construction, but it has the advantage of being more easily controlled. It can be started before the engine is started and can be speeded up or down according to the needs shown by the vacuum-gauge, which is attached to the condensing-chamber.

Fig. 174 shows a vacuum-gauge. Its inner mechanism is

FIG. 174.—Vacuum-gauge.

similar to that of the common steam- or air-gauge. It is graduated to read "inches of mercury."

The pressure of the atmosphere is 14.7 lbs. per square inch. This pressure will support a column of mercury 30 inches high, that is, 14.7 lbs. is equivalent to 30 inches of mercury. $\frac{30}{14.7} = 2 +$, that is, 1 lb. pressure is equal approximately to 2 inches of mercury as read by the gauge. Thus, the gauge-reading may be reduced to pounds by dividing by 2. It is impossible to make a perfect vacuum and in

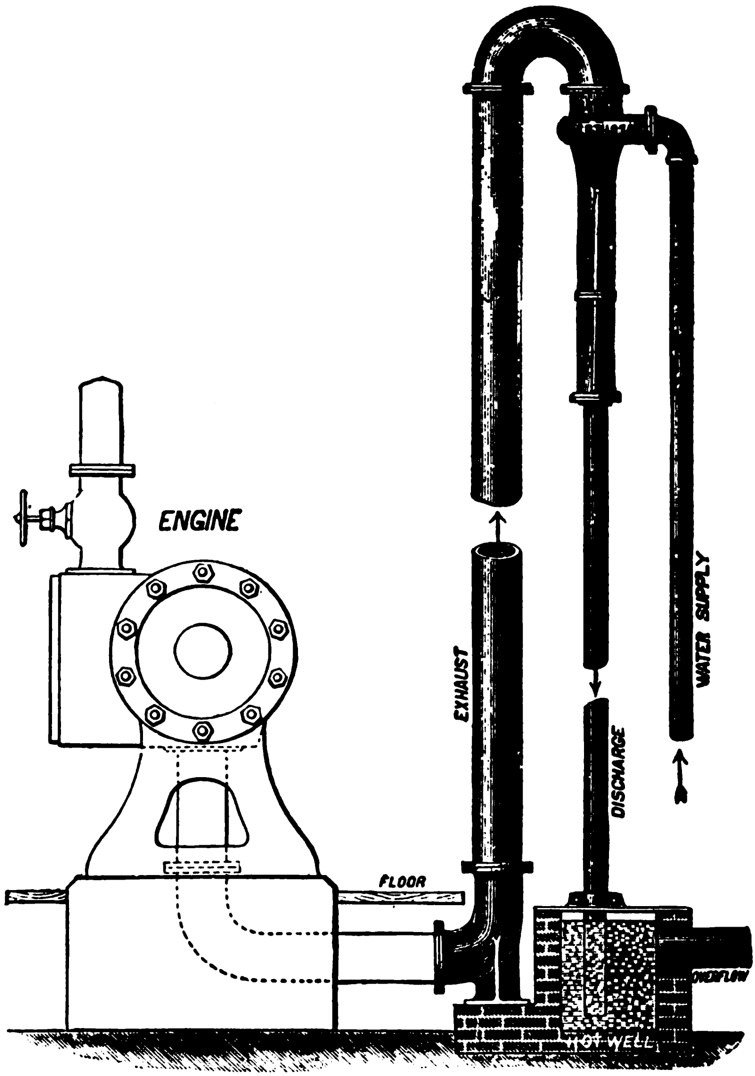


FIG. 175.—Bulkley Condenser.

the case of the condenser 26 inches of mercury or about 13 lbs. pressure is about the average vacuum produced. The perfection of the vacuum will of course depend upon the efficiency of the air-pump.

The Siphon Condenser.

The atmospheric pressure will support a column of water over 32 feet high. This principle is sometimes used to make a condensing-plant without an air-pump.

The principle is applied by the use of a pipe 34 feet high, at the top of which is placed an arrangement similar to an injector through which the exhaust steam and cooling water are drawn by atmospheric pressure. Fig. 175 shows this principle applied in the Bulkley Condenser.

The exhaust steam passes downward through the goose-neck at the top of the apparatus and through the inner cone surrounded by an annular cone of water. The steam is condensed here and falls with the condensing water, entraining the air as it falls. This it will be seen requires no air-pump, but the injection- or circulating-pump is still necessary.

Fig. 176 represents an indicator-card taken from the low-pressure cylinder of a condensing-engine. Here it is noticeable

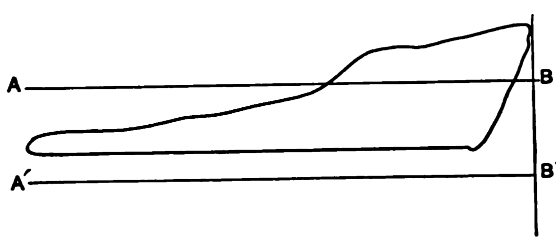


FIG. 176.

that the atmospheric line, AB, is considerably above the line of counter-pressures, its distance above the same corresponding approximately to the pressure due to the vacuum.

PROBLEMS.

1. A vacuum-gauge gives a reading of 27 inches. Find the equivalent pressure in pounds per square inch.

2. In the indicator-card shown in Fig. 176 find the approximate vacuum reading in inches of mercury, the scale of the indicator being 40.

CHAPTER XX.

VALVES AND VALVE-GEARING.

IN Chapter XV in the description of the simple engine, mention was made of the fact that the engine was operated by a D slide-valve. This valve is shown in section in Fig. 177,

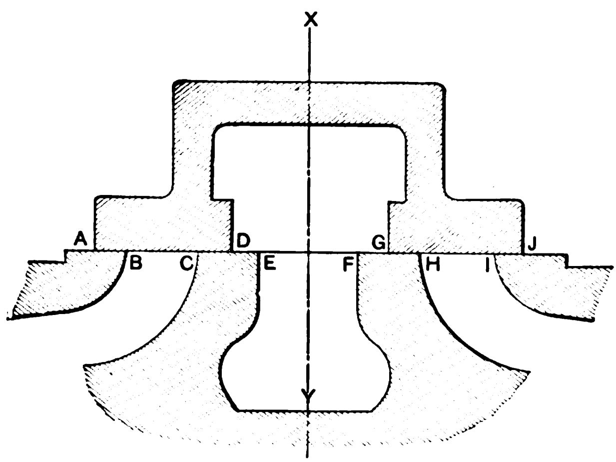


FIG. 177.—Slide-valve.

together with a portion of the valve-seat. The spaces *BC* and *HI* are the steam-ports leading to the cylinder, *EF* is the exhaust-port, and *DG* is the exhaust-chamber in the valve. The portions of valve, *AB* and *IJ*, projecting beyond the outside edge of the ports, when the valve is in its middle position, are known as the *steam-lap* or outside lap of the valve. The portions, *DC* and *HG*, projecting into the exhaust-chamber beyond the inside edge of the port are the *exhaust-lap* or inside lap of the valve. The object of these laps will be explained later.

Lead.—It has been explained in Chapter XV how the valve is moved by the eccentric so as to admit steam to and exhaust it from the cylinder. The manner in which the valve is set will now be shown. Suppose the piston to be just at the end of its stroke. The crank-pin will be just passing over the dead-centre, and will be moving slower in the direction of the centre line of the engine than at any other portion of the stroke. It is desirable that steam be admitted to the cylinder as quickly as possible; to do this it is necessary that the valve should be moved quickly, thus opening the steam-port wide in the shortest possible time. If the eccentric is placed 90° ahead of the crank on the crank-shaft, in the direction of rotation, it will be moving its fastest in the direction of the centre line of the engine, at the instant the crank is moving slowest. If the eccentric was fixed in this position of 90° ahead of the crank, the valve would be in mid-position, the centre line of the valve coinciding with the centre line of the exhaust-port, when the engine was on its dead-centre. Then before steam could be admitted to, say, the port *HI*, Fig. 177, the valve would have to move the distance of the steam-lap, *IJ*. But it has been stated that steam was to be admitted as quickly as possible to the cylinder; therefore the eccentric is advanced still further on the crank-shaft, until the point *J* is just over or beyond the point *I*. The distance the point *J* is advanced beyond *I* is termed the *lead* of the valve. The angle through which the eccentric is advanced ahead of the crank over 90° is termed the angle of advance. Thus, steam will enter the port *HI* just before the crank passes the dead-centre, or just before the piston reaches the end of the stroke; the amount of lead determines how much before. If the angle of advance is such that the points *J* and *I* coincide when the crank is passing the dead-centre, the lead will be zero. The amount of lead is fixed by the angle of advance, and can be altered by moving the eccentric.

Lap.—The steam-lap determines the point of cut-off of the engine. The greater the lap, the earlier the cut-off will occur. If there were no steam-lap on the valve, that is, if when the valve was in mid-position, the point J and the point I were to coincide, the valve would admit steam during the entire stroke of the piston. The valve is wide open when the eccentric is passing its dead-centre. If there is no lap or lead the valve will be in mid-position and just closed when the crank passes the dead-centre, the eccentric being but 90° ahead of the crank in this case. Now if the valve be given steam-lap it is easily seen that when the valve is wide open, the point J will be nearer the point I than would be the case if there were no lap. Hence the valve will close the port before it reaches mid-position. By making the lap great enough it is possible to admit but a small amount of steam to the cylinder, the cut-off occurring soon after the beginning of the stroke. The extreme case is obtained if the lap is made so great that the valve does not open at all, thus allowing no steam whatever to enter the cylinder. If such a case were possible the cut-off would be said to occur at the beginning of the stroke.

The object of the exhaust-lap is to increase the compression of the steam in the cylinder. It acts in the same manner as the steam-lap, closing the port before the end of the stroke. By closing the port before the end of the stroke some steam is trapped in the cylinder, and as it cannot get out it is compressed by the motion of the piston. In the valve shown in Fig. 177, if the steam were compressed above the pressure of the steam in the steam-chest, the valve might be lifted from its seat. In many cases therefore the valve is held in place by a plate which presses against it and is fastened to the steam-chest. The valve, called in this case a balanced valve, slides between this plate and the seat.

The lap on each end of the valve is not always the same. If the lap, both steam and exhaust, were the same the cut-off and compression on each end of the cylinder would be different. This is due to the angularity of the crank. It is usual in the design of engines to equalize the cut-off and compression by making the steam- and exhaust-lap on the two ends different. The method of equalizing the cut-off and compression will be shown in the chapter on "Valve-diagrams."

Setting an Eccentric.—First put the engine on a dead-centre. This may be done by bringing the piston almost to the end of the stroke. Make a mark on the cross-head and another on the guide, close beside the first one. Next make a mark on the fly-wheel. This mark should be opposite a pointer which remains stationary when the engine revolves. When these marks are all made, revolve the engine past the dead-centre, until the mark on the cross-head is once more opposite the mark on the guide. Mark the fly-wheel once more opposite the pointer. If the middle point of the arc on the fly-wheel between the two marked points be brought opposite the pointer, the engine will be on the dead-centre. The eccentric is next brought 90° ahead of the crank in the direction of rotation of the engine. Then it is given an angle of advance until the desired lead is obtained, the steam-chest cover being removed to make the valve visible. The eccentric should be temporarily fastened until the engine has been put on the opposite dead-centre, and it has been ascertained if the leads on both ends are equal. If they are not, they should be made so by adjusting the length of the valve-rod and then the proper amount of lead is given by turning the eccentric on the shaft in a direction depending upon whether a greater or a less lead is desired.

If it is desired to run an engine in the opposite direction to which it is already running, the eccentric should be loosened and rotated on the shaft until it is as much behind the crank as it was formerly ahead of it. The engine should be put on the dead-centres and the lead equalized, just as if the valve was being set for the first time.

Reversing Gears.—There are a number of valve-gears in existence, both for reversing and for varying the cut-off. The simplest of these is the Stephenson link, shown in Fig. 178.

The gear consists of the link *CD*, which is connected with the two eccentrics *A* and *B* on the shaft *S* by the eccentric-rods *CA* and *BD*. A block *E* slides in the link. A rod *EF* joins this block to a rocker *HGF*. The valve-rod *HI* is attached to the other end of the rocker which rotates about *G*. One eccentric is the "go-ahead" eccentric; the other is the reversing eccentric. If the rocker *HGF* were not interposed, *B*

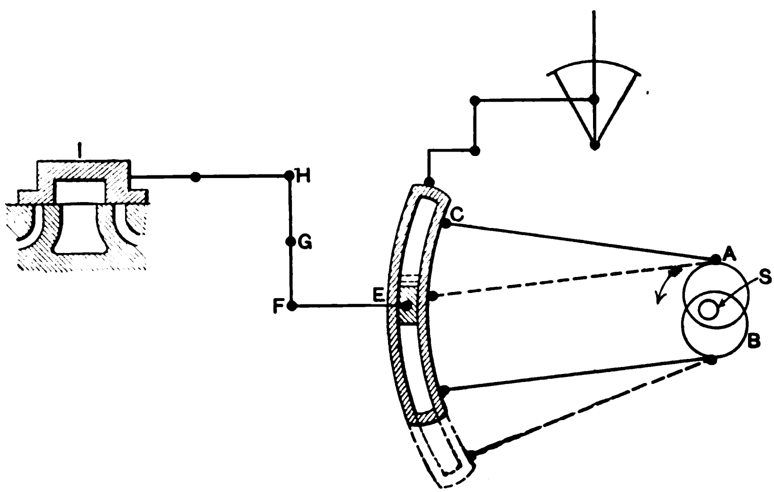


FIG. 178.—Stephenson Link.

would be the forward eccentric, but the relative direction of motion of the eccentric and valve-rods is reversed by the rocker. In the position shown by the full lines, the block is at the middle of the link; the two eccentrics neutralize each other and the piston cannot move. To move the engine either way the link is raised or lowered as may be desired, by means of the system of levers shown. If the link is in the position shown by the dotted lines the engine will run forward, that is, in the direction of the arrow. The cut-off will occur at the latest time possible with this combination. The nearer the block *E* is to the centre of the link, the earlier will be the cut-off. This system is most extensively employed on American locomotives and on some marine engines.

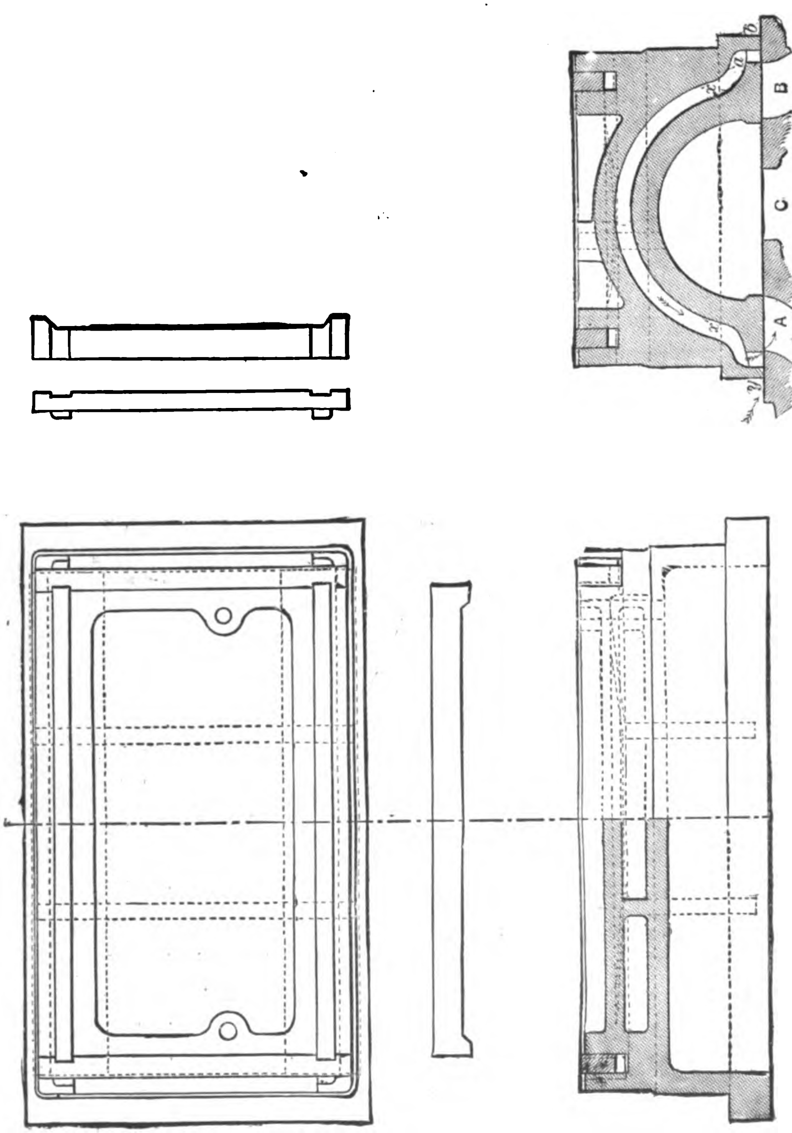


FIG. 179.—The Allen Valve as used on Baldwin Locomotives.

Another form of link is the Gooch link. In this contrivance the link remains stationary and the block is moved. There are many other forms of valve-gear for operating the D-valve, but lack of space forbids their description in this work.

The piston-valve has already been described in Chapter XVI. There are a number of modifications of the D-valve which are used. They will be described briefly below.

The Allen Valve.—This is an ordinary D-valve having a steam-passage around the outside of the exhaust-chamber as shown in Fig. 179. This passage is cut through the steam-lap and so communicates with the port. When the valve is open the maximum amount, steam enters the port from outside as in an ordinary slide-valve, and also by means of the passage. The passage should be equal to one half the port-opening. The advantage of this valve is that it works more quickly than an ordinary D-valve and accomplishes the same results with half the travel.

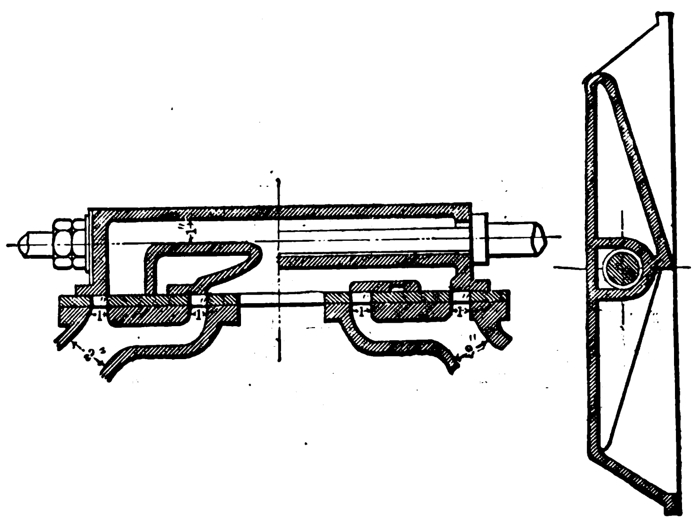


FIG. 180.—Double-ported Slide-valve.

The Double-ported Valve.—This valve, shown in Fig. 180, accomplishes the same results as the Allen valve. One half the steam enters the port from the outside of the valve; the other half enters through the passages cut in the sides of the valve and leading to an auxiliary port in the valve-seat, as shown. The exhaust steam enters the exhaust-chamber in two places, one half coming over the live-steam passage, the rest coming directly into the chamber. The area of the steam-passage in the valve at its outside should be equal to half the area of the auxiliary port. It should become smaller toward the centre until it is zero at the centre. The valve should be high enough to take care of all the exhaust steam that flows over the steam-passage.

Gridiron Valve.—This consists of an arrangement of transverse openings and bars, covering a like arrangement in the valve-seat. There is usually a valve for each end of the cylinder. This valve gives a wide opening with a very short travel.

HIGH SPEED AUTOMATIC CUT-OFF VALVES.

The valves of this class are very numerous. Only a few will be described.

The Meyer Valve is shown in Fig. 181. It consists of a plain D-valve with passages outside of the steam-lap leading

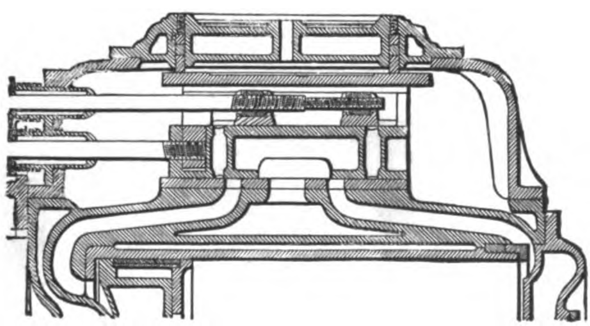


FIG. 181.—The Meyer Independent Cut-off Valve.

to the top of the valve. These passages are covered by blocks which slide on the valve, and which are driven by a separate eccentric. Steam is allowed to enter the ports only through these passages. The distance which separates the blocks can be varied at will by turning a hand-wheel. This wheel turns a right- and left-hand thread which causes the blocks to approach or recede from each other according to the direction of rotation. The blocks can be varied so as to be totally inoperative or to prevent steam from entering the cylinder at any portion of the stroke.

The Buckeye Valve.—This valve is shown in Fig. 151. The valve is held to its seat by balance-pistons which are extended by coiled springs. Steam enters at *D* and passes to the chamber *I* in the valve. Ports cut in these chambers admit steam to the cylinder. These ports are covered by sliding-blocks as in the Meyer valve. These blocks are driven by a separate eccentric. The valve-rod of these blocks passes through the main valve-rod, which is hollow. The cut-off is varied by the eccentric driving the blocks, which is rotated on its shaft by the governor, for variations in load, as will be explained later.

The Ideal Engine Valve has been illustrated in Fig. 153. It is of the hollow-piston type. The valve is driven by a slotted eccentric, which is shifted by the governor on variation of the speed. This alters the travel of the valve, thus making the engine automatic in its action.

SLOW SPEED AUTOMATIC CUT-OFF VALVES.

Corliss Valves.—A Corliss engine is operated by rotating valves instead of slide-valves. Two valves control the admission and two the exhaust. These valves are shown in section in Fig. 156. The two upper valves are the admission-valves; the two lower are the exhaust-valves. The valves are rotated by links from the wrist-plate shown on the cylinder of the engine, Fig. 155. The wrist-plate is rotated by the eccentric-rod, through a rocker, as shown in the figure. The rods joining the valves to the wrist-plate may be altered in length so as to change the admission cut-off, release, and compression

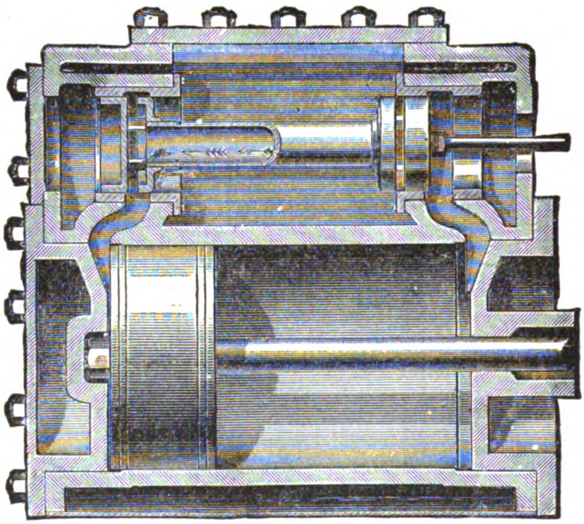


FIG. 182.—Armington and Sims Piston-valve.

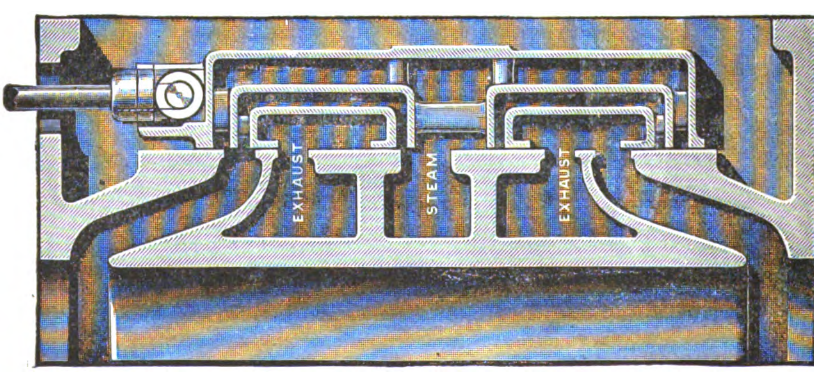


FIG. 183.*—Giddings Valve.

* This valve takes steam internally as shown in the figures.

as may be desired. The admission-valves are opened by a latch connected to the valve-rods. This latch is shown in Fig. 184. A piece carrying two arms, *CD* and *CA*, rotates about the centre *C*. *C* is a point on the axis of the valve. Swung on a pivot at *D* is a trip with two arms, *ED* and *EF*.

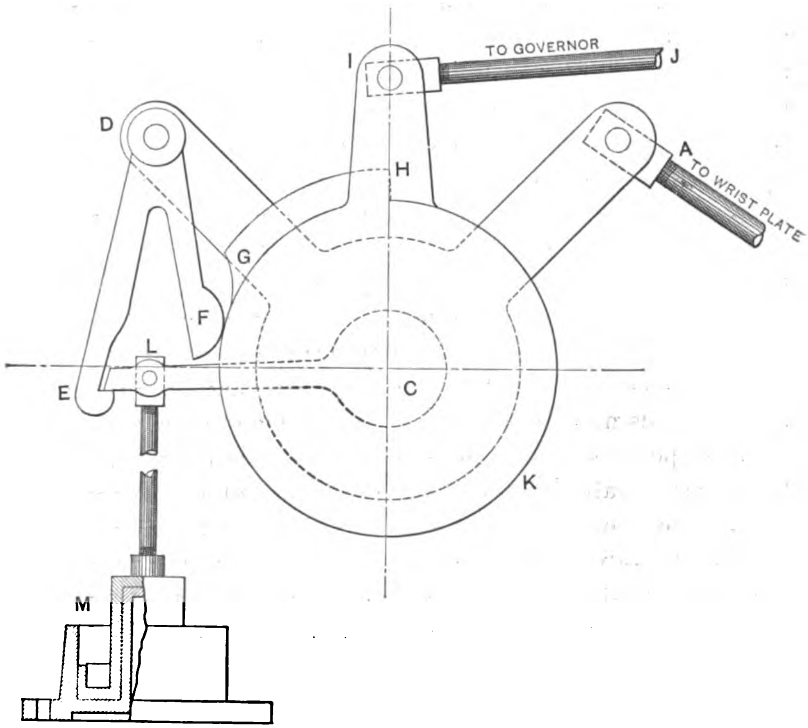


FIG. 184.—Releasing-gear of Corliss Valve.

At *E* is a notch which engages a lever *CL*, which rotates the valve. The extremity, *F*, of the other arm of the trip, when the piece *ACD* is rotated, moves along the circumference of a circular piece, *HGK*, which also is centred at *C*. On the circumference of this piece is a projection, *HG*, which joins the circumference by means of a concave curve at *G*. The valve is connected to a dash-pot at *M* by the rod *LM*. The piece *ACD* is joined to the wrist-plate by means of the valve-rod at A. The piece HGK carries an arm CI. At the point I of this arm a rod connecting with the governor is attached. Now the wrist-plate moves the piece ACD, causing the trip EDF to rise. The notch at E being engaged with the lever LC, the valve is rotated and the port opened. When the trip has been raised a certain distance the projection at F on the arm DF strikes the projection HG. The point F is forced away from C and the trip is rotated about D as a centre. This disengages the notch E from the lever LC, and the released valve is immediately closed by the dash-pot pulling LG to its original position. As the speed is varied the governor moves the rod IJ, thus turning the piece HGK about C and altering the position of HG. This will make the cut-off occur earlier or later as the case may be. The exhaust-valves are connected rigidly with the wrist-plate, and having no releasing devices are positively driven. The valve-rods are in two pieces. The pieces are joined by a turnbuckle with a right and left thread, so that the rods may be lengthened or shortened at will.

A dash-pot is shown at M, Fig. 184. Its piston is an air-tight fit, but a valve not shown in the figure admits a limited quantity of air under the piston. The object of the dash-pot is to close the valve almost instantly as soon as it is released by the trip. It also affords a cushion so that the parts of the valve-mechanism are not violently jarred by the sudden closing.

The governor of the Corliss engine is of the fly-ball type. When the engine speeds up the balls rise, thus moving a lever which is attached to the rods moving the piece GHK in Fig. 184.

Greene Engine.—This engine, like the Corliss, has two valves for admission, and two for exhaust. In the Greene engine, however, the valves are flat slides instead of being rotary. The operating mechanism is shown in Fig. 185. AB is a sliding-block, operated by the eccentric. It carries two tappets C and D. These tappets strike toes E and F on the rocking levers G and H respectively, which move the valves.

If the block AB be moving in the direction shown by the arrow, the tappet D will engage the toe F, rotating it and thus moving the valve. The other tappet will pass under the toe E; the toes are beveled on the under side and thus raise in their sockets, when the beveled side of the tappet comes in contact with them. When the tappet has passed the toe will drop of its own weight. The toe F moves in the arc of a circle.

