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STEAM ENGINES

ENGINEERING EDUCATION SERIES

STEAM ENGINES

PREPARED IN THE
EXTENSION DIVISION OF
THE UNIVERSITY OF WISCONSIN

BY
E. M. SHEALY
ASSOCIATE PROFESSOR OF STEAM ENGINEERING
THE UNIVERSITY OF WISCONSIN

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PREFACE

This book on Steam Engines was written to be used as a textbook for correspondence students in the University of Wisconsin Extension Division. It is the third of a series of three textbooks designed for those students who are pursuing a general course in Steam Engineering, the other two being "Steam Boilers" and "Heat."

In this course in Steam Engines we aim to teach the fundamental principles underlying the operation of the steam engine and to do this in as simple and nonmathematical a manner as possible. This is particularly true with those parts which deal with thermodynamic principles. Enough of the practical features of steam engine operation has been given to illustrate the principles, and it is hoped that operating engineers who take this course will be able to supplement from their own experience other applications of the principles presented.

That part of the course dealing with valve gears has been made more complete than other sections because our experience shows that operating engineers usually do not understand the valve gear mechanism of their engines as well as they do other parts.

Most of the material in the chapter on Lubrication was furnished by Mr. R. P. Tobin, Chief of the Technical Department of the Vacuum Oil Company and we take this opportunity to express our thanks for his aid. We wish to take this opportunity also to thank Mr. J. C. White, Chief Operating Engineer for the State of Wisconsin for very valuable suggestions as to the scope of the course and the outline to be followed, also for many useful hints and suggestions about writing the course, and for a careful and critical reading of the manuscript.

E. M. SHEALY.

MADISON, WIS.,
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STEAM TURBINES

STEAM ENGINES

CHAPTER I

PRINCIPLES OF THE STEAM ENGINE

Elementary Principles.—In the steam engine, heat energy is changed into mechanical energy. The pressure of steam is due to the heat which it contains. The steam pressure acts upon the engine piston, causing it to move, and thus changes the heat energy of the steam into mechanical energy. The kind of motion produced is a backward and forward motion of certain parts of the engine. This kind of motion is called a *reciprocating motion* and the parts of the engine which have this kind of motion are called *reciprocating parts.* Other parts of the engine change the reciprocating motion into a rotary motion and thus the engine may turn a flywheel continuously and transmit the motion to other machines.

Figure 1 is a drawing of a steam engine, simplified in order that its principles may be more readily understood. Practically all steam engines operate upon the same principles, hence this explanation will serve for all classes of engines. Only the frame, cylinder, piston, piston rod, crosshead, connecting rod, crank, shaft, and flywheel are shown here. The frame and cylinder are stationary parts and the others are moving parts. The piston, piston rod, and crosshead have a reciprocating motion, and the crank, shaft, and flywheel have a rotary motion.

The piston is moved backward and forward by the pressure of the steam which is admitted first to one end of the cylinder and then to the other. The supply of steam is controlled by means of a valve operated by the engine so as to open and close at the proper time. This valve, which is an important part of the engine mechanism, is not shown in Fig. 1, but will be illustrated and described later.

Referring to Fig. 1, imagine steam to be admitted to the right-hand end of the cylinder. The pressure of the steam acts upon the piston and moves it to the left. The distance which the piston travels from left to right or from right to left is called the *stroke.* The motion of the piston is transmitted through the piston rod to the crosshead, which has the same motion as the piston. One end of the connecting rod is connected to the crosshead and moves in a straight line with it. The other end of the connecting rod is connected to the crank pin, which moves in a circle about the center of the shaft; therefore, the connecting rod changes the straight line motion of the crosshead into the rotating motion of the shaft and flywheel.

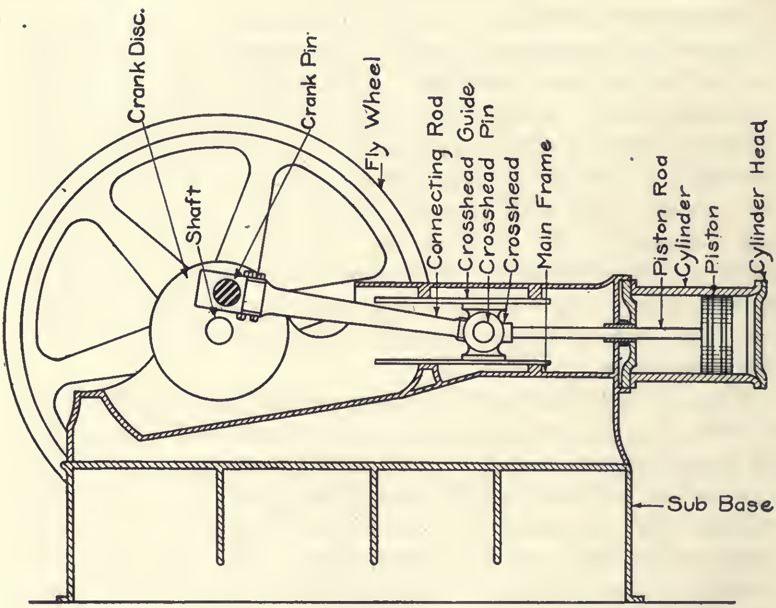


Fig. 1.

When the steam pressure forces the piston of Fig. 1 to the left, the crank pin rotates through the top half of its circular path, moving in the direction of the arrow. As the piston moves to the left, more steam enters the cylinder and maintains a constant pressure upon the piston. At a certain point in the stroke of the piston the valve closes and stops the supply of steam to the right-hand end of the cylinder. This point in the stroke is called the *point of cut-off* or simply *cut-off.* From the point of cut-off to the end of the stroke the steam behind the piston expands and its pressure diminishes. At the end of the stroke from right to left, connection between the right-hand end of the cylinder and the exhaust pipe is opened and the steam begins to be exhausted from the cylinder. The point in the stroke at which steam begins to be exhausted from the cylinder is called *release.*

If the piston should stop just at the end of either stroke, the piston rod, connecting rod, and crank would be in a straight line, and the engine would be *on center* or *on dead center,* as it is sometimes called. In this position steam pressure acting on the piston could not move it, since the pressure would be simply transferred to the bearings of the shaft and there would be no turning effect. However, after completing a stroke, the motion of the parts of an engine is sufficient to carry it past center and the steam pressure will then move the piston forward.

At the beginning of the stroke from left to right, the valve admits steam to the left-hand end of the cylinder while keeping open the connection between the right-hand end and the exhaust pipe. The steam pressure now forces the piston towards the right, and the crank is forced through the bottom half of its circular path, still in the direction of the arrow, thus causing the shaft and flywheel to turn continuously in the same direction. As the piston moves towards the right, the low pressure steam in the right-hand end of the cylinder is forced out by the piston. Steam continues to be admitted to the left-hand end of the cylinder until the point of cut-off in this stroke, after which the steam is expanded behind the moving piston until the end of the stroke, when exhaust commences from the left-hand end. Just before the piston completes its stroke from left to right, the connection between the right-hand end of the cylinder and the exhaust pipe is closed and the steam then remaining in the right-hand end of the cylinder is compressed in order to furnish a cushion for the returning piston. The point in the stroke at which the exhaust passage is closed is called compression.

When an engine passes through a regular series of operations and returns at regular intervals to its starting point, it is said to perform a *cycle.* The parts of the cycle are called *events.* The events in the cycle of a steam engine are *admission, cut-off, release,* and *compression.* The part of the cycle between the point of admission and the point of cut-off is called *admission;* the part between the point of cut-off and the point of release is called *expansion;* the part between the point of release and the point of compression is called *exhaust;* and the part between the point of compression and the point of admission is called *compression.*

The series of operations, admission, expansion, exhaust, and compression occurring in one end of a cylinder make up the cycle for that end. If the cycle is performed in only one end of the cylinder the engine is said to be *single-acting,* but if the cycle is performed in each end of the cylinder, as in the engine described above, the engine is said to be *double-acting.* Nearly all steam engines are double-acting, since a double-acting engine has about twice the power of a single-acting one of the same size. In a double-acting engine the cycles in both ends of the cylinder are being performed at the same time, admission and expansion of one cycle occurring in one end of the cylinder at the same time that exhaust and compression of the other cycle are occurring in the other end of the cylinder.

The end of the cylinder which is towards the shaft or flywheel is called the *crank end,* and the opposite end, or the one furthest from the shaft or flywheel, is called the *head end* of the cylinder. The stroke of the piston from the head end of the cylinder to the crank end is called the *forward stroke* and the stroke from the crank end to the head end is called the *return stroke.*

When the piston is at the end of its stroke it does not touch the head of the cylinder, a small amount of clearance between them being necessary. The space between the head of the cylinder and the piston (when it is at the end of its stroke), together with the volume of the ports, up to the face of the valves, is called the clearance volume, or simply the *clearance.* The clearance is expressed as a percentage of the volume displaced by the piston during a single stroke. For example, the clearance of an engine may be 12 per cent. If a $20'' \times 24''$ engine is under consideration (meaning an engine having a cylinder 20 in. in diameter with a 24 in. stroke) the area of its piston is

$$.7854 \times 20^2 = 314.16 \text{ sq. in.}$$

and the volume displaced during a single stroke is

$$314.16 \times 24 = 7539.84 \text{ cu. in.}$$

$$\text{or } \frac{7539.84}{1728} = 4.305 \text{ cu. ft.}$$

The clearance volume of this engine is

$$12 \text{ per cent. of } 4.305 \text{ or}$$

$$.12 \times 4.305 = .5166 \text{ cu. ft.}$$

Parts of the Steam Engine.—The different parts of a steam engine in their relation to each other are shown in Fig. 2, which represents a common form of steam engine. In this view the

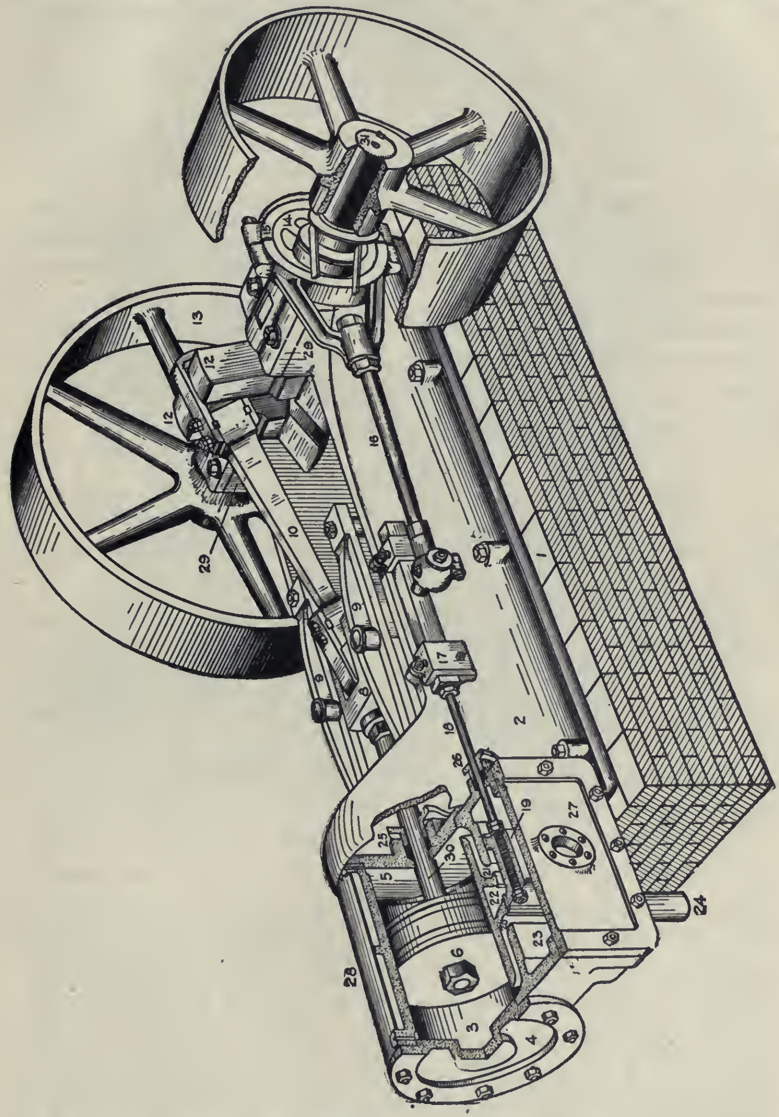


FIG. 2.

cylinder is shown cut away in order to illustrate its interior construction.

In Fig. 2, 1 is the foundation of the engine; 2 is the frame;

3 is the cylinder; 4 is the head end cylinder head; 5 is the crank end cylinder head; 6 is the piston; 30 is the piston rod; 8 is the crosshead; 9 and 9 are the crosshead guides; 10 is the connecting rod; 12 and 12 are the cranks, this being a center crank engine; 31 is the shaft; 13 is the flywheel; 14 is the eccentric; 15 is the eccentric strap; 16 is the eccentric rod; 17 is the valve stem guide; 18 is the valve stem; 19 is the valve which, in this case, is a slide valve; 20 and 21 are the steam ports; 22 is the exhaust port; 23 is the steam chest, which is connected to the steam supply pipe; 24 is the exhaust pipe, which is connected to the exhaust port; 25 is the piston rod stuffing box; 26 is the valve stem stuffing box; 27 is the steam chest cover plate; and 28 is the lagging or covering for the cylinder.

Classification of Engines.—Steam engines may be divided into three classes depending upon their type of valve or method of controlling the speed, and these classes include practically all kinds of engines. These three classes are:

The plain slide valve engine
The automatic high speed engine
The Corliss engine

Any of the above types may be classified in several other ways, among which are:

According to the position of the cylinder, as *horizontal* and *vertical;*

According to the number of cylinders in which the steam is expanded, as *simple, compound, triple expansion,* and *quadruple expansion;*

According to the manner of handling the exhaust steam as, *condensing* and *noncondensing.*

A horizontal engine is one whose cylinder is placed in a horizontal position as illustrated in Fig. 2, while a vertical engine is one having the cylinder placed vertically and directly above the shaft as shown in Fig. 3. Some of the largest engines are a combination of horizontal and vertical, having one horizontal and one vertical cylinder, and the connecting rods from each of these connected to a single crank. This arrangement is used in order to develop a large amount of power in a small space.

A simple engine is one in which the steam is expanded in only one cylinder. In a compound engine the steam is first expanded in one cylinder and the exhaust from this cylinder is led to a second cylinder where it is expanded further. In a triple expansion engine the total expansion of the steam is divided into three parts, each being performed in a separate cylinder, while in a quadruple expansion engine the total expansion of the steam is divided into four parts, each being performed in a separate cylinder. The general names of *multiple expansion* or *compound* are used to designate any engine in which the expansion is performed in more than one cylinder. The reasons for dividing the expan-

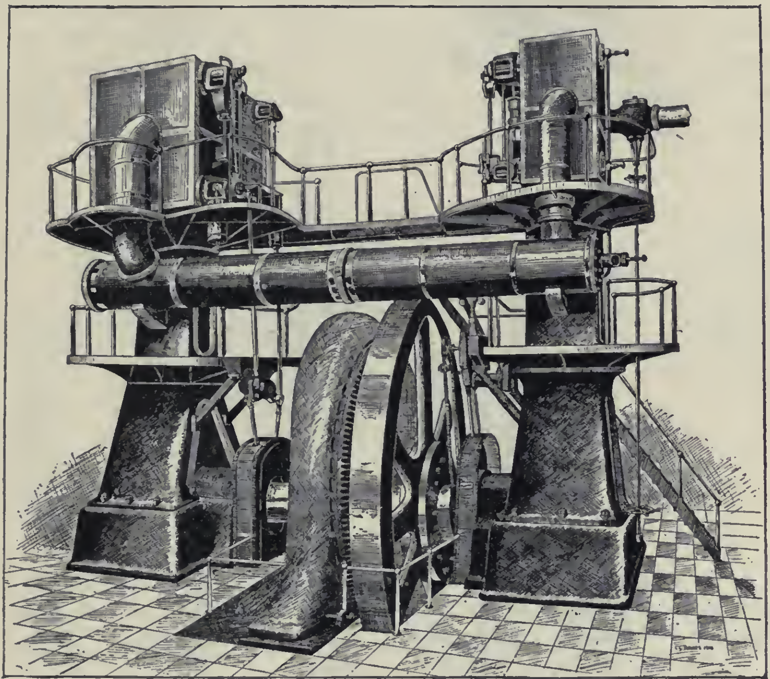


FIG. 3.

sion of the steam into parts and also the construction of multiple expansion engines will be taken up in a later chapter.

A condensing engine is one in which the exhaust steam is changed into water. Since the water thus formed occupies less space than the exhaust steam the back pressure against which the piston must make its return stroke is reduced. In a noncondensing engine the exhaust steam is turned into the atmosphere and the piston must return against the pressure of the atmosphere plus enough pressure to force the exhaust steam through the exhaust pipe and ports. This pressure may amount to from

18 to 20 lbs. per sq. in. absolute, or 3 to 5 lbs. per sq. in. above atmospheric pressure.

The Plain Slide Valve Engine.—This type of engine is named from the kind of valve which is used to distribute steam to the

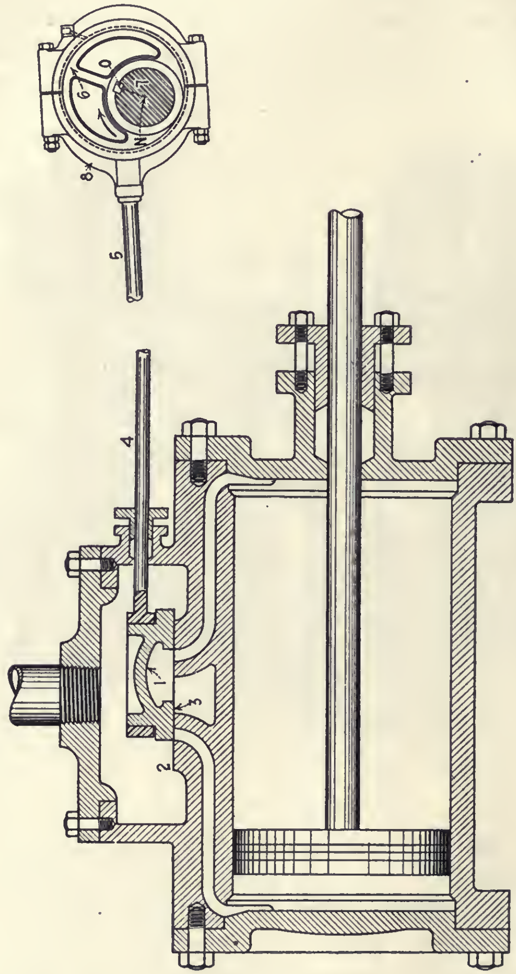


Fig. 4.

two ends of its cylinder, this type of valve being called a *slide valve.* Figure 2 illustrates a common form of plain slide valve engine. The slide valve mechanism is also illustrated in Fig. 2 where it is shown as a part of the engine. The valve and mechanism is again shown in Fig. 4, but in this case the parts are placed differently than in Fig. 2 in order to better show the operation of the mechanism. In Fig. 4 the valve is marked 1; 2 is called the valve seat; 3 is the face of the valve; 4 is the valve rod; 5 is the eccentric rod; 6 is the eccentric; 7 is the shaft; and 8 is the eccentric strap.

The slide valve moves backward and forward over both steam ports and the exhaust port, and is thus able to control the supply of steam to both ends of the cylinder and also the exhaust from both ends. The valve is given its backward and forward motion by the eccentric which is fastened by set screws to the shaft. The eccentric consists of a circular disk of iron having its center, O, in Fig. 4, at some distance from the center of the shaft N. The center of the eccentric thus moves in a circle about the center

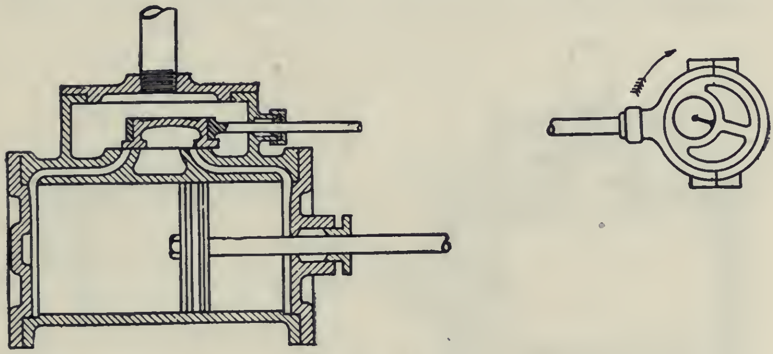


Fig. 5.

of the shaft and in this way has a motion similar to that of a crank with a length NO. A crank which would give the same motion as the eccentric is shown by the dotted lines. The motion of the eccentric is transmitted to the eccentric rod by means of a strap, 8, which passes around the eccentric. The eccentric rod corresponds to a connecting rod and it changes the circular motion of the eccentric into a reciprocating motion, which is transmitted to the valve by the valve rod.

The action of the slide valve can best be explained by considering the series of operations which occur in one end of the cylinder, remembering that similar operations are occurring in the other end but at a different time. In Fig. 4 the piston is at the head end of the cylinder and just beginning its forward stroke, the shaft turning in the direction of the arrow.

In the position shown in Fig. 4, the valve is opening to admit steam to the head end of the cylinder. The steam pressure moves the piston towards the right and the valve is opened wider, which allows steam to flow into the cylinder more freely. The valve soon reaches the end of its travel towards the right and begins to move towards the left, closing the head end steam port. Figure 5 shows the valve just as it closes the head end steam port which cuts off the supply of steam to the cylinder. It will be seen from this figure that the piston has still some distance to go before completing its forward stroke. Figure 5 shows the position of the valve and piston at cut-off. The valve continues to move towards the left, keeping the steam port closed, and the steam expands behind the piston, pushing it towards the right. By the time the piston reaches the end of its forward stroke the inner edge of the valve begins to uncover the head end steam

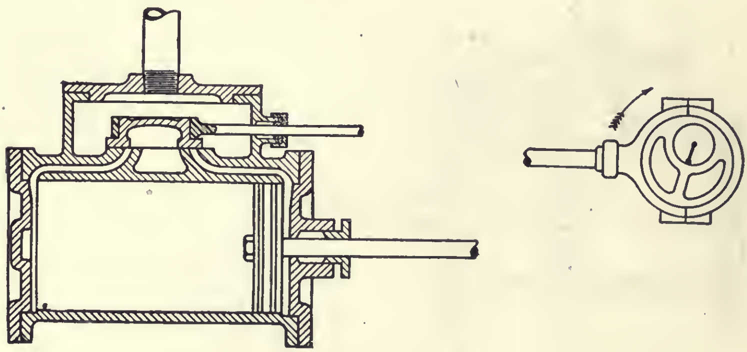


Fig. 6.

port, as shown in Fig. 6, and gives the event known as release. This opens communication between the head end of the cylinder and the exhaust port; then if the steam still has any pressure above that of the atmosphere, this pressure immediately drops to the exhaust pressure.

Steam is now admitted to the crank end of the cylinder and the piston moves towards the left, pushing the spent steam in the head end of the cylinder into the exhaust pipe. This part of the return stroke gives exhaust from the head end of the cylinder. The valve continues to move towards the left, opening the exhaust wider and wider, until it reaches the end of its travel, when it begins to move towards the right and to close the exhaust port. When the piston has reached the point in its return stroke shown in Fig. 7, the valve has moved far enough to the right to close the exhaust port. From this point to the end of the return stroke, the exhaust passage remains closed, and the piston compresses the steam which remains in the head end of the cylinder so that at the end of the stroke the clearance volume of the cylinder is filled with high pressure steam. This completes the cycle in the head end of the cylinder.

By referring to Figs. 4, 5, 6, and 7 it will be seen that the valve is so constructed that at the same time admission and expansion are occurring in the head end of the cylinder, exhaust and compression are occurring in the crank end; and at the same time that exhaust and compression are occurring in the head end, admission and expansion are occurring in the crank end. Thus the cycles for both ends of the cylinder are performed in their

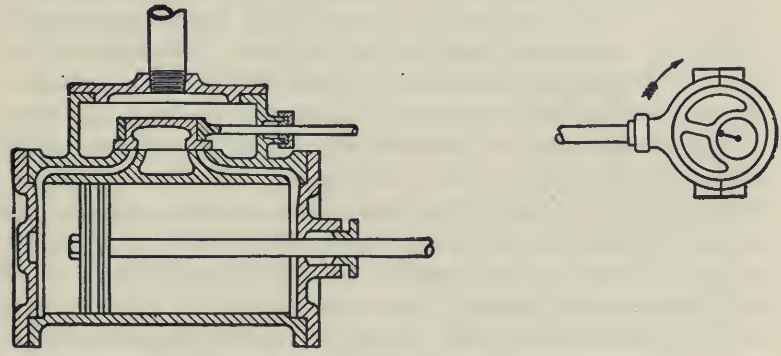


FIG. 7.

proper order and the engine made to run continuously by means of a single slide valve and a single eccentric.

Speed Regulation.—The speed of a steam engine may vary in two ways and from two different causes. First, there may be a variation during each stroke on account of the changing steam pressure in the cylinder. In the first part of the stroke when the steam pressure is high there is a tendency for the speed to increase and in the last part of the stroke when the steam pressure has been reduced by expansion there is a tendency for the speed to be lower. This variation of speed is entirely independent of the load carried by the engine and it would occur whether the load upon the engine was large or small.

The variation of speed during a single stroke is counteracted by the action of the flywheel. It is a well-known fact that when a heavy object is moving it is difficult to change its speed. This property of a body is called its inertia. The inertia of the flywheel is used to counteract the changes of speed during a single stroke. The flywheel absorbs energy during the first part of the stroke when the speed tends to increase and gives it out again during the last part of the stroke when the speed tends to decrease. A heavy flywheel will keep the speed of an engine steadier than a light one and weight concentrated at the rim is more effective than weight nearer the hub, hence flywheels which are intended to steady the speed are usually made with a very heavy rim.

Besides the variation of speed mentioned above, there is also a variation due to changes in the load which the engine carries. Changes in the load affect the speed for a longer period than a single stroke and they cannot be controlled by the action of the flywheel. If the amount of steam supplied to the cylinder at each stroke is constant and the load increases, the speed of the engine will decrease until the power developed in the cylinder balances the load on the engine, and if the load decreases the speed increases until the power developed in the cylinder again balances the load. The load on most engines is varying all the time and, as it is desirable to keep the speed constant, some means must be provided for varying the amount of steam supplied to the engine according to the load it is carrying, so that when the load increases the amount of steam supplied will be greater and when the load decreases the amount of steam supplied will be less. This is called *governing* the engine, and the mechanism for controlling the steam supply is called a *governor*.