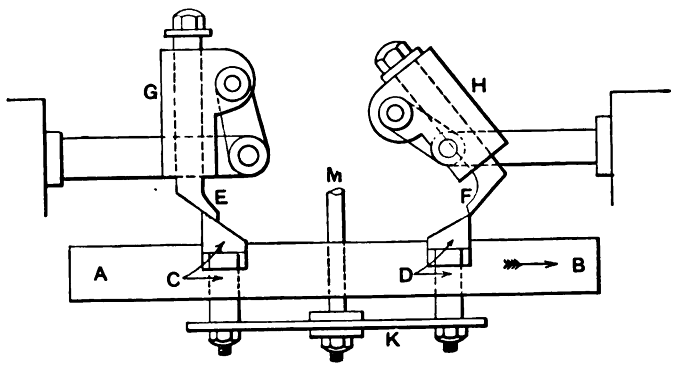


FIG. 185.—Greene Engine Valve-gear.

Hence it will disengage the tappet after a certain time and drop into its original position. On the return stroke of the block the operation will be reversed, F rising in its socket and E rotating. The cut-off is regulated by a fly-ball governor which raises or lowers the plate K to which the tappets are attached. The lower the plate is, the earlier the toes will disengage, thus making the cut-off earlier also.

HIGH-SPEED ENGINE-GOVERNORS.

By the use of Fig. 187 it may be shown that by making the angle of advance less and the travel greater, the cut-off occurs later.

Also by making the angle of advance greater and the travel less the cut-off occurs earlier.

This principle is applied in the construction of governors for automatic cut-off engines, an example of which is shown in Fig. 154.

The eccentric is not fast to the shaft but swings about it from a pivot *S*.

The eccentric proper is fastened to the diamond-shaped plate *ST*, which is held firmly in place by the spring *E*. *a* is the centre of the shaft and *h* is the centre of the eccentric disk.

Hence $ah =$ the eccentricity.

When the wheel increases its speed, the ball *C* moves towards the rim of the wheel by reason of centrifugal force.

This causes *H* and *T* to move to the left and with it the centre *h* of the eccentric.

That is, *h* moves towards *a*, thus making the eccentricity less and making the cut-off occur earlier, which will tend to cause the engine to run slower. When the engine begins to run slow the ball *C* moves toward the shaft and the cut-off is made later, thereby increasing the steam-supply and the speed of the engine.

This is but one of many forms of high-speed engine-governors. All or nearly all operate by shifting the eccentric. Those having but a single valve move the eccentric in a slot as above. Those having multiple valves, as the Buckeye engine, rotate the eccentric on the shaft.

CHAPTER XXI.

VALVE-DIAGRAMS.

THE object of a valve-diagram is to show the relations between steam-lap, exhaust-lap, lead, travel of the valve, distance from mid-position and port-opening, points of admission, cut-off, release, and compression at a glance.

There are several styles of valve-diagrams, all of which are more or less convenient, but the author prefers the Zeuner diagram on account of its simplicity.

In Fig. 186 let AA' be the stroke of the engine drawn to a

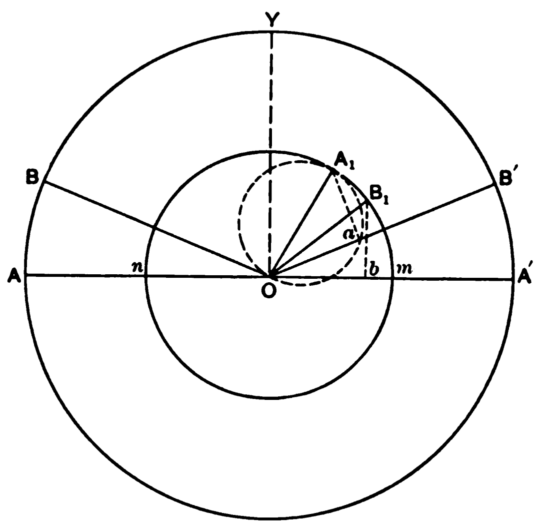


FIG. 186.

reduced scale and likewise mn the travel of the valve. Let YOA_1 be the angle of advance.

Then when the crank is at A the centre of the eccentric is at A_1. Suppose the crank moves to B. To find the position of the eccentric, since the crank and eccentric are both keyed to the same shaft, make the angle $A_1OB_1 = AOB$. B_1 is the position of the eccentric for crank position B. Drop a perpendicular from B_1 to AA', intersecting it at b, then Ob is the distance of the valve from mid-position and which was found in a somewhat laborious manner.

One of the principal objects of the valve-diagram is to shorten the process of finding the distance from mid-position.

The following explains the principle of the Zeuner diagram.

In the same figure draw an imaginary crank OB' moving at the same rate but in the opposite direction of the real crank B. To this line draw a perpendicular from A_1, the point on the line of the angle of advance intersecting it at a.

It is already known that Ob is the distance of the valve from mid-position and by similar triangles the line Oa may be proved equal to it, so that the distance from mid-position may be found by drawing this imaginary crank, and perpendicular to it a line from the point A_1 on the angle of advance. The distance from the centre of the valve circle to the intersection of these two lines being the required distance.

This principle may be made still more convenient by taking OA_1 as the diameter of a circle, which circle is shown dotted in the diagram.

This circle will pass through the point a (a circle on the hypothenuse of a right-angled triangle as a diameter passes through all three points of the triangle).

Hence drawing this circle we have the Zeuner diagram in its most simple form. In the practical use of the Zeuner diagram it is only necessary to draw the eccentric circle on the line of the angle of advance and the imaginary crank. Fig. 187 shows the complete diagram. YOA_1 is the angle of advance, OA_1 the eccentricity, D_1 the point of cut-off, C_1 the point of admission.

As was explained above, the distance from the centre to

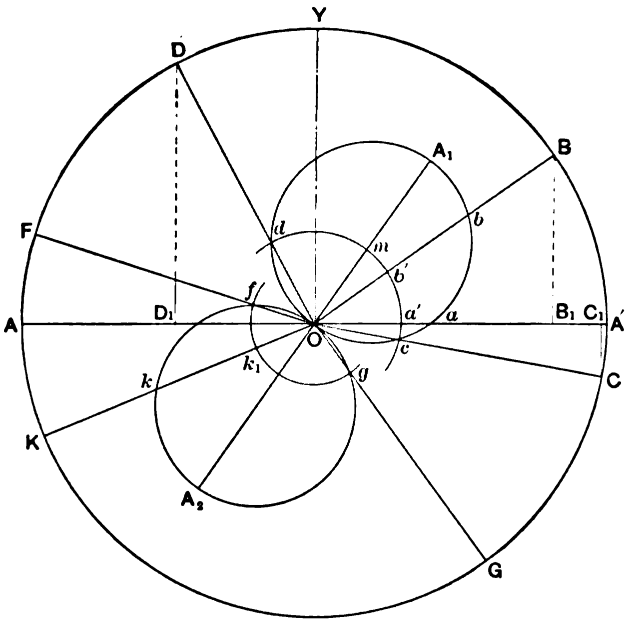


FIG. 187.—Zeuner Diagram.

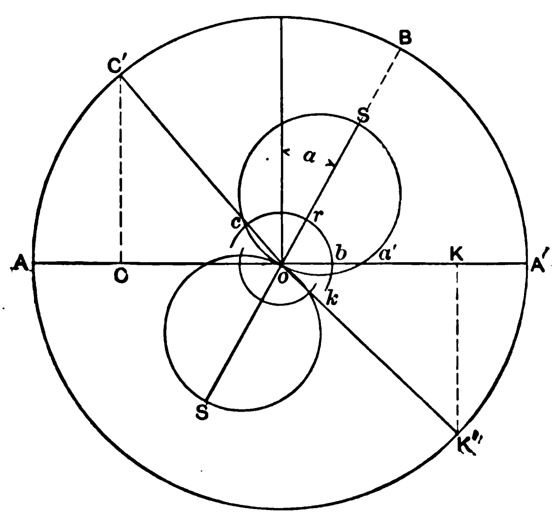


FIG. 188.

the intersection of the eccentric circle with the imaginary crank is the distance of the valve from mid-position as Ob, for the imaginary crank OB. The port-opening is equal to the distance from mid-position minus the steam-lap.

For convenience a steam-lap circle is drawn, as in the figure, for subtracting graphically.

That is, for any crank position the port-opening is equal to the distance cut off between the lap circle and the eccentric circle on the imaginary crank as $b'b$ in the figure. $a'a$ is the port-opening when the imaginary crank is at A', the end of the stroke, hence it must be the lead.

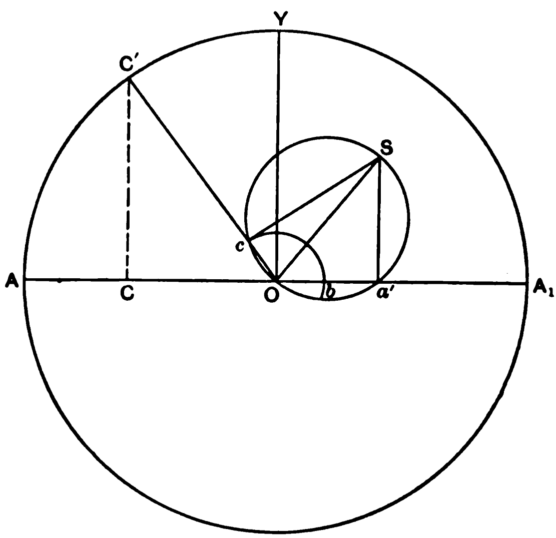


FIG. 189.

At the intersections C and C' the distance between the two circles or port-openings is zero.

The only two points with which the steam side of the valve is concerned which have no opening are admission and cut-off. Hence, drawing a line through O and the intersection c to C, and drawing the perpendicular CC_1, we find C_1 the point of admission.

In a like manner the points of cut-off, release, and compression may be found or *vice versa*, having given these points, the steam-lap, exhaust-lap, or eccentricity may be determined.

The angle DOC between the positions of cut-off and

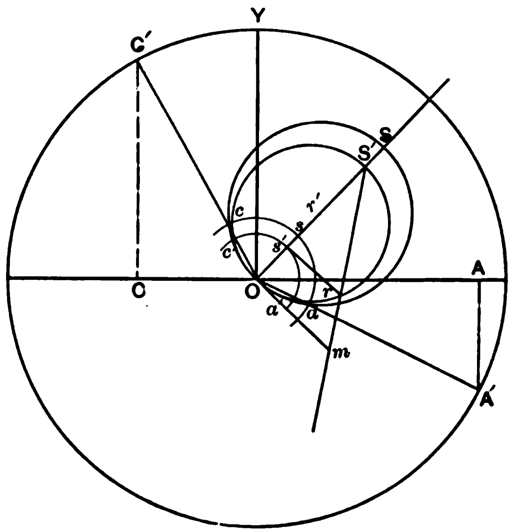


FIG. 190.

admission is bisected by the line of the angle of advance, that is, $DOA_1 = COA_1$. Likewise $FOA_2 = GOA_2$. Og is the exhaust-lap, F the crank position for release, and G the crank position for compression. C is called the crank position for admission.

PROBLEMS.

1. Given the eccentricity E, the angle of advance, the point of cut-off, and the point of closing of the exhaust. Find the steam-lap, exhaust-lap, lead, and the greatest possible opening of the port.

In Fig. 188 let AA' represent the stroke of the piston, OS the given eccentricity, α the angle of advance, $A'C$ the distance passed over by the piston before cut-off occurs, C' the crank position for cut-off, AK the distance passed over by the piston before the exhaust is closed or compression occurs.

On OS, the eccentricity as a diameter, describe the eccentric circle $OcSa'$. Oc is the radius of the steam-lap circle crb.

From what has been explained before, the port opening for any crank position is the distance along that crank line measured between the steam-lap circle and the eccentric circle.

$A'O$ is the crank position for the dead centre; hence ba', its port opening = lead.

At c the port opening becomes zero, which is known to be the case for cut-off. Hence Oc is the steam-lap.

When the imaginary crank moves to B, the port opening is rS.

This is the largest possible port opening, since it is measured along the diameter, and a diameter is greater than any of the chords of a circle.

Upon So extended describe the circle kOS intersecting OK', at k. Ok is the exhaust-lap.

2. Given the steam-lap, point of cut-off, and the lead to find the eccentricity, and the angle of advance.

In Fig. 189 let Ob be the given steam-lap, C the point of cut-off, $C'O$ the crank position for cut-off, and ba' the lead.

With a radius = to the steam-lap Ob, describe the steam-lap arc.

From preceding constructions it is known that the eccentric circle must pass through the points O, c and a'. To do this, draw cS perpendicular to OC' and $a'S$ perpendicular to AA_1 intersecting at S. OS is the diameter of the eccentric circle and YOS is the required angle of advance.

3. Given the point of cut-off, the point of admission, width of port, and the overtravel of the valve, to find the eccentricity, the lap, the lead, and the angle of advance.

In Fig. 190 let C and A be the points of cut-off and admission respectively.

The overtravel of the valve is the maximum port opening of the valve — the width of port or the greatest distance that the steam edge of the valve is beyond the inner edge of the port. Bisect the angle $C'OA'$ in the line OS, and YOS, will be the angle of advance, and the centre of the eccentric circle will be somewhere on OS. Take any point r' on this line as the centre of a trial-valve circle and describe the circle $Oc'S'a'$ intersecting OC' and OA' at c' and a'.

Draw the lap-circle $C''s'a'$, then $s'S'$ should = the given width of port plus the given overtravel if the trial-circle equals the actual circle. Suppose $s'S'$ is less than the width of the port plus the overtravel.

Draw any chord through S' as $S'r =$ to the width of the port plus the overtravel. Join rs' and through O draw Om parallel to rs'.

Then $S'm$ is the actual eccentricity.

With $\frac{S'm}{2}$ as a radius, describe the actual eccentric circle $OcSa$. Also draw the lap-circle csa.

Then OS is the eccentricity, Oc the steam-lap, and YOS the angle of advance.

4. In a valve-gear the angle of lead is 10°, cut-off at three quarters stroke, and the travel of the valve is 3 inches.

Find the angle of advance, the steam-lap, and the lead.

5. In a valve-gear the angle of lead is 12°, cut-off is at two thirds stroke and the lead is a quarter inch.

Find the angle of advance, the eccentricity, and the steam-lap.

NOTE.—The eccentricity is proportional to the lead, hence in the above problem find the angle of advance, then take any eccentricity and by construction find its lead under the conditions imposed by the problem.

Then $\frac{\text{given lead}}{\text{trial lead}} = \frac{\text{required eccentricity}}{\text{assumed eccentricity}}$ from which the actual or required eccentricity may be determined by solving. Then proceed as usual for the other unknown quantities.

6. In a valve-gear release occurs at 96 per cent of stroke, compression at two thirds of the return stroke, and the eccentricity is one and a half inches.

Find the angle of advance and the exhaust-lap.

7. Let the eccentricity in a valve-gear be two and a half inches, the angle of advance 30°, the point of cut-off at three quarters stroke, and the point of release at 96 per cent of the stroke.

Find the steam-lap, the exhaust-lap, and the lead.

8. The steam-lap in a valve-gear is one half inch, the exhaust-lap one quarter inch, cut-off occurs at two thirds of stroke, and the lead is $\frac{3}{16}$ inch.

Find the point of admission, the eccentricity, and the angle of advance.

9. Show by a valve-diagram what changes must be made in the eccentricity and the angle of advance in order to make cut-off occur earlier or later, the lead remaining constant.

NOTE.—It should be remembered that in a given valve-gear the steam-lap is a constant quantity.

10. Would it be practicable to make cut-off occur at one half stroke?

Demonstrate by use of a valve-diagram, noticing the size of the steam-lap and the port opening.

CHAPTER XXII.

ROTARY ENGINES AND STEAM-TURBINES.

To this type of engine belong all those engines having a rotary piston instead of the reciprocating piston, as in the ordinary engine.

The steam enters the cylinder directly from the boiler without passing through the intermediate steam-chest.

The area of the piston against which the steam presses is really an enlargement of the crank-pin. The piston must make a steam-tight fit with the circumference and the ends of the cylinder, and the steam must be controlled so as to exert its pressure on the surface of the piston and then be allowed to escape. There must therefore be the pressure of the steam on one side and the pressure of the atmosphere on the other side. The arrangement used to take the place of the valve in the ordinary engine for separating the steam and exhaust openings is usually called the abutment.

Every rotary engine must have an abutment and a piston.

Rotary engines may be divided into two classes,—those in which the abutment and piston interchange their functions and those in which the abutment remains abutment and the piston remains piston.

Fig. 191 is an example of the former class.

The steam from the boiler enters at *A* and the exhaust steam passes out through *B*.

In the position shown in the figure, *D* is acting as the piston and *C* is the abutment. The pressure on the two long ends of *C* having equal leverage on each end, there is no turning effect produced by *C*, hence it acts only to control the steam. However, the steam-pressure coming against the lower end of *D* rotates it in a direction opposite to the hands

THE ENGINE.

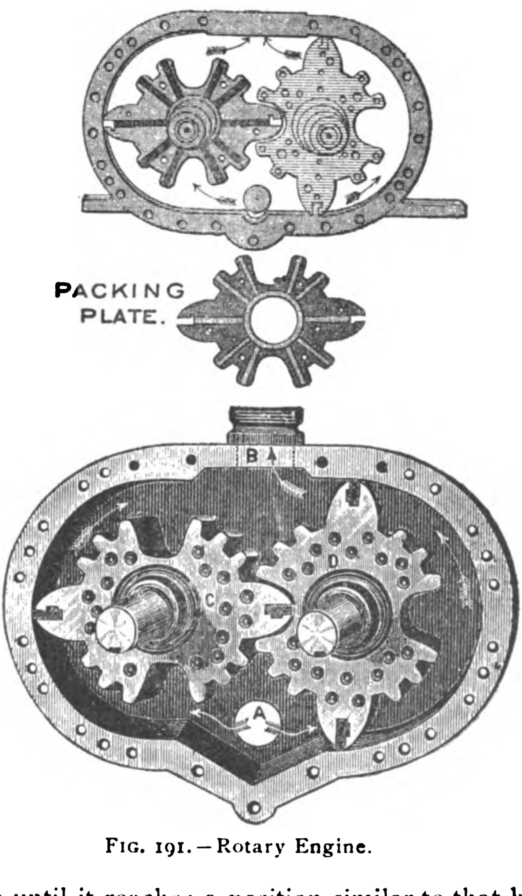


FIG. 191.—Rotary Engine.

of a watch until it reaches a position similar to that held by *C*, when it becomes abutment and *C* becomes piston. It is seen that the two wheels have projections in their perimeters gearing with each other. The engine has only one shaft, but by this gearing together the effort of both wheels is utilized. In this design the steam-joint between the piston and cylindrical part of the casing is made by the use of packing which is pressed out radially by the means of springs. The steam-joint between the piston and the flat sides of the casing is made by the use of radial packing-strips as shown in the figure.

In this design it is evident that there is no expansion of steam except from the boiler. It is also noticeable that there is to the right of D a large *clearance-space*, containing steam which exerts no effort whatever.

Fig. 192 illustrates the type of rotary engine in which the abutment and piston do not change their functions. C is the

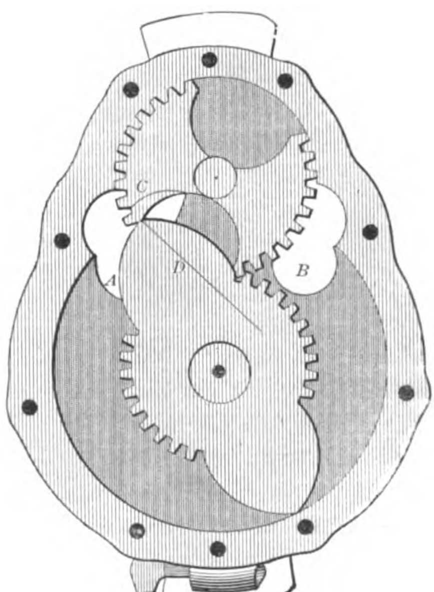


FIG. 192.

abutment and D is the piston. Steam here acts to produce motion in a direction contrary to the hands of a watch because there is less pressure to the right of the lower part than to the left.

The advantages of the rotary engine may be summed up as follows:

There are no reciprocating parts, and for that reason no resistance caused by the inertia of heavy mechanism. There is no dead-point, which is evident from either one of the figures shown.

Their construction being simple and without the necessity of converting reciprocating motion into rotary motion, it occupies small space.

There being no valve-gearing, the friction is small and the cost is light.

It requires very little skill to operate.

The disadvantages are as follows:

The revolving piston is hard to pack so that it will not leak, as the pressure changes for different parts of the revolution, causing a strain which soon wears away the packing.

As has been stated before, there is necessarily a large clearance-space requiring steam which exerts no rotative action. There is no expansion of steam. It is evident then for the last two reasons the rotary engine is very uneconomical in the use of steam.

The expansive action of steam which cannot be obtained with the single rotary engine is sometimes effected by placing two or more of them on the same shaft, the steam from the first one being exhausted into the next one of a larger diameter, and so on.

By this means economy may be practised to some degree.

Owing to the large waste of steam these engines are not used to a large extent. Rotary engines formerly had their largest application on steam fire-engines, driving rotary pumps; but even for this purpose they have been largely displaced by reciprocating engines, driving reciprocating pumps.

STEAM-TURBINES.

The steam-turbine is similar to the rotary engine in that it produces a direct rotary motion without any reciprocating parts.

It consists of a small turbine-wheel which runs by steam as the ordinary turbine does by water. It is interesting to note the fact that the first engine of which history gives us any knowledge was Hero's engine known as a "reaction-wheel" about 120 B.C. In 1629 Branca proposed one similar to it, the form of which is substantially appropriated in some of the modern makes of turbines. These two turbines (Hero's and Branca's) furnished the principal features of the two types of modern steam-turbines.

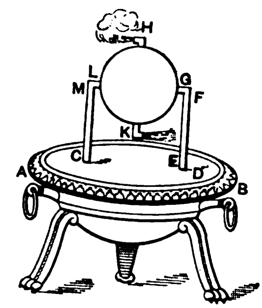


FIG. 193.—Hero's Engine.

The rotary speed of the modern steam-turbine exceeds greatly that of the ordinary reciprocating engine; 10,000 revolutions per minute is an ordinary speed. The efficiency of the

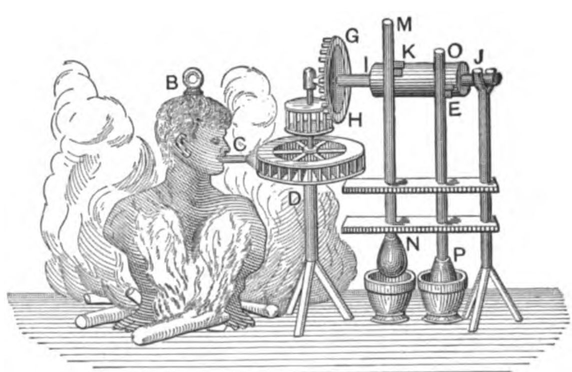


FIG. 194.—Branca's Engine.

steam-turbine is computed by Prof. Thurston to be about 87 per cent, without allowing for waste. The steam-turbine has the advantage of requiring very small floor-space as compared with the reciprocating engine; radiation of heat from the heated surfaces of the engine is small and the condensation losses caused by the constant changing of temperature in cylinders is here avoided. It is comparatively simple in principle and inexpensive in construction and by good designing may be made to attain great economy in the use of steam.

Fig. 193 represents Hero's engine, probably the first engine ever made. Its essential feature is a steam reaction-wheel. Steam is admitted into the sphere through the hollow axles or trunnions. As the steam escapes into the atmosphere the sphere revolves in a direction opposite to that of the flow of the steam from the nozzles, which motion is due to the reaction of the atmosphere.

Fig. 194 represents Branca's steam-engine, A.D. 1629. It consists of a wheel containing vanes upon its circumference, against which a jet of steam acts directly.

The principles contained in one or the other of these two constructions, principally the latter, have been used in the modern steam-turbines.

The steam-turbine has the advantages of the reciprocating engine in the fact that it has no reciprocating parts. Besides

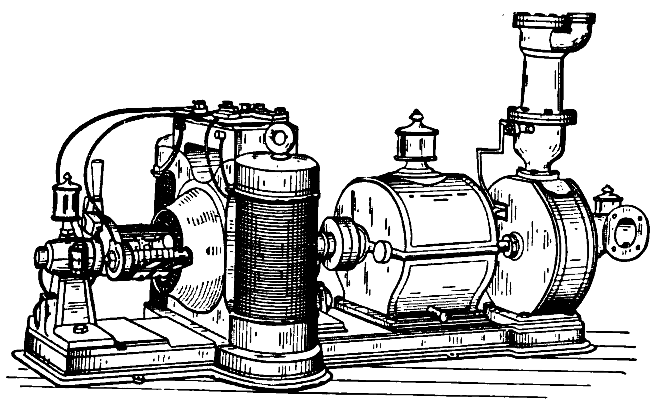


FIG. 195.—De Laval Steam-turbine Direct Connected to Dynamo.

this it has the greater advantage of being very economical in the use of steam. The direct-acting principle of the Branca type is shown in the De Laval turbine shown in Figs. 195 and 196, the former being a De Laval turbine connected to a dynamo. Fig. 196 shows the turbine-wheel and shaft together with the supply-tubes, partly in section in order to show the

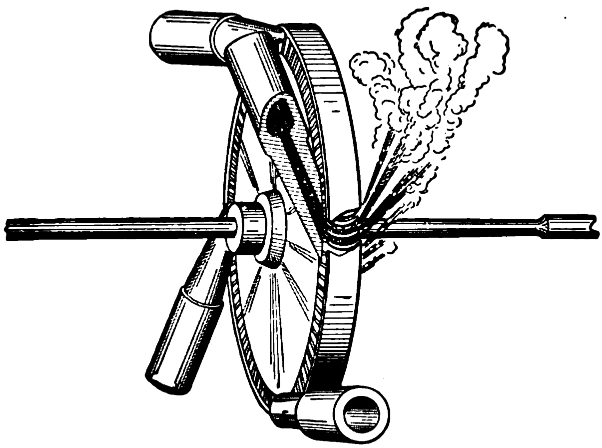


FIG. 196.—De Laval Turbine.

construction. The steam enters the wheel at one side and escapes from the other, the whole wheel being encased as shown in Fig. 195.

The reaction principle, a modification of that contained in

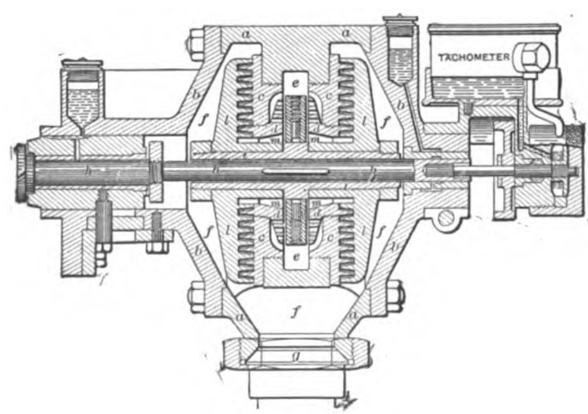


FIG. 197.—Dow Turbine.

Hero's engine, is shown in the Dow Turbine, Figs. 197 and 198. Fig. 197 is a section through the plane of the shaft-axis and Fig. 198 is a cross-section, the scale being twice that in Fig. 197. The action of the steam is best seen in the cross-section view of Fig. 198. The dark part represents opening and the light part represents metal (steel).

FIG. 198.—Runner and Guides of Dow Turbine.

Beginning with the inner series, every alternate one is a revolving turbine; the others are guide-plates.

The flow of steam is outward radially, entering at the middle in Fig. 198. The steam passes through the opening in the first inner turbine and strikes the adjacent guide-plate, and reacting causes the wheel to turn. The steam passes on toward the circumference, the guides causing it to strike the vanes at the proper angle. It is to be noticed that the openings become larger outwards, allowing the steam to expand according to its pressure. The drawing shows the method used for keeping the bearings cool by means of cold water.

The steam is continually expanded in the turbine without the condensation which is unavoidable in the reciprocating engine.

The steam-turbine has all the advantages of the rotary engine besides those enumerated above.

Capacity and Speed of Turbines.—The high speed of these turbines makes the horse-power developed large. It is not possible to use an indicator upon it, hence it becomes necessary to use the dynamometer for measuring the power developed. The efficiency of steam-turbines as far as steam-consumption is concerned is high. Tests have developed the fact that a consumption as low as 16 lbs. of steam per hour per H.P. is possible.

PROBLEMS.

1. The power developed by a turbine was measured by a Prony brake. The reading given by the scale was 50 lbs., the length of the arm was $1\frac{1}{4}$ feet, and the number of revolutions per minute 1000. Find the horse-power.

2. The exhaust steam of the turbine in the above test was condensed in a surface condenser and then weighed. The water used by the turbine during one hour was thus found to be 1375 lbs. Find the weight of water per hour per H.P.

APPENDIX TO PART II.

APPENDAGES TO ENGINES.

THE articles described in this chapter are used with engines in order to effect their best working. They may or may not be used; in fact, it will be found that many engines are run without some of them, at least.