The governor of a steam engine may control the speed by changing the steam pressure or by changing the volume of steam admitted to the cylinder during the period of admission. In the first method the governor operates a throttle valve placed in the main steam pipe where it enters the steam chest and the partial closing of this valve reduces the steam pressure. By this means the governor controls the steam pressure acting upon the piston to agree with the load on the engine. This kind of governor is called a *throttling* governor.

In the second method mentioned above the governor is arranged so as to change the point of cut-off and thus change the volume of steam admitted to the cylinder to agree with the load. When the load increases the governor makes the point of cut-off occur later in the stroke thus admitting more steam to the cylinder, and when the load decreases the point of cut-off occurs earlier, admitting a smaller volume of steam to the cylinder.

An engine whose speed is regulated by throttling or reducing the pressure of the steam supply uses a large amount of steam in proportion to the work it performs, or is inefficient, because the full pressure of the steam is used only when the load is greatest and for any smaller load a portion of the steam pressure is wasted. The method of governing in which the volume of steam admitted to the cylinder is changed is more economical because all of the steam admitted to the cylinder is used at the full boiler pressure.

In the plain slide valve engine the position of the eccentric determines the part of the stroke at which cut-off occurs. Since the eccentric is fastened to the shaft the point of cut-off occurs at a fixed point in the stroke, therefore the speed is governed by the throttling method.

The plain slide valve engine is usually designed to run at slow or medium speeds, with a stroke somewhat greater than the diameter of the piston. Most of them are of small size, since they are uneconomical in the use of steam. They are simple in construction and cheap in cost, hence are much used where only a small amount of power is needed and where expert attendance is not obtainable. With ordinary care, they last a long time and do not easily get out of order. This type of engine uses from 35 to 60 pounds of steam per hour for each horsepower developed.

Automatic High Speed Engines.—The type of engine known as the automatic high speed engine is also a slide valve engine, but its valve differs somewhat from that used in the plain slide valve engine and it also differs in other details of construction.

The valves used on the automatic high speed type of engine are of much better construction than those used on the plain slide valve engine. One kind of valve commonly used on these engines is illustrated in Fig. 8. This kind of valve is known as a balanced valve because it has a balance plate which prevents the steam pressure from acting on the back of the valve. The plain slide valve, which has no balance plate, has the steam pressure in the steam chest acting on the entire area of the back of the valve. As the area of the valve is large, the valve is pressed against its seat with an enormous pressure, requiring a large amount of work in moving the valve. The large area of the valve also makes it difficult for oil to get between the valve and its seat to lubricate it. Some valves have balance plates which cover about

80 per cent. of the area of the valve, leaving 20 per cent. of its area upon which the steam pressure may act. This part of the steam pressure is sufficient to keep the valve properly seated

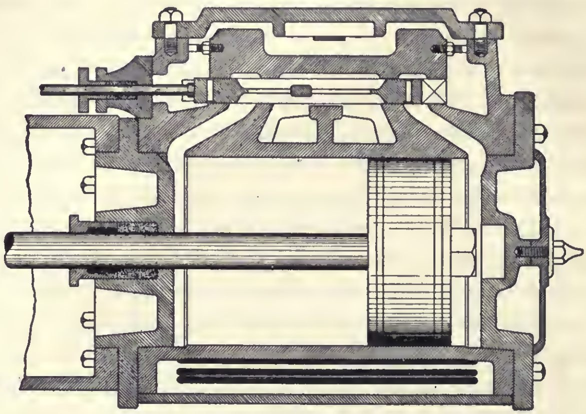


FIG. 8.

and is not enough to cause undue friction. The balance plate is adjustable to allow for wear of the valve, being held in position by screws which press against the sides of the steam chest and hold it in place.

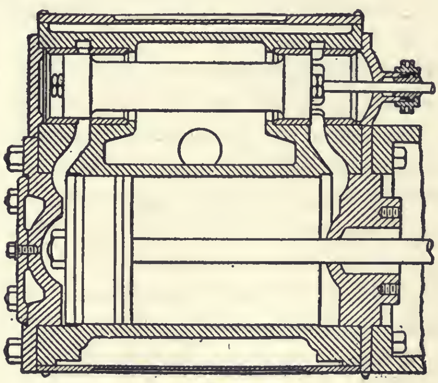


FIG. 9.

Another kind of valve, commonly used on automatic high speed engines, is illustrated in Fig. 9. This form of valve is made in the shape of a spool and is like a slide valve which has been curved into a cylindrical form. Its motion is the same as that of the plain slide valve. In the valve shown here, which is called a piston valve, steam enters the steam chest at the central part of the valve and is admitted to the cylinder past the inner edges of the valve, exhaust taking place past the outer edges. This is the reverse of the manner in which a plain slide valve admits and exhausts steam. The steam ports completely surround the valve so that a large port opening is secured with a small movement of the valve. This valve does not require a balancing plate because the steam pressure acts on all sides of the valve equally, thus making it perfectly balanced.

In the plain slide valve engine the volume of steam admitted to the cylinder at each stroke of the piston is constant, the speed of the engine being controlled by changing the admission pressure

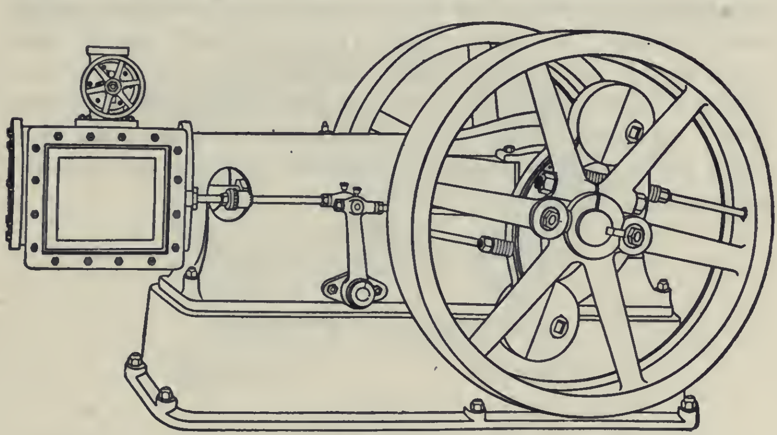


Fig. 10.

of the steam. In the automatic high speed engine, the admission pressure of the steam is constant, the speed being controlled by changing the point of cut-off, and thus regulating the volume of steam admitted to the cylinder to suit the amount of work being done. The latter is the more economical method of controlling the speed because the full steam pressure is utilized all the time. The point of cut-off is changed in the high speed engine by means of a governor which is attached to the eccentric in such manner that the eccentric may be shifted around on the shaft. The mechanism for doing this will be fully described in the chapter on governors.

Figure 10 is a side view of an automatic high speed engine, and serves to show the proportion of its parts. It will be noticed that the engine is self-contained, that is, it rests on a bedplate which forms a foundation for it. These engines are usually short, the parts being grouped closely. As compared with a plain slide valve engine, the automatic high speed engine has a shorter cylinder and connecting rod in proportion to the diameter of the cylinder. It is made in these proportions because a piston speed of about 500 to 700 feet per minute is desirable for all classes of engines and in order to secure this piston speed the length of stroke must be short if the number of revolutions per minute is large.

The automatic high speed engine is made for speeds up to about 350 revolutions per minute and in size up to about 600 horsepower. It has a close speed regulation at all loads and is, therefore, well adapted for direct connection to electric generators, a class of work which requires high speeds and close speed regulation. These engines are also often connected to line shafting by means of belts, and used for general power purposes. They are more efficient than the plain slide valve engine, using from 30 to 40 pounds of steam per hour per horsepower.

CHAPTER II

CORLISS AND OTHER ENGINES

Corliss Engines.—The Corliss engine is an entirely different type from either the plain slide valve or the automatic high speed type. This type of engine, like the others, is named from its type of valve, which is known as the Corliss valve.

The Corliss valve is cylindrical in shape and is placed across

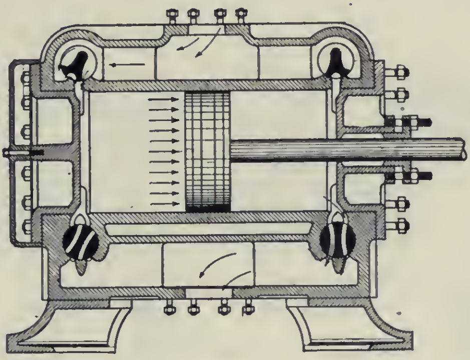


Fig. 11.

the cylinder instead of parallel with it. There are four of these valves for each cylinder, one admission valve for each end of the cylinder and one exhaust valve for each end. Each of these valves rotates about its axis instead of moving backward and

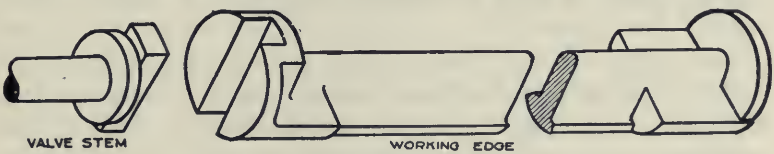


Fig. 12.

forward parallel with its axis, as does the piston type of slide valve. The Corliss valve does not turn through a complete revolution, but oscillates back and forth through an angle only large enough to uncover the port.

A cross section of a cylinder fitted with Corliss valves is shown in Fig. 11, and a view of one of the valves removed from the cylinder is shown in Fig. 12. It will be observed that, by this arrangement of valves, the ports are made short and the clearance reduced. The ports extend across the cylinder and are about equal in length to the diameter of the cylinder. Steam pressure acts upon the backs of the valves keeping them pressed against their seats, but the friction is small since the valves are small and they travel only a short distance in rotating through a small angle. The travel of the valve, in order to gain a full port opening, is sometimes further decreased by having two openings through the valve, or making them "double ported" instead of having the steam pass only one edge of the valve. By having

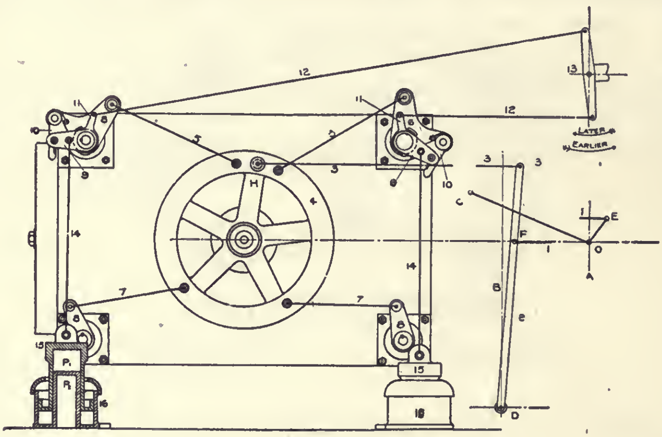


Fig. 13.

two ports through each valve, the valve need travel only one-half as far for a given port opening as would a single ported valve.

The Corliss valve is operated from an eccentric, the same as a slide valve, but it differs from the slide valve in that the admission valves are connected to the eccentric only at those times when they are being opened. The exhaust valves are connected to the eccentric at all times, but the mechanism for moving them is such that they move through a very small angle, thus reducing friction.

The mechanism by which the Corliss valve is operated is shown in Fig. 13 in which some of the rods are represented by single lines in order to make the drawing clearer. As shown in this drawing, the admission valves are at the top of the cylinder and the exhaust valves at the bottom.

The eccentric rod is connected at *F* to the rocker arm *B*, which is pivoted at *D*, so that the end, *G*, of the rocker arm swings back and forth through a small angle. A wheel, 4, called a wrist plate, which is free to turn, is placed on the side of the cylinder, and a point, *H*, on its rim is connected by a rod to the upper end, *G*, of the rocker arm. The wrist plate, therefore, oscillates back and forth through the same angle as the upper end of the rocker arm. All of the valves are connected to the wrist plate by means of the rods 5, 5, and 7, 7, and take their motion from it.

The exhaust valves are fitted with round stems which pass through stuffing boxes in the side of the cylinder, so they are

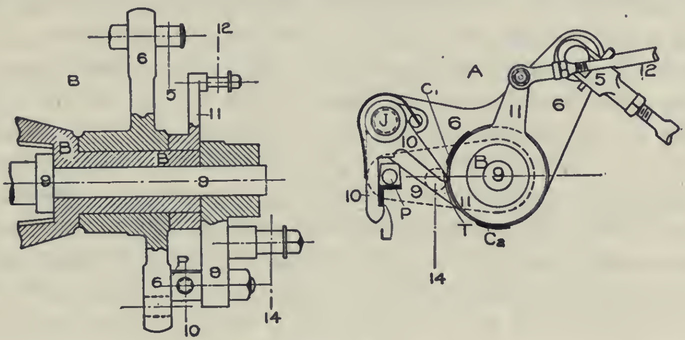


Fig. 14.

free to turn without allowing steam to leak past them. To the ends of the exhaust valve stems are fitted cranks, 8 and 8, which are connected by rods, 7 and 7, to the rim of the wrist plate. The oscillation of the wrist plate causes these cranks to move back and forth through a small angle, thus opening and closing the exhaust valves.

The admission valves are not connected directly to the wrist plate, but are arranged so that they are connected to it while the valves are being opened, and disconnected while the valves are being closed. The device by which this is accomplished is shown in detail in Fig. 14. In this figure, the round valve stem projecting through its stuffing box is shown at 9 in the figure at the left. The end of the valve stem is supported in a yoke, *B*, called a bonnet, which is bolted to the side of the cylinder. The end of the bonnet is finished round and upon it are fitted two cranks, 6 and 11, which are free to turn. The crank 6 is L-shaped and is called a bell crank. One arm of the bell crank is connected by a rod, 5, to the wrist plate; the other end carries a V-shaped hook, 10, called a steam-hook, which is pivoted at J. The end of the valve stem carries a crank, 9, which is keyed to it beyond the end of the bonnet. The end of this crank has upon it a small square steel block, P, and there is another small steel block, L, upon the end of the hook-claw. As the wrist plate oscillates back and forth the bell crank, 6, moves over far enough so its steel block, L, hooks under the steel block, P, on the valve stem crank. As the bell crank moves back, the valve stem crank is pulled with it, thus opening the valve. At a certain point in the backward movement of the bell crank the governor causes the steam-hook, 10, to release the valve stem crank and the valve is closed, which causes steam to be cut off from that end of the cylinder. The crank, 11, is carried by the bonnet and is free to turn upon it. This crank is connected to the governor by the rod 12. The crank 11 has a small steel block, C_1, fastened on its hub, and the position of this steel block is determined by the position of the governor which is connected to the crank 11. As the V-shaped hook-claw moves backward after picking up the valve stem crank, the end, T, of the hook-claw strikes the steel block on crank 11 and causes the hook-claw to release the valve stem crank. Referring to Fig. 13, it will be seen that the wrist plate is near the extreme of its travel towards the right and that the hook-claw on the right-hand end is just ready to pick up the valve stem crank. At the same time the valve on the left-hand end has been opened and is ready to be released by the governor.

The admission valves are made to close quickly by means of a dashpot, which is shown at 15 in Fig. 13. The dashpot consists of a cylinder 16 and a piston P_2. The piston is connected by the rod 14 to the valve stem crank so that as this crank is raised upon opening the valve, the piston in the dashpot is also raised. Raising the dashpot piston causes a vacuum in the dashpot which exerts a powerful suction on the piston and closes the valve quickly when it is released, thus cutting off the supply of steam suddenly.

It will be seen from the above description of the valve mechanism that the method of governing the speed of the Corliss engine is by changing the volume of steam admitted to the cylinder at each stroke which is the most economical method of governing.

The automatic high speed engine uses this method of governing, also; but in this case the arrangement of the valve mechanism is such that when the point of cut-off changes, the point of compression changes also, as will be seen after this type of valve mechanism is studied in a later chapter. In the automatic engine the valve closes somewhat gradually which causes the admission pressure to decrease gradually up to the point of cut-off, while in the Corliss engine the valve opens wide at admission, remains wide open during admission, and closes suddenly at cut-off, thus giving full steam pressure upon the piston during the entire admission. At light loads the valve does not open full port area, but it generally opens enough to give full initial pressure. Also, since the steam valves are independent of the exhaust valves, the point of compression does not change when the point of cut-off changes in regulating the speed. On some Corliss engines the exhaust valves are not only independent of the steam valves but they are operated from a separate eccentric. The reasons for this will be explained fully in a later chapter.

There are many modifications of the Corliss valve mechanism, but the principle is the same in all of them, that is, the valve is opened quickly by the eccentric, is disengaged from the eccentric at the proper time by the governor, and closed quickly by a vacuum dashpot.

The valve mechanism is the distinctive feature of the Corliss engine, as aside from it this engine differs but little from any other type of slow speed engine. The invention of the Corliss valve mechanism is the greatest development that has taken place since the invention of the present form of steam engine by James Watt in 1769. The high efficiency of the Corliss engine is due to its form of valve mechanism more than to anything else as it permits an early cut-off with large expansion of the steam and does not throttle the steam pressure during admission or exhaust.

A general view of a Corliss engine is shown in Fig. 15, which serves to give an idea of the proportion of its parts. It will be observed that this type of engine has a somewhat longer cylinder in proportion to its diameter and also a longer connecting rod than other types of engines. This gives the whole engine an appearance of considerable length in proportion to its height.

The complicated valve mechanism used on the Corliss engines makes it necessary to run them at a relatively low speed in order

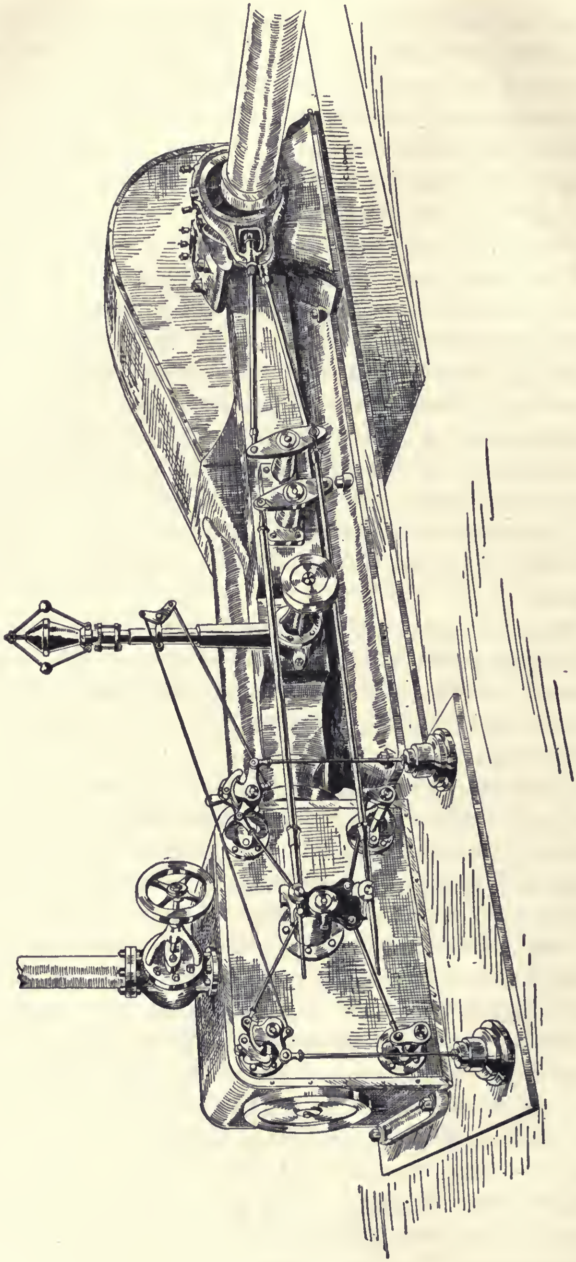


Fig. 15.

for the various parts to adjust themselves and for the dashpot piston to work properly. In order to maintain a proper piston speed with the slow rate of revolution, the cylinder is made long in proportion to its diameter. Corliss engines are rarely made to run at a higher speed than 100 to 125 revolutions per minute, and in the larger sizes the speed is even slower than this.

The Corliss engine is the most efficient type of steam engine. It rarely uses over 25 pounds of steam per hour per horsepower and in the larger sizes its steam consumption is much less than this. This type of engine is made in sizes from 100 to 12,000 horse power. The Corliss engine is particularly adapted to running mills and for other power purposes on account of its smooth running qualities and its close speed regulation. The speed of the Corliss engine is controlled by changing the point of cut-off to suit the load, thus controlling the volume of steam admitted to the cylinder, the admission pressure of the steam remaining constant.

Besides the three main types of engines mentioned above there are various modifications of these types which are used extensively and which are important on this account. Some of the principal ones of these are described below.

Nonreleasing Corliss Engine.—The nonreleasing Corliss engine is a type of medium and high speed engine which has been developed in recent years. It is a combination of the automatic high speed and the Corliss engines and it is used in the same kind of service as the automatic engine, that is, for direct connection of electric generators and for general power purposes by belting to line shafting. On account of its high speed this engine has the general shape and proportions of the automatic high speed engine, as will be seen from Fig. 16, which shows one of these engines.

Any engine which runs at a high speed must have a positive connection between the eccentric and valve because there would not be time for a disengaging mechanism to operate properly. For this reason the nonreleasing Corliss engine has its admission valves, which are of the Corliss type, connected directly to a reach rod which is operated by an eccentric rod from the eccentric connected to the governor. With the Corliss type of valve connected directly to the governor eccentric, the nonreleasing Corliss engine has some of the characteristics of the Corliss engine and some of the automatic high speed engine.

The method of governing the speed of the nonreleasing Corliss engine is the same as that used with the automatic high speed engine, that is, the point of cut-off is changed by shifting the eccentric which operates the admission valves around on the shaft. This does not change the point of compression because the exhaust valves are operated by a separate eccentric which is fastened to the shaft. This gives a steam distribution to the cylinder even better than that given by the automatic high speed engine with its slide valve, because release and compression can be fixed at the most advantageous points and they remain unchanged when the cut-off is changed. A double ported valve, which reduces the necessary movement of the valve for the same port opening and also further reduces the friction, is used in some of these engines. The type of valve used on this engine permits

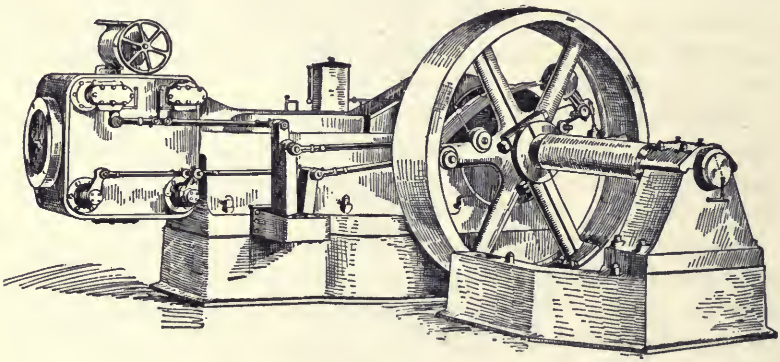


Fig. 16.

short ports which reduce the clearance volume and thereby increase the economy of the engine. However, at least 6 per cent. clearance is needed for quiet running.

In a slide valve engine the same valve is used both for admission and exhaust. The exhaust steam, having a lower temperature than the admission steam, chills the valve so that when the next admission occurs, a part of the admission steam is condensed by coming in contact with the cooler valve. This condensation, though small is avoided in the nonreleasing Corliss engines since there are separate admission and exhaust valves.

Since the nonreleasing Corliss engine combines some of the advantages of the Corliss engine with those of the automatic high speed engine, its steam consumption per horsepower per hour is slightly greater than that of the Corliss engine but less than that of the automatic high speed engine. It is made in sizes up to about 600 horsepower and for speeds up to about 350 revolutions per minute.

The Locomotive.—The locomotive, familiar to everyone, is a type of plain slide valve engine, differing from the ordinary plain slide valve engine in having the valve mechanism arranged so that the direction of rotation may be reversed at will. This type of valve mechanism will be studied in detail in a later chapter. Instead of having a single engine the locomotive has two complete engines, one on each side connected to the main driving shaft by cranks placed 90° apart so the locomotive may be started from any position even if one engine is on dead center.

The valves used on locomotive engines are of the balanced type, some of them being flat valves with balance rings or plates on the back, and some being cylindrical or piston valves which are completely balanced. In both cases the usual location of the valve is on top of the cylinder where it is accessible to the engineer.

Unlike the engines perviously mentioned, the locomotive has no flywheel, its function being performed by the driving wheels and by the weight of the boiler resting on them. Neither has the locomotive engine a governor since it is not intended to run at constant speed. Its speed is controlled by hand to suit the load which the engine is pulling. There are two means of regulating the speed: first by means of a hand-operated throttle valve which controls the pressure of the steam admitted to the cylinders; and second, by changing the point of cut-off, which may be done by means of the reversing mechanism.

Until recent years the locomotive consisted of two simple engines. The demand for greater power has brought about the development of the compound locomotive in which the steam is first expanded in a high pressure cylinder and then in a low pressure cylinder, just as in the stationary compound engine. In some locomotives the high pressure cylinder is on one side and the low pressure cylinder on the other. This arrangement requires some provision whereby high pressure steam may be admitted to the low pressure cylinder in order to start when the high pressure side is on center. Another common arrangement of the cylinders is to place one high and one low pressure cylinder on each side of the locomotive, thus making the locomotive consist of two complete compound engines. In this arrangement of cylinders, the high pressure cylinder is placed directly above and parallel to its low pressure cylinder, the piston rods from both cylinders connecting to a single crosshead. This gives a more compact and powerful engine than the other arrangement of compound cylinders.

A locomotive is a complete power plant in itself consisting of both boiler and engine, and, in some cases, of a feed water heater and superheater also. The amount of power developed is large compared with the size of the boiler and engine, being as much as 2000 Hp. in some cases. When it is considered that this power is sometimes developed with a steam consumption of about 20–24 pounds of steam per horsepower per hour, the efficiency of these machines is wonderful.

Marine Engines.—Engines used on steamships form another distinct class of steam engines. These are also of the plain slide valve type and, like the locomotive, are provided with a mechanism for reversing the direction of rotation of the engine. Marine engines are vertical and are invariably multiple expansion in order to secure a large amount of power within a small space. The use of several cranks also gives a more uniform turning effort to the shaft. The engines are compound, triple or quadruple expansion depending largely upon the size of the engine. The cylinders are placed directly above the crank shaft with their axes vertical. In the larger vessels two propellers are used, each one on a separate shaft driven by a separate engine.

There is no need for a governor on a marine engine as the resistance offered by the water to the revolving propeller increases as the speed increases and this prevents the engine from racing. Slower speeds are secured by partly closing a throttle valve which reduces the pressure of the steam supplied to the cylinders.

CHAPTER III

PARTS OF THE STEAM ENGINE

The Frame.—The frame of an engine supports all of the working parts and holds them in their proper relative positions. The form of the frame, especially on the larger sizes of engines, is determined by the type of engine and the purpose for which it is to be used; thus a Corliss engine used on rolling mill work would have an entirely different kind of frame than would the same type of engine when used for general power purposes, such as supplying power for a factory.

The frame of the automatic high speed engine is the simplest of all engine frames. This frame, as shown in Figs. 10 and 16, is made to rest on a cast-iron sub-base instead of on a masonry foundation, hence the bottom edge of the frame is made in the form of a rectangle to fit the base. As most of these engines are of the center crank type, the whole frame is of rectangular shape, but is smaller at the top than at the bottom, as this shape is best adapted to supporting the bearings at each side, and to leaving space between them for the crank. Such engines usually employ splash lubrication for the crank pin and crosshead, that is, oil is placed inside the frame so that the crank can splash into it at each revolution and throw some of the mixture on the rubbing parts. This makes it necessary to shape the frame so as to contain the oil and also to cover the guides, connecting rod, and cranks to prevent the oil from being splashed out. The rectangular shape of the frame is well adapted to splash lubrication, as the frame itself makes a trough for containing the oil by merely extending the casting across the bottom. The cast-iron bottom of the frame serves not only to contain the oil but strengthens the frame laterally. Even when splash lubrication is not employed, provision must be made for catching the oil that drips from the various bearings as it would soon destroy masonry foundations or soak into wooden floors around the engine.

The cylinder of the high speed engine usually overhangs the frame, hence some provision must be made for fastening it to the frame. This is done by a ring of bolts at the crank end of the cylinder. These bolts are not used for aligning the cylinder with the frame and do not act as dowel pins; they are used only to hold the cylinder close up to the frame. The cylinder is aligned by means of a projection which fits accurately into a bored recess. In most cases the shoulder is on the frame and the recess is in the cylinder, but sometimes this arrangement is reversed, the shoulder being on the cylinder and the recess bored in the frame. The bolts for holding the cylinder may be outside the frame, or they may be inside and just at the end of the guides. The outside bolts are to be preferred because they are easier to reach in case the cylinder is to be removed for repairs.

The frame of the Corliss engine has experienced decided changes in shape within recent years. Formerly the girder frame, as shown in Fig. 17, was most commonly used. The girder frame takes its name from the fact that the part of the frame between the main bearings and the cylinder does not rest directly on the foundation but acts as a girder, supported by a stand or legs placed about the middle of its length. The cross section of the girder frame, between the guides and the main bearings, is in the shape of the letter T to give it greater stiffness for the weight of metal in it. The guides form a part of the frame itself, being either bored to a circular shape or planed to a V-shape, hence this part of the frame is stiffened by the guides. The girder frame has the advantage of being light, and for small engines, particularly where the load is fairly uniform, it is very satisfactory; but for heavy and widely varying loads a more rigid and stronger frame is desirable.

The latter class of service has brought into use on the larger sizes of Corliss engines the "heavy duty" frame, one form of which is shown in Fig. 18. This frame is built in one piece from the cylinder to the main bearings and is box shaped to allow it to rest squarely on the foundation throughout its length, but it is cut away on the outside from the guides towards the main bearing. The cylinder is bolted to the end of the frame and is supported independently. The guides are formed in the frame itself and the frame is formed into a complete circle at both ends of the guides to give greater strength and stiffness. In some heavy duty frames, the part forming the guides is made separate from the part containing the main bearing, being made in the shape of a barrel and bolted to the rear section and to the cylinder, but not resting directly on the foundation.

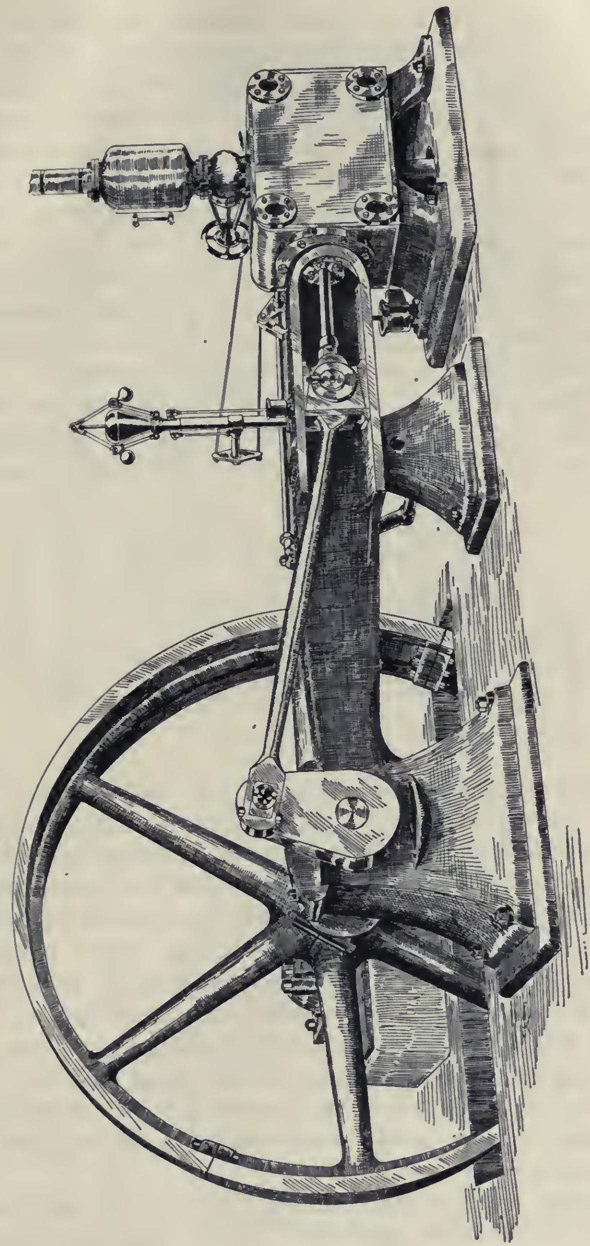


Fig. 17.

The frames of vertical engines are usually A-shaped, as shown in Fig. 19. In the larger sizes they are made in two parts, the upper part or guide section being in one piece and bolted to the lower part or housing. The bottom of the frame is usually a rectangle in shape to permit it to rest squarely on the foundation. A web is cast entirely across the bottom to catch oil and to prevent it from soaking into the foundation. The cylinders are supported by the guide section and are bolted to it. In some marine engines, which are always of the vertical type, the A-shaped frame is modified, the cylinders being supported directly on steel columns and the guides bolted between them. This construction is used to secure lightness with strength.

The Cylinder.—A 20″ × 24″ cylinder of a locomotive is shown in Fig. 20. This one was chosen here because it is the cylinder of a plain slide valve engine, and, with its long ports and short

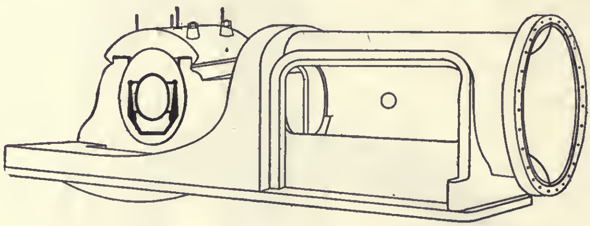


Fig. 18.

valve face represents one extreme; the short-ported Corliss engine is the other extreme. The cylinder body is a continuous shell except where the ports cut through it; and around each end there is a stout flange to which the cylinder heads are bolted.

It will be noticed that the cylinder is bored out to two different diameters, the ends being from ⅛″ to ¼″ larger in diameter than the central portion. The central portion is called the "bore" and the larger end portion the "counterbore." The length of the bore is such that the piston rings slightly overtravel it in order to prevent wearing shoulders in the bore. The counterbore is made large enough to allow reboring the cylinder two or three times to bring it back to a true cylindrical form after it has become worn by the piston.

The heads are cast with a shoulder which is turned to a close fit with the counterbore, and they are then bolted on. The head end head is recessed to receive the nut which holds the piston on the piston rod, thus reducing the necessary clearance; and the

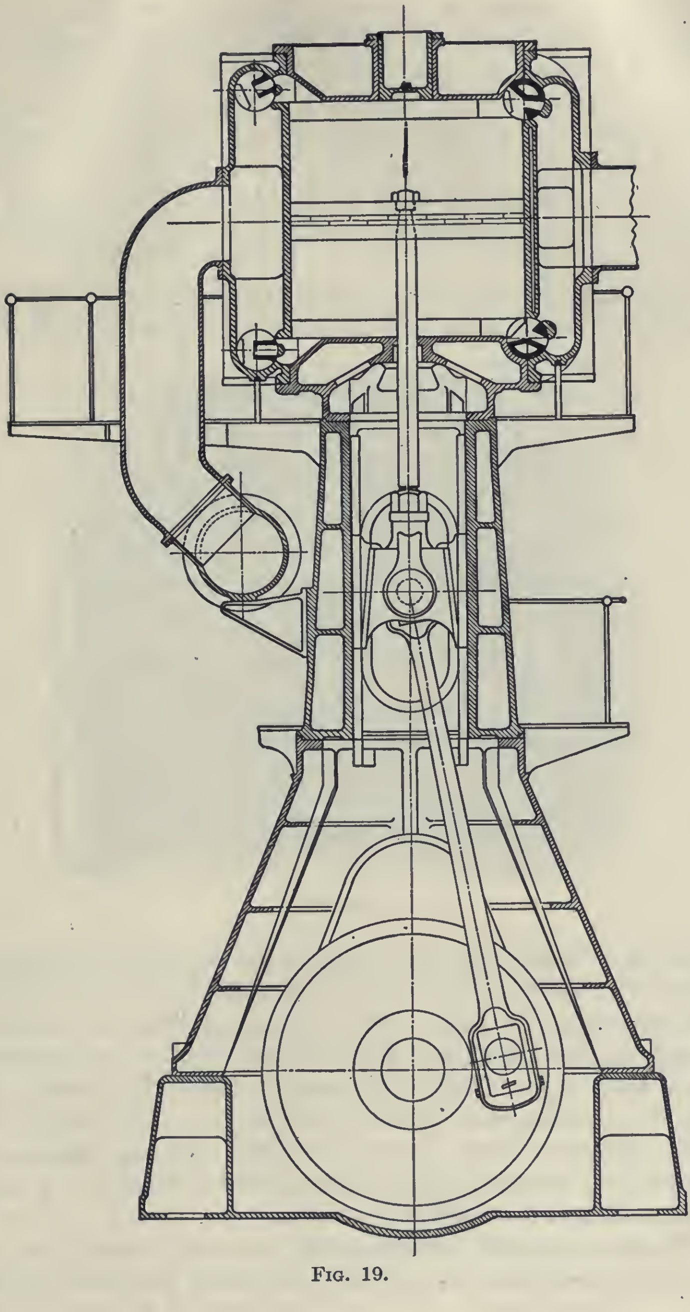


Fig. 19.

crank end is recessed on the outside to form a stuffing box. In this case, both heads are made thin, and ribs are cast on the outside of them to give greater strength. The walls of the cylinder are practically uniform in thickness, and, in order to accomplish this, recesses are left on the outside of the cylinder where necessary. Uniform thickness of the walls of a steam cylinder is desirable in order to prevent unequal strains in the cylinder, due to expansion caused by the heat to which it is subjected.

That part of the casting forming the valve chest and steam passages is much more complicated than the other parts of the

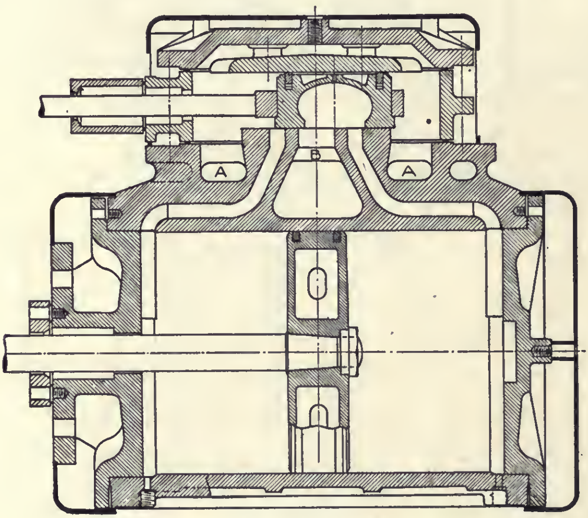


Fig. 20.

cylinder on account of the necessary provision for fastening the cylinder to the saddle, and conducting the live steam to it and the exhaust steam away from it. The valve seat consists of a raised rectangular table with the steam and exhaust ports leading up to its surface. The steam passage enters through the saddle and then divides, entering the bottom of the steam chest on each side of the valve seat at A and A. The exhaust passage lies between the steam passages and it is of such width that a web B is cast across it to brace its side walls.

The entire outside of the cylinder and valve chest is covered with planished sheet iron to give a smoother and neater appearance to it. The space around the barrel of the cylinder, between the cylinder and covering, is filled with nonconducting material such as asbestos, to reduce the loss of heat by radiation.

The cylinder shown in Fig. 21 represents a usual type of construction for automatic high speed engines having flat valves. As compared with the cylinder shown in Fig. 20, this one has a larger diameter in proportion to its length, has shorter ports due to the greater width of valve, and the cylinder casting is somewhat simpler. The cylinder is counterbored, as is the case with all steam engine cylinders, and in this case the heads are

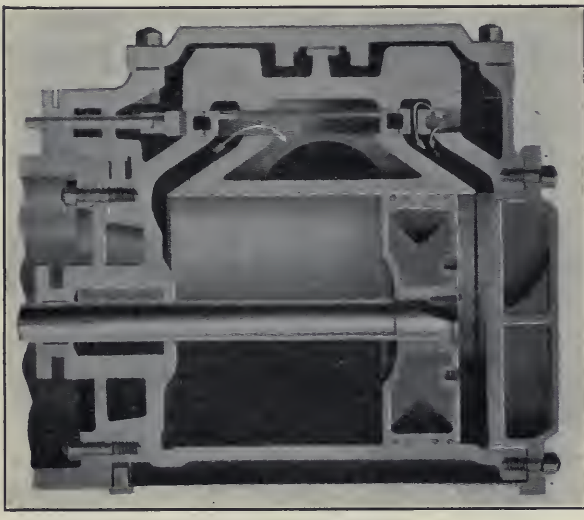


Fig. 21.

recessed to allow the ports to end behind the piston instead of ending flush with the cylinder walls. The heads are set into the cylinders and bolted on. On account of the different methods of fastening the piston to the piston rod, the head end head is not recessed but forms a plane surface to agree with the face of the piston. This head is made double with an air space between for insulation. The crank end head is recessed on the outside for the stuffing box, which extends slightly into the cylinder. The piston is cored out to fit the projection on the head, and thus to reduce the clearance. The clearance is further reduced by the short ports. The exhaust port in this cylinder is wide and shallow, and its walls do not, therefore, require bracing.

A Corliss engine cylinder is shown in Fig. 22. The bottom of the cylinder, which is rectangular in shape and flat, rests on its own bedplate and is bolted to it. The two admission valves are placed at the top of the cylinder and the two exhaust valves at the bottom, all being at the ends of the cylinder. The axes of the valves are placed across the cylinder, and the valve chambers are of rectangular form so that the cylinder has a square cornered appearance. In later types of engines the tendency is to round the valve chambers to conform to the shape of the valve, as shown in Fig. 11. The steam chamber S is cored out of the top of the cylinder and extends all the way across it, giving a flat top to the cylinder. As this chamber contains steam at boiler pressure, webs are cast in it to give a greater strength.

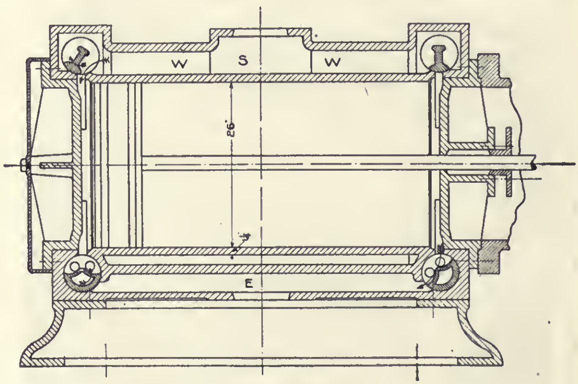


FIG. 22.

The steam chamber is formed right upon the walls of the cylinder so it may act as a steam jacket to this part of the cylinder. The exhaust chamber, E, which contains steam at a lower temperature, is separated from the cylinder walls by a cored chamber. The exposed portions of the valve chambers are usually polished to decrease the radiation of heat, and the other parts of the cylinder are covered with nonconducting material.

In the cylinder shown here the heads are cast very thin and are strengthened by webs cast on the outside of them, the head end head being covered by a cover plate to give a neater appearance. Both heads are cast with plain inner surfaces, and both faces of the piston are made flat to correspond. This allows the piston to be brought very close to the cylinder heads and thus reduce the clearance. The steam and exhaust ports are partly cored out of the heads so there may be some piston surface exposed to pressure when the piston is at the end of its stroke.

The placing of the exhaust valves in a Corliss engine at the lowest point of the cylinder allows water to drain through them when the cylinder is being warmed up preparatory to starting the engine. In a slide valve engine the valve is placed at the side or on top of the cylinder, and drain valves or cocks must be placed in the bottom of the cylinder.

The cylinder of a four valve high speed engine, shown in Fig. 23, is constructed the same as a Corliss cylinder, the principal difference being the greater diameter in proportion to the length.

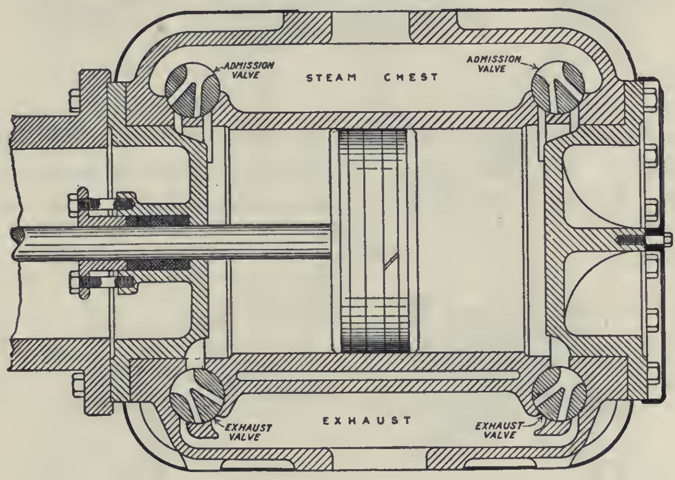


Fig. 23.

The ports and valves of these engines are often made double in order to secure a greater opening with a small valve travel, which is desirable when the speed of the valve is high. The clearance in the cylinders of these engines is relatively greater than that in the Corliss engine because, while the piston may be brought as close to the head, the volume between the piston and head will be greater because of the larger diameter. In some makes of four valve high speed engines and also in some large Corliss engines an effort is made to reduce the clearance to a minimum by placing the valve chambers in the cylinder heads, as shown in Fig. 24; this reduces somewhat the length of ports and thereby reduces the clearance.

The cylinder of a large marine engine is shown in Fig. 25. The peculiar feature about this cylinder is the shape of the heads and the fact that the cylinder is fitted with a liner. The piston is

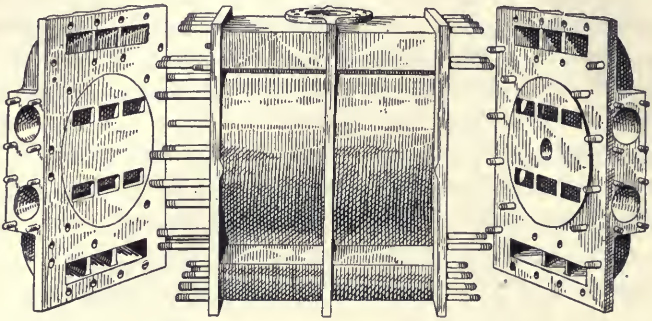


Fig. 24.

cone-shaped to conform to the shape of the heads, this shape being adopted to aid the drainage of water of condensation into the ports. Having a liner in the cylinder, as indicated at A, simplifies

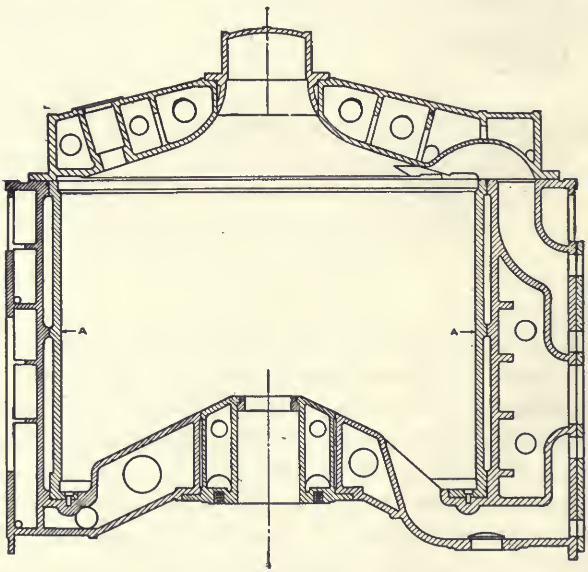


Fig. 25.

the work of casting the cylinder since the liner is cast separately, and permits of a sound and close grained liner being obtained while the cylinder proper is made of softer iron. The liner is fastened to the cylinder at the bottom by sunk head bolts in the inward projecting flange, the top being turned to a close fit. The ports may be either cut through the liner or carried around it; in this case they are carried around it. In order to secure lightness and strength, the heads are cast with double wall chambers which are strengthened by webs. This method of construction also reduces loss of heat from the cylinders since there is an enclosed air space next to the cylinder walls.

The Piston.—An engine piston must meet the following

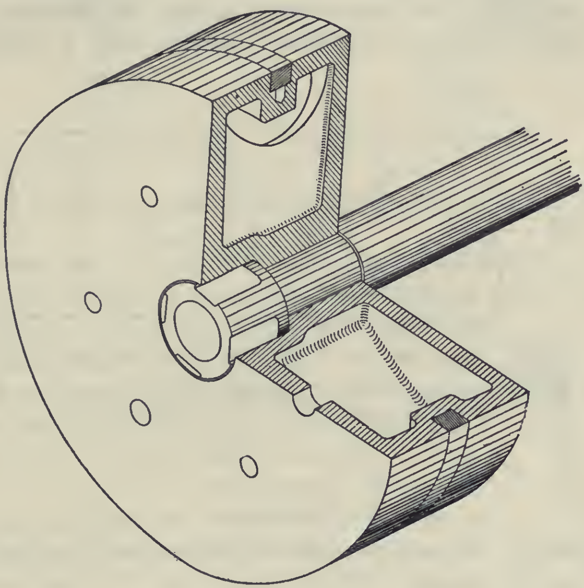


Fig. 26.

conditions: it must have enough strength to withstand the steam pressure acting upon it and yet be no heavier than necessary, as its weight controls its inertia and causes it to wear the cylinder, especially if the cylinder is horizontal; it must have a broad rim or working face, especially in horizontal engines, in order to have plenty of rubbing surface and reduce wear; it must be constructed to prevent the leakage of steam past it. The last result is secured by having the piston a little smaller than the bore of the cylinder and closing the gap between piston and cylinder by means of rings sprung into grooves in the piston.

The box piston is by far the most common type used in cylinders up to 24 inches in diameter. One of the simplest forms of box piston is shown in Fig. 26. This piston consists of a simple hollow casting with flat faces of uniform thickness on both sides and with a hub cast into its center for receiving the piston rod. The piston is strengthened by casting webs across it so as to divide it into a number of compartments. A hole is cast into each of the compartments through which the core may be removed, after which it is drilled and tapped to receive a plug.

In order to fasten the piston rod in the piston, the hub of the piston is bored smaller than the rod, and the end of the rod is turned to fit the hub which rests against the shoulder on the rod. The two are then fastened together with a countersunk

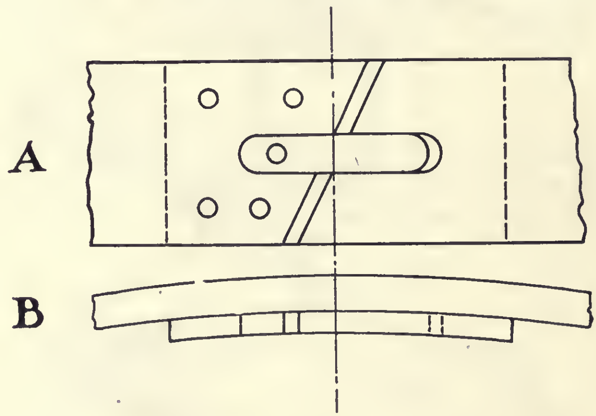


Fig. 26*a*.

nut screwed on the rod, the outer end of the nut being flush with the face of the piston.

This piston is supplied with a single heavy packing ring placed at the center. It is common, however, to place two lighter rings on the piston, one near each edge. In some cases the piston is supplied with four light rings in two grooves placed near each edge. The rings are turned out of cast iron and are made a little larger in diameter than the bore of the cylinder. They are then sawed through diagonally, only one cut being in a ring, and the ring is then sprung on the piston. The spring in the ring, since it is larger than the cylinder, keeps it pressed outward against the walls of the cylinder and prevents steam from leaking past it. When a single ring is used, a clip is placed at the cut in the ring to prevent steam from leaking through, as shown in Fig. 26*a*, but when the piston has more than one ring no clip is used the rings being simply placed on in such manner as to break joints. Instead of the faces of the box piston being flat, they are often

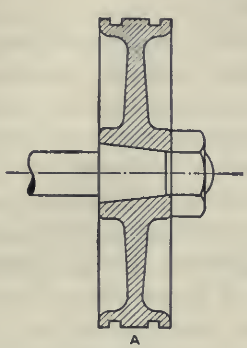


Fig. 27*a*.

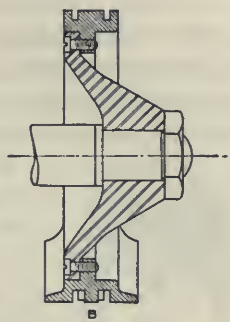


Fig. 27*b*.

shaped to conform to projections on the inside of the cylinder heads.

Two examples of locomotive pistons are shown in Fig. 27. In both of these designs a special effort is made to secure strength and lightness. The one shown at *A* is made entirely of cast iron of a simple *T* section and with two packing rings in the rim. This piston is forced on the tapered end of the piston rod by means of a nut. The piston shown at *B* is made in two parts, a central conical part made of steel, and a cast-iron rim which is bolted to the central portion by means of countersunk bolts. A peculiar feature of this piston is that the bottom, which carries the weight, has a much broader wearing surface than the top. This piston is pressed against a shoulder on the piston rod by means of a nut screwed on the end of the rod. The end of the rod has a slight taper to secure a better connection between the rod and the hub.