Lubricator.—Owing to the pressure of the steam within an engine-cylinder, an ordinary oiler as applied to common bearings cannot be used to lubricate the piston- and slide-valve for the reason that the steam-pressure will not allow the oil to enter. For this purpose lubricators are used which force the

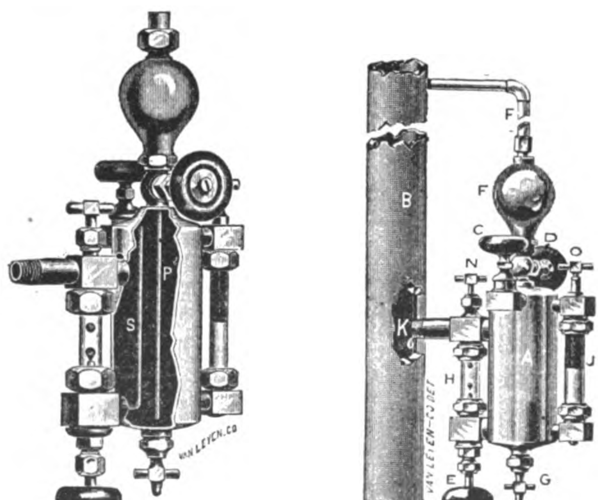


FIG. 199.—The "Detroit" Lubricator.

oil in by steam-pressure. They are usually connected to the steam-pipe just above the point where it enters the steam-chest. The illustration shows a sight-feed lubricator in section, and the manner of connecting it.

B is the steam-pipe just above the steam-chest. *A* is a chamber containing cylinder-oil. *F* is a pipe connecting the top of the lubricator to the steam-pipe above, usually about 3 feet long. *K* is another connection with the steam-pipe. *J* is a glass tube which shows the level of oil in *A*.

The operation of the apparatus is as follows: The chamber *F* provides for the condensation of steam which enters through the pipe *F*. This water of condensation passes down through the valve *D* and through the tube *P* and discharges into the bottom of the oil-chamber *A*. This chamber is filled with oil before starting. A drop of condensed water enters the bottom of *A* and displaces an equal bulk of oil, because of its greater specific gravity, and this drop of oil is forced out through *S*, past the regulating-valve *E*, making its appearance in the feed-glass *H*, through *K*, and then to the cylinder and valve.

Separator.—A considerable quantity of steam is condensed in the steam-pipe between a boiler and its engine.

Water is also carried over in the steam, due to the disturbances on the surface of the water in the boiler.

As water is incompressible, it is dangerous to an engine to pass water through the cylinder.

In order to catch this water which is in the steam and prevent it from entering the engine, a separator is attached to the steam-pipe near the engine.

The accompanying illustration shows, in section, a separator which is very simple in principle as well as construction. It is attached to the steam-pipe near the engine, so that the steam-passage is as indicated by the arrows.

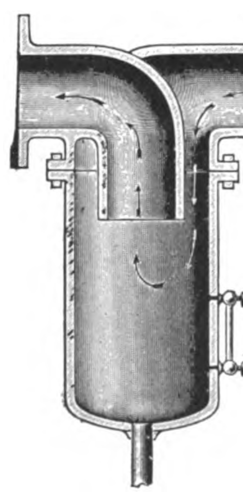


FIG. 200.
Steam-separator.

The water in the steam drops to the bottom, while the steam passes on and out. The water is drained off from the bottom.

Oil-separators. — Steam-plants which use condensed exhaust steam for feed-water require some arrangement for

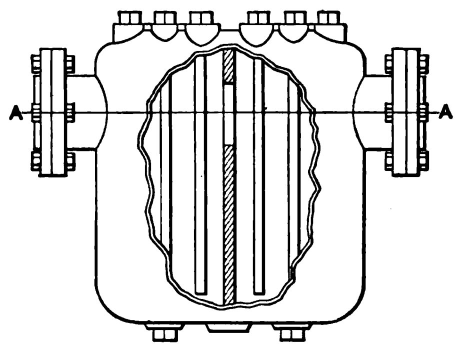


FIG. 201.—Oil-separator.

separating oil and water from the exhaust steam which has cylinder-oil in it. Oil is very prejudicial to steam-production, as it will combine with any dirt or mud in the boiler-water, to form a scale, which settles on the tubes and retards the passage of heat to the water. If this oil-scale becomes too thick it may cause the tubes or shell of the boiler to burn out.

An illustration of an oil-separator is here given.

PART III.

PUMPS, GAS-ENGINES, WATER-POWER, COMPRESSED AIR, ETC.

CHAPTER XXIII.

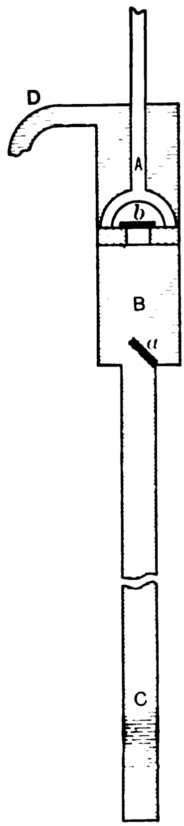
PUMPING-MACHINERY.

FORMERLY pumps were used only to raise water. At the present time, however, pumps are used for many other purposes; such as feeding water against steam-pressure into a boiler, furnishing water-pressure for hydraulic machines such as a hydraulic press, for irrigating, for forcing water or other fluids through long lines of piping for water-works supply, and others too numerous to mention.

A pump usually consists of a reciprocating plunger or piston working within a cylinder, but it may be a revolving wheel with buckets upon its periphery or something resembling buckets such as vanes producing an inductive action.

The reciprocating pump is of two kinds: The suction or atmospheric pump, and the force-pump. The principle of action of these two classes of pumps is shown by diagrams in Figs. 202 and 203.

The suction-pump theoretically can lift water through 34 feet, practically through about 28 feet; that is, the height of a column of water which the atmosphere will sustain. Therefore for wells deeper than 28 feet some other pump must be used. In Fig. 202, *C* represents a pipe from the water to the top of a well. The piston *A* has a valve in it which opens upward. At the bottom of the working-barrel B is another valve opening upward. When the piston is raised a is opened and b is closed, a volume of water being drawn into the pipe equal to the displacement of the piston. When A descends b



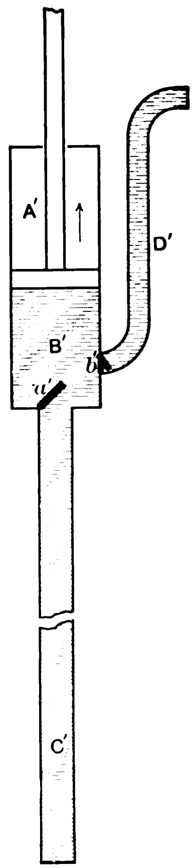


FIG. 202. FIG. 203.

is opened and a is closed. In the figure the piston is on its upward stroke. This is the form used with the old-style wooden pump.

Fig. 203 is a diagram showing the principle of the force-pump. In this case, a solid piston is used having no valve in it. The valves are shown at a' and b'. On the up stroke a' is opened and b' is closed. On the down stroke a' is closed and b' is opened. The water being forced out by the piston or plunger through the pipe D' to any height, depending upon the force which is applied to the piston. Here C' must not be over about 28 feet, or it will be impossible to raise the water

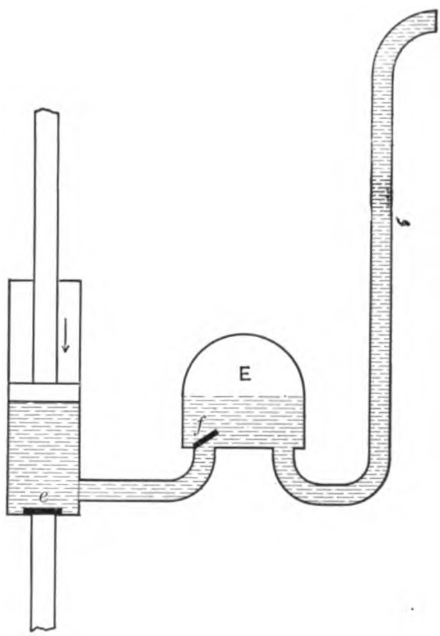


FIG. 204.

to the pump-cylinder by means of suction. This form requires a much stronger construction than the suction-pump, owing to the increased height through which the water may have to be raised. It is evident that the flow of water from the delivery-pipe D' will be in spurts because of the periodic movement of the piston. This may be avoided by the use of the *air-chamber* placed in the delivery-pipe which is practically an enlargement of the delivery-pipe. The diagram in Fig. 204 shows this arrangement, which is identical with that of Fig. 203, except for the addition of the air-chamber E. When water is forced into E the air is compressed so that after a few strokes the water begins to issue from G in a steady stream, being forced from the air-chamber by the compressed air, during the interval between the strokes of the pump.

These diagrams mentioned heretofore are all single action. Fig. 205 shows a double-action pump-piston. As has been stated before, suction-pumps are used only for lifting water or other fluids through small heights, usually for drawing water from shallow wells. For all cases in which the fluid is to be

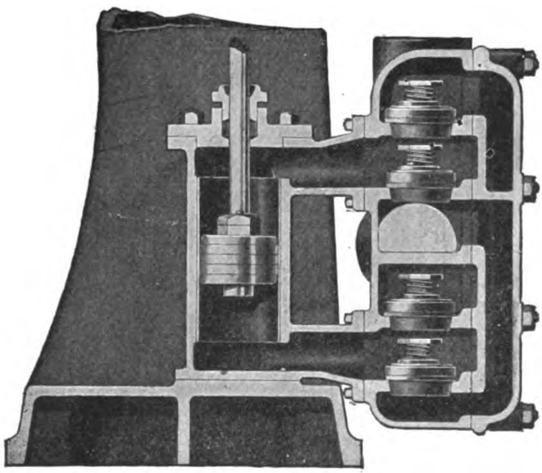


FIG. 205.—The Deane Double-acting Force-pump.

raised to great heights or against great pressures the force-pump is used.

The manner for giving motion to the water-piston is different for different pumps, but steam is the most common method of supplying the force, that is, a steam-engine is used. The following is a classification of modern pumps. First, according to the height of the pump above the water-supply: Suction- and force-pumps. Second, according to the means of giving motion to the water-piston:

$$\text{Steam-pumps} \begin{cases} \text{single.} \\ \text{compound.} \\ \text{duplex.} \end{cases}$$

Power-pumps $\left\{\begin{array}{l}\text{single.}\\ \text{duplex.}\\ \text{triplex.}\\ \text{centrifugal.}\end{array}\right\}$ Belted or Geared.

Electric pumps.
Hot-air pumps.
Gas-pumps.
Hydraulic pumps.
Windmill-pumps.

In the steam-pump one or more steam-cylinders are used containing a reciprocating piston, operated by a slide-valve. A single steam-pump is one having one steam-cylinder and one water-cylinder. A compound steam-pump is one having two steam-cylinders (as in a compound engine) driving a water-piston.

A *duplex pump* is one having two single pumps side by side on the same bed delivering into a common pipe and usually supplied by a common suction-pipe. In this construction the two steam-pistons usually move in opposite directions, so that the slide-valve for one steam-cylinder may get its motion from the piston-rod of the other by means of suitable connection, the D-valve always moving in a direction opposite to that of its piston.

Power-pumps are not connected direct to the source of the power but are connected to a revolving shaft by belts or gearing, this shaft receiving its motion from some motor separate from the pump. Power-pumps are single, duplex, or triplex, according to the number of water-pistons.

Electric pumps are those driven by electric motors. The high speed of the motor is reduced to the slow speed required by the water-piston by means of gear-wheels.

Gas-engines are sometimes the means of running pumps. Again as with electric pumps, the high speed of the gas-engine is reduced to the necessary slow speed of the pump by means of gear-wheels or belts.

A *hydraulic pump* is similar to the steam-pumps, with the exception that water at a great pressure is used as the motor-power instead of steam. In fact, some of the steam-pumps with slight alterations may be used as hydraulic pumps. This arrangement is practicable only where a large amount of water is at hand and at a high pressure, due to natural causes.

The pumps included in the above classification are not necessarily water-pumps. They may be used in pumping any liquid. Petroleum is an example of a fluid other than water which is pumped through many miles of piping by means of these various pumps.

Nearly all the pumps driven by steam or other power are force-pumps. The pumps referred to in the following pages will be force-pumps unless it is expressly stated otherwise.

Power-pumps are fitted either with pistons or plungers in

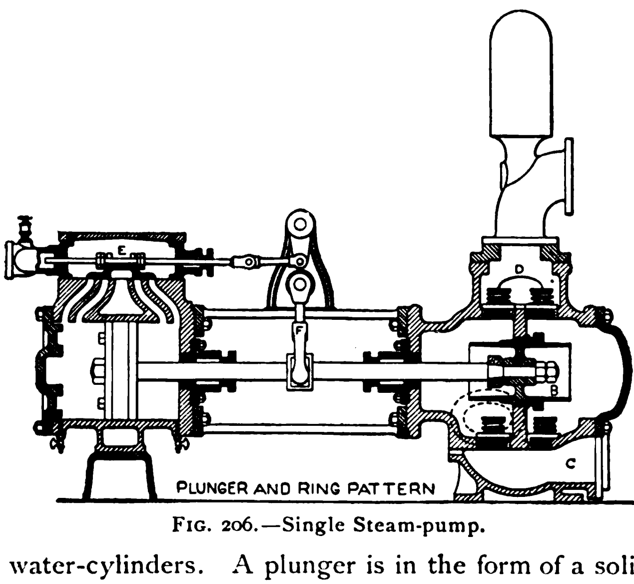


FIG. 206.—Single Steam-pump.

their water-cylinders. A plunger is in the form of a solid rod, suitably packed, while a piston is more in the shape of a disk. A plunger is shown in the pump in Fig. 207.

Fig. 206 shows a double-acting single pump as manufactured by the Worthington Hydraulic Works. The supply of water enters through C, and is delivered through D. The pump-piston is shown at B working in a bored ring. The piston is made somewhat longer than the stroke. The discharge-valves, near D, open upwards and consist of brass disks held down by springs. The supply- or suction-valves, near C, also open upwards. To show the operation of pumping, suppose the piston B makes a stroke toward the right. The left-hand suction-valve will be opened by the suction and the left-hand discharge-valve will be closed by means of the tension of its spring as well as the pressure of the water above it. Likewise the right-hand suction-valve will be closed and the force of the plunger-stroke will cause the right-hand discharge-valve to open and discharge an amount of water equal to the displacement of the piston. From this point it passes through the air-chamber and out through the discharge-pipe in a steady stream. The steam part of the pump is made plain by the drawing. The steam-piston is on the same piston-rod with the water-piston. Its motion is controlled by the D-valve E. The D-valve is connected by a valve-rod and small arms to an arm F which swings with the stroke of the piston. This arrangement takes the place of the eccentric in the steam-engine, necessarily by reason of the fact that there is no rotary motion of a shaft in pumps to which an eccentric could be attached. This type is designed for light service such as filling railroad-tanks, oil-tanks, etc. Fig. 207 shows the construction of a pump designed for heavy pressures such as are necessary for *hydraulic machinery*, as hydraulic cranes, cotton presses, hydraulic riveting- and punching-machines, and hydraulic presses of all kinds, where high pressures are required. It is of the duplex type and differs from the pump shown in Fig. 206 mainly in that it has water-plungers instead of pistons, the plungers entering the cylinder-head full size and being made of brass. There are four single-acting plungers, two of them shown in the cut, and the other two behind them, excluded from view in the drawing. The pressure which this

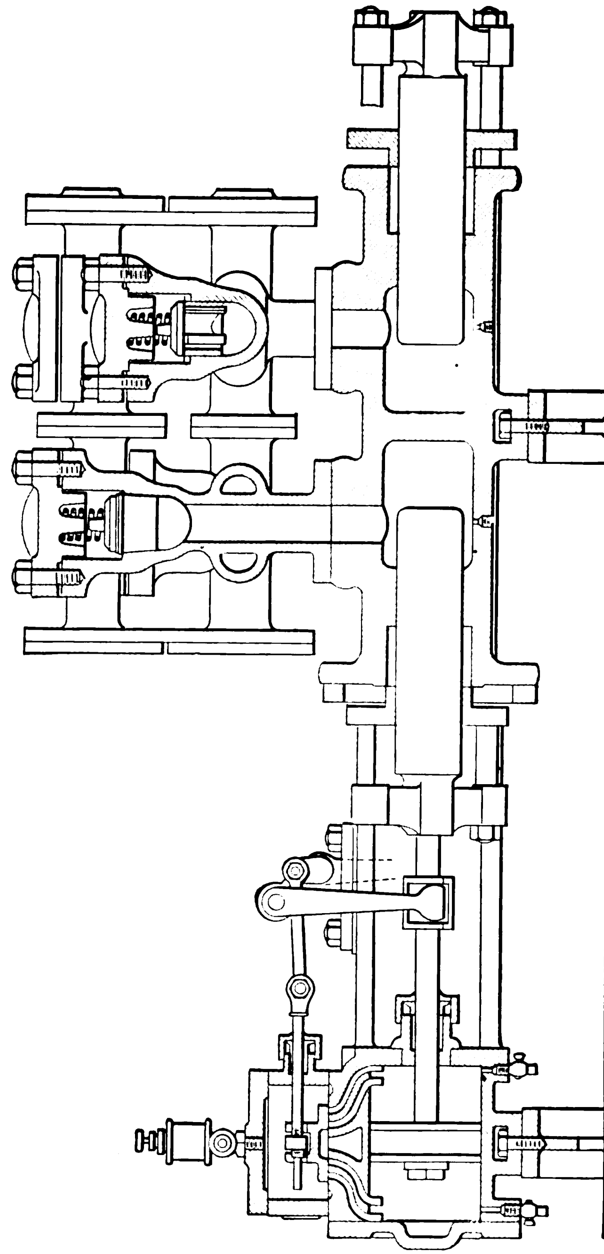


FIG. 207.—High-service Pump.

pump is capable of producing depends mainly upon its speed. The reason for making the plungers single-acting is that by dividing the work done between four plungers instead of two the strain on each piece is made less and hence the pieces may be made lighter. In this way large pressures are carried

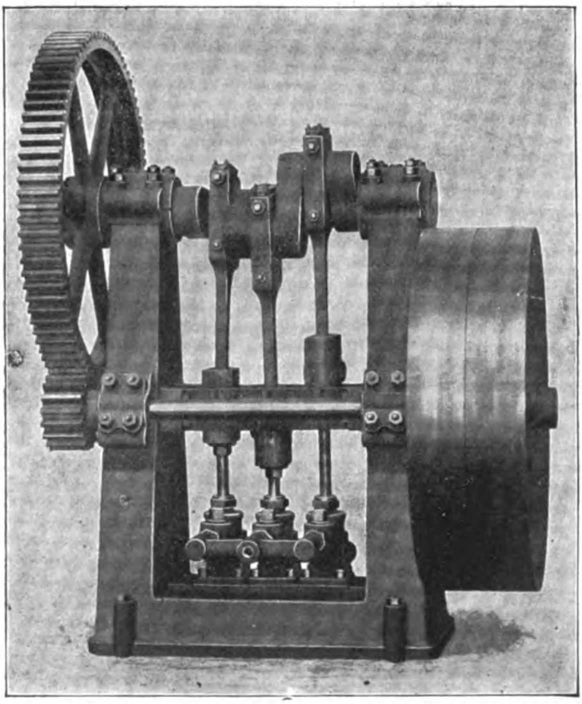


FIG. 208.—Power-pump.

by a number of small parts rather than by a few heavy ones.

It has just been stated that the water-pressure which is maintained by the pressure-pump just described depends largely upon the speed of the pump. This would naturally cause the pressure to be very uncertain as far as regularity is concerned. In hydraulic plants also where hydraulic machines draw off water-pressure sometimes at irregular intervals the pressure in the pump delivery-pipe is very irregular.

For the purpose of giving a uniform pressure the steam-

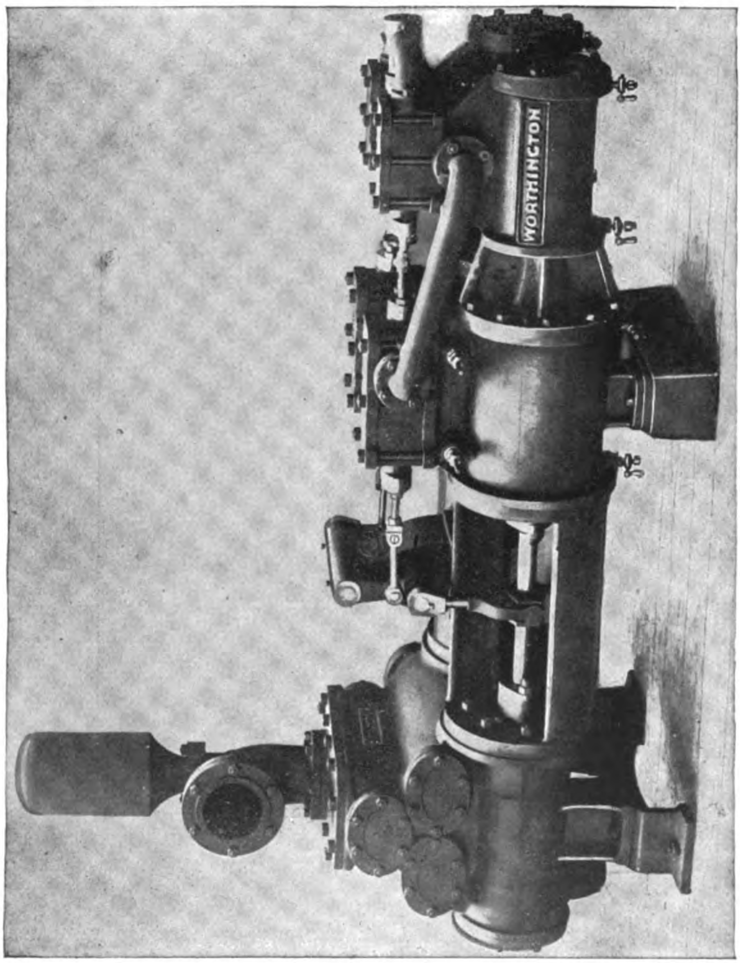


FIG. 209.—Compound Duplex Pump.

accumulator shown in Fig. 210 is used in connection with pressure-pumps.

It consists of a large steam-cylinder in which the piston *A* works, into which the steam from the boiler enters as shown.

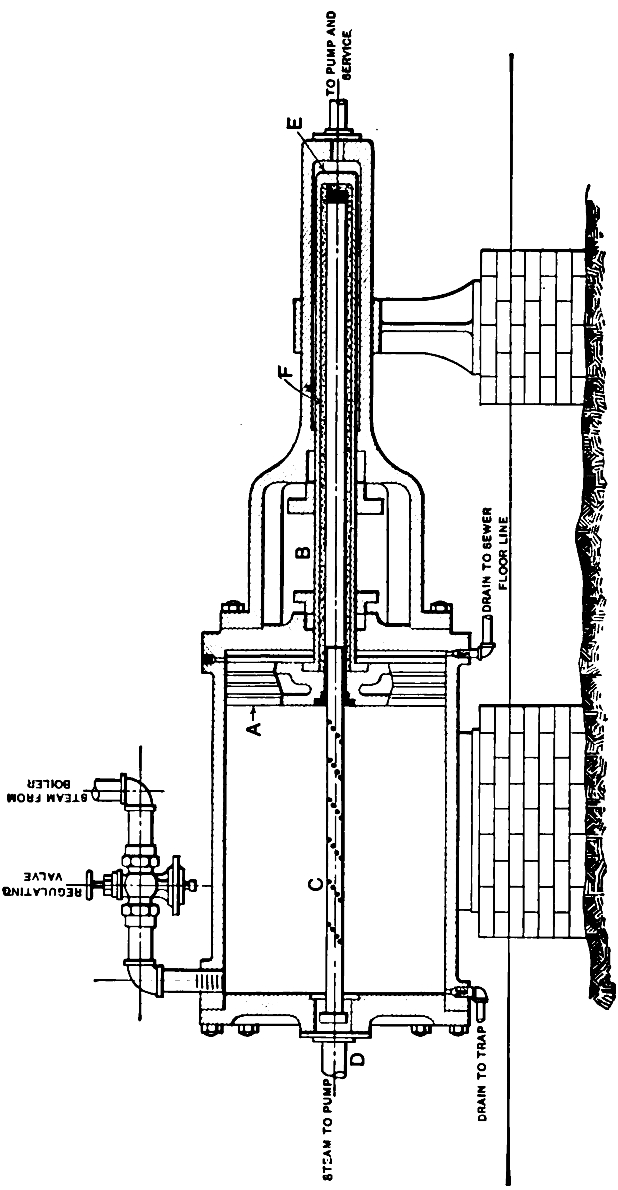


FIG. 210.—Accumulator.

The supply-steam for the pump, which is not shown in the cut, passes from the steam-cylinder of the accumulator through the perforations in the pipe C and out at D. The end E of the accumulator is in direct connection with the delivery-pipe of the pump. F is a ram which is bolted to the piston A and moves with it, passing through two stuffing-boxes, one in the steam cylinder-head and one in the ram cylinder-head. The perforated regulating-pipe C is stationary and enters through the piston A and into the ram. When the piston A moves to

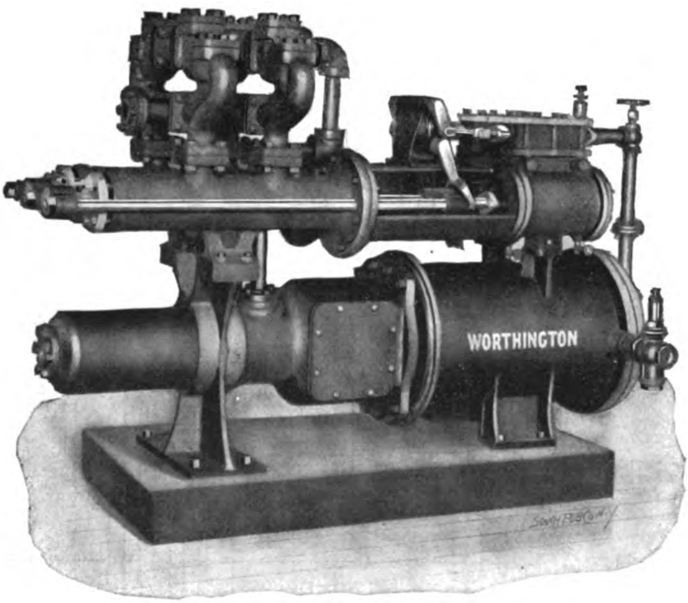


FIG. 211.—Accumulator and Pump.

the left it covers the holes in C. To explain the regulation of the steam-supply to the pump in order to meet the variations of consumption we will suppose that the pump has been running fast, causing the pressure in the delivery-pipe to rise; the high pressure of water at E causes the ram and steam-piston to move to the left. This causes a number of holes in C to be closed, thereby decreasing the quantity of steam admitted to the pump and slowing its speed. If the water-pressure is very great the piston *A* will be pushed far enough to cover all the perforations in *C* and thereby stop the pump entirely. When the pressure at *E* begins to fall, due to the slowing up of the pump or to the drawing off of water by some machine, the ram and piston move to the right, uncovering the perforations, so that steam is admitted to the pump, thus increasing the speed according to the requirements of the service. Fig. 211 shows an accumulator of this type built in connection with a pressure-pump.

Figs. 212 and 213 show a plan often used in deep water-wells, in which the water fails to flow to the surface of the ground. It consists of a vertical steam-engine placed directly over the top of the well and a single-acting vertical pump placed in the well at such a depth that the plunger is submerged or within suction distance of the water. The plunger is attached to the piston-rod by means of a rod made up of sections. In Fig. 212 *A* is the steam-cylinder containing a reciprocating piston, *B* is the steam-chest, *C* is the piston-rod, and *D* the valve-rod. *a b* is a rocker-arm pivoted at *c*. The end *b* is attached by means of a short link to the piston-rod at *e*. The end *a* is attached directly to the valve-rod. When the piston makes a stroke in one direction this arrangement causes the valve to move in an opposite direction; *E* is the discharge-pipe. The steam from the boiler enters at *F*. Fig. 213 shows an enlarged view of the water-plunger and the working-barrel. *AB* is the plunger to the top of which is attached the pump-rod. It works in a brass cylinder *D* which is fastened to the casing of the well and which is called the working-barrel. *E* is called the foot-valve. It is fastened to the bottom of the working-barrel. The valves in the foot-valve and the plunger consist of metal balls which are seated in corresponding openings in the seats. When the plunger makes an upward stroke, water is drawn through the foot-valve, the ball being raised. At the same time the valve in the plunger is closed, raising the water above it. On the

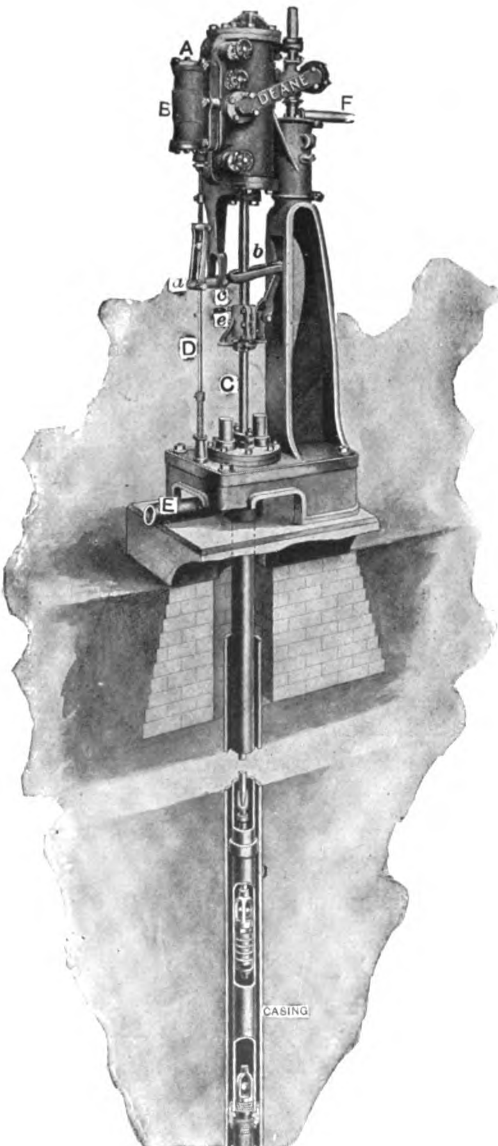


FIG. 212.

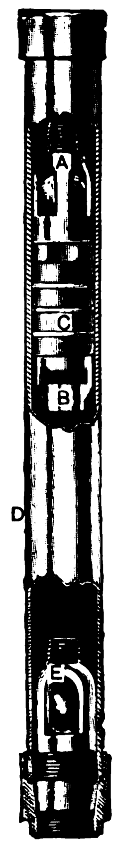


FIG. 213.

downward stroke the foot-valve is closed and the plunger-valve is opened. The discharge-pipe *E* may or may not pass through an air-chamber.

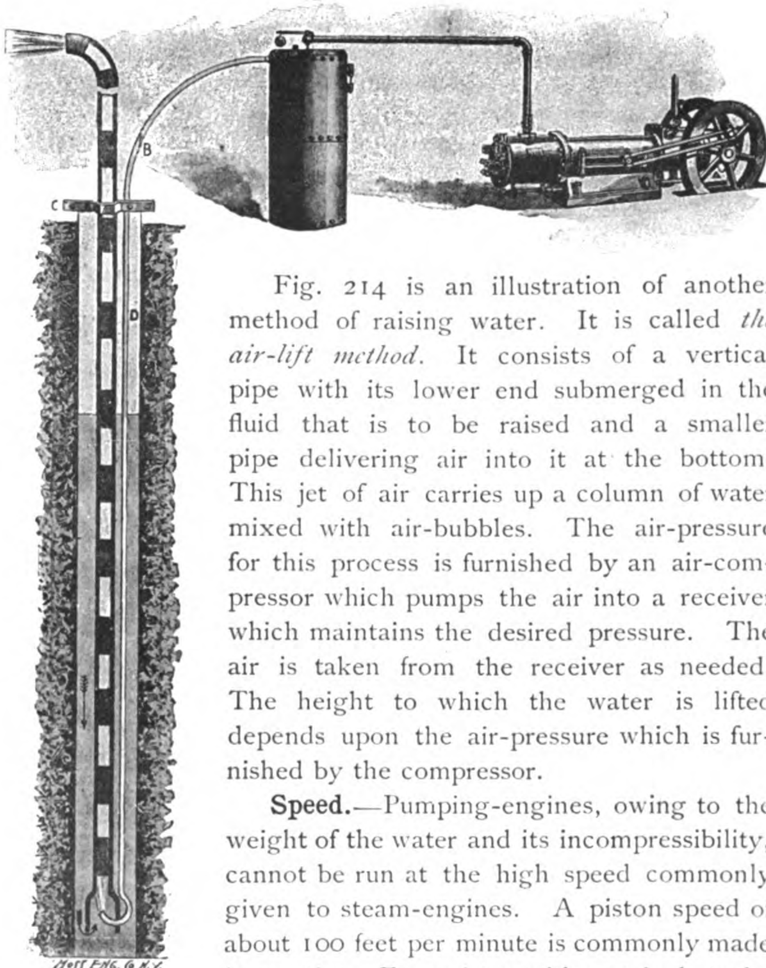


FIG. 214.—Air-lift.

Fig. 214 is an illustration of another method of raising water. It is called *the air-lift method*. It consists of a vertical pipe with its lower end submerged in the fluid that is to be raised and a smaller pipe delivering air into it at the bottom. This jet of air carries up a column of water mixed with air-bubbles. The air-pressure for this process is furnished by an air-compressor which pumps the air into a receiver which maintains the desired pressure. The air is taken from the receiver as needed. The height to which the water is lifted depends upon the air-pressure which is furnished by the compressor.

Speed.—Pumping-engines, owing to the weight of the water and its incompressibility, cannot be run at the high speed commonly given to steam-engines. A piston speed of about 100 feet per minute is commonly made in practice. For a piston with a 12-inch stroke this would give 50 double strokes per minute.

Area of Water-valves.—The water-valves of a pump should be made amply large so that the velocity of the water in passing through them will not be too great. They should have an area large enough to permit the passage of water through them at a rate of not over 250 feet per minute. The water-valve is usually made of a rubber disk or of composition brass which is held upon its seat by a coil-spring.

Water-piston.—Water-pistons are made tight usually by hemp or other fibrous packing. The fact that most water-pistons are constantly in contact with water also insures their proper working and lubrication. They are often made of brass in order better to stand the wearing and decomposing effect of different kinds of water.

Cylinders.—For the pressure carried in high-pressure pumps it is necessary to make the cylinder-walls comparatively thick. A brass lining is often used also, which is better adapted for contact with water than is iron.

Government.—Ordinary steam-pumps, of which Fig. 206 is a type, are governed usually by hand, by throttling the steam with the steam-valve at the entrance to the steam-chest. The speed is too slow to make an accurate centrifugal governor. Fig. 215 is a sectional cut of the Mason Pump-governor. It consists mainly of a cylindrical shell, or reservoir, filled with oil or glycerine. The plunger *AA* is connected through the arm *I* to some reciprocating part of the pump or engine, and works in unison with the strokes of the pump, thereby drawing the oil up through the check-valves *DD* into the chambers *JJ*, whence it is forced alternately through the passages *BB*, through another set of check-valves into the pressure-chamber *EE*. The oil then returns through the orifice *C*, the

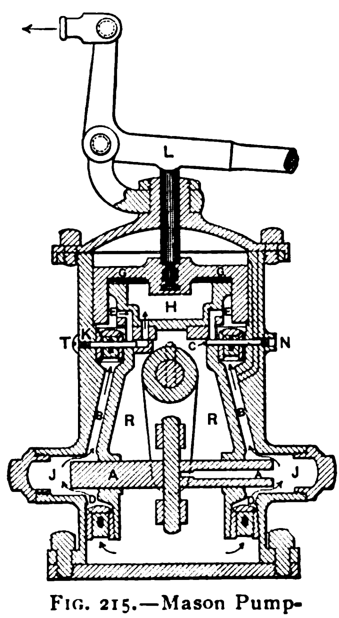


FIG. 215.—Mason Pump-governor.

size of which is controlled by a key inserted at N, into the lower chamber, to be repumped as before. In case the pump or engine works more rapidly than is intended, the oil is pumped into the chamber EE faster than it can escape through the outlet at C, and the piston GG is forced upward, raising the lever L with its weight and throttling the steam. In case the pump runs slower than is intended, the reverse action takes place; the weight on the end of the lever L forces the piston GG down and more steam is let on. As the orifice at C can be increased or diminished by adjusting the screw at N, the governor can be set to maintain any desired speed. The piston GG fits over the stationary piston, forming an oil dash-pot, thereby preventing fluctuation of the governor. This dash-pot is fed from pressure-chamber E through a passage which is controlled by an adjusting-screw T, which is set by a screw-driver (after removing the cap-screw K). It requires no further attention after once adjusted.

Water-pressures.—A pressure-gauge similar to that used for steam-pressures is used to show the pressure of water. Fig. 216 shows a combination water-pressure gauge, having two sets of graduations, one showing the pressure per square inch; the other showing the height of water in feet. This gauge may be placed in any pipe having water-pressure or on a tank, stand-pipe, or reservoir.

Capacity.—A water-lifting arrangement is usually rated according to the number of gallons it will lift in unit time, and not by horse-power, though of course the horse-power could easily be calculated. To find the number of gallons pumped per hour multiply the displacement of the piston per stroke in cubic feet by the number of strokes per hour, and by 7.48, the number of gallons in a cubic foot. Let S be the stroke in inches, A the area of the piston in square inches, and N the number of strokes per minute. Then the capacity in gallons per minute is

$$\frac{SAN}{1728} \times 7.48.$$

This is the theoretical capacity made upon the assumption that at each stroke a quantity of water is pumped equal to the displacement of the piston or plunger, and allowing nothing for leakage. The length of the stroke varies constantly, also,

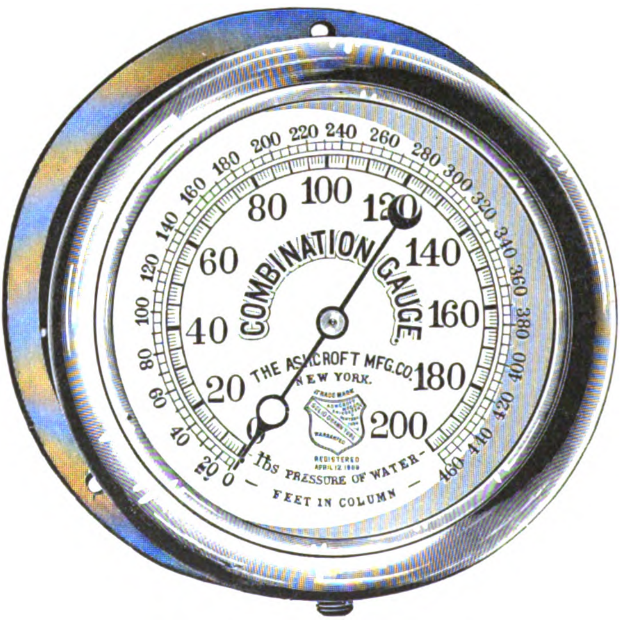


FIG. 216.—Pressure-gauge.

which would vary the quantity of water pumped in a given time.

Meter.—The actual volume of water pumped may be measured by a water-meter, shown in Fig. 217. It is placed so that the water discharged by the pump passes through it. It is used in any place where water-supply or consumption is to be measured. The internal arrangement of the Worthington meter is shown in longitudinal section in Fig. 217 and in transverse section in Fig. 218. The plungers *AA* are closely fitted in parallel rings. The water passes through the inlet and port *I*, and is admitted under pressure into the chamber *D*, at one end of each plunger alternately, while the connection is made between the chamber at the other end of the outlet. Thus the plunger in moving displaces its volume, discharging it through its outlet. The arrangement is such that the stroke

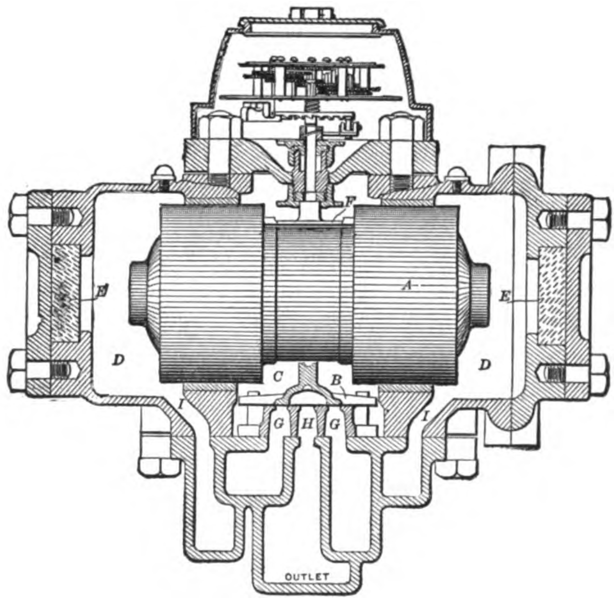


FIG. 217.—Water-meter.

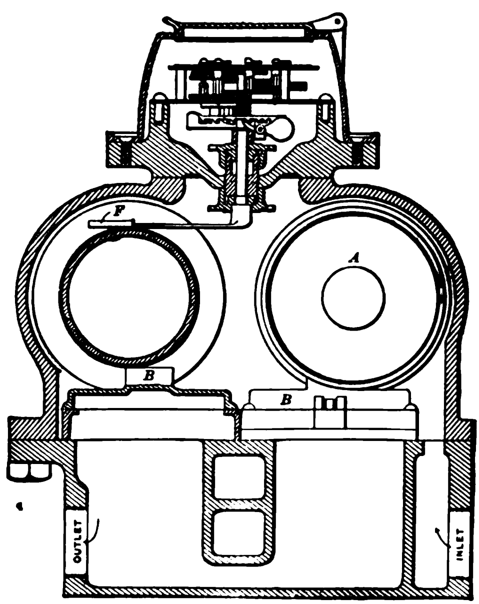


FIG. 218.

of the plungers alternate, the valve actuated by one admitting water to the other. The plungers are brought to rest at the end of the stroke by the rubber buffets *EE*. One plunger imparts a reciprocating motion to the lever *F*, which operates the counter mechanism through the spindle- and ratchet-gear as shown. Thus it will be seen that the counter is arranged to move the dial-pointers once for every four strokes or displacements, and that water cannot pass through the meter without registration, for, in order to pass through, it must displace the plungers, and, therefore, be recorded by the movement of the lever and counter mechanism; nor can there be an over-registration, because the plungers cannot move unless the fluid is displaced.

Duty.—The duty of a pumping-engine means the number of foot-pounds of work done by the pump for every 100 lbs. of coal burnt in the boiler-furnace. It is practically the same as "efficiency."

To find the "duty" as defined above:

Let P be pressure of water in pounds per square foot in the supply-pipe of the pump just before entering the cylinder, determined by dividing the pressure-gauge reading by 144;

P_1 be the pressure of water in pounds per square foot in the discharge-pipe as the water leaves the cylinder, ascertained as before;

h = difference of level between the above two gauges in feet;

W = weight of steam used per hour in pounds;

w = weight of coal burned per hour in pounds;

Q = number of cubic feet of water pumped per hour.

The head of water h may be reduced to pounds per square foot by multiplying by 62.4, the weight of a cubic foot of water.

Then the total pressure of water per square foot upon the water-piston =

$$P_1 - P + (h \times 62.4);$$

$P_1 - P$ representing the pressure required to overcome the friction and the resistance due to the head in the pump, and $h \times 62.4$ a small pressure which is due to the difference of level of the gauges and which has to be added; then

$$\{P_1 - P + (h \times 62.4)\}Q$$

is the number of foot-pounds of work done per hour. Dividing by w we have

$$\{P_1 - P + (h \times 62.4)\}\frac{Q}{w}$$

as the number of foot-pounds done for every pound of coal burned.

The "duty" is 100 times this, by the definition of duty =

$$\{P_1 - P + (h \times 62.4)\}\frac{100 \times Q}{w}.$$

NOTE.—$Q = NLA$, in which N = number of strokes per hour, L = length of stroke in feet, and A = area of piston in square feet. Hence, multiplying by Q is equivalent to multiplying the total pressure on the piston by the distance moved by that pressure per hour.

A high-duty pumping-engine is one which gives an exceedingly high number of foot-pounds per every 100 lbs. of fuel burned. This is attained by using the steam expansively. That is, by a compound or triple-expansion steam-end which of course gives a higher efficiency than with a single steam-cylinder.

The Indicator.

The pressure within the water-cylinder of a pump may be obtained as with the steam-cylinder with the Indicator. Figs. 219 and 220 are facsimiles of cards taken from a small pump of the type shown in Fig. 206. Fig. 219 was taken from the water-cylinder and Fig. 220 from the steam-cylinder. The difference in the height of the two cards is due to the fact that the scale of the indicator-spring on the steam-end was No. 36, while that of the water-end was No. 48.

By reference to the steam-card it will be noticed that the steam was working non-expansively, that is, steam entered the cylinder during the whole stroke. By working up these

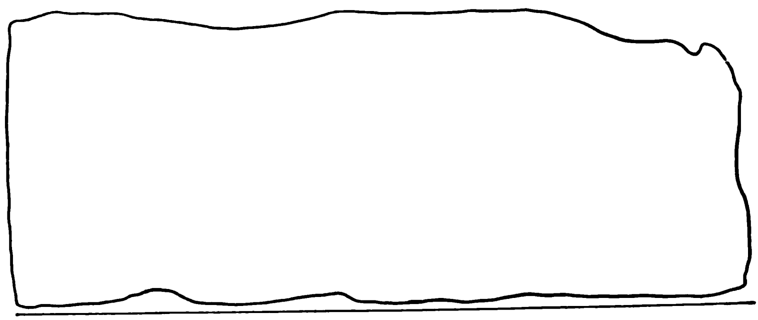


FIG. 219.—Indicator-card from Water-cylinder of Pump.

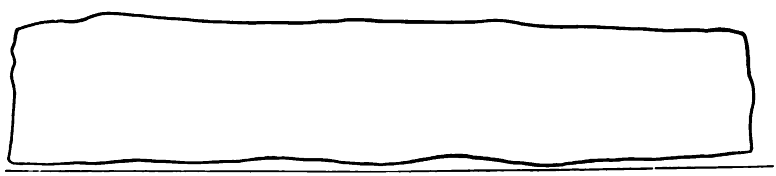


FIG. 220.—Indicator-card from Steam-cylinder of Pump.

two cards it will be found that less work was done on the water-piston than on the steam-piston. This difference represents the work lost in friction.

PROBLEMS.

1. A pipe whose cross-section is one square foot extends vertically 100 feet into the air. What is the total pressure on the bottom of the pipe? What is the pressure per square inch?

Note.—The weight of a cubic foot of water is 62.4 lbs.

2. Find the pressure per square inch on the bottom of a stand-pipe having the same height as the above, but having a cross-section of 20 square feet. Compare results.

3. At the sea-level the pressure of the air is 14.7 lbs. It will also support a column of mercury 29.92 inches high or a column of water 34 feet high. Find pressure per square inch due to a column of mercury 1 inch high and a column of water 1 foot high.

4. The plunger of a pump makes 100 strokes of 24 inches per minute, and is 6 inches in diameter. Find the number of gallons pumped per minute and per hour.

5. Find the number of foot-pounds of work done per hour in the above if the water is pumped into a tank 50 feet above the water source.

6. A pump discharges 100 cubic feet of water per minute. What should be the area of the two discharge-valves through which the water passes in order that the velocity of the water through them shall not be more than 250 feet per minute?

7. Sixty cubic feet of water is pumped per hour, during which time 15 lbs. of coal is burned in the furnaces for it. The pressure in the discharge-pipe as shown by the pressure-gauge is 90 lbs., while the pressure in the suction-pipe is zero, and the two gauges are on the same level. Find the duty of the engine.

8. Work up the card shown in Fig. 220, and find the indicated H.P. of the steam-cylinder, the stroke being $4\frac{1}{2}$ inches, the diameter of cylinder 5 inches, and making 40 strokes per minute.

9. Find indicated H.P. of water-cylinder, stroke $4\frac{1}{2}$ inches, diameter of cylinder $3\frac{5}{16}$, and 40 strokes per minute.

10. Find indicated H.P. lost in friction.

11. Determine the efficiency of the pump.

CHAPTER XXIV.

GAS-ENGINES.

THE Gas-engine differs from the steam-engine in that the whole process of transformation of the heat-energy of the fuel into mechanical work is carried out within the engine itself. Gas is introduced into a cylinder, containing a piston; it is compressed and then ignited or exploded, and the expansion of the gas due to its burning gives the piston a forward impulse.

In the steam-boiler furnace it is necessary to give air to the fuel in order to furnish enough oxygen to support combustion. The same is true of combustion in a gas-engine. A certain quantity of air is mixed with the gas before it is ignited in the cylinder. If the chemical constitution of a gas is known, the volume of oxygen necessary for making the proper explosive mixture can be calculated.

Pressures and Temperatures of Exploding Gas.—With a mass of any perfect gas confined within a closed vessel the absolute temperatures and pressures are proportional to each other, according to the law of Charles.

The following figures are the results of experiments by Dugald Clerk with different mixtures of coal-gas and air.

Mixture.		Maximum Pressure above Atmosphere. Pounds per Square Inch.	Temperature of Explosion calculated from observed Pressure.	Theoretical Temperature of Explosion if all Heat were evolved.
Gas.	Air.			
1 vol.	14 vols.	40	1483.8° F.	3237.8° F.
1 "	13 "	51.5	1892.4	3473.6
1 "	12 "	60	2196.6	3736.4
1 "	11 "	61	2228	4042.4
1 "	9 "	78	2835.6	4838
1 "	7 "	87	3151.4	6033.2
1 "	6 "	90	3257.6	6931.4
1 "	5 "	91	3293.6	
1 "	4 "	80	2903	

The temperature before explosion was 30.6° F.

The temperatures in the fourth column are derived from the law of Charles: the volume of a perfect gas at a constant pressure is proportional to the absolute temperature, or if the volume is constant the pressure is proportional to the absolute temperature and the absolute temperature is proportional to the pressure, or

$$t_1 = \frac{p_1}{p_0} t_0,$$

in which t_1 is the absolute temperature corresponding to the pressure p_1, p_0 is atmospheric pressure, and $t_0 = 491°$, the absolute temperature corresponding to 32° F. The figures in the fifth column are those that would be obtained theoretically if all the gas were perfectly burned, the volume remaining constant, and there being no loss of heat by conduction into the walls of the cylinder. The fact that the temperatures due to the observed pressures are so much lower than the theoretical temperatures indicates that the combustion is not perfect at the time of the explosion, when the maximum pressure is observed.

From the above table we find that coal-gas will give temperatures of explosion of from 1480° F. to 3300° F., depending upon the dilution of the mixture. It is also seen that a mixture of 1 vol. of gas and 4 vols. of air gives a lower pressure than a mixture of 1 vol. of gas and 6 vols. of air. Gas and air in the proportion of 1 to 5 gives about the maximum pressure when coal-gas is used.

CLASSIFICATION OF GAS-ENGINES.

Modern gas-engines may be divided into two great classes, viz., those in which the piston receives an impulse due to the explosion of gas once for every four strokes, that is, for every two revolutions, and those in which the piston receives an impulse for each revolution of the crank-shaft.

The former is called the *four-cycle* or Otto type; Otto having been the first to make practical engines of the four-cycle type.

The latter is called the *two-cycle* type. It is now rarely used.

Most gas-engines are single-acting.

In all gas-engine practice it has been found that the highest efficiency is attained by compressing the gas in the cylinder before igniting it. In some early engines two cylinders were used, one of which was used for compressing the gas, which was then introduced into a power-cylinder and suddenly ignited, the resulting explosion driving the piston forward.

The Otto type uses the same piston for compression and for power. Its operation is as follows: During the first outward stroke the cylinder is charged with gas and air; on the first inward stroke this mixture is compressed. At the beginning of the second outward stroke, the compressed mixture is ignited and the piston is driven forward. During the second inward stroke the burned gases are exhausted into the atmosphere.

Fig. 221 shows a sectional plan of the original engine invented by Otto. *A* is the cylinder which is closed at one end only, doing away with the stuffing-box of the ordinary steam-engine. *B* is the piston connected directly to the crank by a connecting-rod. This does away with the piston-rod and cross-head as used on steam-engines. *C* is the compression-space which is not traversed by the piston. *I* is the admission- and ignition-port, communicating alternately with the gas and air admission-port *K*, and the flame-port *L* in the slide *M*. *N* is the covering holding the slide to the cylinder-face and carrying in it an external flame for lighting the movable one in the flame-port *L*. The exhaust-valve is seen at *O*.

P is the cam-shaft driven by the use of bevel-gears from the crank-shaft, which operates the admission, the exhaust, and the igniting apparatus. To start the engine the flame at *T* is lighted; the cock commanding the internal flame being

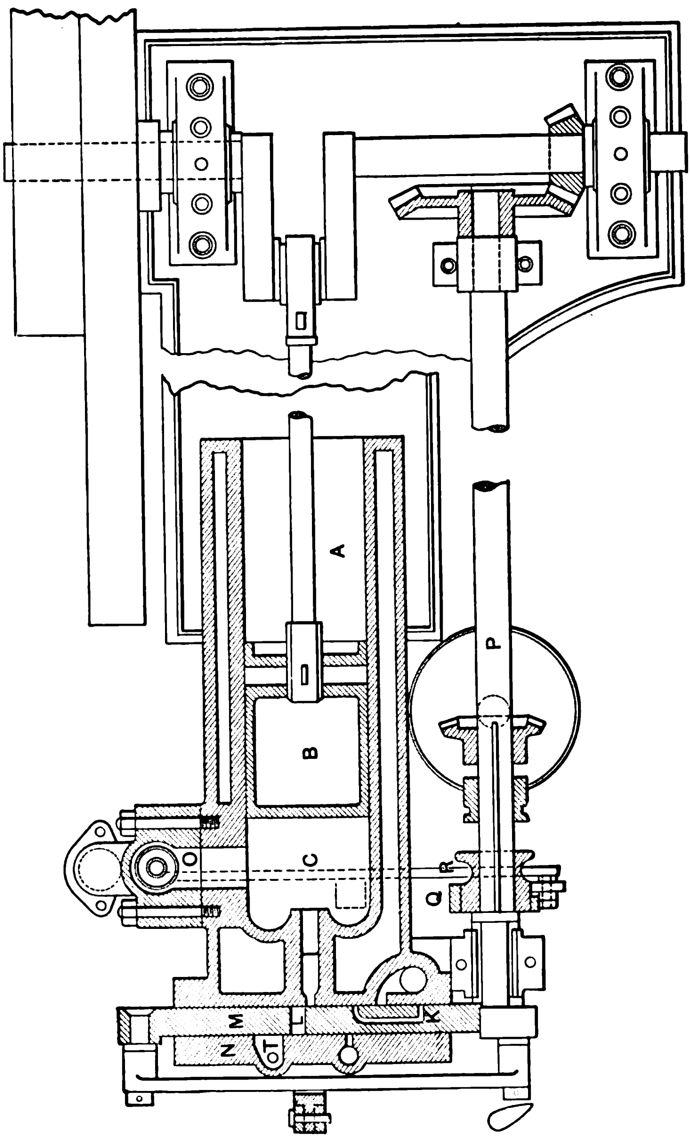


Fig. 221.—Otto Engine.

adjusted, and the gas turned on, a couple of turns at the fly-wheel, by hand, causes ignition and sets the engine in motion.

The regulation of the speed of the engine is made by a centrifugal governor which closes the gas-supply when the engine speed is increased. This causes an explosion to be missed, air only being drawn into the cylinder. When running without load, eight or more revolutions may be made per explosion.

Heavy Construction.

Owing to the fact that only one impulse is given for every two revolutions, it becomes necessary with engines of the Otto type to make all the parts very strong in order to withstand the increased shock per stroke.

The fly-wheel is also made very large in order to carry the engines over the compression stroke, during which no impulse is made.

The objections to the four-cycle just named, that is, the heavy construction necessary, have caused a great deal of study and investigation by gas-engine builders in producing engines making an impulse for every revolution.

The Day gas-engine shown in vertical section in Fig. 222 may be taken as an example of this type.

B is the piston, *C* is the connecting-rod, *D* the crank-pin.

The crank-shaft operates in the closed chamber *E*, which chamber serves as a reservoir for gas and air mixture.

F is the charge inlet-port which admits the charge of gas and air to the cylinder. *G* is the exhaust-port, allowing the discharge of the burned gases.

The action of the engine is as follows: On the up stroke of the piston *B* the pressure of the gases in the chamber *E* is reduced to a little less than atmospheric pressure. When the piston reaches the end of its up stroke, the air inlet-port *H* is uncovered by the lower edge of the piston and air rushes into the chamber *E*, bringing the pressure up to that of the atmosphere.

Gas is also admitted at the same time by means of a valve controlled by the governor, so that *E* is filled with a mixture of gas and air.

On the down stroke of the piston *B* this mixture of gas and air is compressed to a few pounds above atmospheric pressure

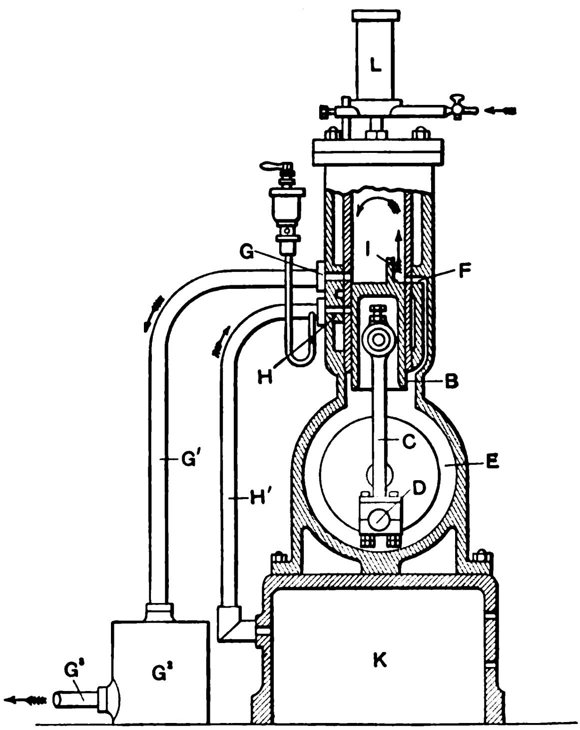


FIG. 222.—Day Engine.

and at the termination of the stroke the port *F* is uncovered by the piston, causing the mixture in *E* to flow into the top part of the cylinder, striking against the baffle-plate *I*, which causes it to flow upward and then downward, as shown by the arrow, expelling the burned gases of the previous stroke.