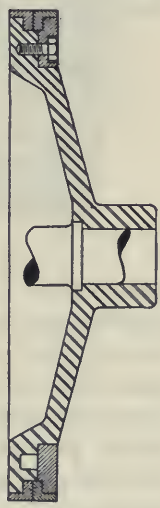


Fig. 28.

The pistons of vertical engines, especially of the marine type, are made as light as possible consistent with proper strength. A wide rim or wearing surface is not necessary on these pistons, since the weight of the piston is not carried by the cylinder. One type of marine engine piston is shown in Fig. 28. This piston is constructed with a steel web, and the entire rim is made of two cast-iron spring rings which are held in place by a follower ring bolted to the steel web.

Corliss engine pistons are usually of the "built up" type, consisting of several adjustable parts bolted together. A piston of this type is illustrated in Fig. 29. The ribbed body of this piston, in which the rod is fastened, is called a spider. The rim is made in two parts, 2 and 3, and is called a "bull ring" or "junk ring." This carries a single heavy packing ring. The bull ring is held in place by a follower plate, 4, which is fastened by tap bolts to the spider. The bull ring forms the wearing surface of the piston and can be adjusted by set screws so as to make the axis of the piston agree with that of the cylinder. The bull ring is made in two parts so that the packing ring may be put in without having to spring it over the piston and also because access may be had to the packing ring without removing the piston from the bore of the cylinder. The piston rod has a taper fit against a collar and is riveted over a heavy washer at the end, after which a key is passed through both rod and hub.

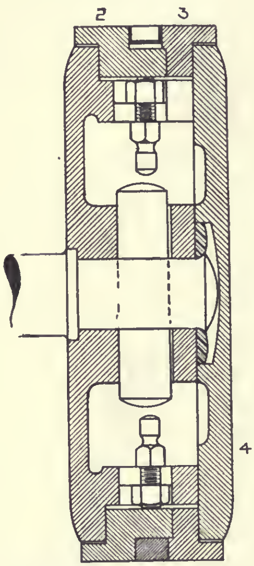


Fig. 29.

Stuffing Box.—Stuffing boxes are used to prevent the leakage of steam at the points where the piston rod and valve rod pass through the cylinder head and valve chest, respectively. As the stuffing boxes at both of these points are constructed alike, a description of one will be sufficient. A stuffing box consists of two parts: first, an annular space surrounding the rod, as shown in Fig. 30; and, second, a cover plate called a "gland" extending into the stuffing box in such manner as to compress the packing material when it is screwed down. Two stud bolts are screwed into the cylinder head, one on each side of the stuffing box, and these extend through the gland and end in a nut so the gland may press upon the packing. The bottom of the stuffing box is most often cut away at an angle, as shown in Fig. 30, but sometimes it is made flat. The hole through which the rod passes into the cylinder must be large enough to accommodate any lack of alignment between the rod and cylinder but must not be large enough to allow any of the packing material to be squeezed through it.

Formerly, braids or strands of hemp soaked in tallow were wrapped around the piston rod and pressed into the stuffing box or packing, but, as increasing steam pressures and temperatures became common, this kind of packing became unsatisfactory. There are now a great variety of packings on the market made of vegetable fiber, asbestos, or rubber in various combinations, and frequently mixed with graphite for a lubricant, these being made of sizes and shapes to fit neatly into the stuffing box. Woven

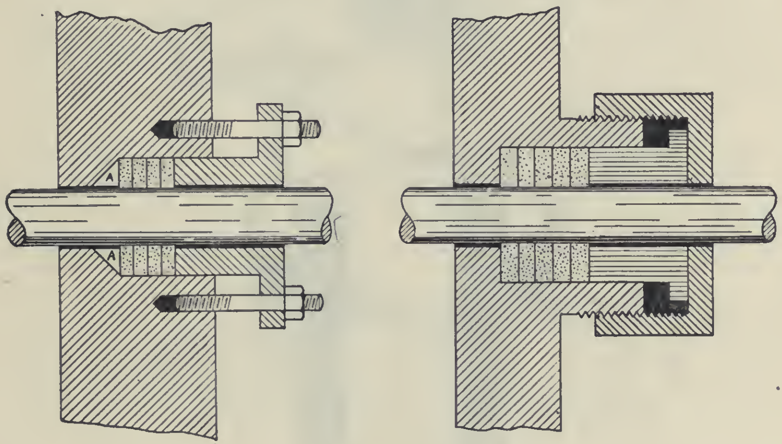


Fig. 30.

packing is often made square in cross section and divided diagonally into two parts so that when the gland is screwed down the packing is pushed out squarely against the rod.

Metallic packing is often used in the stuffing boxes of steam engines, but requires a good alignment of the piston rod to work properly. Metallic packing is usually made in the form of babbitt metal rings, which are pressed against the piston rod, preventing the leakage of steam and causing but little friction. An example of this kind of packing is shown in Fig. 31. The gland, *G*, is merely a heavy cover plate made tight by a copper wire acting as a gasket. The ring 1 presses against the casing 2 and forces the babbitt metal rings 3, 4, and 5 against the rod. These rings are made in segments and placed so as to break joints.

A follower ring 6 is held in place by a heavy spring and keeps the rings in their proper position, but the spring is not depended upon to press the packing against the rod. This is done by the steam pressure acting behind the ring 6 so that the tightness of the packing varies with the steam pressure.

The Crosshead.—The crosshead moves in a straight line between guides and is for the purpose of joining the piston rod to the connecting rod. It, therefore, has two joints: a stationary one

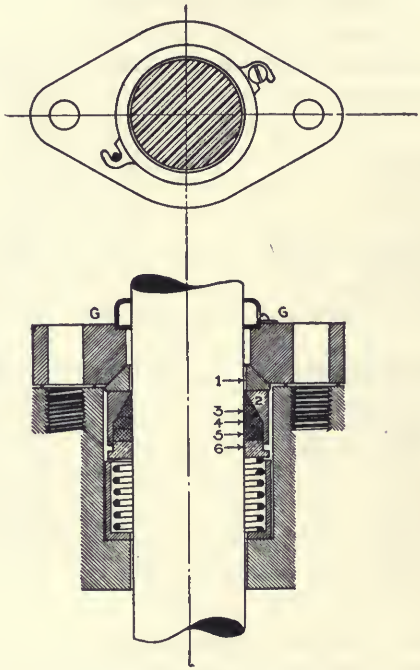


Fig. 31.

between the piston rod and crosshead; and a pin joint between the connecting rod and the crosshead so the connecting rod may be free to move.

There are three general types of crossheads used upon stationary engines, called respectively, the "wing," the "block," and the "slipper" crossheads. The wing crosshead is illustrated by Fig. 32. It consists of a heavy steel or cast-iron block forming three sides of a rectangle and having a heavy "wrist pin" passing between the side pieces or wings. The piston rod is threaded at the end and screwed into the front crossbar of the crosshead, being held securely by means of a lock nut. The wings form the rubbing surfaces, and, to reduce friction, they are often constructed with grooves filled with babbitt metal. The guides consist of four flat bars between which the wings move. The guides are adjustable up and down to take up wear in the crosshead. The wrist pin is usually made of steel, separate from the crosshead, and is held in place by nuts on the ends, or sometimes by means of

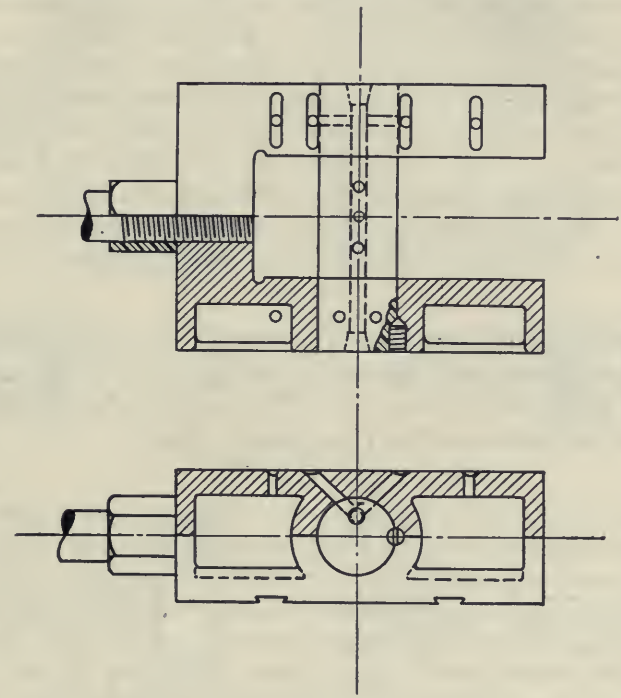


Fig. 32.

a set screw. The wing crosshead is often used on the smaller sizes of engines, especially those of the plain slide valve type.

The block crosshead is most often found on Corliss and automatic high speed engines. As shown in Fig. 33, it consists of a heavy cast-iron block with the wrist pin passing through its center and with the piston rod screwed or keyed into the center at the front. The rubbing surfaces, located at top and bottom, are of circular shape for bored guides, and V-shaped for planed guides. Both top and bottom rubbing surfaces or "shoes" are adjustable and have babbitt metal inserted in grooves or holes to reduce friction. With this type of crosshead the guides are not adjustable but the shoes of the crosshead are adjusted to take up wear and to bring the crosshead into alignment with the piston rod. The shoes are adjusted by means of wedges placed between the shoes and the body of the crosshead and moved by screws fitted with lock nuts to hold the wedge in

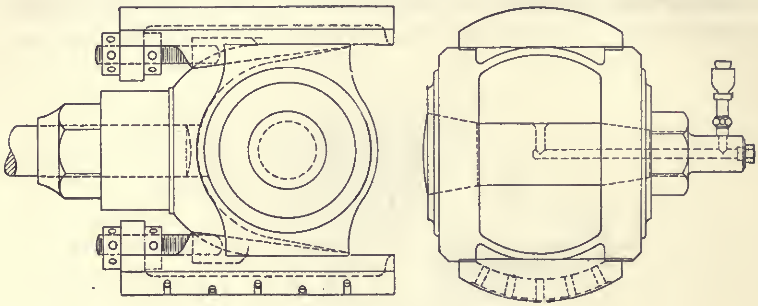


Fig. 33.

position after adjustment. The method of securing the wrist pin in this crosshead is shown by the taper ends of the pin combined with a nut on the end. The method of carrying oil to the rubbing surface of the pin is clearly shown.

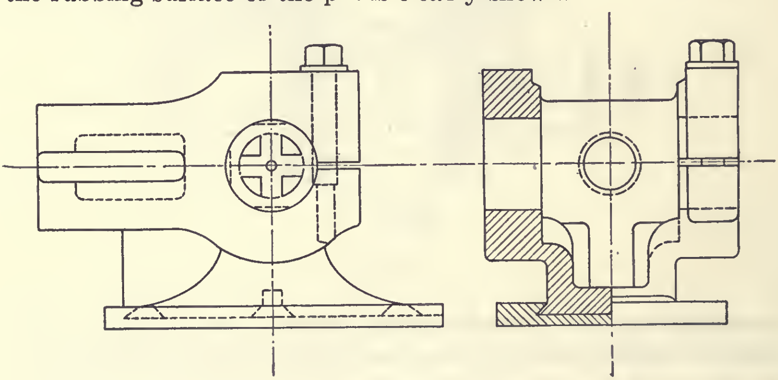


Fig. 33*a*.

The slipper type of crosshead, Fig. 33*a*, resembles the wing type, but differs from it in having the wrist pin and main body of the crosshead placed above the wings. In this case the wings are comparatively thin but the rubbing surface, which is all of the bottom of the crosshead, is broad. The guides consist of a flat planed surface on the engine frame, with a rectangular bar at each side to fit on top of the slipper. These rectangular bars are adjustable to take up the wear of the crosshead. The wrist pin consists of a simple steel cylinder and is clamped between the jaws of the crosshead, which are split and provided with bolts for this purpose. The wrist pin is arranged so it may be turned through 90 degrees as it wears, thus keeping it round.

Connecting Rods.—The connecting rod connects the wrist pin and crank pin and serves to transmit the force acting upon the piston to the crank pin. For one-half of a revolution of the fly-wheel the forces acting along the connecting rod are pushing and for the other half of the revolution they are pulling. The connecting rod consists of an adjustable bearing at each end connected by the shank or rod proper.

The cross section of the rods is made in various shapes, depending upon the type of engine with which they are to be used. For slow running engines of the Corliss type the rod is usually round,

FIG. 34.

being largest at the middle and tapering towards the ends. With engines of higher speed the rod is often shaped like a long cone, tapering towards the crank end and flattened on the sides so as to approach a rectangular cross section as the diameter increases. High speed stationary engines and some locomotives have a rod of rectangular cross section increasing in depth towards the crank end. Many of these engines have the cross section of the rod of an I-shape in order to make them light and strong. Marine engines usually have round rods.

Great variety is found in the construction of the ends of the connecting rod, but they will usually fall in one of the three general classes called respectively the "box" or "solid end," the "strap end," or the "marine end." Often the rod will have one end of the "box" type and the other end of the "strap" type. The box or solid end type of construction is well illustrated in Fig. 34, which shows that the end of the rod is flattened and has a rectangular slot milled into it. Into this slot are placed the two halves of the bearings, which are made of brass or bronze.

These halves are separated by a small space to allow them to be brought closer together as they wear away. The "brasses" are cast with flanges at the sides which fit the sides of the rod to prevent their movement sidewise. Behind one of the brasses is placed a wedge-shaped block with a screw held by nuts at each end passing through it. Adjustment of the brasses is made by turning this screw and moving the wedge-shaped block upward, thus forcing the brasses closer together. By having one of the adjusting wedges placed on the inside of the end and the other placed on the outside of the end, the length of the connecting rod, which is the distance from the center of the wrist pin to the center of the crank pin, is not changed when both ends are adjusted at the same time.

A strap end connecting rod is illustrated in Fig. 35. In this form, the connecting rod ends at the brasses and a separate steel

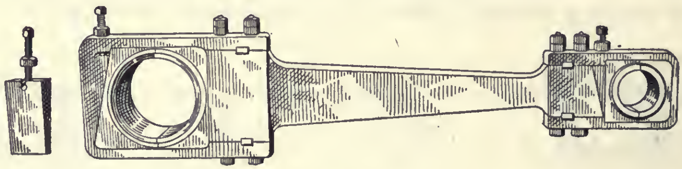


Fig. 35.

strap passes around the brasses and laps over the end of the rod at top and bottom. The strap is fastened to the connecting rod by means of two bolts which pass entirely through both ends of the strap and through the connecting rod and are secured by a nut and lock nut. Keys are inserted between the strap and connecting rod at top and bottom to keep the strap in line with the rod and to relieve the bolts of shear and permit the use of lighter bolts. The brasses are adjusted by means of a wedge in back of one of the brasses, the wedge being moved up or down by a bolt threaded into the brass and passing through the strap with a lock nut on the outside.

Strap end connecting rods are used commonly on locomotives but are dropping out of use on stationary engines. When used on locomotives, the brasses form the bearing against the pins, but for stationary engines the brasses are usually lined with babbitt metal. Locomotive connecting rods have the brasses adjusted by means of a wedge driven down behind one side of the brass and locked in place by means of a set screw.

The right hand end of the connecting rod shown in Fig. 36

is of the marine type; the left-hand end is of the box type, described before. Marine end connecting rods are used on all marine engines and on some vertical stationary ones. The example shown here is from a stationary engine. Those designed for marine engines usually have both ends of the connecting rod of the marine type, instead of only one. In the marine end connecting rod, adjustment of the brasses is secured by means

FIG. 36.

of bolts placed parallel to the connecting rod and passing through shoulders on the rod and on the removable end, as shown. This form of adjusting device permits a short end for the rod, which accounts for its common use on marine engines, where the crank pin passes close to the floor.

Crank and Crank Pin.—The steam pressure acting upon the piston is transmitted through the connecting rod to the crank pin and then through the crank to the shaft. Engines may be divided into overhung crank engines, in which the crank is at the end of the shaft; and into center crank engines, in which the crank is placed at or near the center of the shaft.

An overhung crank for an engine of medium size is shown in Fig. 37. This crank is made in the form of a cast-iron disk, with holes bored to receive the crank pin and the shaft. The crank disk is made thicker opposite the crank pin than it is on the crank pin side in order to counterbalance it. The crank disk is either forced on the shaft and keyed, as shown here, or else shrunk on, in which case a key is unnecessary. The crank pin is usually forced in by hydraulic pressure and then riveted over. Overhung cranks for large slow speed engines are sometimes forged from steel in the shape shown in Fig. 38. This

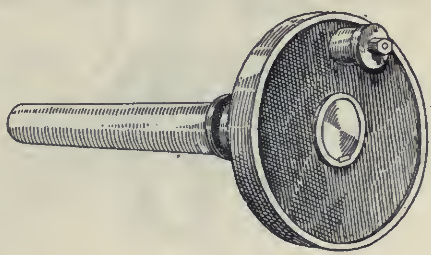


FIG. 37.

shape does not permit of as much counterbalancing as the disk shape, but, on the other hand, slow speed engines do not require as heavy counterweights as do high speed engines.

A crank for a center crank engine is shown in Fig. 39. This

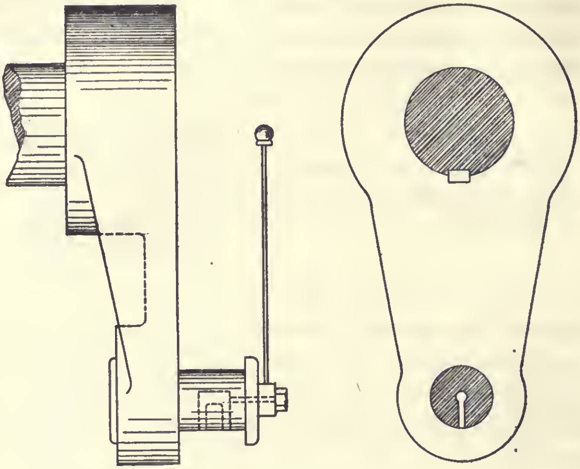


FIG. 38.

consists of two cast-iron disks with counterweights, fastened to the shaft, which is made in two sections, and with the crank pin connecting the two disks. Another form of center crank is illustrated in Fig. 40, in which the cranks, crank pin, and shaft are all in one piece and forged from steel. The counterweights are made of cast iron and bolted to the cranks opposite the crank pin, as shown.

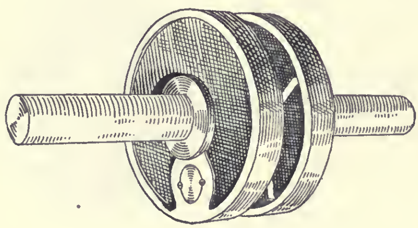


FIG. 39.

Bearings.—There are always two bearings for the shaft on an engine. In center crank engines these two bearings are alike, but in side crank engines the one next to the crank, called the main bearing, is much heavier than the other, or outer bearing.

The bearings must not only support the weight of the shaft and flywheel, but must resist the thrust of the piston, and also resist the pull of the belt, if there is one. The resultant of all of these forces causes the bearings to wear at an angle with the horizontal rather than at the bottom and top. Since the wear comes at an angle to the horizontal, provision must be made for adjusting the bearings at an angle. This is done by dividing the bearing into two parts with the division line between the parts making an angle with the horizontal, or by dividing the bearing into four parts so that adjustment may be made at the side or at the top.

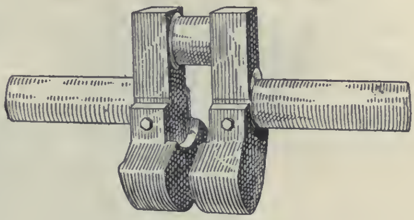


FIG. 40.

Two part bearings, such as are often found on high speed engines, are illustrated in Figs. 41 and 42. In Fig. 41 the bearing is divided along a line making about 45 degrees with the horizontal. The lower part has a flat plate cast on its outer surface which rests in the frame and which may be adjusted vertically by shimming, or placing thin sheets of metal under it. The top half has one flat plate at the side and another at the top. This part of the bearing may be adjusted horizontally by means of bolts passing through the back of the frame and bearing against

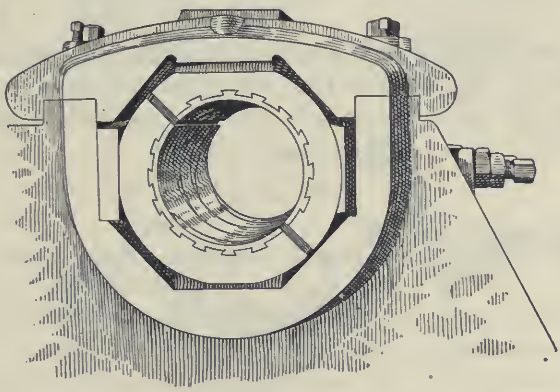


FIG. 41.

the flat plate at the side. The cap rests on the flat plate at the top and is held in place by bolts passing down into the frame. The bearing is lined with babbitt metal having a series of diagonal grooves cut in it for distributing the oil throughout the length of the bearing.

The bearing shown in Fig. 42 is similar to the above except for the method of adjusting. In this bearing the top part is moved horizontally by means of a wedge placed behind it and which may be moved up or down by a bolt extending through the cap.

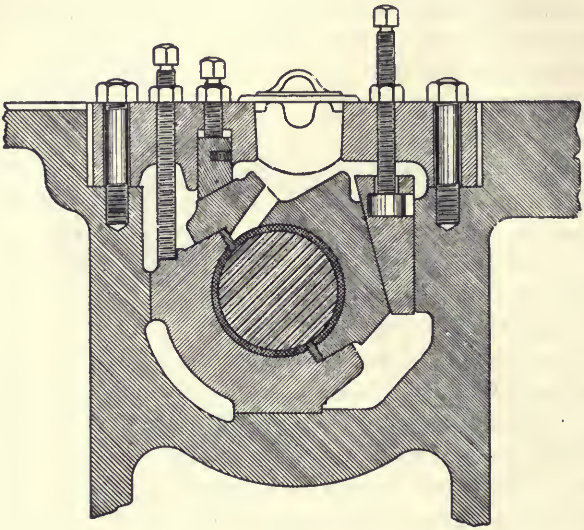


Fig. 42.

Large horizontal engines usually have main bearings of the type shown in Fig. 43. In this type the bearing is divided into four parts, the side pieces being adjustable. One of these side pieces

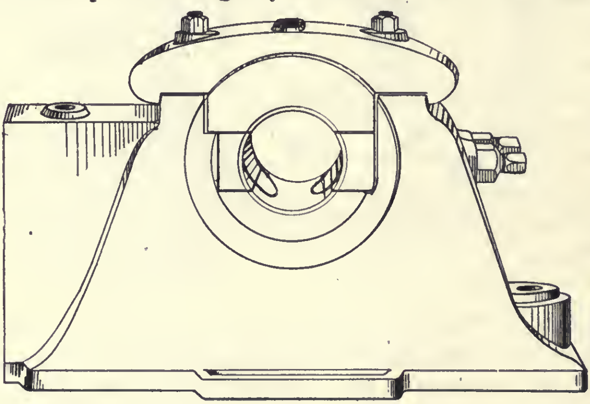


Fig. 43.

is adjusted by shims and the other by set screws passing through the frame. The top part of the bearing is made as a part of the cap, which is held in place by bolts passing down into the frame and which serves to hold the parts of the bearing together. The bearing is lined with babbitt metal with diagonal grooves cut in it. The advantage of this type of construction is that the bearing may be removed without removing the shaft, by taking off the cap, slightly lifting the shaft, and turning the bearing around.

The Flywheel.—The flywheel serves a threefold purpose: It sometimes serves to transmit the power of the engine to other machines by means of belts; to store up enough energy near the middle of the piston stroke to carry the engine past center; and, by storing up energy at one part of the stroke and giving it out

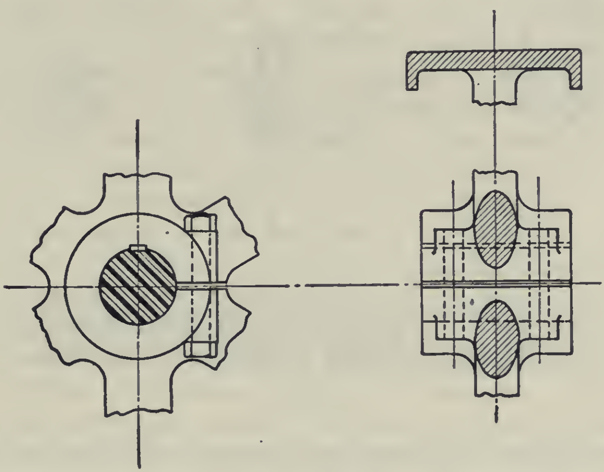


Fig. 44.

again at other parts, to prevent fluctuations of speed during a revolution of the flywheel.

Small engines are usually supplied with two ordinary belt wheels. In slow speed engines one of these wheels is sometimes larger than the other, but in medium and high speed engines both wheels are of the same size and kind. Even when high speed engines are direct connected to generators there is often one flywheel.

Wheels less than 9 feet in diameter are usually cast in one piece, but with the hub split on one side, as shown in Fig. 44, so it may be clamped to the shaft by two bolts, one on each side of the spokes. These bolts are not depended upon to hold the wheel, however, but merely to simplify putting the wheel on the shaft. The hub is held securely to the shaft by means of a close fitting key.

Flywheels between 9 and 16 feet in diameter are commonly made in halves, and larger sizes are divided into a greater number of parts, as a 16 foot piece is about as large as can be shipped on an ordinary flat car. Fig. 45 illustrates the method of joining the halves of a large flywheel. The hub is clamped to the shaft by

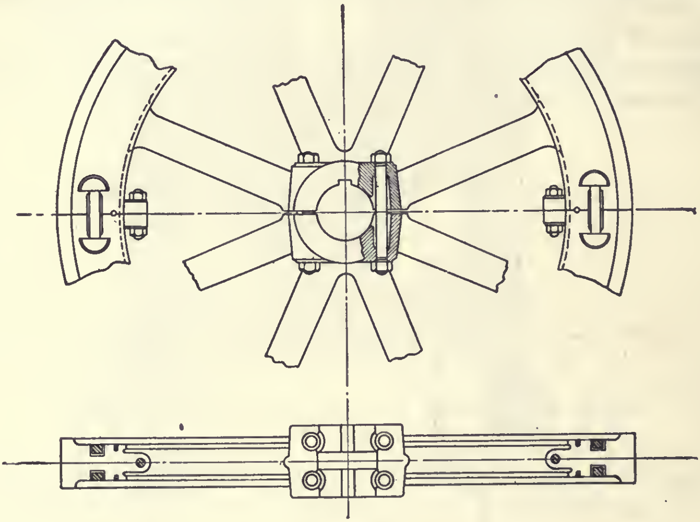


Fig. 45.

bolts extending all the way through on each side of the shaft, after which the hub is keyed to the shaft. This type of wheel is used only as a flywheel and not as a belt wheel, hence, the rim is made narrow and deep in order to concentrate its weight as far from the center of the shaft as possible. The rim is fastened together by inwardly projecting flanges bolted together and by I-shaped bars or links let into the sides of the rim. These links are machined exactly to length between heads and the slots similarly machined. The links are made shorter than the slots by from one in one thousand to one in eight hundred. The links are then heated and expanded until they will go into the slots. Upon cooling, the links contract and hold the halves of the rim together tightly.