When the piston *B* makes its next upward stroke, it compresses this mixture into a small space at the end of the cylinder to a pressure of about 50 lbs. above atmospheric pressure.

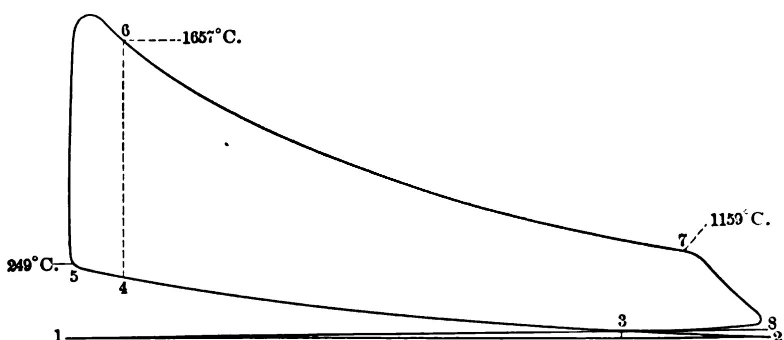
At this point the hot-tube L ignites the compressed mixture and the piston makes a downward stroke due to the impulse of the explosion. By this arrangement an impulse is made for each revolution.

The disagreeable noise of the exhaust-discharge is averted by conducting it first to an exhaust-chamber G_2 and then to the atmosphere by the pipe G_3.

Owing to the very high temperature of exploding gases it is necessary to circulate water around the cylinder, because the cylinder-walls would be so highly heated that the entering gas would be ignited without compression.

Indicator-cards.—The pressure in a gas-engine cylinder may be indicated with an indicator similar to the steam-engine indicator, but a very strong spring must be used in order to reduce the effects of inertia due to the shock of the explosion.

Fig. 223 shows a normal diagram of the work in a gas-engine of the Otto or four-cycle type. The line 1–2 is made



8½ INS. DIA. CYLINDER. 14 INS. STROKE. 160 REVS. PER MIN.

FIG. 223.

while the piston draws in the charge and shows that the pressure falls a little below atmospheric pressure, which is due to the fact that the admission-port offers resistance to entering air and gas.

On the return stroke (first instroke) the line 2 to 5 is made, the pressure rising by compression to the point 5, at which point ignition occurs and the explosion raises the pressure suddenly as shown near 6.

The line 6–7 may be called the expansion-curve (adiabatic expansion), the exhaust-valve opening at 7.

On the second return stroke the exhaust gases are expelled as shown by the line 8–1.

The gas-engine indicator-card will not as a usual occurrence be as regular in outline as the one just shown, because of the effects of the inertia of the parts of the indicator.

The temperatures of the working mixture during the cycle are (for convenience) marked upon the card. The formula for the H.P. is $\frac{PLAN}{33,000}$, in which the factors are the same as for the steam-engine, remembering that N will be the number of impulses, that is, the number of piston-strokes divided by four.

Losses in a Gas-engine.—The principal losses in a gas-engine are: 1st, heat given out to the walls and the jacket-water, and 2d, great heat expelled in the exhaust gases. These losses cannot be avoided but may be made less by careful designing.

The following is the result of a test and calculation made by Thurston upon a gas-engine, which represents the distribution of heat in good gas-engines:

Heat transferred into useful work, 17 per cent.

Heat transferred to the jacket-water, 52 per cent.

Heat lost in the exhaust gas, 16 per cent.

Heat lost by conduction and radiation, 15 per cent.

This shows an efficiency of 17 per cent.

The Working Fluid.—Any fuel that is not in the gaseous state already may be converted into a gas suitable for use in a gas-engine.

Generally speaking, the amount of power that can be derived from any fuel is greater when it is first made into a gas and then used to drive a gas-engine, than when the fuel is burned in the furnace of a steam-boiler producing power for a steam-engine.

The most common gas-fuel is city gas.

The amount used by the engine is measured with a gas-meter.

Gas-producing plants are sometimes arranged which manufacture gas for the direct use of a particular gas-engine.

Fig. 224 shows such an apparatus which is designed to use anthracite coal. The producer at the left is a brick chamber lined with fire-brick.

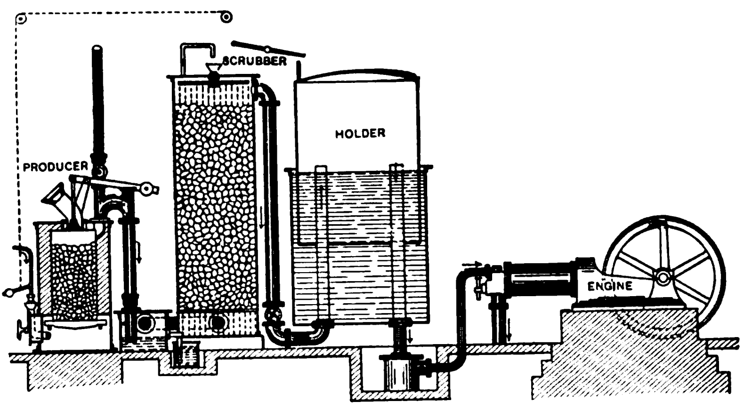


FIG. 224.—Producer Gas-plant.

At the bottom is a furnace and grate. Coal is placed in the upper part and the heat of the furnace drives off the volatile matter in the form of gas, smoke, etc.

From the top of the producer this gaseous matter is carried by a pipe to the bottom of the scrubber. This scrubber is filled with water and other substances, which catch all the impurities and allows pure gas only to reach the top.

From the top of the scrubber the gas is carried by a pipe to the holder, which is sealed with water, the gas being above the surface of the water, from whence it is taken, as needed, by the engine.

VALVES AND VALVE-MECHANISMS.

The majority of modern gas-engines are of the Otto type and the admission- and exhaust-valves are poppet-valves held upon their seats by a spring (see Fig. 228). These valves are operated from the cam-shaft by means of cams, levers, etc. (see Fig. 228). The cam-shaft is operated by the crank-shaft by means of bevel-gears, the axes of the crank-shaft and cam-shaft being perpendicular to each other.

REGULATION.

There are two general methods of controlling the speed of gas-engines for variable loads. 1st, *by varying the number of impulses*, which is called the *hit-and-miss* method, and 2d, by *varying the strength of the impulse*, the number of impulses being the same for each revolution, but the strength of the impulse being varied by different means.

The hit-and-miss method may be carried out in three ways:

1st. By holding the gas-valve closed during the time required for one or more impulses; 2d, by stopping the action of the exhaust-valve, keeping it either open or closed during the idle strokes.

When the exhaust-valve is held open the suction within the cylinder is not sufficient to open the admission-valve. When the exhaust-valve is kept closed during the idle strokes, the products of combustion are kept in the cylinder, which of course keeps the pressure within too great to allow an admission of new mixture.

3d. By cutting off the current from the igniter.

In this case the governor is attached to a switch, which is opened when a speed is reached which is above normal, the charge within the cylinder being alternately compressed and expanded until the speed is decreased to normal and the switch closed.

The *variable-impulse method* may also be carried out by three different methods:

1st. By reducing the proportion of gas in the full mixture, that is, by a partial stoppage of the gas-supply. Of course if the mixture is poor in gas the force of each impulse will be correspondingly small.

2d. By reducing the quantity of mixture admitted without altering the proportion of gas and air. This method is similar to the method used with the steam-engine using a throttling-governor.

3d. By varying the point of the stroke at which admission occurs.

The greatest strength of impulse is obtained by igniting at such a time that the maximum pressure is at the beginning of the stroke.

Making the ignition earlier or later than this will decrease the force of the impulse and hence the speed.

The six methods above named are operated by two classes of governors, viz., the *centrifugal governor* and the *inertia governor*. The centrifugal governor works upon the same principle as that used on throttling steam-engines.

Fig. 229 shows an example.

Fig. 225 shows one form of the inertia governor which is

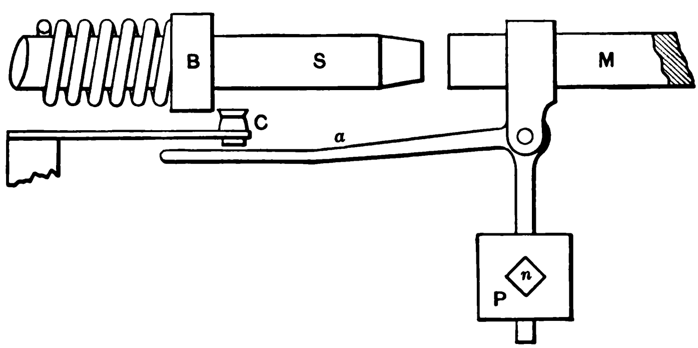


FIG. 225.—Inertia Governor.*

used only with the hit-and-miss method. The valve-stem S is operated from the cam-shaft by means of the slide M.

Every time M strikes S the valve is opened.

To the slide is connected the pendulum P, which swings about a pivot. When the speed is above normal, the pendulum lags behind, causing a to strike the pin C, which causes C to catch the block B and hold the valve open.

* From the "Gas-engine Hand-book by E. W. Roberts."

As soon as the speed decreases enough the arm *a* releases *B*, so that the exhaust-valve is closed and the engine admits fuel for another impulse. This it will be seen is that method, already described, in which the exhaust-valve is kept open during the idle strokes.

IGNITERS.

There are four ways of igniting the charge in a gas-engine:

1. Ignition by means of a naked flame as in the Otto engine, Fig. 221.

2. Contact with a surface which is at a high temperature, as shown in Fig. 222.

The hot-tube is the best example of this class. Fig. 226 shows an old arrangement of this class.

A part of the cylinder is shown at the right.

The wrought-iron tube 1 is heated by the Bunsen flame 2.

The piston at the proper time uncovers the hole 4, and the mixture entering under pressure is ignited.

3. Ignition by means of the flame of an electric arc. With this form of igniter an electric circuit from a battery is closed, by means of contact points which are situated within the compression-space. The breaking and closing of the circuit is effected by means of cams or eccentrics and links operated from the cam-shaft.

This latter method of ignition is probably the most popular method in modern gas- and gasoline-engines.

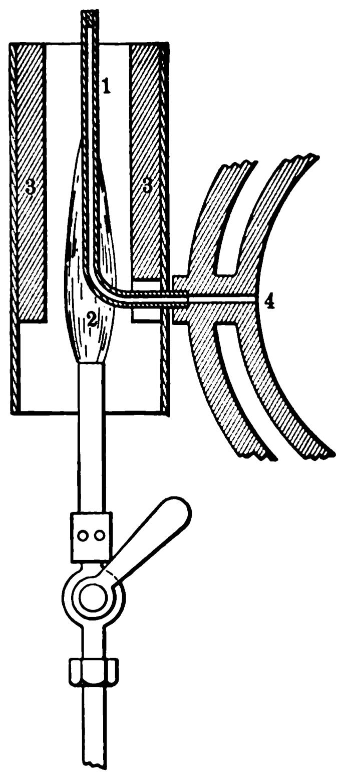


FIG. 226.—Hot-tube.

GASOLINE-ENGINES.

The gasoline-engine is practically the same as the gas-engine except for a few details, and many engines are manufactured which may be used either for gas or for gasoline.

Gasoline is introduced into the cylinder in a finely divided spray by passing a jet of air over the gasoline as it enters. This spray is then compressed and ignited just as if it were a mixture of air and gas.

The gasoline is supplied to the engine, from a tank placed below it, by means of a gasoline-pump, which pump is operated from the cam-shaft by suitable gearing. The other details of gasoline-engines are practically the same as for gas-engines. Fig. 227 shows the general arrangement of the engine and

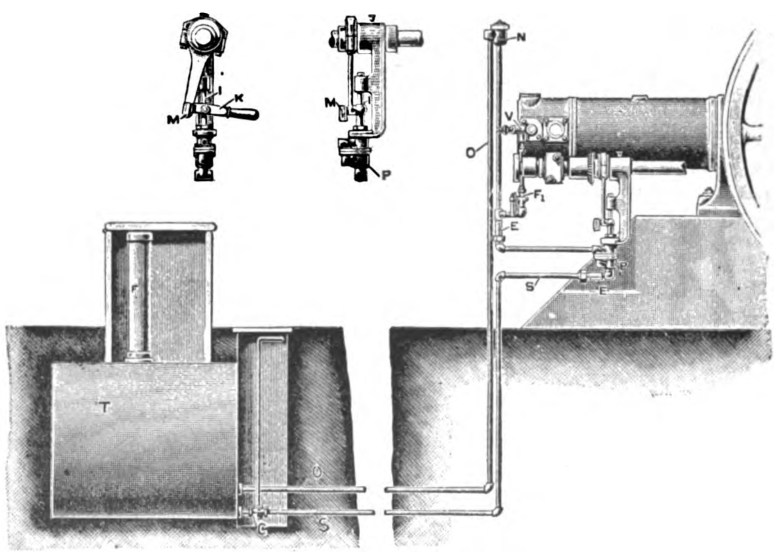


FIG. 227.—Arrangement of Engine for using Gasoline.

gasoline-tank for a stationary plant. *T* is the gasoline-tank, *P* is the gasoline-pump, and *S* is the pipe leading to the pump from the tank.

The pump lifts the gasoline from the tank through the supply-pipe S into the overflow-cup N, containing about $\frac{1}{4}$ to $\frac{1}{2}$ pint according to the size of the engine. From the cup the

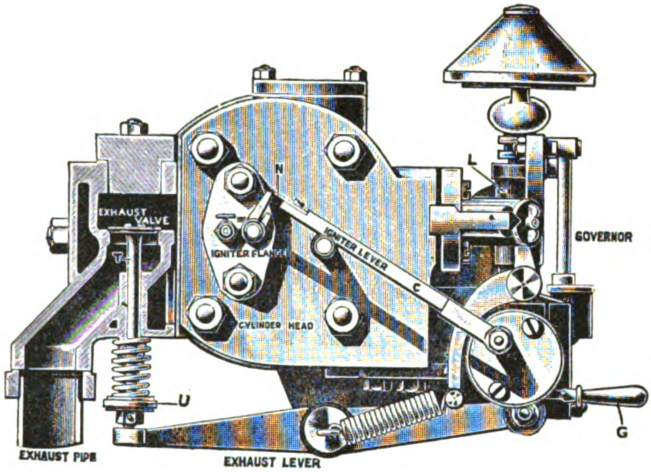


FIG. 228.—Cylinder and Governing Device of the Otto Gasoline-engine.

gasoline is admitted through the gasoline-valve V into the mixing valve of the engine. By means of the overflow-cup N

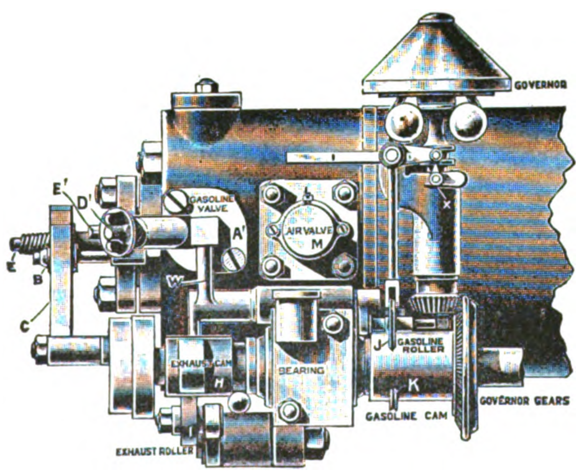


FIG. 229.

and the overflow-pipe O the over-supply of gasoline which the pump is capable of supplying is taken back to the tank. When starting the engine it is necessary to pump gasoline into the cup *N* by hand.

This is done by means of the hand-lever *K*, which is disconnected from the engine by unscrewing the pin *M*. Fig. 228 shows an end view and Fig. 229 a side view of the cylinder and governing apparatus of the Otto gasoline-engine.

OIL-ENGINES.

The oil-engine may be taken as another type of the gas-engine in which another step is added to the process already described for the gasoline-engine, of atomizing the fuel by a jet of air.

In the oil-engine the fuel, which is oil, is not only first *atomized* by the process already described, but it is *vaporized* by passing it through a heated chamber.

By these two operations the oil is converted into a gas, after which it is introduced into the cylinder, compressed, and ignited as in the gas-engine.

The atomizing principle is best shown by the perfume spray-producer shown in Fig. 230.

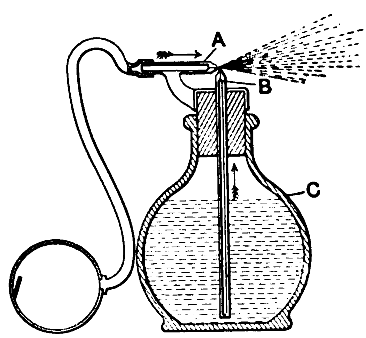


FIG. 230.—Perfume-sprayer.

In this elementary form an air-blast passing from the small jet *A* crosses the top of the tube *B* and creates a partial vacuum within.

The liquid in the bottle then flows up the tube *B* and, issuing at the top through a small orifice, is blown into a fine spray.

This principle is used for spraying gasoline and oil. In the oil-engine this spray is then easily vaporized by heat and then introduced into the engine. The principal fuel used in oil-engines is petroleum. Fig. 231 shows a very simple

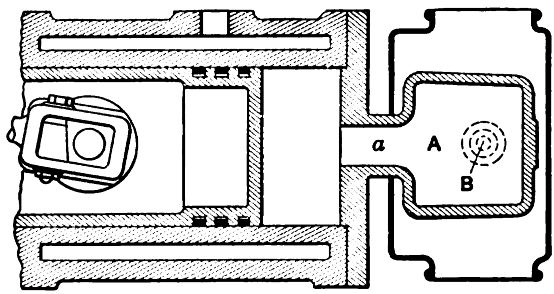


FIG. 231.

vaporizing arrangement for oil-engines as used on the Hornsby-Ackroyd oil-engine. Vaporizing is done by introducing the spray into the combustion-chamber *A*, which is arranged so that the heat of each explosion maintains it at a temperature high enough to vaporize the spray by mere injection upon the hot surfaces, the heat being sufficient to ignite the mixture of vapor and air when it is compressed. To start the engine, the vaporizer is first heated by a separate lamp, the spray is injected into the inlet *B*, and the engine is given a few turns by hand, after which the heat of each explosion furnishes heat for vaporizing the charge for the next impulse.

Fig. 232 shows a section through the vaporizer and cylinder of the Priestman oil-engine. *K* is the cylinder and *E* is the vaporizer.

The oil is sprayed and forced into *E* by means of an air-pump which is operated by the cam-shaft. The vaporizer is kept hot, while the engine is running, by the exhaust gases which leave the cylinder through the exhaust-valve *N* and the port *O*, entering the jacket *P* which surrounds the vaporizer.

The oil-engine is not as economical in the use of fuel as the gas- or gasoline-engine; and besides this it is difficult to design them so that there is no danger of explosion without making them very uneconomical as far as converting heat-units into useful work is concerned.

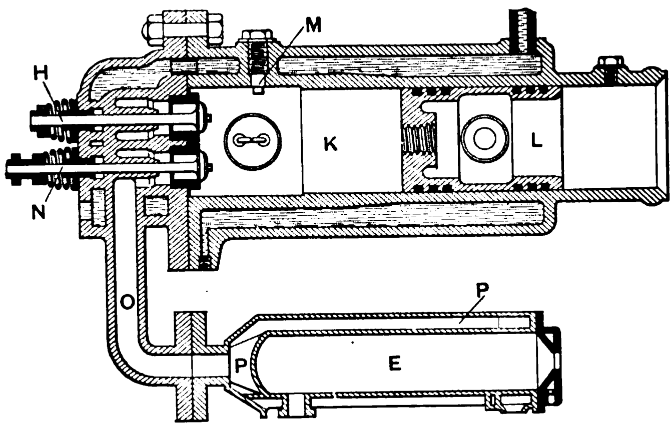


FIG. 232.

The jacket-water for gas-, gasoline-, and oil-engines is supplied by means of a water-pump usually operated from the cam-shaft.

CHAPTER XXV.

WATER-POWER.

WATER-MOTORS are those in which the working fluid is water, corresponding to steam in the steam-engine for producing motion. The pump does not belong to this class, being an instrument which acts upon water rather than one upon which water acts. Water is practically incompressible and consequently inexpansible, hence it cannot be used expansively in water-engines, as is the case with steam, though water is sometimes used in engines similar to those which use steam for the working fluid. Hence an indicator-card taken from a reciprocating water or hydraulic engine would show a maximum pressure line approximately parallel to the atmospheric line.

In the steam-engine the pressure of steam depends upon the quantity of heat supplied to it in the boiler.

Water-pressure depends upon the height of its source and its velocity. Here the term "*head*" is used instead of height in calculating water-pressure; that is, we say that a head of 50 feet is obtained from a natural stream when its source is 50 feet above where its water is applied to the motor.

$$V = \sqrt{2gH}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

is an equation connecting head and velocity, in which V is velocity in feet per second, H the head in feet, and $g = 32.2$.

The head H in feet may be reduced to pounds per square inch by multiplying by 62.4 the weight of a cubic foot of water and dividing by 144. This is equivalent to multiplying the head in feet by $\frac{62.4}{144} = .426$.

The head, instead of being an actual distance between levels, may be caused by pressure as by a pump. A pressure-gauge may be used for reading this pressure or head.

The two terms pressure and head are often used one for the other.

There are two general classes of water-motors, viz.: Those having rotary pistons or runners and those having reciprocating pistons. The former are by far the more common in American practice.

The following is a classification of the principal water-motors:

1. Rotary
   - Water-wheels
     - Overshot wheel.
     - Undershot wheel.
     - Breast-wheel.
   - Turbines
     - Parallel flow.
     - Radial flow
       - Outward.
       - Inward.
     - Mixed flow.
   - Motors
     - Impulse.
     - Jet.
2. Reciprocating Hydraulic Engines.

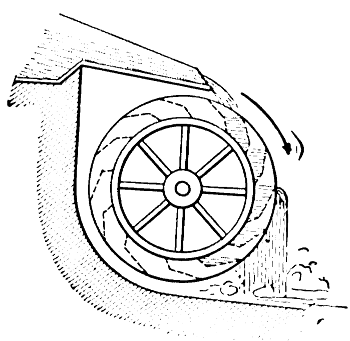


FIG. 233.

The simplest and first used of the water-motors is the *water-wheel*, in which the water produces motion by acting directly against vanes or buckets placed upon the circumference of a wheel. Fig. 233 shows an *overshot wheel.*

The circumferential speed of the wheel will be the same as the velocity of the water, which from (1) is $V = \sqrt{2gH}$, H being the head. If this velocity in feet per second be multiplied by the pressure on the vane due to the weight of the water, we have the foot-pounds of work done. Suppose that the cross-section of the stream of falling water is B square feet; then $B\sqrt{2gH}$ cubic feet is the volume and its weight is $62.4B\sqrt{2gH}$ lbs. per second. Multiplying the weight by the distance through which it falls, H, we have $62.4B\sqrt{2gH} \times H = 500.69BH^{\frac{3}{2}}$ ft.-lbs. of work per second and $500.69BH^{\frac{3}{2}} \times 60$ ft.-lbs. per minute. Dividing this by 33,000, we have, approximately, the horse-power developed by the stream =

$$\frac{500.69BH^{\frac{3}{2}} \times 60}{33,000} = .9103BH^{\frac{3}{2}}$$

This is usually called the water H.P., because it is the horse-power which the stream is capable of producing with an ideal water-motor, that is, in which there are no losses such as friction, etc. Fig. 234 shows an *undershot wheel* in which the water passes under the wheel.

Fig. 235 shows a *breast-wheel.* In this case the water strikes the wheel at or near the level of the axis. These wheels are furnished with vanes or buckets of such shape that

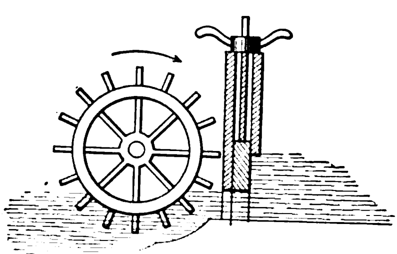


FIG. 234.—Undershot Wheel.

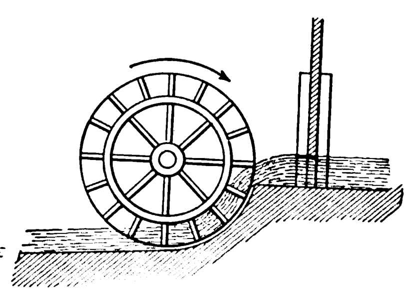


FIG. 235.—Breast-wheel.

not only will they admit water without hindrance, but also return it to the lowest possible point on the wheel. The flow of water is controlled by large gate-valves. These wheels require comparatively large amounts of water for their running, hence their efficiency is small. For the plants of the present time the wheel would necessarily be so large that its construction and operation would be impracticable. For these reasons they have largely gone out of use.

TURBINES.

In a parallel-flow turbine the water enters and leaves the turbine in a line parallel to the axis. In the radial-flow turbine the water enters and leaves the turbine on radial lines, flowing inward or outward according to the class of turbine. In a mixed-flow turbine the water enters radially and leaves in a line parallel to the axis.

The outward-flow turbine included in the classification is shown in principle in Fig. 236. In the centre are a number of fixed curved guides which direct the water against the curved vanes or buckets of the wheel, causing it to rotate in

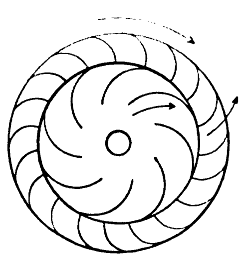


FIG. 236.—Outward-flow Turbine.

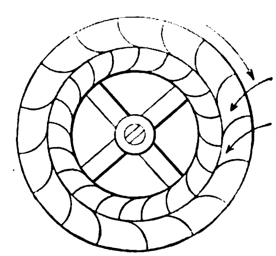


FIG. 237.—Inward-flow Turbine.

the direction shown by the arrow. The action of the water in the turbine is also illustrated by the arrows.

Fig. 237 shows the principle of the inward-flow turbine. In this instance the water enters from the outside, being directed by the curved guides against the vanes or buckets of the wheel and leaves at the interior. The curves of the wheels and the guides are designed such that a maximum efficiency may be obtained.

The larger number of American turbines belong to the mixed-flow type.

Fig. 238 shows a turbine of this type. The water flows from the outside inward through the openings between the guides, strikes the wheel or runner and flows out through the

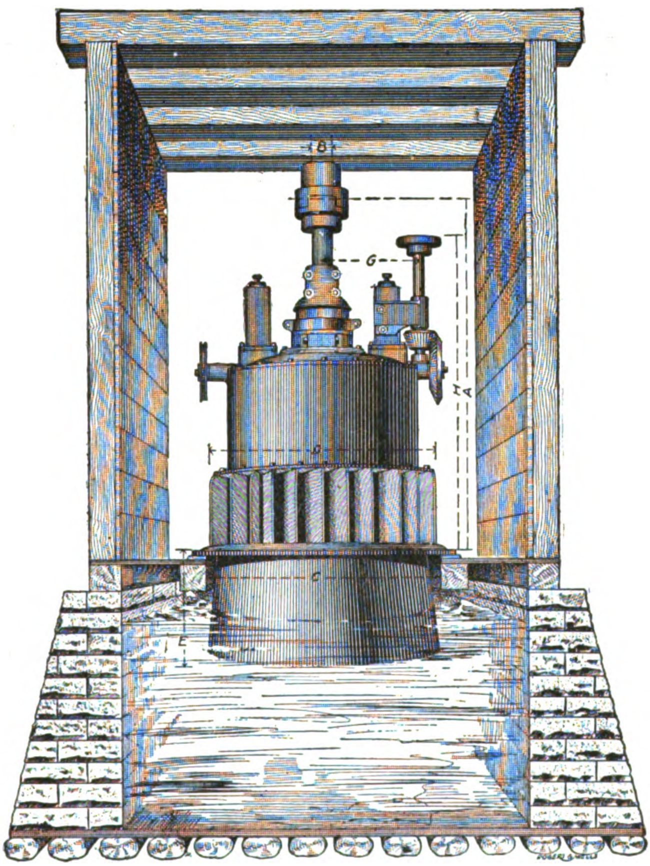


FIG. 238.

bottom. The vanes join runners at the bottom by means of vertical curves. The reaction of the water on these vanes causes the wheel to revolve.

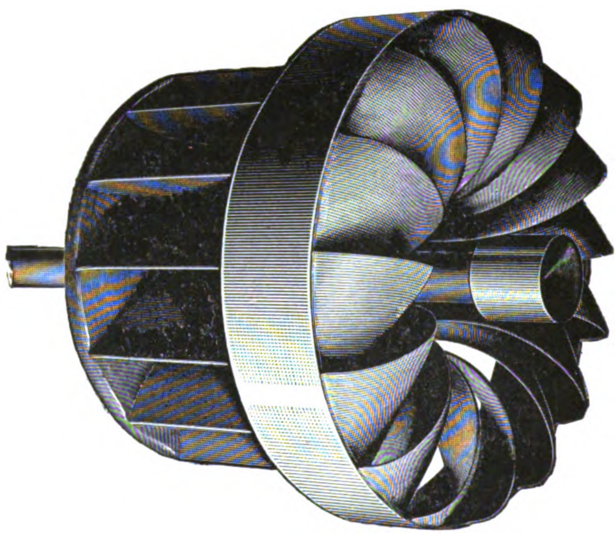


FIG. 239.—The New American Turbine-wheel.

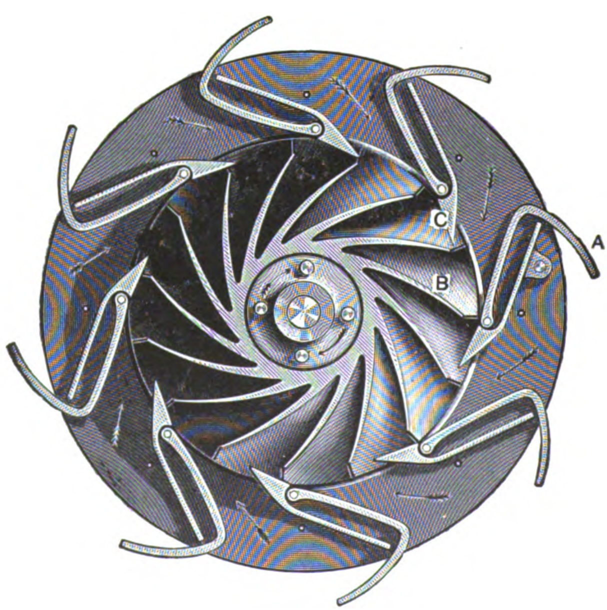


FIG. 240.—Runner-case and Guides.

Fig. 239 shows the runner and shaft which when in operation are vertical. Fig. 240 shows a horizontal section of the runner-case and guides. *C* is a guide, *B* is a vane of the runner, usually called a *bucket*, and *A* is a gate. The inflow of water into the machine is controlled by these gates. The opening between the guides may be closed by means of the gates, and the machine stopped. The efficiency of a water-turbine of this type depends largely upon the curves of the guides or chutes and the buckets. By giving them the proper curves the least resistance and greatest working effect is produced.

Runner.—The runner of this turbine is a solid casting, though in some makes the buckets are of steel, moulded to a cast hub and rim. The very high speed requires that it be made very strong.

Transmission.—The power developed by the turbine is transmitted to machinery from the runner-shaft by suitable gearing.

Manner of Applying Water.—Fig. 238 shows a method used for conducting water to and from the wheel. The wheel is placed in this case over an opening in the "head-race" or flume-floor so that the water has to pass through the turbine in order to get into the "tail-race" which is below this floor. The level of the water in the "tail-race," is kept constant. Sometimes the pipe leading to the tail-race, called a suction-tube, is made very long in order to produce suction. The limit of course would be 28 feet.

Fig. 241 shows a turbine located in an iron suction-tube about 20 feet above the tail-water level. From this it is readily seen that the turbine is entirely in water while in operation. That part of the flume *A* above the turbine is the head-race, that part in which the wheel is contained is the pit, and that part below the turbine is the tail-race. *C* is the draft- or suction-tube.

The water for the turbine is controlled in its passage through the head-flume by means of large gate-valves. Fig. 242 shows one operated by a crank and gearing or by

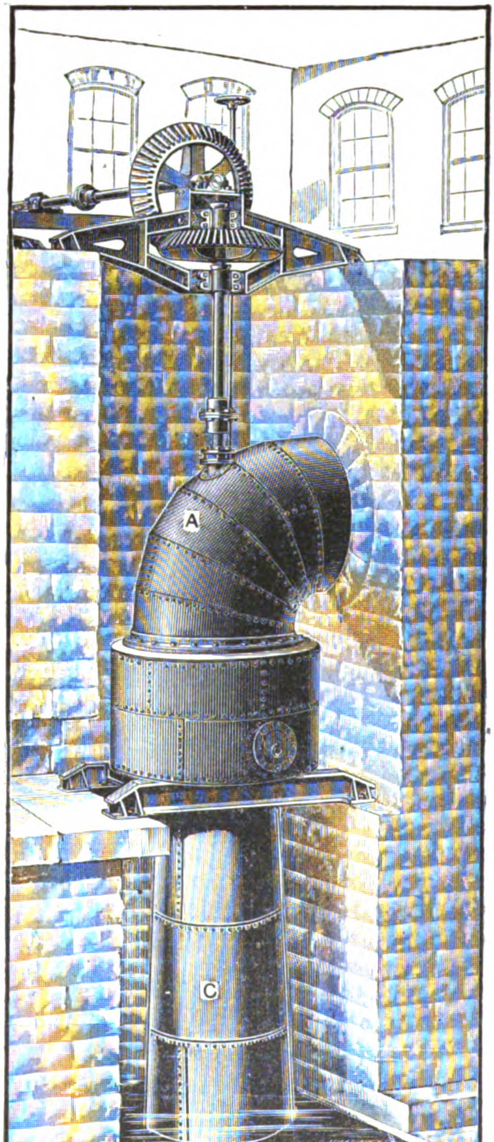


FIG. 241.—Suction-tube (New American Turbine).

hydraulic power. By means of this valve the water may be shut off from the machine for examination, repairs, etc.

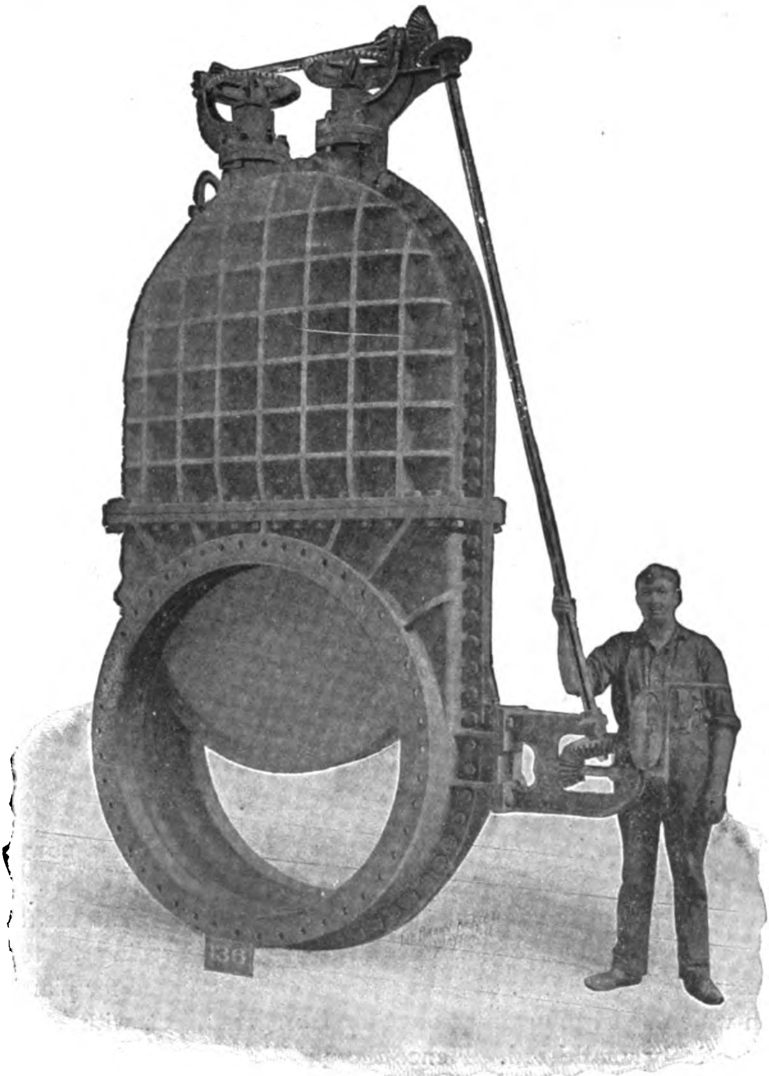


FIG. 242.—Gate.

Fig. 243 shows the arrangement of a turbine plant (Victor). *A* is the head-race constructed of masonry. *B* is the head gate-valve. C is the head-flume. D is the shaft to which the wheels are fastened (the turbines are not seen, being covered by the iron casing). E is the draft-tube, and F is the tail-race, constructed also of masonry. The head is the vertical distance from the water-level in the tail-race to the level of the supply-water.

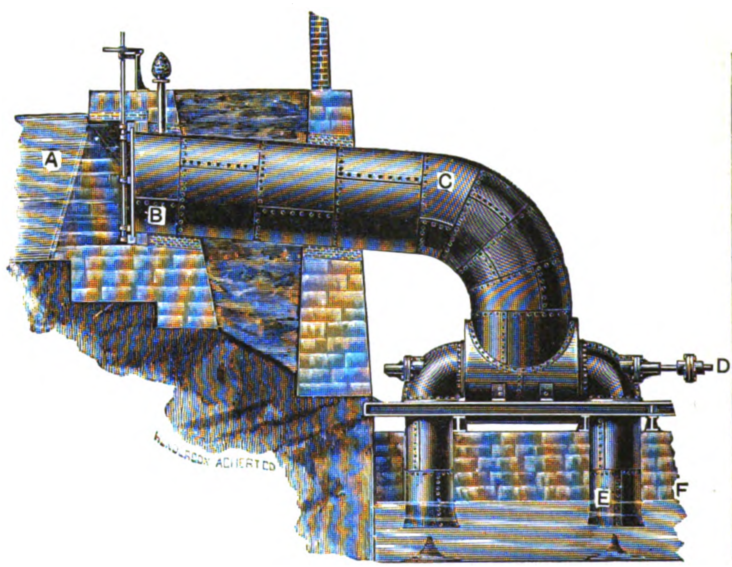


FIG. 243.—Turbine Plant.

RATING OF TURBINES.

Water-turbines are rated according to the horse-power they will develop.

For the same head and quantity of water different makes of turbines will give different horse-power, according to the excellence of design. These turbines are designed for plants in which a large quantity of water is at hand, but with comparatively small head. Hence the quantity of water is a factor which affects the horse-power. The head used for these is usually not over 100 feet. It has already been explained how to find the water horse-power. The horse-power which the wheel really develops will always be less than this. The efficiency $= \frac{\text{wheel H.P.}}{\text{water H.P.}}$, is 60 to 90 per cent, according to make of wheel.

The wheel horse-power may be found by means of a dynamometer.

SPEED OF TURBINE.

The velocity of the wheel is due to the velocity of the water. Theoretically the velocity of the wheel would be the velocity of the water, but actually this is not the case. A turbine attains its greatest efficiency when the velocity of the turbine is one half that of the water driving it. The actual speed can be gotten by a test with a speed-indicator.

REGULATION.

The speed of the turbine is regulated by closing the water-supply to the turbines by some automatic or hand-apparatus. Fig. 244 shows the Snow centrifugal governor for water-turbines. It has a foundation of its own and is connected to the turbine-shaft by means of a belt to the pulley A. R is the shaft which operates the valve which closes the water-supply. If the turbine is running at normal speed the shaft R is not revolved by the governor. If the speed is increased, however, above the normal the balls rise and the governor-spindle is lowered thereby causing S and R to revolve in such a direction as to close the gates.

If the speed is decreased below the normal the governor-balls descend and operate a mechanism which turns S and R in such a direction as to open the sluice-gates. By means of a hand-wheel the shaft R may be operated by hand.

SETTING.

The head-race should be made sufficiently large to prevent a diminution of the head by friction, etc. It is usual to make the head-race of such cross-section that the water will not flow through it faster than 100 feet per minute.

Making the head-race too small is equivalent to reducing the head.

The wheel-pit, that is, where there is no draft-tube, should be of sufficient depth that it will not produce a reaction of the water against the under side of the wheel.

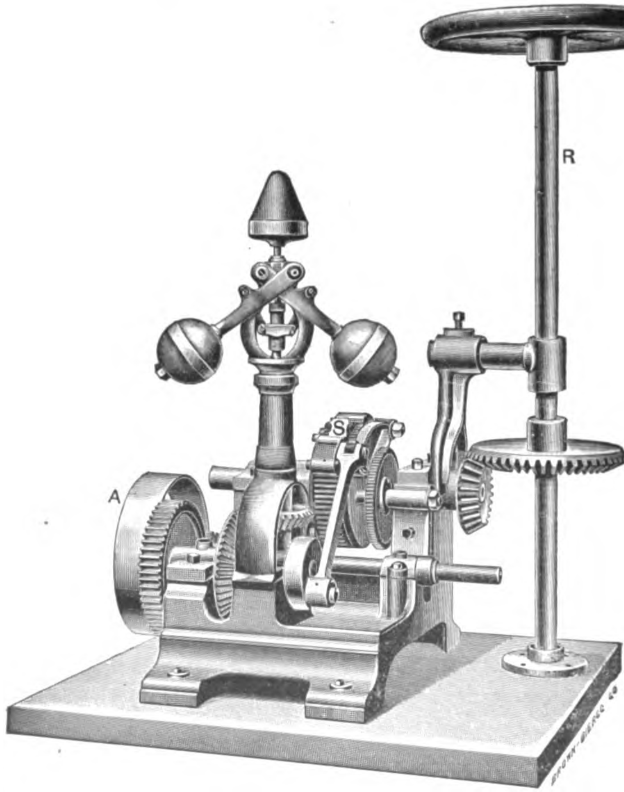


FIG. 244.—Snow Centrifugal Governor.

An additional loss of power is also caused by the fact that a portion of the head is consumed by forcing the water out of the pit when the outlet is of insufficient size. The pit should be lined so that its bottom and sides will not be damaged by erosion.

The tail-race should also be both wide and deep.

The flume should be large also, in fact, all the water-passages should be large enough to prevent loss of head on account of friction.

DRAFT-TUBES.

This is an air-tight tube used in constructions where the wheel-pit is very deep. Its lower end should dip 2 or 3 inches below the surface of the standing tail-water. It is never longer than 28 or 30 feet, and is usually less. It is sometimes very short, say 2 to 3 feet, simply for the purpose of carrying the water so that it is out of reach of the lower timbers of the penstock.

IMPULSE- OR JET-WHEELS.

This class consists of those water-motors utilizing small volumes of water with high heads by the use of a jet escaping from a nozzle and striking against buckets which are placed upon the circumference of a wheel, thus producing motion.

An example of this type is found in the Pelton motor-wheel, Fig. 245, mounted upon a wood frame. The water-supply issues in a jet from the nozzle and strikes the buckets of the wheel.

The supply is controlled by a valve.

The speed is kept constant under varying loads by means of a centrifugal governor which closes or opens the supply according to the needs of the load. The action of this governor is the same in effect as that described for the turbine. The discharge or tail-water leaves the motor at the bottom. Fig. 246 shows this wheel mounted in an iron case, which arrangement makes the setting of the motor a more convenient matter.

A Pelton motor-wheel is shown also in Fig. 254, driving an air-compressor. With a low head and large volume of water the arrangement shown in Fig. 247 is used, in which three nozzles direct as many jets against the same wheel.

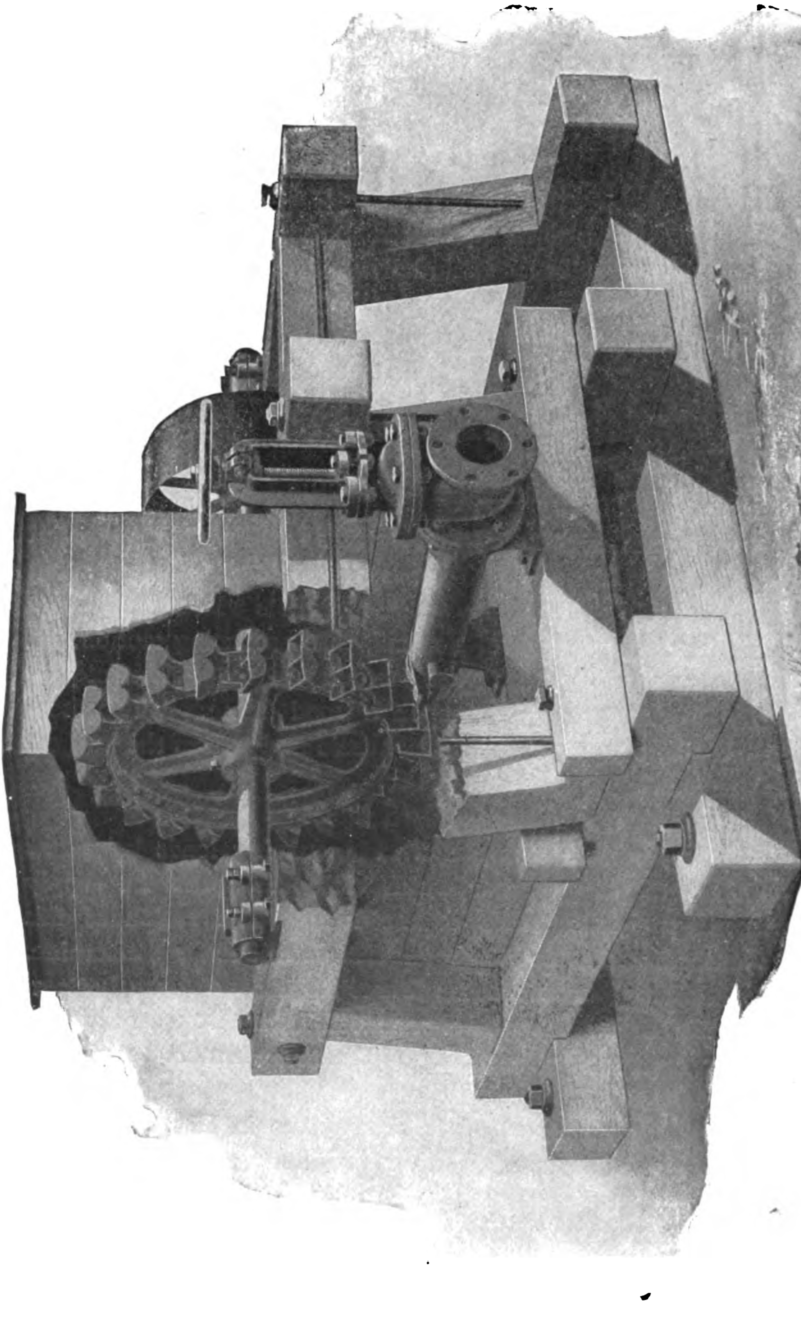


FIG. 245.—Standard Pelton Wheel Mounted on Wood Frame.

The motors require practically the same setting as turbines.

The horse-power of these wheels may be determined by means of a dynamometer.

The efficiency $= \dfrac{\text{wheel H.P.}}{\text{water H.P.}}$ is comparatively high.

The simplicity of this arrangement makes it very desirable.

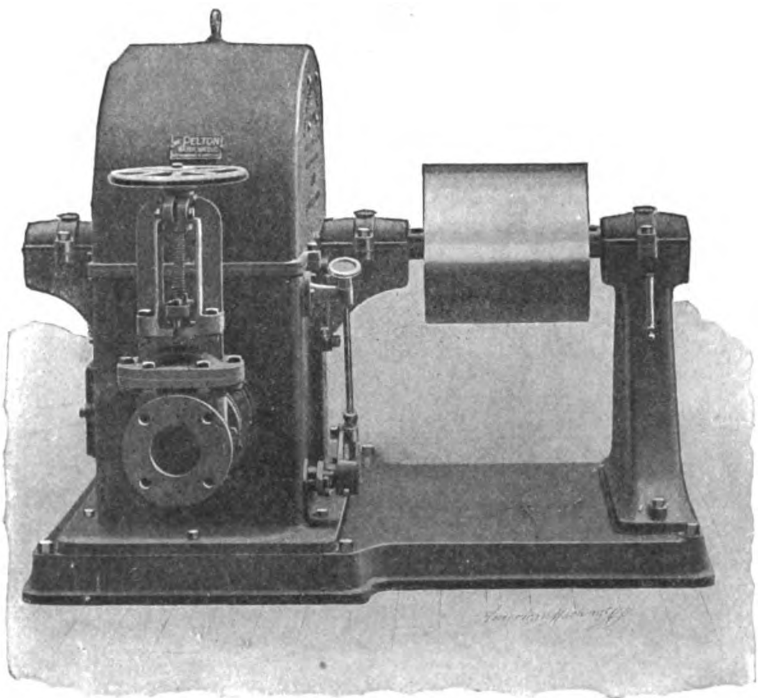


FIG. 246.—Pelton Water-motor.

HYDRAULIC PUMP.

It has been stated already that steam-engines with slight modifications may be used as water-engines, in which case water is used as the working fluid instead of steam.

Reciprocating water-engines are necessarily of slow speed because of the incompressibility of water and its large amount of friction in passing through pipes as compared with steam.

Fig. 248 shows a pump driven by water-pressure. Its construction does not differ materially from that of the steam-

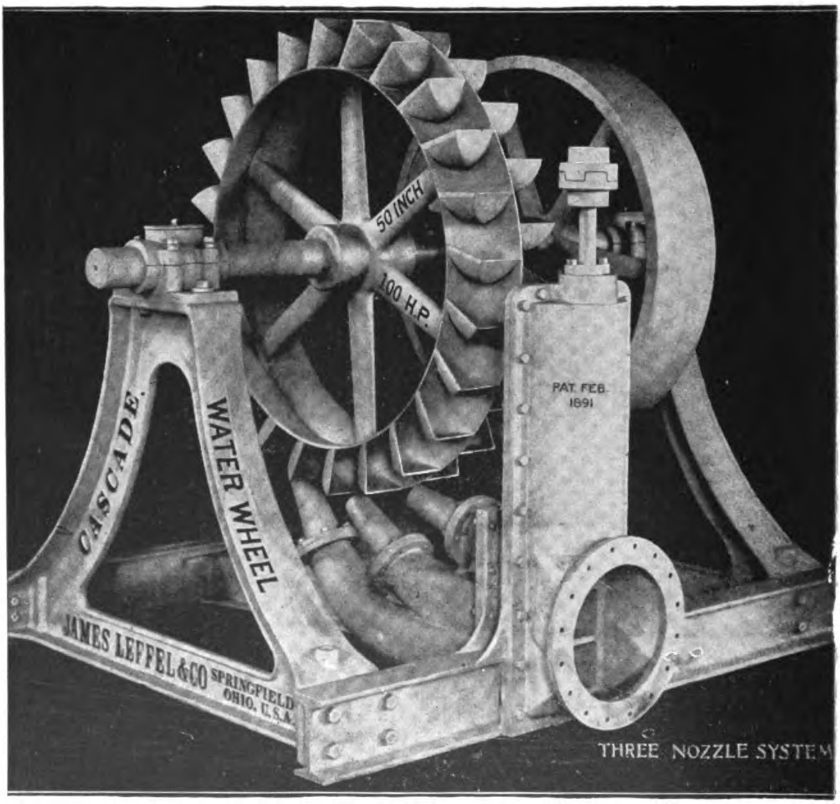


FIG. 247.—Multiple Jet-wheel.

pump except that the driving cylinders are provided with extra large passages suitable for the prompt and easy flow of water.

Loss of Head.—In any water-engine the pressure upon the piston or buckets is theoretically found by the formula already given, $H = \frac{v^2}{2g}$. However, it is found that part of this pressure

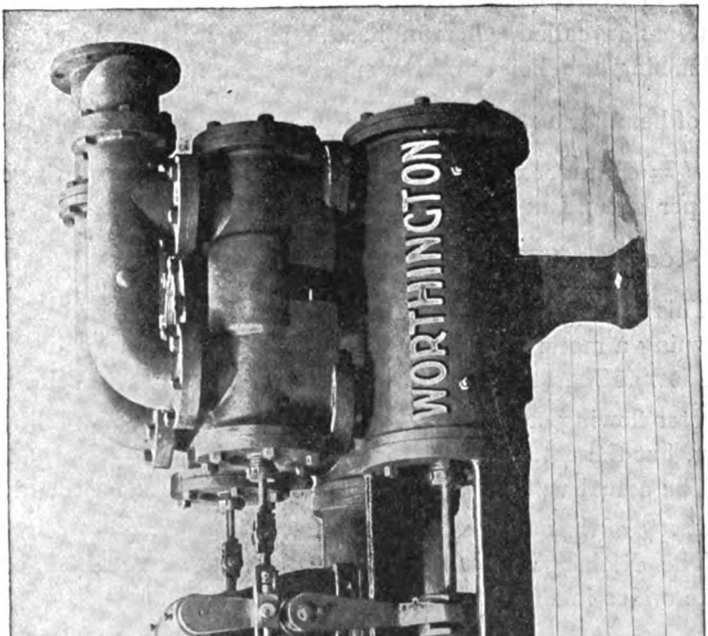


FIG. 248.—Water-engine.

is lost in overcoming friction on its way through pipes, etc. This loss is called "loss of head," since it is equivalent to diminishing the height of the supply above the motor. This loss of head for a straight pipe is $h = f\frac{4lv^2}{d2g}$, in which l = length and d = diameter of the tube, both in feet. v = velocity in feet per second and f = a coefficient which Weisbach gives to be .00644, which makes $h = \frac{.01288lv^2}{dg}$.

This loss must be subtracted from the measured head when designing a motor to do given work.

Flow of Water Through Orifices.—The theoretical velocity of water flowing from an orifice is $\sqrt{2gH}$, or that of a falling body, but the actual velocity is less than this.

The actual velocity at the plane of the orifice is generally about .97 of that due to head, that is, $.97\sqrt{2gH}$.

The discharge then would be $Q = Av$, where A = area of the orifice in Fig. 249.

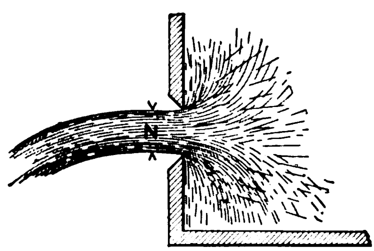


FIG. 249.

The jet contracts on leaving the orifice as shown at N.

This contracted area is called the "Vena Contracta."

In calculating the discharge the velocity is multiplied by the area of the cross-section of the jet at this point instead of the area of the orifice. This area is usually about .64 of the area of the orifice. Hence multiplying the velocity $.97\sqrt{2gH}$ by the area $.64A$, we have $Q = .64A \times .97\sqrt{2gH} = .62A\sqrt{2gH}$ as the discharge volume. By means of this formula the size of pipe, head, and velocity may be compared.

Tables, however, are given by authorities which take in consideration more minutely the difference of flow due to the smoothness or roughness of pipes, whether larger or smaller, etc., to which the author refers the student.

MEASURING THE POWER OF STREAMS.

The principal method used is that of obtaining the velocity of the stream, as a creek or river, and then multiplying it by the area of the cross-section of the stream.

Let $Q = Av$, in which Q is the discharge in cubic feet per minute and A the area of the cross-section in square feet, and v the velocity in feet per minute.

A may be found approximately by taking the depth at regular intervals across the stream, the distance from the edge to the first sounding being one half that of the other intervals.

By adding these different depths and dividing by the number of soundings the mean or average depth is obtained. Multiply this average depth by the width at the surface and the result is approximately the area of the cross-section of the stream.

v is found in different ways.

A simple method sometimes used is that in which a float is placed on the stream and the time it requires to pass over a known distance noted.

The velocity of the float will be that of the water at the surface of the stream, the velocity being less toward the bottom than at the top. The mean or average velocity is generally taken in practice as about .80 of the surface velocity.

Another simple method of finding the surface velocity is by means of a Pitot tube. This is shown in Fig. 250.

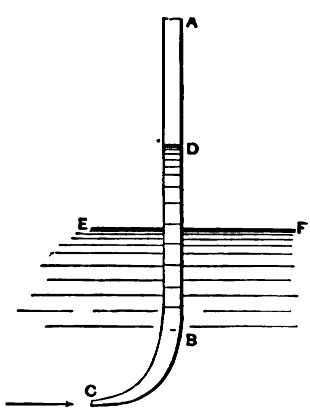


FIG. 250.—Pitot Tube.

The velocity of the water passing the opening at C causes the water to rise in the tube to a height varying with the velocity of the water. Let h be the height of the column of water in the tube. The velocity is theoretically $v = \sqrt{2gh}$.

For small streams the quantity of water which the stream is capable of supplying is determined as shown in Fig. 251.