CHAPTER IV

HEAT, WORK, AND PRESSURE

Force.—If a weight is held in the outstretched hand, a downward pull on the hand will be experienced. Unless this pull is resisted, the hand will be moved downward. The pull on the hand in this case is called a *force*. A force always tends to produce motion and it may, therefore, be defined as that which produces motion or tends to produce motion. If a force is applied to any stationary object, the object will move unless the force is resisted or opposed by another force equal in amount but opposite in direction. If a moving object is not acted upon by any force, or is acted upon by forces which are equal in amount but opposite in direction, the object will continue to move with a uniform velocity; but if the forces acting upon the moving object are not equal in amount and opposite in direction, the velocity will increase as long as such forces are applied.

In a steam engine, the piston is acted upon by the force of the steam pressure which causes the piston to move, and the motion is transmitted to the flywheel. The velocity of the flywheel will increase until the forces opposing its motion are equal in amount to those causing it to move, after which it will move with uniform velocity.

A force is measured by the number of pounds with which it pulls; thus a force of 10 pounds is the downward force exerted by a weight of 10 pounds, or the force necessary to lift a weight of 10 pounds.

A force acting upon the crank pin or rim of the flywheel produces what is called a *torque* or *twisting moment.* A torque cannot be measured in pounds because its amount depends both upon the force applied at the circumference of the circular path of the force and also upon the distance from the center to the point at which the force is applied. In the case of the engine crank, the torque depends both upon the actual force applied to the crank pin and upon the length of the crank from the center of the shaft to the center of the crank pin. The amount of the torque is expressed in foot-pounds and is equal to the number of pounds of force applied at the circumference multiplied by the radius of the circular path, in feet. If a force of 1000 pounds is applied to a crank pin which is two feet from the center of a shaft, the torque or twisting moment will be

$$1000 \times 2 = 2000 \text{ foot-pounds}$$

Work.—When force is applied to an object in such a manner as to cause the object to move, work is performed. If the object does not move, no work is performed no matter how large the applied force may be. Steam admitted behind the piston of a steam engine causes the piston to move, hence the steam performs work upon the piston. If the piston is blocked so that it cannot move and steam is admitted behind it, no work will be performed because, while the steam pressure may be as great as before, the piston in this case does not move.

The amount of work performed is always equal to the force applied, expressed in pounds, multiplied by the distance through which the object moves, expressed in feet. The unit of work is the foot-pound and is the amount of work done when a force of one pound moves through a distance of one foot.

Example.—How much work is performed when a steam pressure of 80 lb. per sq. in. acts upon the piston of a steam engine which is 18 inches in diameter and which moves it through a distance of two feet?

Solution.—The area of the piston is

$$.7854 \times 18^2 = 254.5 \text{ sq. in.}$$

The force acting on the piston is

$$254.5 \times 80 = 20{,}360 \text{ lb.}$$

The work performed when this force moves through a distance of two feet is

$$20{,}360 \times 2 = 40{,}720 \text{ ft.-lb.}$$

Work and torque should not be confused with each other, even though both are expressed in foot-pounds. In the case of work there is motion; in the case of torque there is no motion.

Energy.—Energy is the ability to do work. Water falling over a waterfall is able to perform work, hence falling water possesses energy. Steam performs work in a steam engine, hence steam possesses energy. Anything which is capable of performing work, or producing motion by overcoming a force, possesses energy.

There are several kinds of energy such as mechanical energy or the energy of motion, electrical energy, heat or thermal energy, and chemical energy; and any of these different kinds of energy may be changed into any other kind. In a power plant the coal which is burned under the boilers contains chemical energy, due to the various chemical substances of which it is composed. When the coal is burned, its chemical energy is changed into heat energy. The heat energy causes the water in the boiler to form steam and the steam containing the heat energy is carried to the cylinder of a steam engine. The heat energy which the steam contains is changed, in the cylinder of the steam engine, into the mechanical energy of the moving piston. The mechanical energy of the moving piston is transferred through the connecting rod and crank to the shaft. A direct-connected generator on the shaft changes the mechanical energy into electrical energy and this electrical energy may be changed again into mechanical energy by means of a motor.

Energy cannot be measured directly but it may be measured by the effects which it produces. The different kinds of energy mentioned above produce different effects, hence each kind has a different unit which is based upon the effects produced by this kind of energy. The unit for mechanical energy, or the energy of motion, is the foot-pound the same as for the unit of work, since the effect of mechanical energy is to produce work. Any object which is capable of performing one foot-pound of work is said to contain one foot-pound of energy.

Heat.—In the study of steam engines we are concerned principally with *heat energy*, or simply *heat*, as it is commonly called, since the work performed in the cylinder of a steam engine comes from the heat which is contained in the steam.

Both heat energy and mechanical energy are energies of motion but there is this difference between them; that mechanical energy is a motion of a body taken as a whole while heat energy is a motion of the particles of which a body is composed, aside from any motion which the body as a whole may have.

All substances are composed of very small particles called molecules, which are so small that they cannot be seen, even by the aid of the most powerful microscope. The molecules are in constant motion, vibrating back and forth, yet held together by a force of attraction which they have for each other and which keeps them vibrating within certain limits. Since the molecules are too small to be seen, and vibrate through a very short distance, they apparently cause no motion of the body as a whole, even though they are vibrating back and forth among themselves at a very rapid rate. The energy of these vibrating molecules is *heat energy*, or what is called *heat.*

If a nail is placed on an anvil and struck rapidly with a hammer, the nail becomes hot or its temperature is increased, and, if the blows are struck faster, the temperature of the nail becomes higher; that is, the temperature of the nail depends upon the rapidity of the hammer blows. In vibrating back and forth, the molecules of a substance strike each other a great many blows, hence the temperature of the substance depends upon the rapidity with which the molecules vibrate, in the same way that the temperature of the nail in the above example depends upon the rapidity of the hammer blows, and the faster the molecules vibrate the higher the temperature of the substance will be. When heat is applied to a substance, the molecules receive energy and vibrate faster, hence, heating a substance increases its temperature.

A substance may exist as a solid, a liquid, or a gas, depending upon its temperature and the amount of heat which it contains. At the lower temperatures a substance will be in the form of a solid; as it is heated to higher temperatures, it changes into a liquid; and as it is heated to still higher temperatures, it changes to a gaseous state. When a solid substance is heated, its molecules vibrate faster and faster and the temperature of the substance increases. The faster the molecules vibrate the longer their path tends to be, hence, the more nearly they come to breaking down the attraction which the molecules have for each other and which preserves the shape of the solid substance. But this partial breaking down of the force of attraction between molecules allows the substance to increase in size. This explains the expansion of substances when they are heated.

As the solid substance is heated to a higher temperature, the vibration of the molecules becomes so fast and the blows which they strike become so hard that the attraction between the molecules is partially broken down and the substance takes a liquid form, in which it does not retain a definite shape, but the molecules are free to move with respect to each other. Even in the liquid form there is a certain amount of attraction between the molecules; enough to keep them in one body.

As heat continues to be applied to the liquid, its molecules vibrate faster, it increases in temperature, and continues to expand. Some of the molecules near the surface of the liquid vibrate with enough force to break through the surface and then go into the space above the liquid; some of these returning, others remaining in the outer space. This is the effect known as evaporation. Finally the temperature of the liquid becomes so high that great numbers of the molecules pass into the space above the liquid and boiling begins.

The only difference between evaporation and boiling is that evaporation takes place only at the surface while in boiling the liquid is changed into a vapor both at the surface and in the body of the liquid, usually along the surface through which the heat passes into the liquid. This part of the vapor is formed in bubbles, which, being lighter than the liquid surrounding them, rise to the surface and burst. In a vapor or gas the attraction between the molecules has been completely broken down and they are free to move anywhere within the vessel which contains them, hence, a gas always expands and fills completely any vessel in which it is placed. The molecules of a gas are vibrating rapidly and, as there is no attraction between them, they move in straight lines from one end or side of the vessel to the other. In this way they are continually striking blows against the sides of the containing vessel. Just as the particles of water in a stream from a hose exert a pressure upon any object which the stream strikes, so the blows of the molecules against all sides of a vessel produce a pressure upon them and cause the gas to expand if the vessel is enlarged.

Temperature.—Temperature should not be confused with heat. Temperature is only one of the *effects* of heat, and is not heat itself. Temperature is a measure of the rapidity of vibration of the molecules and, therefore, is a measure of the intensity of heat.

Unit of Heat.—Heat energy cannot be measured directly but is measured by its effects. The most common effect of heat is increasing the temperature of a substance, and as water is one of the most common substances, the effect of heat in raising the temperature of water is used in measuring quantity of heat. The unit of heat is called the British Thermal Unit (abbreviated

B.t.u.) and is taken as the quantity of heat which is required to raise the temperature of one pound of water from 62° to 63° F., this temperature being chosen because the amount of heat required to change the temperature of one pound of water one degree varies slightly at different temperatures. However, this variation is so slight that it may be neglected for most practical purposes, and the unit of heat taken as the quantity of heat which is required to raise the temperature of one pound of water one degree, without reference to any particular temperature.

Mechanical Equivalent of Heat.—Since heat energy is capable of performing work there must be a numerical relation between heat and work. It has been found by experiment that one British Thermal Unit (B.t.u.) is equivalent to 778 foot-pounds of work and, therefore, also to 778 foot-pounds of mechanical energy.

Specific Heat.—Experiment shows that different substances require different amounts of heat to raise their temperature one degree. Thus, one pound of cast iron requires .1189 B.t.u. to raise its temperature one degree, while one pound of lead requires .0305 B.t.u. to raise its temperature one degree. The number of heat units required to raise the temperature of one pound of any substance one degree is called the specific heat of that substance. On this basis, the specific heat of water is one.

Power.—Power is the *rate* of doing work. *Work* and *power* should not be confused. Work does not take into account the length of time required to perform it, while power does. In order to raise a weight of 4400 pounds through a distance of 60 feet, the amount of work required is $4400 \times 60 = 264{,}000$ ft.-lbs., and this amount of work is the same whether the weight is lifted in one minute or in one hour, but the *power* required to raise the weight will be greater for the shorter time in which the weight is raised than for the longer time.

The unit of power adopted in engineering work is called the horsepower (abbreviated Hp.) and is the performance of 33,000 foot-pounds of work in one minute. This is equivalent to the performance of 550 foot-pounds of work in one second, or of 1,980,000 foot-pounds in one hour. The 264,000 foot-pounds of work mentioned in the example above would require:

$\frac{264{,}000}{33{,}000} = 8$ horsepower if performed in one minute, or

$\frac{264{,}000}{1{,}980{,}000} = 0.133$ horsepower if performed in one hour.

Since 778 foot-pounds of work are equivalent to one British Thermal Unit, one horsepower is equivalent to $= \frac{33,000}{778} = 42.42$ B.t.u. per minute, or

$\frac{1,980,000}{778} = 2545$ B.t.u. per hour, in round numbers.

Example.—In a certain power plant 400 pounds of coal are burned each hour. The coal has a heating value of 12,000 B.t.u. per pound, of which the engines utilize 10 per cent. How much power is developed by the engines?

Solution.—Heat liberated by the burning coal per hour $= 400 \times 12,000 =$ 4,800,000 B.t.u. Heat utilized by the engines per hour $= 4,800,000 \times .10$ $= 480,000$ B.t.u. Horsepower equivalent of 480,000 B.t.u. per hour $= \frac{480,000}{2545} = 188.6$ Hp.

Atmospheric Pressure.—The earth is surrounded by a body of air which exerts a pressure upon everything upon the surface of the earth, and this pressure must be taken into account in nearly all calculations dealing with pressure. The pressure exerted by the air is due to its weight and amounts to 14.7 lb. per sq. in. at sea level. If the atmospheric pressure is measured at a point above sea level, as on a mountain, it will be less than 14.7 lb. per sq. in., because the weight of air above this point is less. The following table shows the atmospheric pressure at various elevations above sea level:

Elevation above sea level in feet	Atmospheric pressure lbs. per sq. in.
0	14.70
1,000	14.20
2,000	13.67
3,000	13.16
4,000	12.67
5,000	12.20
6,000	11.73
7,000	11.30
8,000	10.87
9,000	10.46
10,000	10.07

Besides varying with the elevation above sea level, the atmospheric pressure also varies slightly with the weather, but the variation from this cause is not very great.

Vacuum.—In engineering work a vacuum is a space in which the pressure is less than atmospheric pressure. An absolute vacuum is a space in which there is *no* pressure. It is almost impossible to produce an absolute vacuum, hence in most engineering work we have to deal with a partial vacuum, in which there is some pressure although not so much as atmospheric pressure.

Barometer.—The atmospheric pressure may be measured by an instrument called a *barometer*. A simple form of barometer may be constructed as follows: A glass tube about 32 inches long, closed at one end, is completely filled with mercury. The finger is then held over the open end of the tube to prevent the mercury from spilling and the tube is quickly inverted and the open end placed in a cup of mercury, as shown in Fig. 46. If the finger is then removed from the end of the tube, the mercury will sink in the tube until it stands at a certain height H above the surface of the mercury in the cup, depending upon the atmospheric pressure.

Fig. 46.

The space above the mercury in the tube is as near a perfect vacuum as can be produced. The full atmospheric pressure acts upon the surface of the mercury in the cup, hence the mercury in the tube will stand at such height that its weight will just balance the pressure of the atmosphere on an area equal to that of the cross section of the tube. Since a cubic inch of mercury weighs .4908 pound, the height of the mercury in the tube, in inches, multiplied by .4908 gives the atmospheric pressure in pounds per square inch. Thus, if the height H of the mercury is 29 inches, the atmospheric pressure is $29 \times .4908 = 14.23$ lbs. per sq. in.

Absolute and Gage Pressures.—Gages which are used for indicating steam pressure are constructed so they read the amount of pressure above that of the atmosphere; that is, they do not read the true pressure or pressure above an absolute vacuum, but instead have their zero point at the atmospheric pressure. For this reason it is necessary to add atmospheric pressure to that indicated by the gage in order to find the true pressure or pressure above an absolute vacuum. The pressure indicated by a gage is called *gage pressure* and pressure measured above an absolute vacuum is called *absolute pressure*. The absolute pressure is equal to the atmospheric pressure plus the gage pressure.

Example.—At a certain place in which the height of the mercury in a barometer stands at 28.5 inches, the steam gage on a boiler reads 110 lbs. per sq. in. What is the absolute pressure of the steam in the boiler?

Solution.—The atmospheric pressure equals

$$28.5 \times .4908 = 13.98 \text{ (practically } 14.0\text{) lbs. per sq. in.}$$

The absolute pressure in the boiler equals

$$14 + 110 = 124 \text{ lbs. per sq. in.}$$

When the number of pounds pressure is given without stating whether it is gage or absolute pressure, it is usually understood to be gage pressure.

Measuring Vacuum.—A vacuum is measured by attaching it to a mercury column, similar to a barometer, and reading the

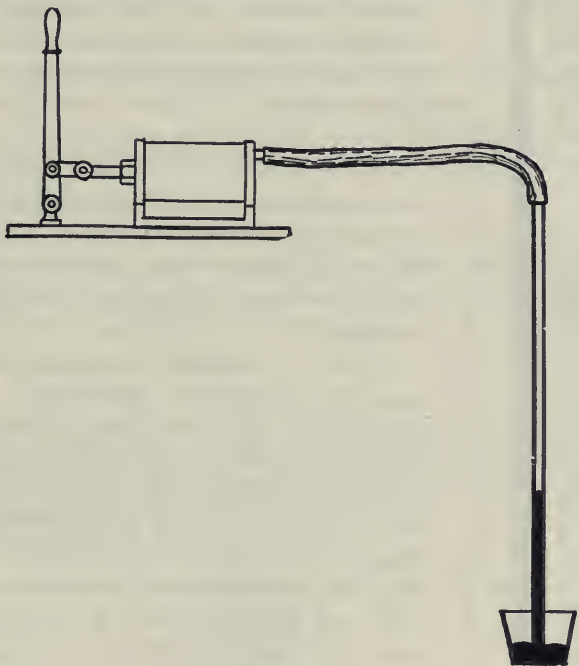


Fig. 47.

number of inches of mercury which it will support. If the top end of the glass tube in Fig. 46 is opened so the atmospheric pressure can act on the surface of the mercury in the tube, the mercury will immediately fall to the same level as that in the cup. If, now, the top of the glass tube is attached to an air pump, as shown in Fig. 47, and the pressure in the tube reduced below that of the atmosphere, then the mercury will rise in the

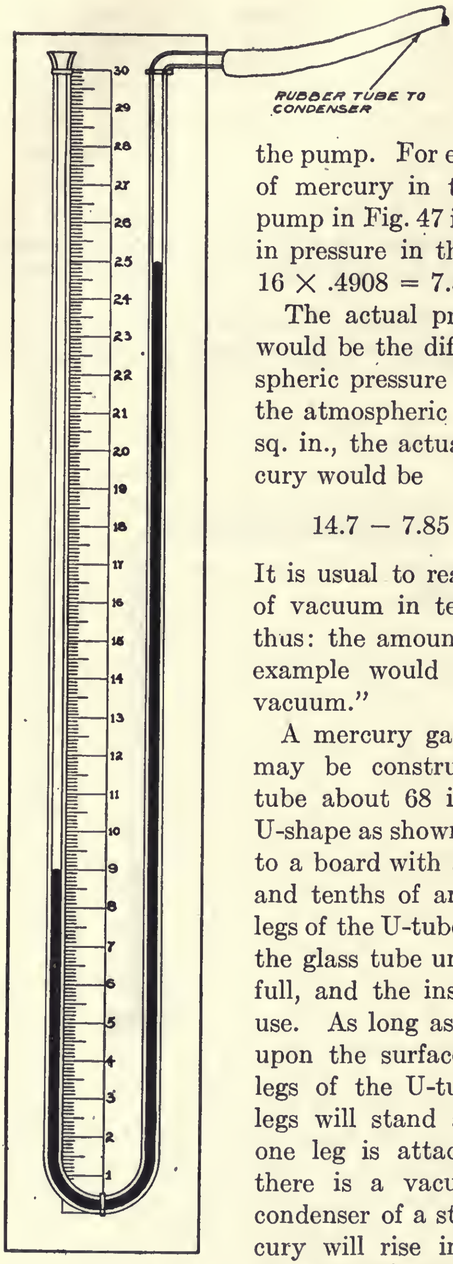


FIG. 48.

tube to a height which balances the difference in pressure between the atmosphere and that in the pump. For example, suppose the height of mercury in the tube attached to the pump in Fig. 47 is 6 inches. The reduction in pressure in the tube then amounts to $16 \times .4908 = 7.85$ lbs. per sq. in

The actual pressure above the mercury would be the difference between the atmospheric pressure and 7.85 lb. per sq. in. If the atmospheric pressure were 14.7 lb. per sq. in., the actual pressure above the mercury would be

$$14.7 - 7.85 = 6.85 \text{ lb. per sq. in.}$$

It is usual to read and express the amount of vacuum in terms of inches of mercury thus: the amount of vacuum in the above example would be called "16 inches of vacuum."

A mercury gage for measuring vacuum may be constructed as follows: A glass tube about 68 inches long is bent into a U-shape as shown in Fig. 48, and is attached to a board with a scale divided into inches and tenths of an inch placed between the legs of the U-tube. Mercury is poured into the glass tube until the legs are about half full, and the instrument is then ready to use. As long as atmospheric pressure acts upon the surface of the mercury in both legs of the U-tube, the mercury in these legs will stand at the same height, but if one leg is attached to a vessel in which there is a vacuum, as, for example, the condenser of a steam engine, then the mercury will rise in this branch and fall an equal amount in the other branch. The difference in height of the two columns of mercury will then represent the difference in pressure between the atmosphere and that in the condenser. For example, suppose the difference in height of the two columns of mercury is 16 inches. The reduction in pressure in the leg attached to the condenser is

$$16 \times .4908 = 7.85 \text{ lbs. per sq. in. below that of the}$$

atmosphere. If the pressure of the atmosphere is 14.7 lbs. per sq. in., the actual pressure in the condenser will be

$$14.7 - 7.85 = 6.85 \text{ lbs. per sq. in.}$$

The amount of vacuum in this example would be called "16 inches of vacuum."

Another form of vacuum gage, and the one most commonly used, is constructed like a pressure gage, except it reads pressures below atmosphere instead of above, and instead of reading in pounds per square inch it reads in inches of mercury; its readings, therefore, have the same meaning as those of the two vacuum measuring devices previously described.

It should be noted that the same amount of vacuum as indicated by a vacuum gage does not always mean the same thing. Thus, in New York, which is at sea level, a vacuum gage on a condenser which reads 22 inches of vacuum shows that the absolute pressure in the condenser is

$$14.7 - (22 \times .4908) = 4.1 \text{ lbs. per sq. in.,}$$

while in Butte, Montana, which is approximately 5000 feet above sea level and where the atmospheric pressure is about 12.2 lbs. per sq. in., a vacuum of 22 inches would show that the absolute pressure in the condenser is only

$$12.2 - (22 \times .4908) = 1.4 \text{ lb. per sq. in.}$$

CHAPTER V

PROPERTIES OF STEAM

Formation of Steam.—Suppose we have an open pan containing water at a temperature of 32° and that the atmospheric pressure is 14.7 lb. per sq. in. If the pan is placed over a fire, the temperature of the water begins to rise as soon as the water absorbs heat. The temperature of the water will continue to increase until it reaches 212° when small bubbles of steam will begin to form at the bottom of the pan and rise to the surface of the water and burst, liberating the steam which they contain. It will be noted that the temperature of the water does not rise above 212° even though we continue to apply heat to it. If we apply heat faster, the boiling occurs more rapidly. If we apply heat slower, the boiling occurs slower, but the temperature remains constant at 212°. If we place thermometers in both the water and the steam rising from the water, both of them will indicate 212°, showing that the temperature of the steam is the same as that of the water with which it is in contact. The temperature of the steam cannot be raised above that of the boiling water as long as it remains in contact with the boiling water, since any attempt to do so will simply result in boiling the water faster. The steam may, however, be collected and removed from the presence of water and then heated to a higher temperature.

If we should measure the amount of heat supplied to the water while its temperature was increasing from 32° to 212°, we would find that 180 B.t.u. had been supplied for each pound of water in the pan. If we should measure the amount of heat supplied to the water *after it had reached* 212°, we would find that 970.4 B.t.u. had been supplied for each pound of steam formed. We would also find, upon measurement, that while one pound of the water occupies a volume of only .0155 cubic feet, the volume of the steam formed from it occupies a volume of 26.79 cubic feet or about 1700 times the volume of the water from which it was formed. The large quantity of heat absorbed by the water after it has reached the boiling temperature is utilized in breaking down the attraction of the molecules for one another and in increasing the volume of the steam from that of the water to that of the steam, this increase of volume taking place against the pressure of the water and the pressure on the surface of the water, which, in this case, is the atmospheric pressure of 14.7 lb. per sq. in.

If the water, in the above example, had been heated in a closed vessel under a pressure greater than 14.7 lb. per sq. in., the temperature at which boiling occurred would have been greater than 212°, and if the pressure in the vessel had been less than 14.7 lb. per sq. in., the temperature at which boiling occurred would have been less than 212°, showing that the temperature at which water boils depends upon the pressure acting upon the surface of the water.

In the case of the water heated in an open pan, mentioned above, it is to be noted that the total amount of heat required to form one pound of steam may be divided into two distinct parts; first the amount of heat absorbed by the water in raising its temperature from 32° to the boiling temperature, and, second, the amount of heat required to change the water into steam after it has reached the boiling temperature. The first of these is called the "*heat of the liquid*" or sometimes the "*sensible heat of the steam*," because this part of the heat is sensible to the touch or affects a thermometer. The second part of the heat mentioned above is called the "*latent heat*" or the "*latent heat of evaporation*," because it is not sensible heat but is latent. In the case of water heated under a pressure of 14.7 lb. per sq. in. the heat of the liquid amounts to 180 B.t.u. and the latent heat to 970.4 B.t.u. The sum of these two quantities or $180 + 970.4 = 1150.4$ B.t.u. is the total quantity of heat supplied in forming one pound of steam from one pound of water at an initial temperature of 32°. The name of "*total heat*" is given to this quantity.

Steam Tables.—The boiling temperature, heat of the liquid, latent heat, total heat, and volume of steam formed at various pressures have been found from experiment, or calculated, and this information is placed in tables, called *steam tables*, one of these tables being found at the end of this chapter. It will be noted that this table is headed "Properties of Saturated Steam." By *saturated steam* is meant steam which has the same temperature at which it was formed. Steam in contact with the water from which it was formed is saturated steam. The various quantities given in the steam table are for *one pound weight of dry steam*, hence to find these quantities for any other weight than one pound, it is necessary to multiply the values given in the table, except those of temperature and weight of 1 cu. ft. of steam, by the actual weight of the steam.

The properties of steam depend on its absolute pressure. The pressure offers a certain amount of resistance to the expansion of the water into steam and it is the amount of this resistance which determines the temperature of evaporation and other quantities. Consequently, the absolute pressure is the first item given in the table.

For convenience, the corresponding gage pressures are given in the next column, assuming an atmospheric pressure of 14.7 lb. per sq. in. In case the barometer shows an atmospheric pressure very different from this, it is best to add the actual atmospheric pressure to the gage pressure and thus get the absolute pressure, which should then be used for finding the properties of the steam. In using properties of steam at pressures below that of the atmosphere, it is especially desirable to calculate the absolute pressure from barometer and vacuum gage readings rather than to use the vacuum gage reading in the gage pressure column of the table. An example will show what a difference this will make.