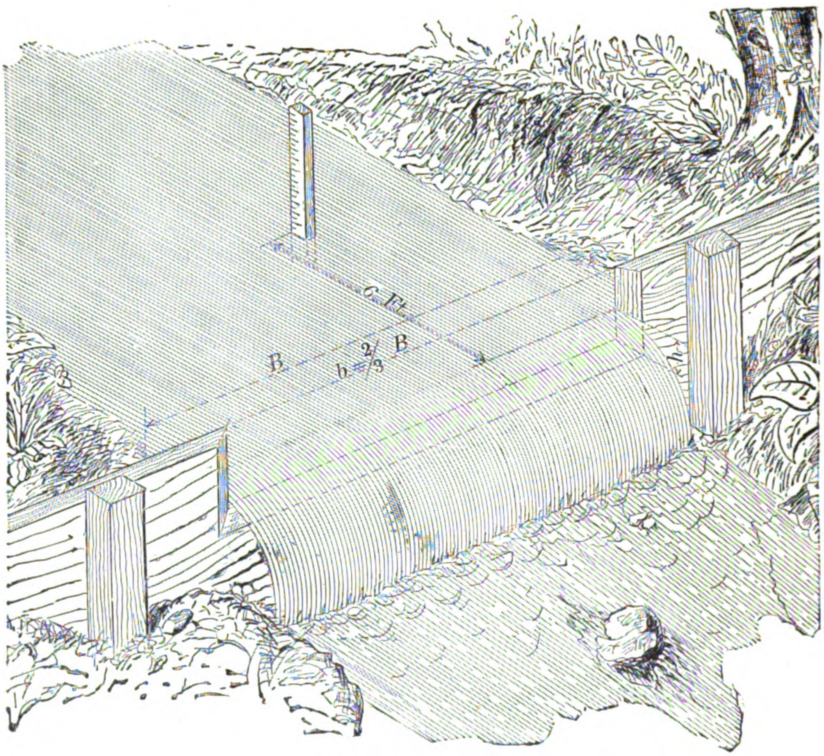


FIG. 251.—Weir.

This method is called the "Weir-dam Measurement" and is more exact than the method just shown.

A notch is cut in a thick plank, the length of the notch being about two thirds the width of the stream. Stakes are driven in the ground and this plank is held by them so that the water of the stream passes through the notch. To find the depth of the stream flowing through the notch, proceed as follows:

In the stream about 6 feet above the dam drive a stake in the channel and mark a point on it level with the bottom of the notch. Then when the water is all flowing over the dam, mark the water-level on the stake. The distance between these two marks is the depth of the stream. Then find the width of the notch in feet, which gives the cross-section of the stream. The quantity of water flowing in the stream is then determined by the following formulæ, given by Francis:

$$Q = 3.33lh^{\frac{3}{2}}$$

if the notch is taken the full width of the stream;

$$Q = 3.33(l - .1h)^{\frac{3}{2}}$$

if the notch begins at one edge of the stream and does not extend completely across;

$$Q = 3.33(l - .2h)h^{\frac{3}{2}}$$

if the notch is in the middle of the stream, and does not reach either edge.

In each case Q = quantity of water flowing in cubic feet per second; l = length of weir-notch in feet; h = head of water on the crest in feet, measured as shown above.

PROBLEMS.

1. The power of a certain stream is utilized by a Pelton water-wheel 100 feet below the source. Find the pressure per square inch due to the head; also the pressure per square foot.

2. Find the velocity of the jet which strikes the buckets in feet per second, neglecting friction, in the above.

3. Find the actual volume of water in cubic feet discharged per hour by a head of 150 feet through a pipe, the nozzle of which is 4 inches in diameter, taking the friction into account.

4. A stream is 20 feet wide and 12 inches deep. Its mean velocity is 400 feet per minute. Find the horse-power it is capable of producing.

5. A certain water-turbine uses 4536 cubic feet of water per minute with a head of 50 feet. Its developed horse-power is 342 as measured by a dynamometer. Find the efficiency.

6. A pipe 12 inches in diameter and 600 feet long supplies water to a water-wheel. Find the loss of head in feet due to friction, by Weisbach's formula.

7. If the difference in level of the two ends of the pipe in the above is 500 feet, find the effective head.

8. Suppose that the water rises to a height of 6 inches in a Pitot tube due to the velocity of the water. Find the velocity of the water.

9. The dimensions of a weir are $l = 42$ inches; the weir is in the middle of the stream. The depth of water on the weir is 16 inches. Find the quantity of water flowing in the stream.

10. Suppose that in problem 9 the weir was the full width of the stream, 72 inches. Find h in this case.

CHAPTER XXVI.

COMPRESSED AIR.

COMPRESSED air is used principally as a means of transmitting power through moderately long distances. In a common steam-plant considerable steam is condensed in the pipes before it reaches the motor. The greater the distance the greater will be these losses. For very great distances the use of steam for power transmission becomes impracticable, and either electric transmission or compressed-air transmission must be used.

The power of a waterfall or that of a steam-engine may be used for compressing air up to a high pressure, and this air is then conducted through pipes to the air-motor—which may be similar to a common steam-engine—or may have other forms, according to the kind of work to be done. Since the air after losing the heat given to it by compression is not subject to further cooling by the atmosphere as in the case of steam, the air may be transmitted through many miles of piping without any other loss except that due to friction and possible leakage. Another example of the use of compressed air is that of mining operations. The power-plant of the mine must be above ground. Underground there are a great number of rock-drills and locomotives for hauling ore through the mine, which must be operated by some means. Steam is objectionable because it will have to be exhausted into the interior of the mine, producing damp, etc. Besides this the loss due to condensation would be large because of the distance between the boiler and the motor. By means of steam or other power air may be compressed at the entrance to the mine and conducted through pipes with small losses to the motors. The exhaust into the mine starts a current of air towards the mine-entrance, thus tending to rid the mine of poisonous or objectionable gases.

MANNER OF COMPRESSING AIR.

Air is taken from the atmosphere and given a high pressure by means of an air-compressor.

Air-compressors may be classified according to the manner in which the air-pistons are given their motion, as steam-driven, belt-driven, and water-driven. Compressors are also classified according to the number of air-pistons, as single-stage compressors and double-stage, or compound, compressors, corresponding to the simple and compound steam-engine.

The steam-driven compressor consists of an ordinary steam-engine usually having the air- and steam-pistons fastened to the same piston-rod. Double-stage compressors may be either tandem or cross-compound, similar respectively to tandem and cross-compound steam-engines.

In belt- or gear-driven compressors the driving power may be taken from a shaft, instead of being actuated directly by steam- or water-power. The shaft of the compressor carries a large pulley or gear-wheel, taking power from the motor-shaft by means of a belt or gear-wheels.

The water-impulse compressor is shown in Fig. 254. The impulse-wheel is mounted directly upon the main shaft of the machine. The wheel is put in motion by turning on a jet of water from the pipe at the bottom. The force of the water against the buckets produces rotary motion. The force of the jet is controlled by means of the hand-valve at the left of the cut.

The process of compression in a double-stage compressor takes place as follows: The air is drawn into the low-pressure or intake cylinder and is partially compressed, after which it

A, low-pressure steam-cylinder.
B, high-pressure " "
C, " air-cylinder.
D, low-pressure " "
E, intercooler.

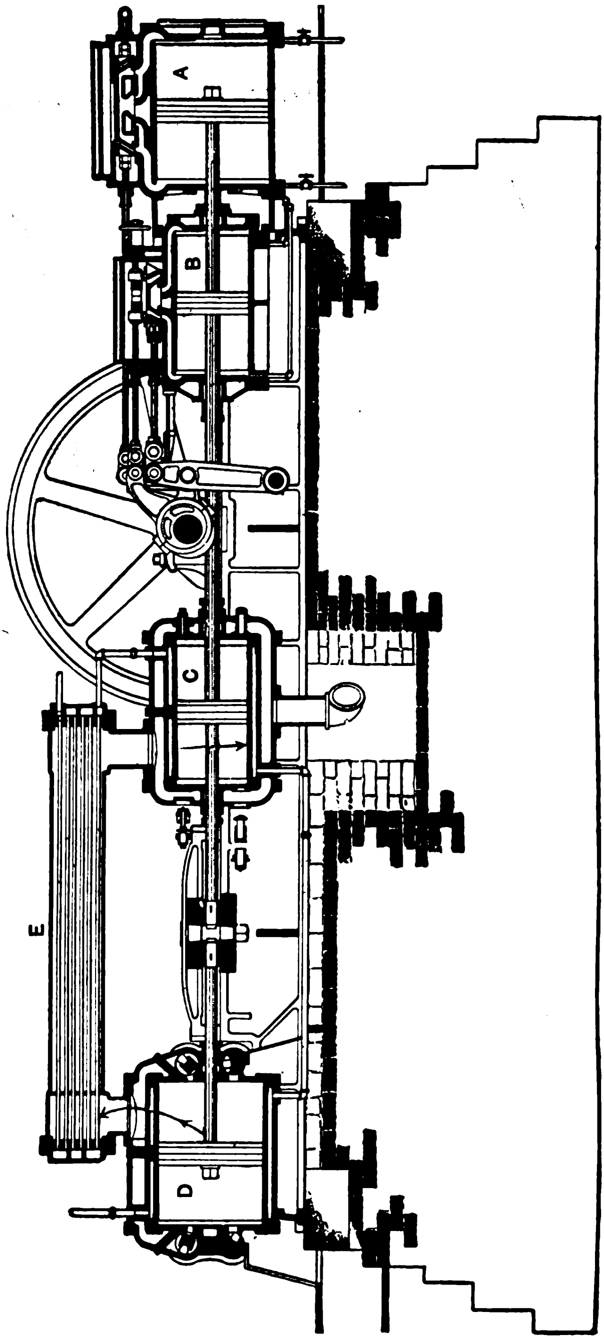


Fig. 252.—Steam-driven Air-compressor.

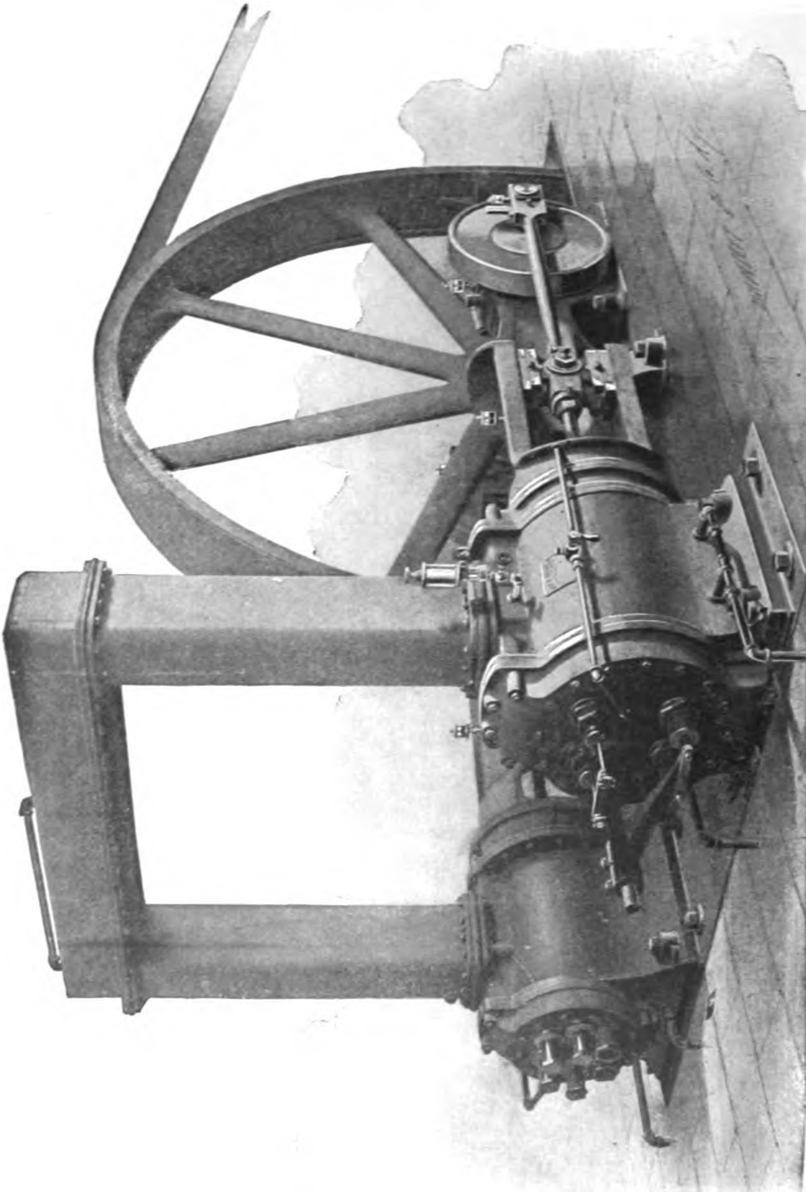


FIG. 253.—Rand Belt-driven Air-compressor. Cross-compound Air cylinders.

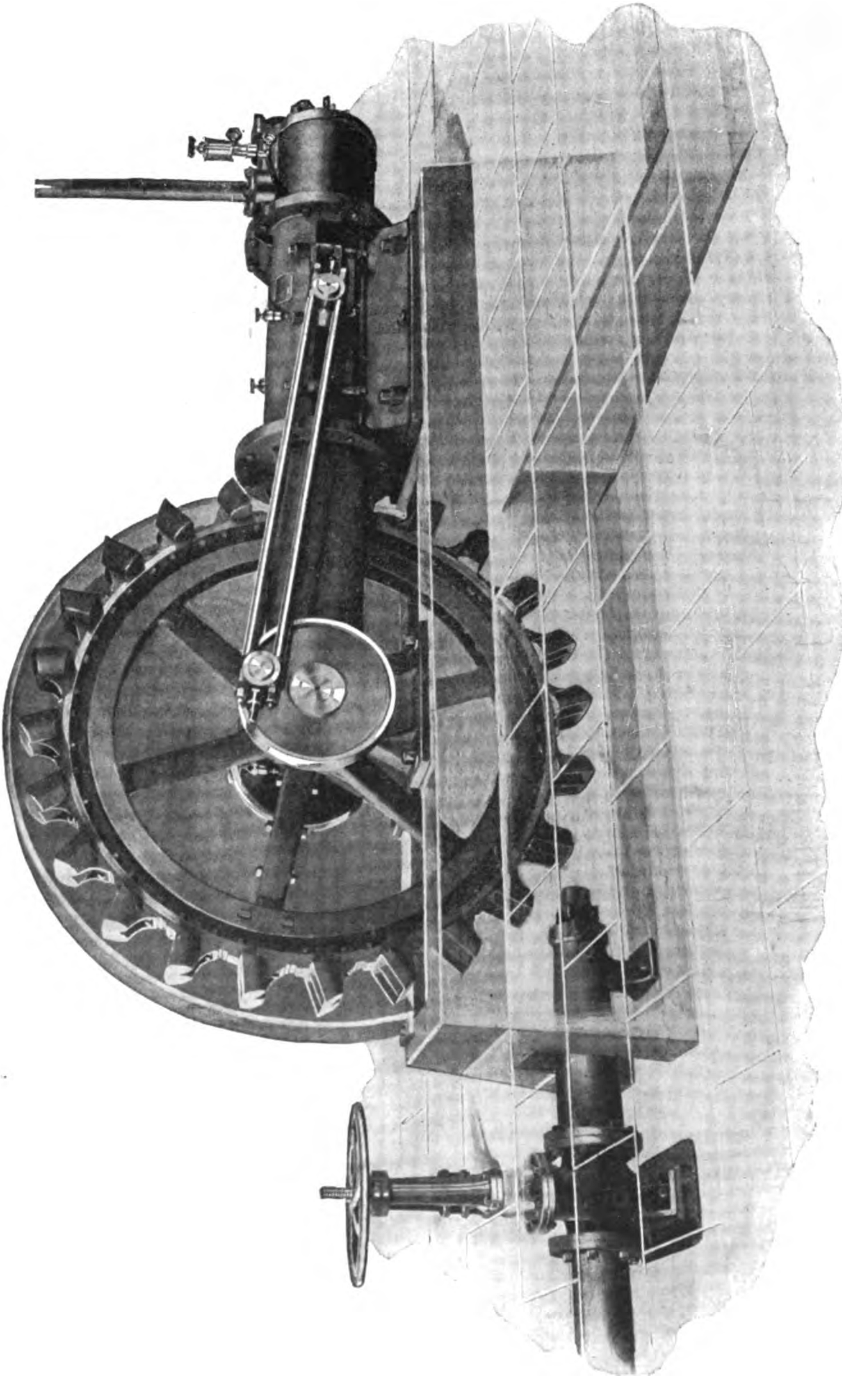


FIG. 254.—Pelton Water-impulse Air-compressor.

passes through an intercooler, where the heat of compression is taken away by means of cold water. It is then drawn into the high-pressure cylinder, where it is compressed to a higher pressure, and then it passes to an air-receiver and thence through the pipe-line to the motors.

Details of Compressors.—The general arrangement of the parts of an air-compressor is similar to that of a steam-engine with a few exceptions, which will be explained.

Cylinders and Valve-chest.—The valves and steam-chest are the same as for a steam-engine. Usually, the Corliss valve is used, though some manufacturers use poppet-valves for admitting the air. Fig. 252 shows the Corliss valve used in the intake air-cylinder of the Norwalk Compressor and poppet-valves in the high-pressure air-cylinder. The higher pressure in this cylinder makes it possible to use the poppet-valve more successfully without chattering, whereas in the low-pressure cylinder the low pressure and the spring would cause a constant vibration of the valve against its seat, thus producing an objectionable noise. When air is compressed its temperature rapidly increases. In order to keep the temperature of the cylinder-walls from being unduly heated, the air-cylinders are usually surrounded with a space through which water is circulated. The following table shows the theoretical increase of temperature of compression without jacketing:

Temperature of air before compression,				60°	90°
"	" compressed to		15 lbs.	177°	212°
"	"	"	30 "	255°	294°
"	"	"	60 "	369°	417°
"	"	"	90 "	455°	507°
"	"	"	120 "	524°	580°

Formerly the temperature due to compression was lowered by means of an injection of cool water, but its use has been discontinued for the following reasons:

1st. The presence of water on the inside of the cylinder may make it unsafe to run the compressor at speeds which would otherwise be safe and proper.

2d. With the use of water in the compression-cylinder it would be impossible to lubricate the surface of the cylinder. With water in the cylinder and the dirt and grit which usually accompanies it, the piston and cylinder surface wear more rapidly.

THE INTERCOOLER.

The *intercooler* is used only on compressors which have at least two stages of compression. It consists of a large chamber E, Fig. 252, situated between the low-pressue and high-pressure cylinders, which has small pipes in it through which cool water circulates. As already stated, the low-pressure piston compresses the air and raises its temperature. The object of the intercooler is to reduce the air-temperature before it enters the high-pressure cylinder. This reduces the power required to finish the compression, because the reduction of the temperature reduces the pressure and thereby lessens the power of steam required to run the compressor. Imagine a cylinder of any capacity, say 2 cubic feet, containing air at a temperature of say 369° at a pressure of 60 lbs. per square inch. Then if by any means we can cool. the temperature down from 369° to 255°, the pressure is reduced from 60 lbs. to 30 lbs., according to the table. Then the air at 30 lbs. pressure can be further compressed with less power than air at 60 lbs.

The heat produced by compression will be dissipated sooner or later; if not in the compressor, then in the pipe after it leaves the compressor, so that the final pressure will be approximately the same whether the cooling is in the compressor itself or after it leaves it.

Ratio of Volume of Air-cylinders.—The ratio of the volume of the low-pressure air-cylinder to that of the high-pressure cylinder will be: $R = \frac{D^2}{d^2}$; where R is the ratio and D and d the diameters of low- and high-pressure pistons. R differs with different sizes and makes, but is usually taken from 2 to 3.

Fly-wheel.—By taking indicator-cards from both the steam- and air-cylinders of a compressor it will be found that at the beginning of a stroke the pressure in the steam-cylinder is at a maximum (practically boiler-pressure) and the pressure in the intake air-cylinder is at a minimum. This evidently tends to make a very unbalanced state of running, the effect being to run fast at the beginning of the stroke, with a gradual slowing up toward the end of the stroke. In order to balance the compressor and partly counteract this effect the fly-wheels are usually made very heavy, and in fact the whole compressor is made massive in order to withstand the fluctuating strains.

The Air-receiver.—The air-receiver is not a part of the compressor itself, but it is a necessity in any compressed-air system. It is a large tank into which the air passes after leav-

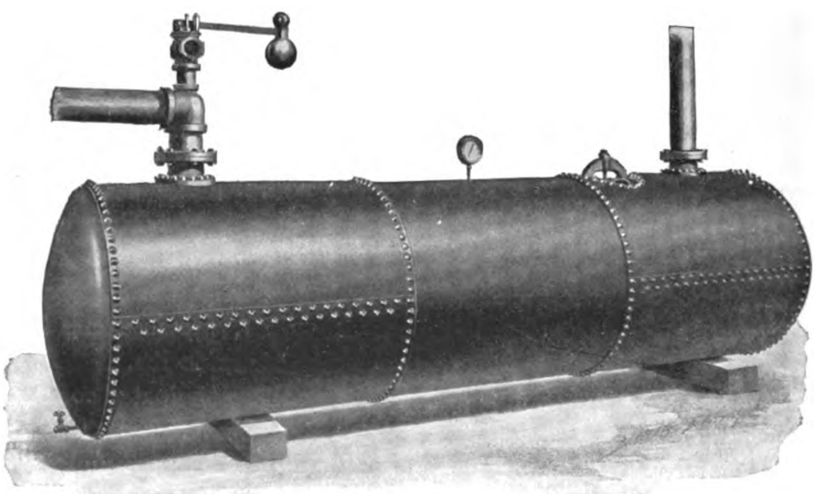


FIG. 255.—Air-receiver.

ing the compressor. It corrects the pulsating effect caused by the periodic expulsions of the air from the compressor into the line of piping. If the distance from the compressor to the motor is great it is best to use two air-receivers, one near the compressor and one near the motor. The first receiver is preferably placed far enough away from the compressor so that a great part of the heat of compression may pass off to the atmosphere through the pipe before the receiver is reached.

Usually about 50 feet is sufficient. Fig. 255 shows a receiver made by the Rand Drill Company. Another use of the receiver is to catch any moisture in the air that may be condensed by the cooling and allow it to be drawn off from the bottom of the receiver. Additional moisture may be deposited in the pipe-line and carried to the second receiver, where it may be drawn off before reaching the motor.

Regulation.—Compressors are usually governed by a ball- or throttling-governor which works in combination with an automatic regulator so that the steam is throttled more or less according to the air-pressure, so that the machine runs just fast enough to supply the demand for air. In this way no more air is compressed than is used, thus economizing the power. Fig. 256 shows the regulating arrangement used on the Rand compressor. It consists of an ordinary ball-governor placed on the steam-pipe with the addition of a small air-cylinder which has a pipe-connection with the exit-pipe of the compressor. When the pressure of air becomes excessive the piston of the air-regulator is pushed up and this in turn closes the governor-valve and diminishes the speed of the compressor. If the air-pressure becomes low the large ball at the right of the figure causes the governor-valve to open and the speed of the compressor to be increased.

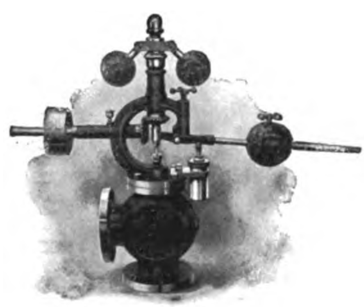


FIG. 256.

MOTORS.

Compressed air may be substituted for steam in almost any of its uses and it can be used in any steam-driven motor, since the laws of expansion of steam and air are practically the same. Compressed air has the advantage of being more easy of transportation than steam for the reason stated before, that is, it can be taken through long distances in pipes without condensation losses. One of the most important uses of compressed air is in mining operations, the rock-drills being operated by compressed air. Fig. 257 shows the details in section of the actuating mechanism of a rock-drill, manufactured by the Rand Drill Company. The valve is a plain slide-valve, always thrown in the same direction in which the piston is moving,

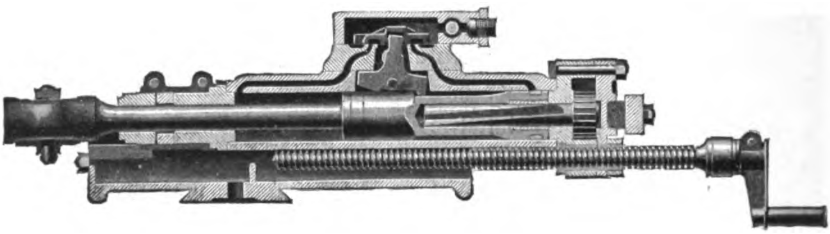


FIG. 257.—Rock-drill.

and is operated by a three-arm rocker which is held in place by a pin. This rocker is placed in a recess of the cylinder between the ends of the double-headed piston, a part of which is in section, and its upper arm engages the valve. As the piston reciprocates it moves the rocker and valve in the direction in which it is going.

Another important use of compressed air is found in the present system of braking trains. In this system a steam-driven compressor, called the air-pump, which compresses a constant air-supply into a main reservoir, is mounted on the locomotive. From this reservoir the air is taken to auxiliary reservoirs, one under each car, by means of the train-pipe. The air is kept at a uniform pressure of 70 pounds per square inch, by means of a governor on the pump, in both the train-pipe and reservoirs. When it is desired to apply the brakes, the engineer by means of the engineer's valve in the cab reduces the pressure in the train-pipe. This actuates a valve, called the triple-valve, placed under each car. The triple-valve on opening allows air to pass into the brake-cylinder under the cars; it moves a piston, which is connected to the brake-shoes by a series of levers. The brakes are released by restoring the pressure in the train-pipe; this causes the triple-valve to open a passage to the atmosphere, from the brake-cylinder, thus allowing the air to escape. A spring returns the piston to its original position. For an emergency stop, the pressure in the train-pipe is reduced suddenly, and the triple-valve opens in such a manner as to allow air from both the train-pipe and the auxiliary reservoir to enter the brake-cylinder, thus applying the brakes with full force.

Compressed air is used for many purposes in railroad shops with as many different means of application, of which limited space forbids a description here.

LAWS OF AIR-PRESSURE.

Air-pressures in a compressed-air system are measured by a common pressure-gauge, similar to a steam-gauge. For convenience the dial sometimes is made to read atmospheres instead of pounds. An atmosphere corresponds to 14.7 lbs. per square inch. The pressure-gauge ordinarily indicates pressures above the atmosphere.

The *Absolute Pressure* is the pressure above a perfect vacuum, or the gauge-pressure plus 14.7 lbs.

Free Air.—This is the term commonly used for air at the atmospheric pressure; that is, of the air which enters the intake-cylinder. The temperature is usually measured by the Fahrenheit scale. *Absolute temperature* is the temperature as indicated by the thermometer plus 461° F. Thus at the temperature 90° by the thermometer, the absolute temperature is $90° + 461° = 551°$. Likewise for temperatures below zero, a temperature of $-20°$ by the thermometer would give $-20° + 461° = 441°$ as the absolute temperature.

The relations of volume, pressure, and temperature of air may be given as follows:

1. The volume of air varies inversely as the absolute pressure when the temperature is constant.

2. The absolute pressure varies directly as the absolute temperature when the volume is constant.

3. The volume varies as the absolute temperature when the pressure is constant.

4. The product of the absolute pressure and the volume is proportional to the absolute temperature.

These laws may be more concisely expressed by means of the equation:

$$\frac{PV}{T} = \frac{P'V'}{T'},$$

in which P, V T are the pressure, volume, and temperature respectively.

The following table shows the weight and volume of air at different temperatures.

VOLUME, DENSITY, AND PRESSURE OF AIR AT VARIOUS TEMPERATURES. (D. K. CLARK.)

Temperature.	Volume of One Pound of Air at Constant Atmospheric Pressure. Volume at 62° F. = 1.	Weight of 1 Cu. Ft. in Pounds.	Pressure of a Given Weight of Air and Constant Volume. Atmospheric Pressure at 62° F. = 1.
Fahrenheit.	Cubic Feet.	Pounds.	Lbs. per Square Inch.
0°	11.583	.0863	12.96
32	12.387	.0807	13.86
40	12.586	.0794	14.08
50	12.840	.0778	14.36
62	13.141	.0760	14.70
90	13.845	.0722	15.49
140	15.100	.0662	16.89
200	16.606	.0602	18.58
250	17.865	.0559	19.98
300	19.121	.0522	21.39
400	21.634	.0462	24.20
500	24.146	.0414	27.01
600	26.659	.0375	29.82

Referring again to the relations of pressure, volume, and temperature as shown in 1, 2, 3, and 4, a diagram, Fig. 258, is produced which will show at a glance conditions as deduced from these laws.

The figures at the left indicate pressures in atmospheres above vacuum, that is, absolute pressures. The corresponding figures at the right denote pressures by the gauge in pounds.

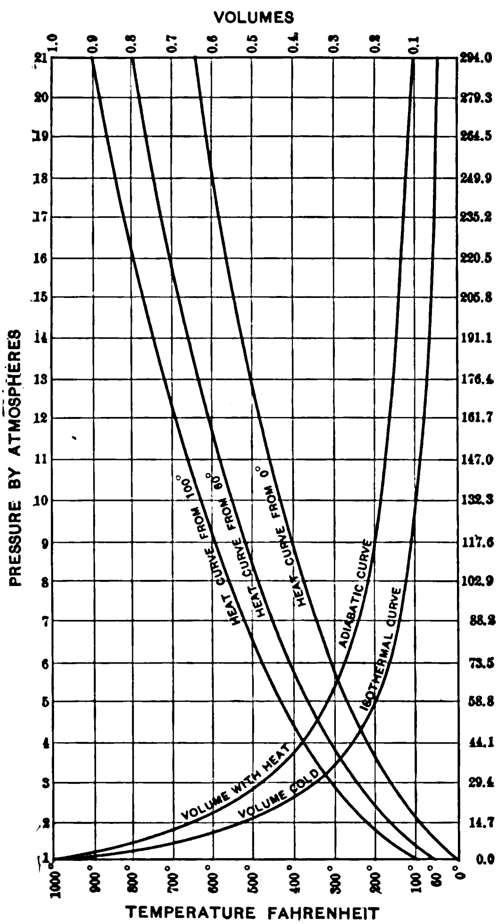


FIG. 258.*

At the top are volumes from one tenth to one. At the bottom, degrees of temperature to 1000° Fahrenheit. The two curves which begin at the lower left-hand corner and extend to the

* From "Compressed Air."

upper right are the lines of compression. The upper one is the "Adiabatic" curve, that which represents the pressure at any point on the stroke, with the heat developed by compression remaining in the compressed air; the lower is the "Isothermal," the pressure curve when the heat of compression is abstracted as fast as compression takes place. The three curves which begin at the lower right-hand corner and rise to the left are heat curves, and represent the increase of temperature corresponding to different pressures and volumes, assuming in one case that the temperature of the air before admission to the compressor is zero, in another 60°, and in another 100°. We see by referring to the adiabatic curve that for a volume of one cubic foot the pressure is 1 atmosphere. Again for a volume of .5 cubic feet the pressure is nearly 3 atmospheres absolute, or say about 28 lbs. by the gauge; and likewise the relation of pressure to volume up to 1 cubic foot are found.

If we compare the compression line from zero with the compression line from 100° it will be noticed that in compressing the air from, say 1 atmosphere to 10 atmospheres, the original difference, which at the start was only 100°, has now become about 200°, and for a pressure of 20 atmospheres about 250°. This shows that it is highly important to supply air to the compression as cold as possible. The greater the temperature of entering air the greater the rise in temperature for equal amounts of compression. Neither the adiabatic or the isothermal curves represent the exact condition of the air, but where there is a system of cooling the air during compression, the lines on the indicator-card may be traced between the adiabatic and isothermal lines. For purposes of calculation, however, the adiabatic may be used, as it is the curve most nearly approached in practice.

Intake-air.—As already stated, the air used for compression should be as cool as possible in order to admit of the best economy. For this reason it is not considered best to use air direct from the engine-room, as it is more or less heated, but it should be taken from outside. The air should also be free from dust, for which reason the outer end of the conduit-pipe should not be too close to the ground.

Volume and Weight of Air per Minute.—The quantity of air required for a compressor depends upon the volume of the low-pressure cylinder and the velocity of the piston.

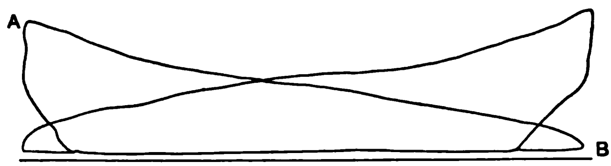


FIG. 259.—Indicator-card taken from the Steam-cylinder of a Compressor.

Knowing the length of the stroke in feet, L, and the area of the low-pressure piston in square inches, A, we have $\frac{LA}{144}$ as the volume in cubic feet swept over by the piston at one stroke. Multiplying this by N, the number of strokes per minute, we have $\frac{LAN}{144}$ as the volume in cubic feet per minute passed over by the piston. This would represent the total volume of air used if there were no clearance. But as this is not generally the case, we make provision for it by multiplying the total displacement per minute by the ratio, $\frac{AB}{EF} = x$, as shown in Fig. 260, which gives $\frac{LANx}{144}$ as the actual volume per minute.

To find the equivalent weight of this volume, multiply by the weight w of a cubic foot of air* as found in the table on page 320. This makes the weight of air per minute $W = \frac{LANxw}{144}$.

* At the temperature of the intake air.

INDICATORS AND INDICATOR-CARDS.

The ordinary steam-engine indicator may be used in determining the conditions of pressure, etc., in the compressor cylinder as in the steam-engine. The accompanying cards were taken by the author from a steam-driven single-stage and single-acting air-compressor.

The card, Fig. 259, was from the steam end and the card, Fig. 260, from the air-cylinder. *A* represents the beginning

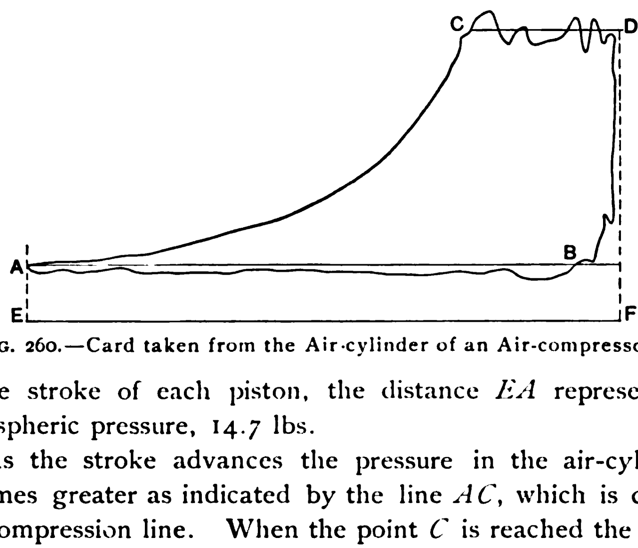


FIG. 260.—Card taken from the Air-cylinder of an Air-compressor.

of the stroke of each piston, the distance *EA* representing atmospheric pressure, 14.7 lbs.

As the stroke advances the pressure in the air-cylinder becomes greater as indicated by the line *AC*, which is called the compression line. When the point *C* is reached the compressed air is released (delivered) from the compressor into the receiver; *FD* represents practically the pressure in the receiver. At *D* the end of the stroke is reached and the return stroke, during which admission is occurring, the pressure being reduced to atmospheric pressure as shown by the line of admission *BA*. By comparing Figs. 259 and 260 the opposite conditions of pressure in the steam- and air-cylinders are made manifest.

Horse-power of Air-compressor.—To find the horse-power from the indicator-card, the same general method is employed as for the steam-engine, that is, of finding the mean height of the card, either by use of a planimeter or by ordinates, then multiplying by the scale of the indicator-spring, which gives Pe, the mean effective pressure. This may then be placed in the

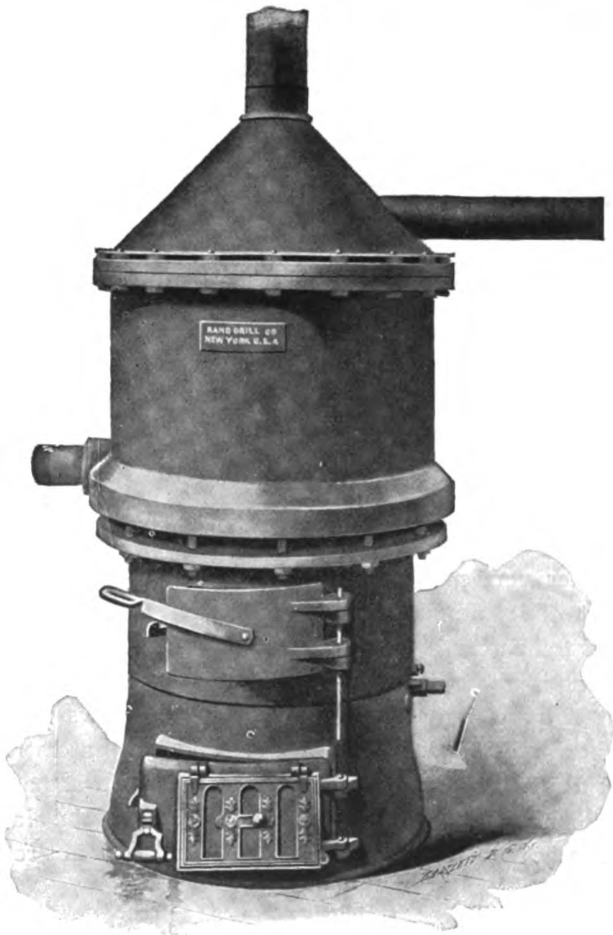


FIG. 261.—Air-heater.

formula $\frac{PLAN}{33,000} =$ H.P. By finding from the cards the H.P of both steam and air ends the friction in the machine may be determined. The air-card will give somewhat less H.P. than the steam-card, the difference being due to friction.

Freezing of the Exhaust in Air-motors.

A great deal has been said in the foregoing upon the fact that air when compressed rapidly attains an increase of temperature, for which reason jacketed cylinders and intercoolers are used in connection with the compressor. On the other hand, when air is expanded, as is the case in any motor, there is a corresponding lowering of the temperature. Now, if the highly compressed air reaches the motor cool, as it is likely to do after passing through long lines of piping, and if the air contains moisture, the expansion will cause the exhaust to have such a low temperature that the exhaust-pipe is liable to become clogged by ice. In fact, quite a great deal of trouble is experienced in this manner. One way to obviate this is to extract all the moisture from the air before it reaches the motor, or at least as much as possible. This can be done by placing two receivers between the compressor and the motor, one near each. The receiver near the compressor catches the free moisture, and the second one drains all the water which has been condensed in the pipe.

Another very practical method is to heat the air close to the point of its admission to the motor. By this means the range of temperature which is required for a given number of expansions is made high enough so that the temperature of the exhaust will be above freezing. Fig. 261 shows a reheater made by the Rand Drill Company. The air enters it at the top and leaves it at the bottom. The air-chamber has a conical shape which allows for the increase of volume of the air as it is heated. By this means the velocity of the air is not retarded.

FRICTION OF AIR IN PIPES.

In an ordinary steam-plant, it is usual, in determining the quantity of steam required, to find the horse-power of the engine and then arrange to supply the steam to develop that horse-power. In compressed-air systems the motor often is so far away from the source of pressure that a great loss of pressure is sustained by the air in passing from one to the other by friction of the air in the pipes, especially if there are many sharp turns in the pipe. The difference found by subtracting the pressure of the air entering the motor from the air leaving the compressor is called the "*difference of head.*" The following formula, given by Mr. Frank Richards, gives the extra head required to overcome the friction in the pipes:

$$H = \frac{V^2L}{10{,}000D^5\alpha};$$

in which D = diameter of pipe in inches;
L = length of pipe in feet;
V = volume of air delivered in cubic feet per minute;
H = head or difference of pressure required to overcome friction and maintain the flow;
α = constant, whose value is found experimentally for different sizes of pipe.

VALUES OF "α" FOR DIFFERENT DIAMETERS OF WROUGHT-IRON PIPE.

1″	.35	5″	.934
1¼″	.5	6″	1
1½″	.662	8″	1.125
2″	.565	10″	1.2
2½″	.65	12″	1.26
3″	.73	16″	1.34
3½″	.787	20″	1.4
4″	.84	24″	1.45

The above formula may also be made use of for calculating the size of pipe required. For this use, however, the handling of the fifth root of D becomes very inconvenient. It is best to use logarithms or to find a table of fifth powers, and, if necessary, interpolate.

PROBLEMS.

1. Find the total piston-displacement (volume swept over by the piston) per minute of a single-acting one-stage air-compressor making 150 revolutions per minute, the stroke being 12 inches, diameter of piston 10 inches.

2. Find the volume in cubic feet of the air required for a double-

acting two-stage compressor making 100 revolutions per minute, the stroke being 24 inches, the diameter of the intake or low-pressure air-cylinder 20 inches, and the clearance 2 per cent.

3. Find the volume in cubic feet of the air required for a double-acting compressor of the two-stage type making 100 revolutions per minute, the stroke being 12 inches, the diameter of the piston being 8 inches, and from which the card shown in Fig. 260 was taken.

4. Reproduce the indicator-card shown in Fig. 259 on tracing-paper, and find the indicated horse-power of the steam end of the compressor from which it was taken, the stroke being 12 inches, bore of cylinder 8 inches, revolutions 125 per minute, and scale of spring 55.

5. Reproduce, as in the above problem, the indicator-card (Fig. 260) from the air-cylinder (single-acting) of the compressor, the stroke being 12 inches, bore of cylinder 8 inches, scale of spring 60, revolutions per minute 125. Find the indicated horse-power of the air-cylinder.

6. From the results found in the above two problems find the H.P. of the steam-cylinder lost in friction. Also the per cent of the indicated horse-power of the steam-cylinder lost in friction. Also determine the efficiency of the machine.

7. Let the air furnished by the compressor in problem 2 be led through a 10-inch pipe a distance of 4 miles to a motor. Find the extra pressure in lbs. per square inch required to overcome the friction.

8. Find the weight of air compressed per minute in problem 3, the temperature of the intake air being 90°.

CHAPTER XXVII.

HOT-AIR ENGINES.

HOT-AIR engines, as the name indicates, use air for the working-fluid. The working effect is produced by the alternate heating and cooling of a body of air.

The air on one side of a piston is suddenly heated, causing it to expand and drive the piston forward.

After it has expanded it is cooled and contracted by some external means, after which it is heated and expanded again. Fig. 262 shows a sectional view and Fig. 263 an elevation of an Ericson Hot-air Engine.

In the sectional view 2 is the power-piston working in the cylinder marked 1, which is open at the top.

3 is another piston of very large volume used to transfer the air from above it to below it and *vice versa*. 6 is a gas-burner which heats the lower end of the cylinder shown at 4.

17 is a water-pump whose piston is operated by the engine. The power-piston 2 and the transfer-piston 3 transform their reciprocating motion to rotary motion, the former through the crank-beam 8 and the latter through the bell-crank 12, better shown in the elevation, Fig. 263.

The piston-rod of the transfer-piston passes through the hollow piston-rod of the power-piston.

The upper end of the cylinder is jacketed so that the water which is pumped all passes out through it, keeping that end of the cylinder cool all the time.

The lower end of the cylinder is surrounded with a non-

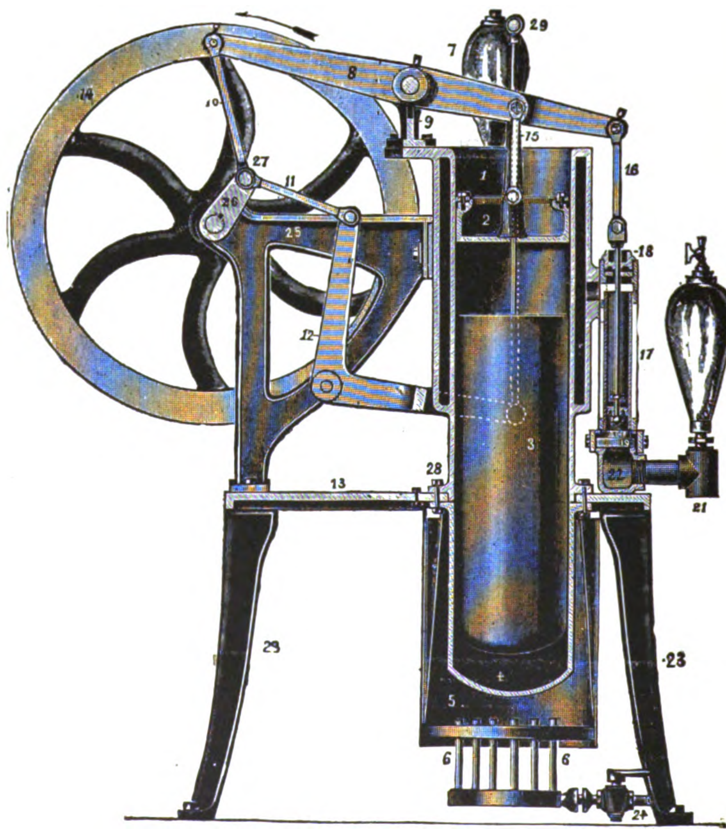


FIG. 262.—Hot-air Engine.

conductor of heat so that it is kept at a high temperature all the time.

The transfer-piston does not fit tight in the cylinder, but

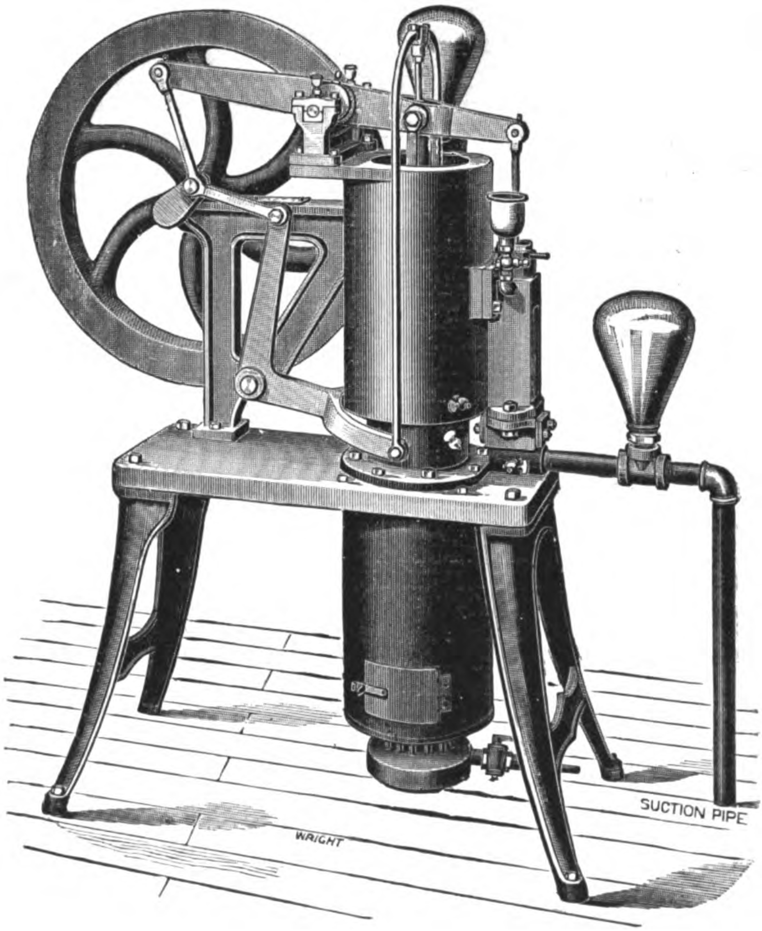


FIG. 263.—Hot-air Engine.

leaves a space around it through which the air may pass to and fro.

The operation is as follows: The lower end of the cylinder is first heated by the gas-burner. The engine must be started by giving it a revolution or two by hand.

The air contained in the cylinder is first compressed in the cool top part, the pistons approaching each other.

This compression causes the cool air to pass by the transfer-piston to the bottom of the cylinder, where it is heated and immediately expands, driving the power-piston upward; the fly-wheel carries the motion to where the expansion by the heat gives the piston another impulse.

It is, plainly, single-acting.

An attachment may be used which will burn wood or other solid fuel instead of gas.

This engine is only used where small quantities of power are required, the running of small pumps being its principal application.

It is especially suited to pumping because of its comparatively low speed. The low speed is due to the fact that it takes some time for the heat to impart its expansive effect to the body of air. The same air is used continuously, and is cooled, compressed, heated, and expanded in the regular order with little noise.

CAPACITY.

The horse-power is found by the formula, $\frac{PLAN}{33,000}$, the same as for the steam-engine, P being obtained from an indicator-card taken from the air-cylinder.

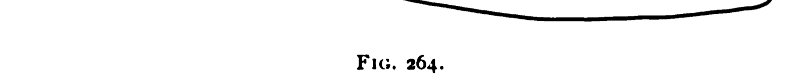


FIG. 264.

The card shown in Fig. 264 was taken from an Ericson Hot-air Engine.

AB is the atmospheric-pressure line. CB is the line of high pressure due to heating of the air in the lower end of the cylinder and made while the power-piston is moving upward; it is noticeable that the end at *B* is lower than the atmospheric line, showing that the air expanded to a pressure lower than atmospheric pressure. *BDC* is made during the downward stroke of the power-piston, showing the gradual compression of the air. Hence *CB* may be called the expansion-line and *BDC* the compression-line.

PROBLEMS.

1. Trace the card shown in Fig. 264, and find the indicated horse-power developed, the revolutions per minute being 47, the bore of the air-cylinder 8 inches, the stroke of the air-piston 4 inches, and the scale of the indicator-spring 10.

2. A pressure-gauge on the discharge-pipe of the water-cylinder in the above test showed a pressure of 32 lbs. per square inch. The diameter of the water-piston is $1\frac{3}{4}$ inches, and its stroke is 9 inches. Find the number of foot-lbs. of work done per minute by the water-piston, remembering that the water-pump is double-acting.

3. Find the horse-power lost in friction, leakage, etc.

4. What is the efficiency of the machine from the conditions given in the previous problems?

TABLE I.

PROPERTIES OF STEAM.

Pressure by the Gauge.	Temperature.	Total Heat above 32°.	Latent Heat.	Volume of 1 Pound 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
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

TABLE I.—*Continued.*

PROPERTIES OF STEAM.

Pressure by the Gauge.	Temperature.	Total Heat above 32°.	Latent Heat.	Volume of 1 Pound of Steam.
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
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

Table I.—*Continued.*

PROPERTIES OF STEAM.

Pressure by the Gauge.	Temperature.	Total Heat above 32°.	Latent Heat.	Volume of 1 Pound of Steam.
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
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

TABLE I.—*Continued.*

PROPERTIES OF STEAM.

Pressure by the Gauge.	Temperature.	Total Heat above 32°.	Latent Heat.	Volume of 1 Pound of Steam.
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

TABLE II.

TABLE FOR WEIRS.*

Inches Depth on Weir.	0	1/8	1/4	3/8	1/2	5/8	3/4	7/8
1..........	0.40	0.47	0.56	0.65	0.74	0.83	0.97	1.03
2..........	1.14	1.25	1.36	1.47	1.59	1.71	1.84	1.96
3..........	2.09	2.12	2.36	2.60	2.64	2.78	2.93	3.06
4..........	3.22	3.38	3.53	3.69	3.85	4.01	4.17	4.35
5..........	4.51	4.68	4.85	5.02	5.25	5.38	5.56	5.74
6..........	5.92	6.10	6.30	6.49	6.68	6.87	7.07	7.27
7..........	7.46	7.67	7.87	8.07	8.28	8.49	8.70	8.91
8..........	9.12	9.33	9.55	9.77	9.99	10.21	10.43	10.66
9..........	10.88	11.11	11.34	11.57	11.80	12.04	12.27	12.51
10..........	12.75	13.15	13.23	13.47	13.72	13.96	14.21	14.46
11..........	14.71	14.96	15.21	15.46	15.72	15.98	16.24	16.49
12..........	16.76	17.02	17.28	17.55	17.82	18.08	18.35	18.62
13..........	18.89	19.17	19.44	19.72	20.00	20.27	20.56	20.83
14..........	21.12	21.40	21.68	21.97	22.26	22.55	22.83	23.13
15..........	23.42	23.71	24.01	24.30	24.60	24.90	25.19	25.50
16..........	25.80	26.10	26.41	26.71	27.02	27.32	27.63	27.94
17..........	28.26	28.57	28.88	29.19	29.51	29.83	30.14	30.46
18	30.78	31.11	31.43	31.75	32.07	32.40	32.73	33.05

* This table gives the number of cu. ft. per minute that will pass over a weir 1 inch wide, and from 1 inch to 18⅞ inches deep. For instance, a weir 1 inch wide and 10⅛ inches deep will pass 13.15 cu. ft. of water per minute.

TABLE III.

FLOW OF COMPRESSED AIR THROUGH PIPES.

Final Pressure at the Point of Delivery—80 Pounds Gauge.

Cubic Feet of Free Air Delivered in Compressed Form at 80 Pounds Gauge-pressure per Minute.

Nominal Size of Pipe in Ins.	1		1¼		1½		2		2½		3		3½		4		4½		5		6		7		8	
Length of Pipe in Feet	50	100	100	300	250	500	250	750	250	750	500	1500	750	2000	1000	3000	1500	3000	2000	4000	2000	5000	3000	10000	3000	10000
Gauge-pressure at the Entrance to the Pipe.																										
80.2	23.7	16.8	35.6	20.7	40.1	28.4	76.0	49.0	143	83	163	95	210	128	263	153	288	211	383	271	558	352	694	379	999	546
80.4	33.1	23.4	49.7	29.0	56.0	39.6	106.1	61.6	200	116	228	132	292	179	368	213	416	295	535	379	781	493	970	530	1397	763
80.6	41.1	29.0	60.9	35.4	68.7	48.6	130.0	75.5	245	142	279	162	358	220	451	262	506	361	656	464	957	605	1189	650	1712	936
80.8	47.0	33.2	70.3	41.0	79.3	56.0	150.1	87.1	283	164	322	187	420	254	521	302	586	417	757	536	1105	699	1373	750	1977	1080
81.0	52.5	37.1	78.7	45.7	88.7	62.8	168.0	97.5	312	184	361	211	463	284	582	338	659	[illegible]	847	600	1260	780	1535	839	2210	1208
81.2	57.5	40.7	86.2	50.1	97.3	68.8	184.2	107.0	348	202	397	229	507	311	642	370	721	511	930	658	1356	857	1685	921	2437	1326
81.4	62.2	44.0	93.2	54.2	105.1	74.3	199.0	115.5	375	218	428	248	549	337	690	400	781	552	1004	711	1464	925	1819	994	2619	1431
81.6	66.5	47.0	99.7	58.0	112.4	79.5	212.9	123.5	402	233	457	266	586	360	738	428	835	591	1074	760	1566	989	1946	1063	2802	1531
81.8	70.6	50.0	105.8	61.5	119.3	84.4	225.8	131.0	426	247	485	282	622	381	783	454	886	627	1139	807	1662	1049	2065	1128	2974	1624
82.0	74.4	52.6	111.5	65.0	125.8	88.9	238.1	138.2	449	261	512	297	657	402	826	479	934	661	1202	851	1753	1107	2178	1190	3136	1714
82.2	78.1	55.2	117.5	68.3	132.0	93.4	250.0	145.1	472	274	537	312	689	422	867	503	1071	694	1262	894	1840	1162	2286	1249	3292	1799

TABLE IV.

VELOCITY OF WATER.

Table giving velocity of water, in feet per second, and the cubic feet of water per minute, to develop one horse-power at 80 per cent duty under heads from 1 to 108 feet.

Head.	Velocity.	Cubic Ft.	Head.	Velocity.	Cubic Ft.	Head.	Velocity.	Cubic Ft.
1	8.02	661.765	37	48.78	17.886	73	68.53	9.065
2	11.34	330.883	38	49.44	17.415	74	69.00	8.943
3	13.89	220.589	39	50.09	16.968	75	69.46	8.822
4	16.04	165.441	40	50.72	16.544	76	69.92	8.707
5	17.92	132.353	41	51.35	16.141	77	70.38	8.594
6	19.65	110.294	42	51.98	15.756	78	70.84	8.484
7	21.22	94.538	43	52.59	15.390	79	71.29	8.377
8	22.68	82.720	44	53.20	15.040	80	71.74	8.272
9	34.06	73.529	45	53.80	14.706	81	72.19	8.170
10	25.36	66.177	46	54.40	14.368	82	72.63	8.070
11	26.60	60.160	47	54.99	14.080	83	73.07	7.973
12	27.78	55.147	48	55.57	13.787	84	73.51	7.878
13	28.92	50.905	49	56.14	13.505	85	73.95	7.785
14	30.01	47.269	50	56.71	13.236	86	74.38	7.695
15	31.06	44.118	51	57.27	12.976	87	74.81	7.606
16	32.08	41.360	52	57.84	12.726	88	75.24	7.520
17	33.07	38.927	53	58.39	12.486	89	75.67	7.436
18	34.03	36.765	54	58.93	12.255	90	76.09	7.353
19	34.96	34.830	55	59.48	12.032	91	76.51	7.272
20	35.87	33.088	55	60.01	11.817	92	76.93	7.193
21	36.75	31.513	57	60.56	11.610	93	77.35	7.116
22	37.61	30.080	58	61.08	11.410	94	77.76	7.040
23	38.46	28.772	59	61.61	11.216	95	78.18	6.966
24	39.29	27.574	60	62.12	11.029	96	78.59	6.893
25	40.10	26.471	61	62.71	10.849	97	79.00	6.822
26	40.89	25.453	62	63.15	10.674	98	79.40	6.753
27	41.67	24.510	63	63.66	10.504	99	79.81	6.685
28	42.44	23.634	64	64.16	10.340	100	80.22	6.618
29	43.19	22.819	65	64.66	10.181	101	80.61	6.552
30	43.93	22.059	66	65.16	10.027	102	81.01	6.487
31	44.65	21.347	67	65.65	9.877	103	81.40	6.425
32	45.37	20.680	68	66.14	9.732	104	81.80	6.363
33	46.07	20.053	69	66.62	9.591	105	82.19	6.303
34	46.77	19.464	70	67.11	9.454	106	82.58	6.243
35	47.45	18.908	71	67.58	9.321	107	82.97	6.185
36	48.12	18.382	72	68.06	9.191	108	83.35	6.127

INDEX.