Suppose a vacuum gage on a condenser shows a vacuum of 27 inches and we want to find the temperature of the steam in the condenser. Without knowing the barometer reading but assuming the atmospheric pressure to be 14.7 lb. per sq. in., we would say that the reduction in pressure in the condenser amounted to $27 \times .4908 = 13.25 +$ lb. per sq. in. and that at a vacuum of 13.25 lb. (-13.25 lb. gage), which corresponds to an absolute pressure of $14.7 - 13.25 = 1.45$ lb. per sq. in., the temperature of the steam (from the steam table) is about 115°.

Now suppose we first look at a barometer and find that it stands at 28 inches. The condenser has more vacuum than we thought it had. The absolute pressure in the condenser is

$$(28 - 27) \times .4908 = .4908 \text{ lb. per sq. in.,}$$

or not quite .5 lb. per sq. in. absolute, and we find from the steam table that the temperature of the steam is a little less than 80° instead of being 115°.

The third column in the table gives the temperatures in which water boils when under the pressures given in the first column. These temperatures are also the temperatures of saturated steam under the given pressures, and likewise the temperatures at which steam under the given pressures will condense.

The total heat required to form steam from water which has an initial temperature of 32° is found in the fifth column of the table. This quantity is the sum of the heat supplied to the water, and the latent heat.

The heat of the liquid, or the heat in the water above 32°, is found in the third column. This is the amount of heat which must be supplied to the water to raise its temperature from 32° to the boiling point. For approximate calculations the heat of the liquid per pound of steam may be found by subtracting 32° from the boiling temperature, since the specific heat of water is nearly 1, but for accurate calculations the heat of the liquid should be obtained from the steam table. The different results obtained by these two methods may be shown as follows: At 165 lb. per sq. in. absolute pressure the boiling temperature is 366°. This would give, by difference of temperature,

$$366 - 32 = 334 \text{ B.t.u.}$$

for the heat of the liquid, while its actual value, from the steam table, is 338.2, a difference of over 1 per cent.

After water is raised to the boiling point, heat must be added to change it into steam. This heat is called *latent heat*, and it varies in amount, decreasing as the pressure of the steam increases. At an absolute pressure of 14.7 lb. per sq. in., the latent heat is 970.4 B.t.u. per pound of steam, and at an absolute pressure of 100 lb. per sq. in. it is 888. B.t.u. The whole amount of the latent heat will be absorbed only when the whole pound of water has been evaporated; also, when one pound of steam is condensed, the full latent heat will be given up by it. If the water is being evaporated at 100 lb. per sq. in. absolute pressure and after reaching the boiling temperature only one-half of the latent heat or

$$\tfrac{1}{2} \times 888 = 444 \text{ B.t.u.}$$

are supplied to the water, then only one-half of a pound of steam will be formed, and conversely, if we extract 444 B.t.u. from a quantity of steam at 100 lb. per sq. in. absolute pressure, only one-half of a pound will be condensed.

All of the quantities given in the steam table are calculated from water at 32°, and in practical problems it is generally necessary to calculate the heat in steam above some other temperature than 32°. Thus if we wish to know how much heat must be supplied to one pound of water at 170° in order to turn it into steam having a pressure of 150 lb. per sq. in. absolute, we must remember that the water already contains

$$170 - 32 = 138 \text{ B.t.u.}$$

Now, since the total heat of steam at 150 lb. per sq. in. absolute is 1193.4 B.t.u., there will have to be supplied only

$$1193.4 - 138 = 1055.4 \text{ B. t. u.}$$

in order to turn it into steam. Since the heat of the liquid at 150 lb. per sq. in. absolute pressure is 330.2 B.t.u., only

$$330.2 - 138 = 192.2 \text{ B.t.u.}$$

need be supplied to the water to bring it to the boiling temperature, but the entire latent heat, 863.2 B.t.u., must be supplied in order to evaporate it into steam.

Interpolation from Tables.—*Interpolation* refers to the method used to find values between those given in the tables, as for example, finding the latent heat at 44½ lb. per sq. in. absolute pressure. The table gives the latent heat for 44 lb. and for 45 lb. but not for 44½ lb., and we *interpolate* to get the value for 44½ lb. which would be halfway between 929.2 and 928.2 or just 929.7. Suppose we wish to obtain the heat of the liquid at 120 lb. per sq. in. gage pressure (134.7 lb. absolute). The table gives 134 lb. and 135 lb., the corresponding values of the heat of the liquid being 321.1 and 321.7. For one pound change in pressure, the heat of the liquid changes

$$321.7 - 321.1 = .6 \text{ B.t.u.}$$

Now, 134.7 is .7 lb. more than 134 or .3 lb. less than 135. We can, therefore, add .7 of .6 to 321.1 or subtract .3 of .6 from 321.7. Either way we get 321.52 as the value of the heat of the liquid at 120 lb. per sq. in. gage pressure.

In interpolating, remember that the latent heat and the volume of one pound of steam *decrease* as the pressure increases and that all other items in the table increase. For most calculations it is sufficiently accurate to take the nearest value given in the table without bothering to interpolate.

Wet Steam.—Saturated steam may be either wet or dry. If wet, it has small particles of water suspended in it, just as in a fog air has particles of water suspended in it. The water which is suspended in wet steam has not been evaporated into steam and has not received the latent heat. It is in the form of water but is at boiling temperature and it has therefore received the entire heat of the liquid. Dry steam has no moisture in it and has received both the entire heat of the liquid and the entire latent heat. The quantities given in the steam tables are for *dry* saturated steam.

In changing water into steam suppose that one pound of water is heated from 32° to the boiling temperature. Up to this point the one pound of water has received the entire heat of the liquid. When boiling commences, suppose that one-half of the pound of water is evaporated into steam and the other half of the pound is thrown up into the steam in the form of fine particles and remains suspended there. The steam has then received only one-half or .50 of the latent heat. If three-quarters of the water had been evaporated into the steam and the other quarter was suspended in the steam in the form of water, the steam would contain three-quarters or .75 of the latent heat.

The total heat above 32° in one pound of dry steam is equal to the sum of the heat of the liquid and the latent heat, or calling the total heat H, the heat of the liquid h, and the latent heat L, then

$$H = h + L$$

The total heat in one pound of wet steam is the sum of the entire heat of the liquid, h, and that fraction of the latent heat which has formed steam, or

$$H = h + qL$$

in which q is the per cent. of the pound of steam which has been evaporated. The quantity, q, is called the *quality* of the steam.

Example.—A pound of water at 32° is heated to the boiling temperature at a pressure of 100 lb. per sq. in. absolute and turned into steam having a quality of 90 per cent., that is, 90 per cent. of the pound of water is evaporated and the other 10 per cent. is suspended in the steam in the form of water. How much heat has been supplied to the steam?

Solution.—For 100 lb. per sq. in. absolute

$$h = 298.3$$
$$L = 888.0$$

6

and in this case $q = .9$

Therefore
$$\begin{aligned} H &= h + qL \\ &= 298.3 + .9 \times 888.0 \\ &= 298.3 + 799.2 \\ &= 1097.5 \text{ B.t.u.} \end{aligned}$$

Observe that if the steam had been dry it would have contained

$$\begin{aligned} H &= h + L \\ &= 298.3 + 888.0 \\ &= 1186.3 \text{ B.t.u.} \end{aligned}$$

Steam usually contains from 2 to 10 per cent. of moisture so that its quality is from 90 to 98 per cent. Steam which has a quality of 98 per cent. or more is called "commercially dry steam."

In case the steam is formed from water having a temperature higher than 32°, the heat which is already in the water above 32° must be subtracted from the total heat of the steam. For example, suppose steam at 150 lb. per sq. in. absolute pressure and having a quality of 95 per cent. is formed from water having a temperature of 170°. The heat already in the water above 32° is

$$170 - 32 = 138 \text{ B.t.u.}$$

hence, only 330.2 − 138 = 192.2 B.t.u. need be supplied to the water per pound in order to bring it to the boiling temperature. Since the quality is 95 per cent. only

$$.95 \times 863.2 = 820 \text{ B.t.u.}$$

is absorbed per pound in evaporating the water. Therefore, this steam has received

$$192.2 + 820 = 1012.2 \text{ B.t.u. per pound}$$

Superheated Steam.—If saturated steam is taken away from the presence of water and heated, its temperature may be raised above that at which it was formed. Steam which has a higher temperature than that at which it was formed is called *superheated steam.* Since superheated steam has received heat above that required to form it into saturated steam, it contains more heat per pound than saturated steam.

The total heat above 32° contained in a pound of superheated steam may be found by adding to the total heat of saturated steam for the same pressure, as found in the steam table, the number of heat units required to superheat the steam, as shown by the following table:

HEAT UNITS REQUIRED TO SUPERHEAT STEAM

Absolute pressure	Degrees of superheat										
	10	20	40	60	80	100	130	160	200	250	300
1	4.9	9.6	18.8	27.9	36.9	46.0	59.6	73.2	91.3	114.0	136.8
10	5.4	10.4	20.1	29.6	39.0	48.4	62.4	76.3	94.9	118.0	141.2
15	5.5	10.6	20.5	30.2	39.7	49.2	63.3	77.4	96.1	119.4	142.9
20	5.6	10.8	20.9	30.7	40.3	49.9	64.1	78.3	97.1	120.6	144.2
30	5.7	11.1	21.4	31.4	41.3	51.0	65.5	79.8	98.8	122.6	146.5
40	5.9	11.3	21.8	32.0	42.0	51.9	66.6	81.1	100.3	124.2	148.3
50	6.0	11.5	22.2	32.5	42.4	52.6	67.4	82.1	101.4	125.6	149.8
60	6.0	11.7	22.5	32.9	43.2	53.3	68.2	82.9	102.4	126.7	151.0
80	6.2	11.9	22.9	33.6	44.0	54.2	69.3	84.2	103.9	128.4	152.9
100	6.3	12.2	23.3	34.1	44.6	55.0	70.2	85.2	105.1	129.7	154.4
130	6.4	12.4	23.8	34.7	45.4	55.8	71.3	86.4	106.4	131.2	156.1
160	6.5	12.6	24.2	35.3	46.0	56.6	72.1	87.4	107.5	132.5	157.5
200	6.7	12.9	24.7	35.9	46.8	57.4	73.1	88.6	108.9	134.1	159.3
250	6.9	13.2	25.1	36.5	47.6	58.4	74.3	89.9	110.4	135.9	161.3
300	7.0	13.5	25.6	37.1	48.3	59.2	75.3	91.0	111.7	137.4	163.0

The use of this table may be illustrated by the following example:

Example.—Determine the number of heat units contained in a pound of superheated steam having a pressure of 130 lb. per sq. in. absolute and having a temperature of 447.4°.

Solution.—By referring to the steam table we see that the temperature of saturated steam at a pressure of 130 lb. per sq. in. absolute is 347.4° and that its total heat is 1191 B.t.u. The degree of superheat is, therefore,

$$447.4 - 347.4 = 100°$$

And the above table shows that for this degree of superheat and for a pressure of 130 lb. per sq. in. absolute the number of heat units required to superheat the steam is 55.8. The pound of superheated steam will, therefore, contain

$$1191 + 55.8 = 1246.8 \text{ B.t.u.}$$

Since superheated steam contains more heat than the same weight of saturated steam it is evident that the superheated steam is also dry.

PROPERTIES OF DRY SATURATED STEAM
from Marks' and Davis' Steam Tables

1 Absolute pressure lb. per sq. in. p	2 Temperature; degrees Fahrenheit t	3 Heat of the liquid per pound B.t.u. h	4 Latent heat of evaporation per pound B.t.u. L	5 Total heat per pound B.t.u. H	6 Volume of one pound cu. ft. v	7 Density or weight of one cu. ft. lbs. d
0.0886	32.0	0.00	1073.4	1073.4	3294.0	0.000304
1	101.83	69.8	1034.6	1104.4	333.0	0.00300
2	126.15	94.0	1021.0	1115.0	173.5	0.00576
3	141.52	109.4	1012.3	1121.6	118.5	0.00845
4	153.01	120.9	1005.7	1126.5	90.5	0.01107
5	162.28	130.1	1000.3	1130.5	73.33	0.01364
6	170.06	137.9	995.8	1133.7	61.89	0.01616
7	176.85	144.7	991.8	1136.5	53.56	0.01867
8	182.86	150.8	988.2	1139.0	47.27	0.02115
9	188.27	156.2	985.0	1141.1	42.36	0.02361
10	193.22	161.1	982.0	1143.1	38.38	0.02606
11	197.75	165.7	979.2	1144.9	35.10	0.02849
12	201.96	169.9	976.6	1146.5	32.36	0.03090
13	205.87	173.8	974.2	1148.0	30.03	0.03330
14	209.55	177.5	971.9	1149.4	28.02	0.03569
15	213.0	181.0	969.7	1150.7	26.27	0.03806
16	216.3	184.4	967.6	1152.0	24.79	0.04042
17	219.4	187.5	965.6	1153.1	23.38	0.04277
18	222.4	190.5	963.7	1154.2	22.16	0.04512
19	225.2	193.4	961.8	1155.2	21.07	0.04746
20	228.0	196.1	960.0	1156.2	20.08	0.04980
21	230.6	198.8	958.3	1157.1	19.18	0.05213
22	233.1	201.3	956.7	1158.0	18.37	0.05445
23	235.5	203.8	955.1	1158.8	17.62	0.05676
24	237.8	206.1	953.5	1159.6	16.93	0.05907
25	240.1	208.4	952.0	1160.4	16.30	0.0614
26	242.2	210.6	950.6	1161.2	15.72	0.0636
27	244.4	212.7	949.2	1161.9	15.18	0.0659
28	246.4	214.8	947.8	1162.6	14.67	0.0682
29	248.4	216.8	946.4	1163.2	14.19	0.0705

Properties of Dry Saturated Steam—*Continued*

1 Absolute pressure lb. per sq. in. p	2 Temperature; degrees Fahrenheit t	3 Heat of the liquid per pound B.t.u. h	4 Latent heat of evaporation per pound B.t.u. L	5 Total heat per pound B.t.u. H	6 Volume of one pound cu. ft. v	7 Density or weight of one cu. ft. lbs. d
30	250.3	218.8	945.1	1163.9	13.74	0.0728
31	252.2	220.7	943.8	1164.5	13.32	0.0751
32	254.1	222.6	942.5	1165.1	12.93	0.0773
33	255.8	224.4	941.3	1165.7	12.57	0.0795
34	257.6	226.2	940.1	1166.3	12.22	0.0818
35	259.3	227.9	938.9	1166.8	11.89	0.0841
36	261.0	229.6	937.7	1167.3	11.58	0.0863
37	262.6	231.3	936.6	1167.8	11.29	0.0886
38	264.2	232.9	935.5	1168.4	11.01	0.0908
39	265.8	234.5	934.4	1168.9	10.74	0.0931
40	267.3	236.1	933.3	1169.4	10.49	0.0953
41	268.7	237.6	932.2	1169.8	10.25	0.0976
42	270.2	239.1	931.2	1170.3	10.02	0.0998
43	271.7	240.5	930.2	1170.7	9.80	0.1020
44	273.1	242.0	929.2	1171.2	9.59	0.1043
45	274.5	243.4	928.2	1171.6	9.39	0.1065
46	275.8	244.8	927.2	1172.0	9.20	0.1087
47	277.2	246.1	926.3	1172.4	9.02	0.1109
48	278.5	247.5	925.3	1172.8	8.84	0.1131
49	279.8	248.8	924.4	1173.2	8.67	0.1153
50	281.0	250.1	923.5	1173.6	8.51	0.1175
51	282.3	251.4	922.6	1174.0	8.35	0.1197
52	283.5	252.6	921.7	1174.3	8.20	0.1219
53	284.7	253.9	920.8	1174.7	8.05	0.1241
54	285.9	255.1	919.9	1175.0	7.91	0.1263
55	287.1	256.3	919.0	1175.4	7.78	0.1285
56	288.2	257.5	918.2	1175.7	7.65	0.1307
57	289.4	258.7	917.4	1176.0	7.52	0.1329
58	290.5	259.8	916.5	1176.4	7.40	0.1350
59	291.6	261.0	915.7	1176.7	7.28	0.1372
60	292.7	262.1	914.9	1177.0	7.17	0.1394
61	293.8	263.2	914.1	1177.3	7.06	0.1416
62	294.9	264.3	913.3	1177.6	6.95	0.1438
63	295.9	265.4	912.5	1177.9	6.85	0.1460
64	297.0	266.4	911.8	1178.2	6.75	0.1482

PROPERTIES OF DRY SATURATED STEAM—*Continued*

1 Absolute pressure lb. per sq. in. p	2 Temperature; degrees Fahrenheit t	3 Heat of the liquid per pound B.t.u. h	4 Latent heat of evaporation per pound B.t.u. L	5 Total heat per pound B.t.u. H	6 Volume of one pound cu. ft. v	7 Density or weight of one cu. ft. lbs. d
65	298.0	267.5	911.0	1178.5	6.65	0.1503
66	299.0	268.5	910.2	1178.8	6.56	0.1525
67	300.0	269.6	909.5	1179.0	6.47	0.1547
68	301.0	270.6	908.7	1179.3	6.38	0.1569
69	302.0	271.6	908.0	1179.6	6.29	0.1590
70	302.9	272.6	907.2	1179.8	6.20	0.1612
71	303.9	273.6	906.5	1180.1	6.12	0.1634
72	304.8	274.5	905.8	1180.4	6.04	0.1656
73	305.8	275.5	905.1	1180.6	5.96	0.1678
74	306.7	276.5	904.4	1180.9	5.89	0.1699
75	307.6	277.4	903.7	1181.1	5.81	0.1721
76	308.5	278.3	903.0	1181.4	5.74	0.1743
77	309.4	279.3	902.3	1181.6	5.67	0.1764
78	310.3	280.2	901.7	1181.8	5.60	0.1786
79	311.2	281.1	901.0	1182.1	5.54	0.1808
80	312.0	282.0	900.3	1182.3	5.47	0.1829
81	312.9	282.9	899.7	1182.5	5.41	0.1851
82	313.8	283.8	899.0	1182.8	5.34	0.1873
83	314.6	284.6	898.4	1183.0	5.28	0.1894
84	315.4	285.5	897.7	1183.2	5.22	0.1915
85	316.3	286.3	897.1	1183.4	5.16	0.1937
86	317.1	287.2	896.4	1183.6	5.10	0.1959
87	317.9	288.0	895.8	1183.8	5.05	0.1980
88	318.7	288.9	895.2	1184.0	5.00	0.2001
89	319.5	289.7	894.6	1184.2	4.94	0.2023
90	320.3	290.5	893.9	1184.4	4.89	0.2044
91	321.1	291.3	893.3	1184.6	4.84	0.2065
92	321.8	292.1	892.7	1184.8	4.79	0.2087
93	322.6	292.9	892.1	1185.0	4.74	0.2109
94	323.4	293.7	891.5	1185.2	4.69	0.2130
95	324.1	294.5	890.9	1185.4	4.65	0.2151
96	324.9	295.3	890.3	1185.6	4.60	0.2172
97	325.6	296.1	889.7	1185.8	4.56	0.2193
98	326.4	296.8	889.2	1186.0	4.51	0.2215
99	327.1	297.6	888.6	1186.2	4.47	0.2237

PROPERTIES OF DRY SATURATED STEAM—*Continued*

1 Absolute pressure lb. per sq. in. p	2 Temperature; degrees Fahrenheit t	3 Heat of the liquid per pound B.t.u. h	4 Latent heat of evaporation per pound B.t.u. L	5 Total heat per pound B.t.u. H	6 Volume of one pound cu. ft. v	7 Density or weight of one cu. ft. lbs. d
100	327.8	298.3	888.0	1186.3	4.429	0.2258
101	328.6	299.1	887.4	1186.5	4.388	0.2279
102	329.3	299.8	886.9	1186.7	4.347	0.2300
103	330.0	300.6	886.3	1186.9	4.307	0.2322
104	330.7	301.3	885.8	1187.0	4.268	0.2343
105	331.4	302.0	885.2	1187.2	4.230	0.2365
106	332.0	302.7	884.7	1187.4	4.192	0.2336
107	332.7	303.4	884.1	1187.5	4.155	0.2408
108	333.4	304.1	883.6	1187.7	4.118	0.2429
109	334.1	304.8	883.0	1187.9	4.082	0.2450
110	334.8	305.5	882.5	1188.0	4.047	0.2472
111	335.4	306.2	881.9	1188.2	4.012	0.2593
112	336.1	306.9	881.4	1188.4	3.978	0.2514
113	336.8	307.6	880.9	1188.5	3.945	0.2535
114	337.4	308.3	880.4	1188.7	3.912	0.2556
115	338.1	309.0	879.8	1188.8	3.880	0.2577
116	338.7	309.6	879.3	1189.0	3.848	0.2599
117	339.4	310.3	878.8	1189.1	3.817	0.2620
118	340.0	311.0	878.3	1189.3	3.786	0.2641
119	340.6	311.6	877.8	1189.4	3.756	0.2662
120	341.3	312.3	877.2	1189.6	3.726	0.2683
121	341.9	313.0	876.7	1189.7	3.697	0.2705
122	342.5	313.6	876.2	1189.8	3.668	0.2726
123	343.2	314.3	875.7	1190.0	3.639	0.2748
124	343.8	314.9	875.2	1190.1	3.611	0.2769
125	344.4	315.5	874.7	1190.3	3.583	0.2791
126	345.0	316.2	874.2	1190.4	3.556	0.2812
127	345.6	316.8	873.8	1190.5	3.530	0.2833
128	346.2	317.4	873.3	1190.7	3.504	0.2854
129	346.8	318.0	872.8	1190.8	3.478	0.2875
130	347.4	318.6	872.3	1191.0	3.452	0.2897
131	348.0	319.3	871.8	1191.1	3.427	0.2918
132	348.5	319.9	871.3	1191.2	3.402	0.2939
133	349.1	320.5	870.9	1191.3	3.378	0.2960
134	349.7	321.1	870.4	1191.5	3.354	0.2981

PROPERTIES OF DRY SATURATED STEAM—*Continued*

1 Absolute pressure lb. per sq. in. p	2 Temperature; degrees Fahrenheit t	3 Heat of the liquid per pound B.t.u. h	4 Latent heat of evaporation per pound B.t.u. L	5 Total heat per pound B. .u. H	6 Volume of one pound cu. ft. v	7 Density or weight of one cu. ft. lbs. d
135	350.3	321.7	869.9	1191.6	3.331	0.3002
136	350.8	322.3	869.4	1191.7	3.308	0.3023
137	351.4	322.8	869.0	1191.8	3.285	0.3044
138	352.0	323.4	868.5	1192.0	3.263	0.3065
139	352.5	324.0	868.1	1192.1	3.241	0.3086
140	353.1	324.6	867.6	1192.2	3.219	0.3107
141	353.6	325.2	867.2	1192.3	3.197	0.3129
142	354.2	325.8	866.7	1192.5	3.175	0.3150
143	354.7	326.3	866.3	1192.6	3.154	0.3171
144	355.3	326.9	865.8	1192.7	3.133	0.3192
145	355.8	327.4	865.4	1192.8	3.112	0.3213
146	356.3	328.0	864.9	1192.9	3.092	0.3234
147	356.9	328.6	864.5	1193.0	3.072	0.3255
148	357.4	329.1	864.0	1193.2	3.052	0.3276
149	357.9	329.7	863.6	1193.3	3.033	0.3297
150	358.5	330.2	863.2	1193.4	3.012	0.3320
151	359.0	330.8	862.7	1193.5	2.993	0.3341
152	359.5	331.4	862.3	1193.6	2.974	0.3362
153	360.0	331.9	861.8	1193.7	2.956	0.3383
154	360.5	332.4	861.4	1193.8	2.938	0.3404
155	361.0	332.9	861.0	1194.0	2.920	0.3425
156	361.6	333.5	860.6	1194.1	2.902	0.3446
157	362.1	334.0	860.1	1194.2	2.885	0.3467
158	362.6	334.6	859.7	1194.3	2.868	0.3488
159	363.1	335.0	859.3	1194.4	2.851	0.3508
160	363.6	335.6	858.8	1194.5	2.834	0.3529
161	364.1	336.2	858.4	1194.6	2.818	0.3549
162	364.6	336.7	858.0	1194.7	2.801	0.3570
163	365.1	337.2	857.6	1194.8	2.785	0.3591
164	365.6	337.7	857.2	1194.9	2.769	0.3612
165	366.0	338.2	856.8	1195.0	2.753	0.3633
166	366.5	338.7	856.4	1195.1	2.737	0.3654
167	367.0	339.2	855.9	1195.2	2.72	0.3675
168	367.5	339.7	855.5	1195.3	2.706	0.3696
169	368.0	340.2	855.1	1195.4	2.690	0.3717

PROPERTIES OF DRY SATURATED STEAM—*Continued*

1 Absolute pressure lb. per sq. in. p	2 Temperature; degrees Fahrenheit t	3 Heat of the liquid per pound B.t.u. h	4 Latent heat of evaporation per pound B.t.u. L	5 Total h at per pound B.t.u. H	6 Volume of one pound cu. ft. v	7 Density or weight of one cu. ft. lbs. d
170	368.5	340.7	854.7	1195.4	2.675	0.3738
171	368.9	341.2	854.3	1195.5	2.660	0.3759
172	369.4	341.7	853.9	1195.6	2.645	0.3780
173	369.9	342.2	853.5	1195.7	2.631	0.3801
174	370.4	342.7	853.1	1195.8	2.616	0.3822
175	370.8	343.2	852.7	1195.9	2.602	0.3843
176	371.3	343.7	852.3	1196.0	2.588	0.3864
177	371.7	344.2	851.9	1196.1	2.574	0.3885
178	372.2	344.7	851.5	1196.2	2.560	0.3906
179	372.7	345.2	851.2	1196.3	2.547	0.3927
180	373.1	345.6	850.8	1196.4	2.533	0.3948
181	373.6	346.1	850.4	1196.5	2.520	0.3969
182	374.0	346.6	850.0	1196.6	2.507	0.3989
183	374.5	347.1	849.6	1196.7	2.494	0.4010
184	374.9	347.6	849.2	1196.8	2.481	0.4031
185	375.4	348.0	848.8	1196.8	2.468	0.4052
186	375.8	348.5	848.4	1196.9	2.455	0.4073
187	376.3	349.0	848.0	1197.0	2.443	0.4094
188	376.7	349.4	847.7	1197.1	2.430	0.4115
189	377.2	349.9	847.3	1197.2	2.418	0.4136
190	377.6	350.4	846.9	1197.3	2.406	0.4157
191	378.0	350.8	846.5	1197.3	2.393	0.4178
192	378.5	351.3	846.1	1197.4	2.381	0.4199
193	378.9	351.7	845.8	1197.5	2.369	0.4220
194	379.3	352.2	845.4	1197.6	2.358	0.4241
195	379.8	352.7	845.0	1197.7	2.346	0.4262
196	380.2	353.1	844.7	1197.8	2.335	0.4283
197	380.6	353.6	844.3	1197.8	2.323	0.4304
198	381.0	354.0	843.9	1197.9	2.312	0.4325
199	381.4	354.4	843.6	1198.0	2.301	0.4346
200	381.9	354.9	843.2	1198.1	2.290	0.437
205	384.0	357.1	841.4	1198.5	2.237	0.447
210	386.0	359.2	839.6	1198.8	2.187	0.457
215	388.0	361.4	837.9	1199.2	2.138	0.468
220	389.9	363.4	836.2	1199.6	2.091	0.478

PROPERTIES OF DRY SATURATED STEAM—*Continued*

1 Absolute pressure lb. per sq. in. p	2 Temperature; degrees Fahrenheit t	3 Heat of the liquid per pound B.t.u. h	4 Latent heat of evaporation per pound B.t.u. L	5 Total heat per pound B.t.u. H	6 Volume of one pound cu. ft. v	7 Density or weight of one cu. ft. lbs. d
225	391.9	365.5	834.4	1199.9	2.046	0.489
230	393.8	367.5	832.8	1200.2	2.004	0.499
235	395.6	369.4	831.1	1200.6	1.964	0.509
240	397.4	371.4	829.5	1200.9	1.924	0.520
245	399.3	373.3	827.9	1201.2	1.887	0.530
250	401.1	375.2	826.3	1201.5	1.850	0.541
260	404.5	378.9	823.1	1202.1	1.782	0.561
270	407.9	382.5	820.1	1202.6	1.718	0.582
280	411.2	386.0	817.1	1203.1	1.658	0.603
290	414.4	389.4	814.2	1203.6	1.602	0.624
300	417.5	392.7	811.3	1204.1	1.551	0.645

CHAPTER VI

INDICATORS

Work Diagrams.—A diagram may be drawn, by means of an instrument called an indicator, which shows the work performed by the steam in the cylinder of a steam engine, and from such a diagram may be calculated the horsepower developed by the engine.

The area of any diagram is equal to the product obtained by multiplying together its two sides. For example, in Fig. 49 the figure *abcd* has one side equal to 6 feet and the other side equal to 3 feet, hence the area of the diagram *abcd* is $6 \times 3 = 18$ sq. ft. In Fig. 50 is a similar diagram, except that one side represents distance in feet and the other side represents force in pounds.

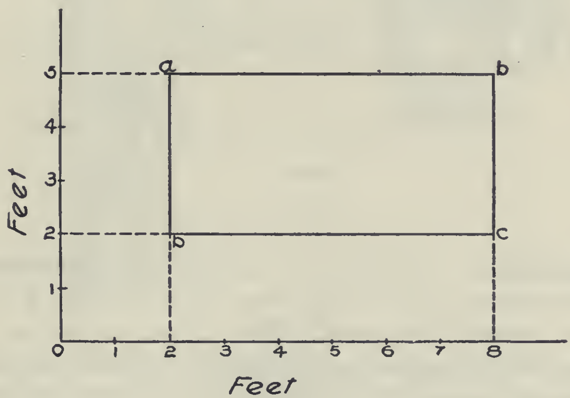


Fig. 49.

The area of this diagram *abcd* is equal to the product of the two sides or $6 \times 3 = 18$, but in this case the area represents foot-pounds, since the product of feet and pounds is foot-pounds or

$$6 \times 3 = 18 \text{ foot-pounds.}$$

Since foot-pounds is the unit of work, the area of the diagram *abcd* in Fig. 50, represents work, or is a work diagram. In a similar manner, if a diagram is drawn so that one side represents distance and the other side represents force (or pressure, which is a force) the area of the diagram will represent foot-pounds of work. This is the same principle upon which an indicator draws a work diagram for the cylinder of a steam engine. The indicator is so arranged that it draws on a sheet of paper a diagram which represents by its height the pressure of the steam in the cylinder and by its length represents the stroke of the piston. Such a diagram, with one side representing distance and the other side representing pressure, shows by its area the work being performed in the cylinder.

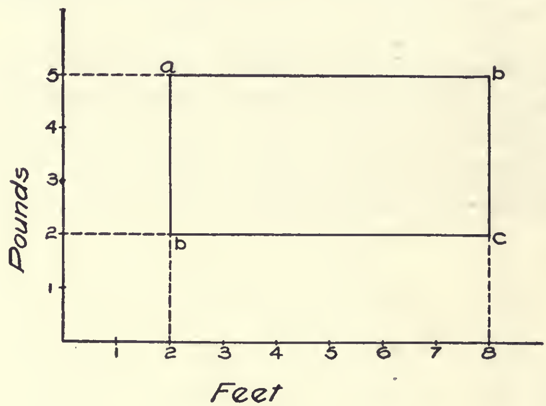


Fig. 50.

The Indicator.—An indicator must record the pressure in the cylinder at every part of the stroke. In order to do this, the indicator consists of two parts, one of which moves in a vertical direction, with the steam pressure in the cylinder and the other which moves in a horizontal direction in unison with the piston. The part moving horizontally in unison with the piston carries a sheet of paper and the part moving vertically in unison with the pressure carries a pencil point. When the pencil point is brought into contact with the paper, a diagram is drawn which shows the pressure in the cylinder at every part of the stroke.

One type of indicator, shown in Fig. 51, is partly cut away in order to show the inside construction of it. The device for measuring the steam pressure consists of a small cylinder 5, fitted with a piston 8, the cylinder being attached to the clearance space of the engine cylinder so that the steam pressure in the clearance space acts upon the under side of the indicator piston. A piston rod 10 is connected to the piston by means of a ball joint in order to give flexibility between the rod and the piston. The piston rod passes through the cylinder head 2, and its end is joined to the pencil arm 16 by means of a short link 14. One end of the pencil arm is pivoted at 18. The other end, which moves over a sheet of paper on the drum 24, carries a pencil point, and is forced to move in a vertical line by the links 14 and 15. The piston is held down by a finely adjusted coil spring and the steam pressure forces the piston upward against the resistance

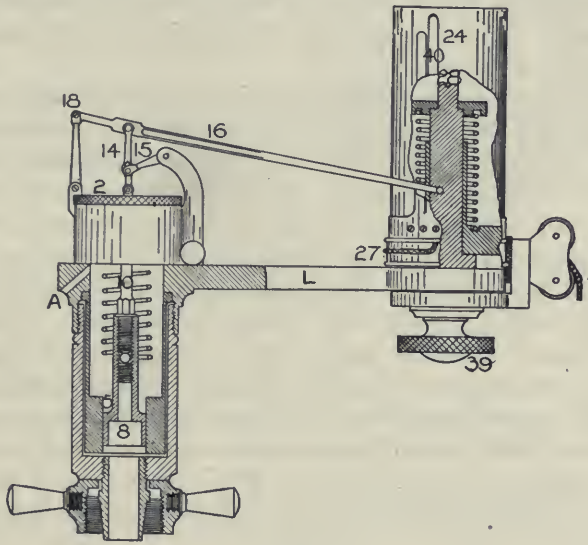


Fig. 51.

of this spring. These springs are interchangeable and several are supplied with each indicator. The number of the spring indicates the number of pounds pressure per square inch which, acting upon the piston, will move the pencil point one inch; for example, if a No. 60 spring is in the indicator it will require a pressure of 60 lb. per sq. in. on the piston to move the pencil point one inch vertically. Therefore, the diagram drawn by the indicator may be measured and the pressure in the engine cylinder at any point of the stroke determined. An indicator spring should always be used with the instrument for which it is intended as its scale depends upon the length of the pencil arm and a slight difference in the length of the arm makes considerable difference in the movement of the pencil, since the movement of the pencil point is usually five times the movement of the piston. The pencil arm, together with its links, is free to turn about the axis of the cylinder so the pencil point may be pressed against the drum or lifted away from it.

The piston is ground to a close fit with the cylinder in order to make a nearly steam tight fit and, at the same time, not cause excessive friction. Any steam that leaks past the piston escapes through the hole *A* in the cylinder and is thus prevented from collecting over the piston and exerting a downward pressure upon it. This piston has a number of small grooves cut in its edge, which serve to hold lubricating oil and also aid in preventing the leakage of steam.

The drum 24, which carries the sheet of paper and which moves in unison with the engine piston, is placed parallel to the indicator cylinder and is located at the end of the arm *L*, which forms a part of the indicator cylinder. The drum is clamped to the end of the arm *L*, by means of the thumb nut 39. A cord wrapped around the base of the drum in the groove 27 serves to turn it about its vertical axis. The cord takes its motion from the crosshead, which has the same motion as the piston. The cord is not usually attached directly to the crosshead because the stroke of the crosshead is greater than the circumference of the drum and would therefore turn it through more than one complete revolution, which would be undesirable on account of the pencil point striking the paper clips, 40. The drum is usually provided with stops which prevent it from turning through a complete revolution.

The circumference of the drum is about 5 inches and it is desirable to have the indicator diagram about 3 to 3½ inches long, while the crosshead may have a stroke of 2 or 3 feet, hence it is necessary to use some device which copies the motion of the crosshead on a reduced scale and to attach the cord to this device, which is called a reducing motion. Various forms of reducing motions will be described later.

On the forward stroke of the crosshead the cord which is wrapped around the drum is pulled outward, turning the drum through a part of a revolution. At the same time a coil spring inside the drum is wound up. This coil spring has one end attached to the drum and the other end to the stationary spindle 28, hence, when the crosshead makes the return stroke the drum turns in the opposite direction, keeping the cord taut and rewinding it in its groove 27.

All the moving parts of an indicator are made as light as possible, to avoid inertia effects, or over-travelling. This is especially necessary with the piston and connected parts, as these move rapidly, and inertia would cause the pencil point to move too high on its upward stroke and too low on its downward stroke. For the same reason, it is advisable to have a rather stiff spring in the indicator, or one which will give a diagram about $1\frac{1}{2}$ to 2 inches high. The number of the spring to use depends both upon the boiler pressure and the speed of the engine; thus with a boiler pressure of 90 lb. per sq. in. a No. 60 spring should be used with a high speed engine, as this will give a diagram with a maximum height of $\frac{90}{60} = 1\frac{1}{2}$ inches. The number of spring to be used with any pressure may be found by dividing the pressure by the desired height of diagram, but it should be remembered that a high speed engine requires a stiffer spring than a slow speed one.

Example.—What number of indicator spring should be used with a boiler pressure of 125 lb. per sq. in. if it is desired to obtain a diagram $1\frac{1}{2}$ inches high?

Solution.—
$$\frac{125}{1\frac{1}{2}} = 83.3$$

As the nearest regular size of spring is 80, this would be used. The regular sizes of springs are 8, 10, 12, 16, 20, 30, 40, 50, 60, 80, 100, 120, 150, and 180.

An indicator is attached to a cylinder by means of a short length of pipe and a quick-opening valve placed just below it. This valve, which is furnished with the instrument, is used so that steam pressure may be cut off when the indicator is not being used, thus reducing wear on the working parts.

All engine cylinders have holes at each end which are bored and threaded for attaching indicators, the holes entering the clearance space so they will not be covered by the piston at any time. If possible, it is best to use two indicators, one attached to each end of the cylinder, as this makes it possible to take diagrams from the two ends of the cylinder at the same time and also permits shorter connections between the indicator and cylinder. A long indicator connection is likely to cause a drop in pressure and thus make the indicator give a false record. Straightway valves are used for the same reason.

A single indicator is sometimes attached as shown in Fig. 52 and used for taking diagrams from both ends of the cylinder, in which case both diagrams are drawn on the same sheet of paper. The principal disadvantage in using a single indicator is the time required to shut off steam from one end of the cylinder and turn

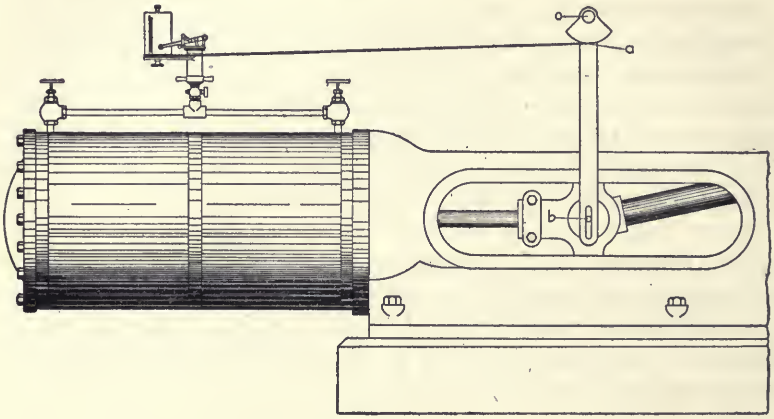


Fig. 52.

it on from the other end, which does not allow diagrams to be taken from the two ends of the cylinder at the same time. The time elapsing between taking the diagrams may, however, be greatly shortened by using a three-way valve, as shown in Fig.

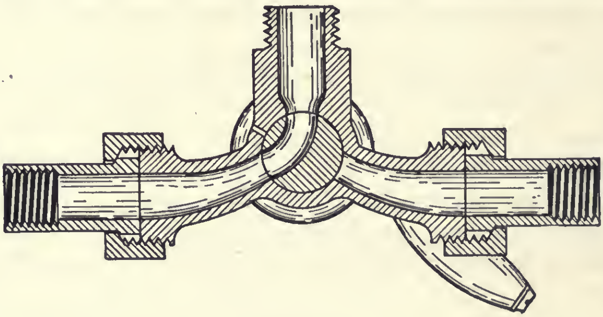


Fig. 53.

53, at the indicator, instead of the straightway valve shown in Fig. 52.

Indicators made by the various manufacturers differ from each other in small details, but most of them have the general form illustrated by Fig. 51. An indicator having a different form of pencil movement is shown in Fig. 54. In this indicator one of the usual movable links is replaced by a slot cut in a plate *G* and the pencil arm has a small roller on its side which moves in this slot. The slot is for the purpose of giving a perfect straight line motion to the pencil point and for securing a uniform proportion between the motions of the pencil point and indicator piston, and it is shaped to secure these results. In this indicator,

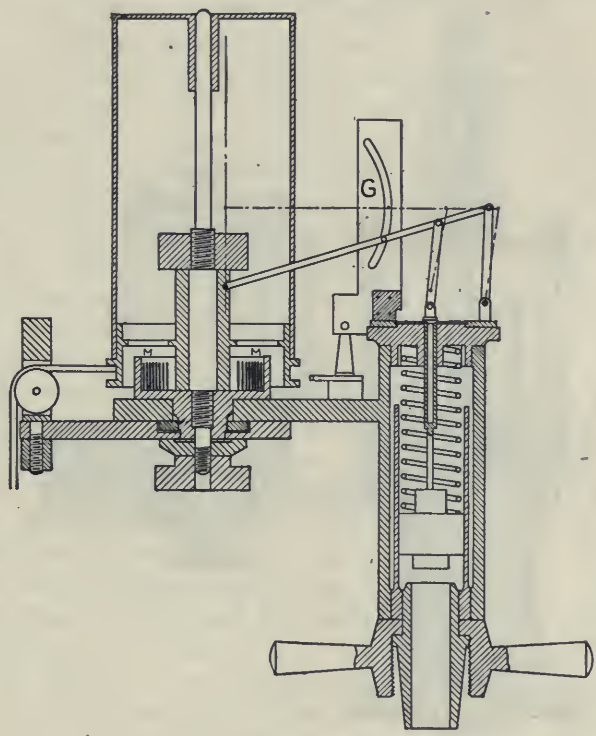


FIG. 54.

the drum spring is a flat coil spring placed at the base of the drum, as shown at *M*.

Fig. 55 shows a form of indicator having the spring outside the cylinder, the piston rod being made longer to hold it. When the spring is placed inside the cylinder and used with high pressure steam, the high temperature to which it is subjected is liable to change its stiffness and hence its scale. The outside spring arrangement is intended to overcome this disadvantage as well as to simplify the operation of changing springs for different pressures. The outside spring is especially adapted to superheated steam.

The indicator shown in Fig. 55 also illustrates another recent improvement in indicators, that is, a drum with which a number of diagrams may be drawn on the same paper. It is designed to use a roll of paper upon which the indicator traces a series of diagrams which will continue until the roll is exhausted, unless

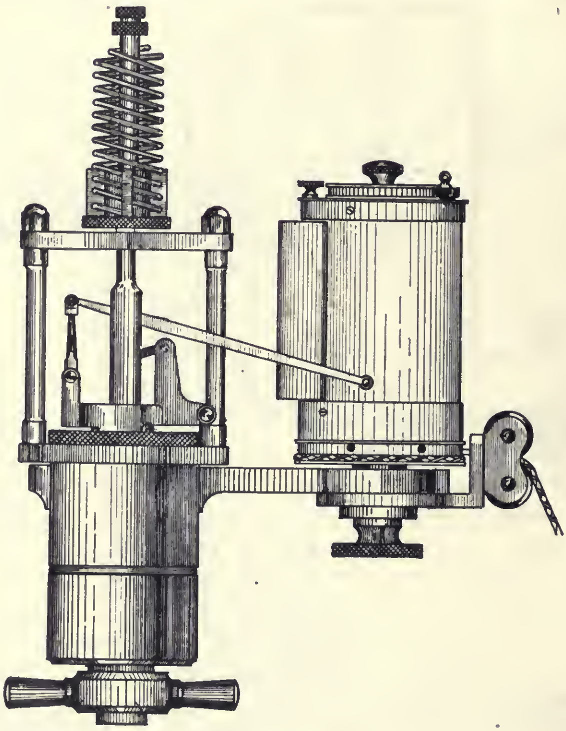


Fig. 55.

interrupted by the operator. The roll of paper is located within an opening in the shell of the drum, thence the paper passes around the outside of the drum and inward to a central cylinder, to which it is attached. Upon the top of the drum is a ratchet wheel which automatically unwinds a small length of paper from the spool and winds it on the inner cylinder, thus giving a series of diagrams which overlap each other slightly, as shown in Fig. 56. These indicators are commonly used with engines in which the load changes rapidly, such as rolling mill engines, because they record the changes in load and also show the action of the engine under such changes. The other ordinary form of indicator is suitable for drawing a diagram during only a single revolution, and if the pencil point is held on the paper for more than a single revolution, the diagrams will be drawn upon each other, making it difficult to distinguish them.

Reducing Motions.—The principal requirement of a reducing motion is that it shall reproduce *accurately* the motion of the crosshead on a small scale. Some reducing motions which are very simple in construction do not reproduce the motion of the crosshead accurately.

A form of reducing motion intended to be attached directly to the indicator is illustrated in Fig. 56*a*. This reducing motion consists of a large pulley around which is wrapped the cord which connects with the crosshead of the engine. This pulley drives a smaller pulley by means of bevel gears which reduce the motion. The smaller pulley drives the indicator drum by means of a cord wrapped around the drum and the smaller pulley. This reducing motion is supplied with several different sizes of the smaller pulley which adapts it to steam engines having piston strokes ranging from 14 inches to 72 inches.

This type of reducing motion will reproduce the motion of the crosshead accurately if cords are used which do not stretch and if the cords are prevented from piling on top of each other as they wrap around the pulleys.

The reducing motion shown in Fig. 52 is one of the simplest and most common forms. It consists of a wooden arm pivoted at the top to a stand which is attached either to the floor or to the frame of the engine. The lower end of this arm has a slot for receiving a pin fitted to the center of the wrist pin. The upper end of the arm has a curved block fastened to it with a groove in its edge in which is placed the cord to the indicator. In order to reproduce accurately the motion of

FIG. 56.

the crosshead the curvature of the block must be such that the distance *ob* from the pivot to the center of the wrist pin divided by the distance *oa* from the pivot to the center of the cord must

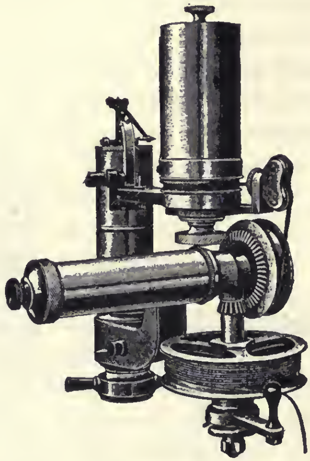


Fig. 56*a*.

be constant at all points of the stroke, and also the pivot must be directly over the center of the wrist pin when the

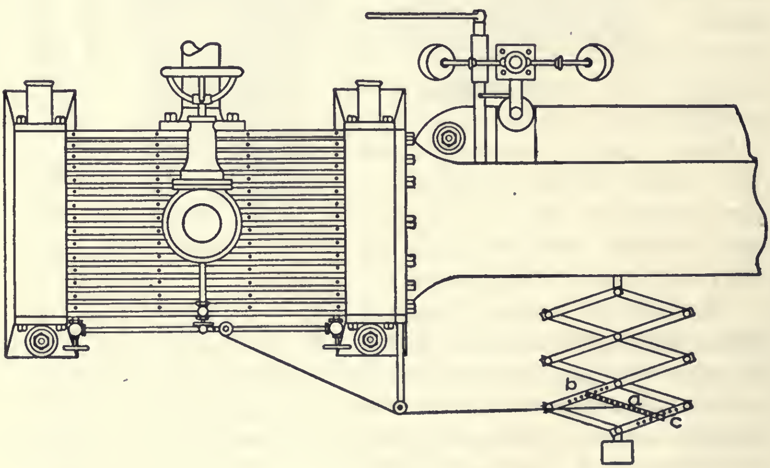


Fig. 57.

crosshead is at the middle of its stroke. Lost motion between the slot and the pin in the wrist pin must also be avoided.

The top view of the engine illustrated in Fig. 57, shows a pantograph reducing motion; one end of the pantograph being attached to the crosshead and the other end to a stand placed on the floor. The indicator cord is attached to the crossbar *a*, which is attached to the bars *b* and *c*. The reducing motion may be used with different lengths of stroke by attaching the crossbar *a* at various points along the bars *b* and *c*, but always fastening it in corresponding holes in *b* and *c*. On account of the large number of joints in the pantograph and the probability of lost motion in them, this form of reducing motion should be made of steel with closely fitting joints. When so made the pantograph reproduces the motion of the crosshead accurately.

In attaching the cord to the reducing motion, it should always

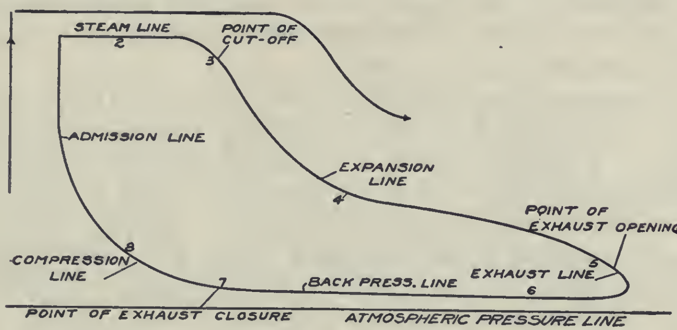


FIG. 58.

be arranged so it will run in a direction parallel to the crosshead, as in Fig. 57, otherwise the motion which the drum receives will not be the same as that of the crosshead. Also the cord must be attached to cross bar *a* on the center line of the pantograph or a correct reduction of the crosshead movement will not be obtained. Several holes should therefore be provided in cross bar *a* for the insertion of the pin so that the latter may be properly located as the bar *a* is moved from one position to another on bars *b* and *c* for different lengths of stroke.

Indicator Diagrams.—In taking an indicator diagram, a paper card, especially prepared for the purpose, is placed on the drum and the cord attached to the reducing motion. Steam is then turned into the indicator and after it is warmed up the pencil point is touched to the drum while the engine is making a single revolution. Steam is then turned off the indicator and the pencil point again touched to the drum in order to draw the atmospheric line. The resulting diagram will be similar to that shown in Fig. 58.

Since the atmospheric line is drawn while only the pressure of the atmosphere is acting upon the piston of the indicator, this line represents the pressure of the atmosphere and it serves as a reference line for other parts of the diagram. Gage pressures may be measured from the atmospheric line, but if absolute pressures are desired, it will be necessary to draw a line of *no pressure* parallel with the atmospheric line and at a distance below it equal to the atmospheric pressure as read on a barometer, and drawn to the same scale to which the diagram is drawn.

The indicator diagram shows the varying pressure in the cylinder for a complete revolution or a forward and back stroke, and anything which affects this pressure also affects the shape of the diagram. It also shows, by its area, the work being performed in the cylinder. The method of calculating the work from the diagram will be given later.

The diagram shown in Fig. 58 is from only one end of the cylinder, the end towards the left, but since it shows all changes of pressure, it gives a record of a complete cycle of the events taking place in that end of the cylinder. These events have been marked on the diagram for reference. The point at which steam is admitted to the cylinder is shown at the left, slightly before the piston reaches the end of its back stroke. As soon as steam is admitted the pressure in this end of the cylinder rises to the full admission pressure. By the time this has occurred the piston has reached the end of its return stroke and is starting on its forward stroke. During the first part of the forward stroke steam is being admitted behind the piston, hence the pressure remains constant during this part of the stroke. The part of the diagram drawn while steam is being admitted is called the steam line. At the end of the steam line is the point of cut-off at which the admission valve closes. On account of the gradual closing of this valve, the pressure changes gradually and the point of cut-off is not sharply defined. After the admission valve closes, the steam in the cylinder expands behind the advancing piston, as shown by the expansion line, the pressure of the steam gradually becoming smaller as its volume increases. Just before the piston has completed its forward stroke the exhaust valve opens and the pressure of the steam quickly drops while the piston is completing its stroke. The exhaust valve remains open during the greater part of the return stroke, and the piston pushes the low pressure steam from the cylinder, giving the exhaust line on the diagram. As the ports and exhaust passages offer a certain amount of resistance to the flow of the exhaust steam, the exhaust line will be above the atmospheric line by a distance which represents a few pounds pressure. This pressure is called the "back-pressure" since it acts against the advancing piston.

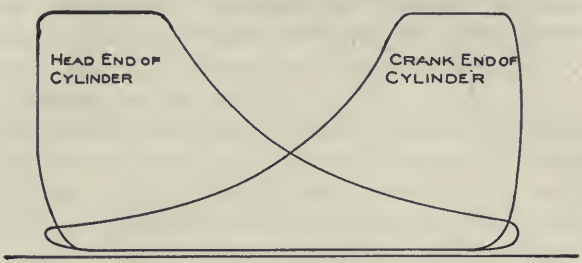


Fig. 59.

Near the end of the exhaust stroke the exhaust valve closes, giving the "point of compression" or "exhaust closure." After the exhaust valve is closed the steam remaining in the cylinder is compressed until the admission valve opens, thus completing the cycle of events for this end of the cylinder.

Similar events occur in the other end of the cylinder but not at the same time. These may be shown on a separate diagram

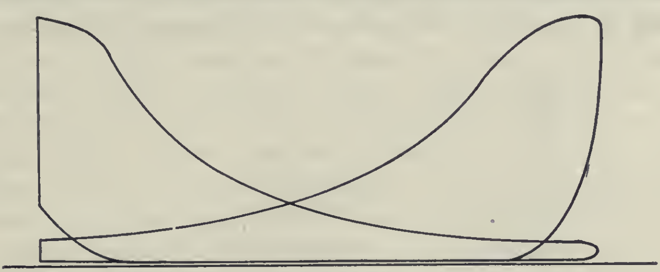


Fig. 60.

drawn with another indicator, or, if a single indicator is used for both ends of the cylinder, both diagrams will be drawn on one paper or "card," and this will show the relative positions of the events. Such a double diagram is illustrated in Fig. 59, which shows that admission and expansion are occurring in the head end of the cylinder while exhaust is taking place from the crank end, and that admission and expansion are occurring in the crank end while exhaust is taking place from the head end.

Besides being used to determine the power developed by an engine, the indicator diagram also shows whether or not the engine and indicator are adjusted properly. Faults in the engine adjustment will be considered in a later chapter. A few of the more common indicator faults will be considered here. Fig. 60 shows diagrams taken by an indicator in which the cord is too long, thus allowing the drum to stop before the crosshead has

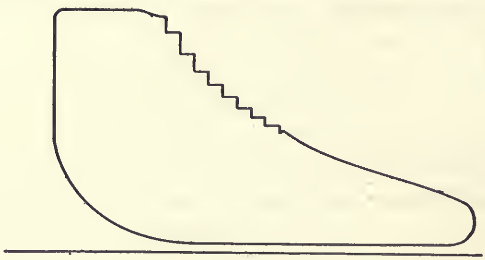


FIG. 61.

completed its stroke. It will be seen that the left-hand ends of both diagrams appear to be cut off, the heel of one diagram and the toe of the other being cut off on the same line. The same fault may be caused by the motion of the crosshead not being reduced sufficiently, but in this case the cord is liable to be broken.

Sometimes the piston of a new indicator will fit too tightly, causing it to stick in the cylinder. The result will show in a

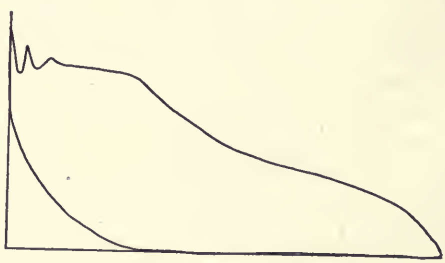


FIG. 62.

stepped expansion line as in Fig. 61. The steps will usually be more distinct near the beginning of the expansion line where the pressure is high. The same fault may be caused in an old indicator by a gummed piston which has not been cleaned and lubricated.

If the spring used in an indicator is too weak for the pressure, the diagram will not only be too high, but its steam line will be wavy, especially near the end, as shown in Fig. 62. Such a wavy line is caused by the vibration of the spring when high pressure steam is first admitted to the cylinder. The remedy for this is, of course, to use a stiffer spring.

Expansion of Steam.—Between the point of cut-off and release the weight of steam in the cylinder remains constant provided there is no leakage of steam either into or out of the cylinder. The steam that is in the cylinder simply expands, that is, its volume increases and its pressure falls.

If we should select a number of points along the expansion line of an indicator diagram and multiply the absolute pressure at each of these points by the volume of steam in the cylinder at that point we would find that the product of this multiplication would be practically a constant number. This being true it is evident that the pressure of the steam falls at the same rate that its volume increases. When the volume of the steam has increased to twice the volume contained in the cylinder at the point of cut-off, the absolute pressure of the steam will be one-half of what it was at the point of cut-off. In like manner, when the steam has expanded so that its volume is 1.5 times its volume at cut-off its absolute pressure will be $\frac{2}{3}$ or $\frac{1}{1.5}$ of what it was at the point of cut-off; and when the steam has expanded so that its volume is 4 times its volume at cut-off, its absolute pressure will be $\frac{1}{4}$ of its absolute pressure at cut-off. It should be observed that the volume of steam which is expanding refers to the total volume of steam which is in the cylinder when cut-off occurs. This volume includes not only the volume of steam taken into the cylinder at each stroke, which is the same as the volume displaced by the piston from the beginning of its stroke up to the point of cut-off, but it includes also the volume of steam in the clearance space when the piston is at the beginning of its stroke.

Example.—Cut-off occurs at $\frac{5}{8}$ stroke in a $10'' \times 12''$ engine having 12 per cent. clearance. What is the total volume of steam in the cylinder at the beginning of expansion?

Solution.—The area of the piston is

$$\frac{10 \times 10}{144} \times .7854 = .5454 \text{ sq. ft.}$$

Since the length of stroke is 12 in. or 1 ft., the piston displacement is

$$.5454 \times 1 = .5454 \text{ cu. ft.}$$

and the clearance volume is

$$.5454 \times .12 = .0654 \text{ cu. ft.}$$

Since cut-off occurs at $\frac{5}{8}$ stroke the volume of steam taken into the cylinder at each stroke is

$$\tfrac{5}{8} \times .5454 = .3409 \text{ cu. ft.}$$

and the total volume of steam at the beginning of expansion is

$$.3409 + .0654 = \underline{.4063} \text{ cu. ft.}$$

The quantities calculated in the above example are shown on the indicator diagram illustrated in Fig. 63. In this illustration the piston displacement is represented by the length of the diagram. The clearance volume *c* which in this case, is 12 per cent. of the piston displacement is represented by the distance between the end of the diagram and the line *ao*, which is the line of no

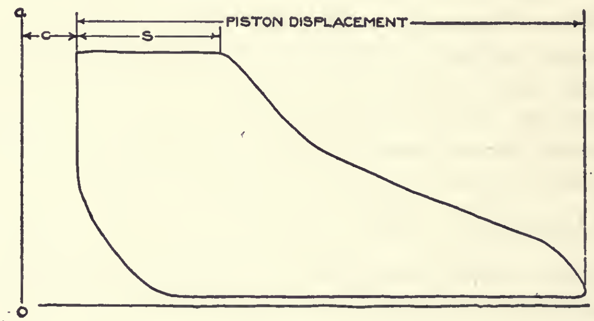


Fig. 63.

volume. The line *ao* is located by making the distance *c* equal to the clearance volume to the same scale that the length of the indicator card represents the piston displacement. For example, suppose the indicator diagram is 2.5 in. long, and this 2.5 in. represents the piston displacement of the above example or .5454 cu. ft. In this case a length of $\frac{2.5}{.5454}$ or 4.583 inches would represent 1 cu. ft. of volume. Therefore the clearance volume *c*, which is .0654 cu. ft. would be represented by a length of $.0654 \times 4.583 = .2997$ in. or practically .3 in. That is, the no volume line *ao* would be drawn .3 in. from the end of the indicator diagram.

Ratio of Expansion.—The *ratio of expansion* is a measure of the number of times the steam is expanded in the cylinder. For example, if there are two cubic feet of steam in the cylinder when cut-off occurs (at the beginning of expansion) and this is expanded to four cubic feet, its volume has been increased *two* times, or its ratio of expansion is two.

Since the increase in volume of steam during expansion is practically in proportion to the decrease in pressure, the pressure of the steam at the end of expansion may be calculated, if the admission pressure and the ratio of expansion are known.

Example.—If the admission pressure is 60 lb. per sq. in. and the ratio of expansion is 4, what will be the pressure in the cylinder at the end of expansion?

Solution.—Pressure at end of expansion

$$= \frac{60}{4} = 15 \text{ lb. per sq. in.}$$

It may be seen from the above discussion and example that for a given admission pressure the final pressure will be lowest with an early cut-off, or large ratio of expansion. Also, if cut-off occurs late in the stroke, the steam being expanded but little, the pressure will be high at the end of the stroke when the exhaust valve opens. In the latter case, the pressure remaining in the steam is wasted, hence an early cut-off or large ratio of expansion is more desirable than a small ratio of expansion.

The number of times which steam may be expanded in a cylinder depends upon the admission pressure of the steam as well as upon the point of cut-off. If this pressure is low and the steam is expanded a large number of times, the final pressure will be carried below the exhaust pressure, forming a loop at the toe of the diagram, and no useful work will be gained from the last part of the expansion. For example, if the admission pressure is 60 lb. per sq. in. and the ratio of expansion is 6, the final pressure of the steam will be 10 lb. per sq. in., which is below atmospheric pressure. If this engine exhausts into the atmosphere, the expansion below 14.7 lb. per sq. in. produces no useful work because, when the exhaust valve opens, the pressure in the cylinder will rise to 14.7 lb. per sq. in. In fact, the pressure in the cylinder at the end of expansion should be a few pounds above the exhaust pressure because, if the steam is expanded completely to the exhaust pressure, the extra work gained is not enough to compensate for the friction of the engine during the last part of the stroke; hence, instead of there being a gain from the expansion of the last few pounds of pressure, there is actually a loss.

An approximate value of the ratio of expansion may be taken as the reciprocal of the fraction of the piston stroke at which cut-off occurs. For example, if cut-off occurs at $\frac{1}{4}$ stroke this approximate value of the ratio of expansion is 4; if cut-off occurs at $\frac{3}{4}$ stroke the approximate ratio of expansion is $\frac{4}{3}$ or $1\frac{1}{3}$. This method of computing the ratio of expansion gives only an approximate value because the clearance volume is neglected.

Whether the ratio of expansion is calculated by the exact method or the approximate method it is necessary to locate the point of cut-off on the indicator diagram. As the point of cut-off is not sharply defined on the indicator diagram it is rather difficult to locate the exact point of cut-off; but it may be done with a fair degree of accuracy by locating the point of cut-off at the point where the downward curve of the admission line meets the upward curve of the expansion line.

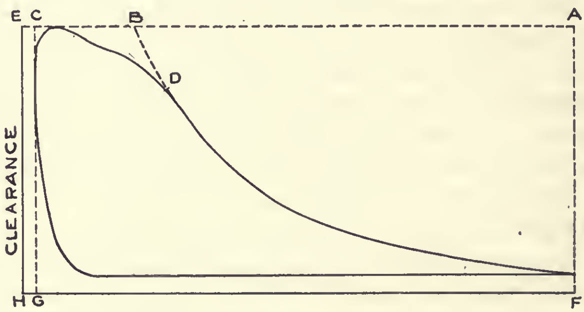


Fig. 64.

The difficulty of locating the point of cut-off exactly on indicator diagrams has led to the use of the *commercial cut-off* in determining the ratio of expansion. The commercial cut-off is located by drawing a horizontal line on the diagram through the maximum admission pressure and extending the expansion line up to meet this line. The intersection of these two lines is the commercial cut-off.

The method of determining the commercial cut-off and from it the ratio of expansion is illustrated in Figs. 64 and 65. The line *EH* is first drawn so that the length *EC* represents the clearance volume to the same scale that the diagram is drawn. The line EA is then drawn through the maximum admissiom pressure and parallel to the atmospheric line. In case the admission line is wavy, as in Fig. 65, the line *EA* is drawn at the average height of the waves. From the point *D* where the expansion line changes direction of curvature the expansion line is extended upward to intersect the line EA at the point B. The point B is the point of *commercial cut-off*. The fraction of stroke up to the commercial cut-off is

$$\frac{BC}{AC}$$

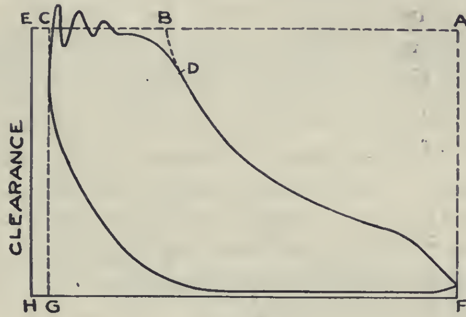


Fig. 65.

and the ratio of expansion is

$$\frac{AC + EC}{BC + EC} = \frac{AE}{BE}$$

These distances may be measured directly on the diagram or they may be calculated from the piston displacement and clearance volume, if the point of commercial cut-off is known.

CHAPTER VII

INDICATED AND BRAKE HORSEPOWER

Mean Effective Pressure.—The area of the diagram, which represents the work being performed in the cylinder, may be found by multiplying together its height and length, having proper regard for the scales of pressure and stroke, but since the diagram is of irregular shape its average height must be used in this calculation. The average height of an indicator diagram is called its *mean effective pressure,* abbreviated M.E.P. Multiplying together the M.E.P. in pounds per square inch, the length of the stroke in feet, and the area of the piston in square inches will give the number of foot-pounds of work performed during the time in which the diagram was made, or one revolution. Multiplying the above product by the number of revolutions per minute will give the number of foot-pounds of work performed per minute. This may be expressed in a formula as follows:

$$W = Plan$$

in which W = the number of foot-pounds of work per minute
P = the M.E.P. in pounds per sq. in.
l = the length of the stroke in feet
a = the area of the piston in sq. in.
n = the number of revolutions per minute (r.p.m.)

Example.—A $20'' \times 24''$ engine makes 240 r.p.m. and the indicator diagrams show a M.E.P. of 63 lb. per sq. in. How many foot-pounds of work is the engine performing per minute?

Solution.—The length of the stroke is $24'' = 2$ ft.

The are of the piston is

$$.7854 \times 20^2 = 314.16 \text{ sq. in.}$$

Hence the work performed is

$$\begin{aligned} W &= Plan \\ &= 63 \times 2 \times 314.16 \times 240 \\ &= 9{,}500{,}198 \text{ ft.-lb. per min.} \end{aligned}$$

The mean effective pressure may be measured directly from the indicator diagram by the method of ordinates or by means of an instrument called a planimeter. The method of ordinates consists in dividing the diagram into a number of parts and measuring its height at each of these parts, taking into account the scale to which the diagram is drawn, that is, the number of the indicator spring.

The ordinate method of measuring the mean effective pressure is illustrated by Fig. 66. The limiting lines at right and left of the diagram are first drawn perpendicular to the atmospheric line, and the space between them divided into ten equal parts. Vertical lines, as shown at 1, 2, 3, 4, etc., running through the center of these spaces are then drawn and the length of each

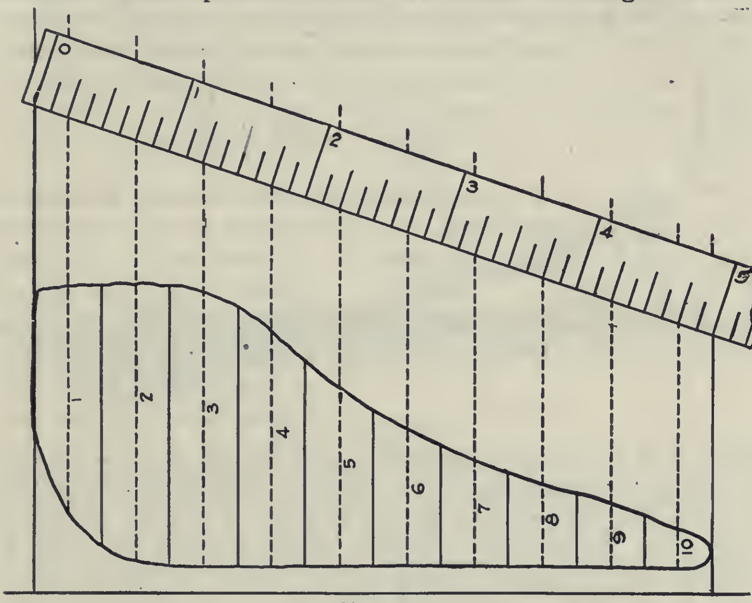


FIG. 66.

between the upper and lower lines of the diagram is measured. Adding these lengths together, multiplying their sum by the scale of the indicator spring, and dividing by 10 will give the average pressure or M.E.P.

To obtain the centers of the ten spaces previously mentioned, a convenient method is to take an ordinary scale and place it as shown in Fig. 66 so that the diagonal length between the limits of the diagram will be exactly 5 inches. Then at the left of the scale point off at ¼ inch, and from there on every ½ inch towards the right of the diagram. The last point will be at 4¾ inches. From these points draw vertical lines through the diagram perpendicular to the atmospheric line.

A convenient method of obtaining the combined lengths of the ordinates is to take a narrow strip of paper and mark on its edge the height of each ordinate. Begin with No. 1 and mark its length on the paper. Then place the mark made for No. 1 at the end of No. 2 and make a mark on the paper at the other end of No. 2 and so on until the combined length of the 10 ordinates is obtained. Measure this combined length with an ordinary inch scale, multiply by the scale of the spring, and divide by 10. The result will be the M.E.P. for the diagram. For example, suppose the combined length of the 10 ordinates in Fig. 66 measures 7.8 inches and the diagram was drawn with a No. 60 spring. The M.E.P. would then be:

$$\frac{7.8 \times 60}{10} = 46.8 \text{ lb. per sq. in.}$$

In a noncondensing engine having insufficient load where the steam is cut off at a very early part of the stroke, a diagram may be obtained similar to the one shown in Fig. 67. In this case the diagram must be treated as two distinct parts, the loop at the toe of the diagram near the end of expansion being treated as negative and the other part being treated as positive. The whole diagram is, in this case, divided into 20 equal parts. From the combined lengths of the positive ordinates which number from 1 to 8, subtract the combined lengths of the negative ordinates, which number from 9 to 20, then multiply the difference by the scale of the spring and divide by 20, which is the number of ordinates. The result will be the mean effective pressure for the whole diagram. For example, if the combined length of the positive ordinates is 4 inches and the combined length of the negative ordinates 1.7 inches, the difference would be 2.3 inches. If the scale of the spring is 60, the M.E.P. would be

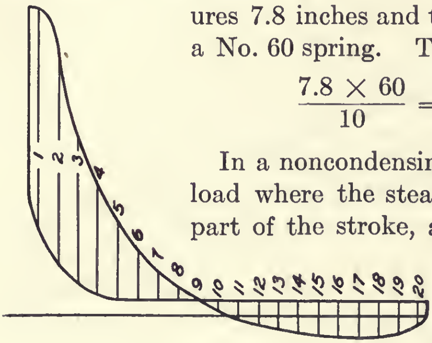


Fig. 67.

$$\frac{2.3 \times 60}{20} = 6.8 \text{ lb. per sq. in.}$$

which is the average pressure for the entire diagram.

The ordinate method of measuring the mean effective pressure, described above, gives only approximate results. The approximation is closer the larger the number of parts into which the diagram is divided, but 10 divisions give results which are close enough for most practical purposes. The most accurate and quickest method of measuring the mean effective pressure is by means of a planimeter and this method should be used if a planimeter is at hand.

There are several makes of planimeters on the market, alike in principle but differing in details of construction. The one shown in Fig. 68 will be found very convenient for measuring mean effective pressure from indicator diagrams. This instrument consists of two arms *AB* and *CD* which are pivoted so they may move with respect to each other. In preparing the instrument for use the two points *E* and *F* on the back of the arm *CD* are set a distance apart equal to the length of the diagram. This adjustment should be made as close as possible by hand and a

Fig. 68.

final and closer adjustment made by means of the screw *G*. The two arms of the planimeter are then held at approximately 90° to each other, the point *H* is placed near the center of the indicator diagram, and the point *J* is pressed firmly into the board to hold it stationary, the small weight *K* being placed on it for the same purpose. The point *H* is next placed at one corner of the diagram, preferably the upper left-hand corner, and the small wheel *M* turned until its zero is opposite the zero on the fixed scale. The instrument is now in position for measuring the M.E.P. of the diagram. This is done by tracing out the diagram with the point *H*, following around the diagram in a clockwise direction. When the point *H* has been moved entirely around the diagram and brought back to its starting point, the number on the wheel opposite the zero point of the fixed scale is read. This number, when multiplied by the scale of the spring and divided by a constant which is usually 40, gives the M.E.P. of the diagram in pounds per sq. in. In case there is a loop in the toe of the diagram, as in Fig. 67 the point is first carried down the expansion line and then around the loop in a counter-clockwise direction. This automatically subtracts the average pressure of the loop from the average pressure of the remainder of the diagram.

Indicated Horsepower.—The rate at which work is performed in the engine cylinder, which is calculated from the indicator diagram, is called the *indicated horsepower* abbreviated I.H.P. After the M.E.P. of the indicator card has been measured, the indicated horsepower may be calculated by the formula:

$$\text{I.H.P.} = \frac{Plan}{33{,}000}$$

in which I.H.P. is the indicated horsepower
P is the M.E.P. in pounds per sq. in.
l is the length of stroke in feet
a is the area of the piston in sq. in.
n is the number of revolutions per minute

The above formula gives the I.H.P. from a single indicator diagram, which is taken from but one end of the cylinder, hence, to find the total I.H.P. for the engine, the I.H.P. must be calculated for each end of each cylinder and their sum taken. It should be remembered that the piston rod occupies a portion of the area of the piston, and, for accurate results, its area must be subtracted from the area of the piston when calculating the I.H.P. for the crank end of the cylinder.

Example.—Calculate the indicated horsepower of a 20″ × 24″ simple engine, running at 240 r.p.m., the M.E.P. for the head end of the cylinder being 48 lbs. per sq. in. and for the crank end 49 lbs. per sq. in. The diameter of the piston rod is 2½ inches.

Solution.—The area of the piston on the head end is

$$.7854 \times 20^2 = 314.16 \text{ sq. in.}$$

The area of the piston rod is

$$.7854 \times 2.5^2 = 4.91 \text{ sq. in.}$$

The area of the piston on the crank end is

$$314.16 - 4.91 = 309.25 \text{ sq. in.}$$

The length of stroke is

$$\frac{24}{12} = 2 \text{ ft.}$$

The indicated horsepower for the head end is

$$\text{I.H.P.} = \frac{Plan}{33{,}000} = \frac{48 \times 2 \times 314.16 \times 240}{33{,}000} = 219.3$$

The indicated horsepower for the crank end is

$$\text{I.H.P.} = \frac{Plan}{33{,}000} = \frac{49 \times 2 \times 309.25 \times 240}{33{,}000} = 220.4$$

The total I.H.P. is

$$219.3 + 220.4 = 439.7$$

The indicated horsepower may be calculated approximately by using the average M.E.P. for the two ends of the cylinder and neglecting the area of the piston rod. The total indicated horsepower is then calculated by the formula:

$$\text{I.H.P.} = 2\frac{Plan}{33{,}000}$$

Applying this formula to the above example would give

$$\text{I.H.P.} = 2\frac{Plan}{33{,}000} = 2\frac{48.5 \times 2 \times 314.16 \times 240}{33{,}000} = 443.2$$

The result is 3.5 horsepower larger than the result obtained by the other method of calculation.

Engine Constant.—In the formula for indicated horsepower it will be noted that, for any particular engine, the part of the formula

$$\frac{la}{33{,}000}$$

is a constant quantity. This quantity is called the *engine constant.* Since the net area of the piston will be different on the head and crank ends, an engine will have a head end engine constant and a crank end engine constant. When the engine constant has once been calculated, the horsepower may be found at any time by observing the M.E.P. and speed and multiplying these quantities by the engine constant.

Brake Horsepower.—The indicated horsepower is the power developed in the cylinder of an engine. This power is transmitted through the piston rod, crosshead, connecting rod, crank and main shaft to the flywheel and a portion of it is lost by friction in the various bearings of the engine. Hence the amount of power delivered at the flywheel will be smaller than that developed in the cylinder. The power delivered to the flywheel is called the *brake horsepower,* abbreviated B.H.P. It may be measured by means of a device called a friction brake, hence the name *brake* horsepower.

A friction brake usually consists of a band which is clamped on the face of the flywheel and which may be tightened so as to produce more or less friction between it and the flywheel. The power of the engine is expended in overcoming the friction of the brake, which is arranged in such way that the pull of the engine upon the brake may be measured. The brake horsepower is calculated from the pull and speed of the engine and the dimensions of the brake.

A common form of friction brake, called a Prony brake, is shown in Fig. 69. This brake consists of a wooden beam *C* and a band *B* made of a number of hard wood blocks fastened to a sheet iron band and passing around the flywheel *A*. The beam *C* has a steel knife-edge fastened to its under side near the end and resting on an iron plate on top of the stand *E*. The stand *E* rests on a platform scale so the pull of the engine upon the brake may be weighed. One end of the band containing the friction blocks is fastened to the beam by passing through it and having

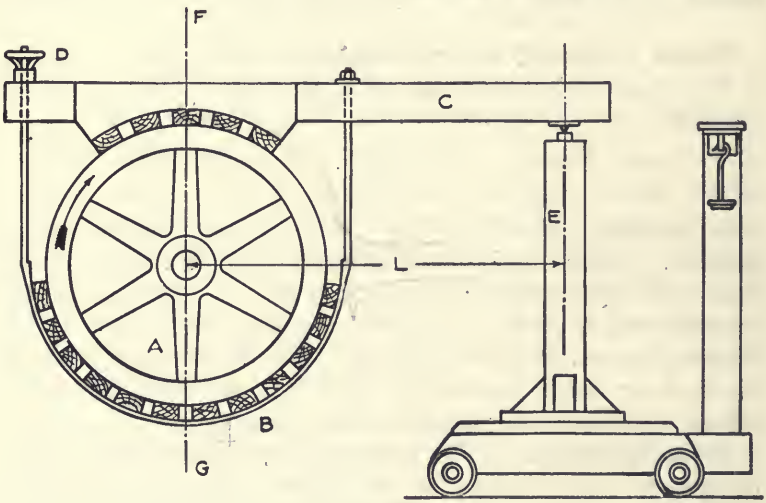


Fig. 69.

a nut on its end. The other end of the band is held by hand wheel *D* so it may be tightened and its friction adjusted. The edges of the flywheel form inwardly projecting flanges so that a stream of water may be run into the flywheel to keep it cool.

Preparatory to using the brake, the distance *L* from the center of the brake to the knife-edge is measured, the stand *E* is weighed, and the unbalanced weight of the brake about the center line *FG* is obtained. This may be done by suspending the brake by a cord at the point *F* while the end of the beam rests on a scale and noting its weight. In using the brake the engine is brought up to full speed and the band tightened as much as possible without reducing the speed. The weight registered on the scales and the speed of the engine are observed at the same time.

The brake horsepower may now be calculated by the formula:

$$\text{B.H.P.} = \frac{2\pi lnW}{33{,}000}$$

in which

B.H.P. is the brake horsepower
l is the length from center of flywheel to knife-edge in feet
n is the number of revolutions per minute
W is the pull of the engine on the brake, in pounds
W = weight registered on scales minus weight of stand minus unbalanced weight of brake
$\pi = 3.1416$

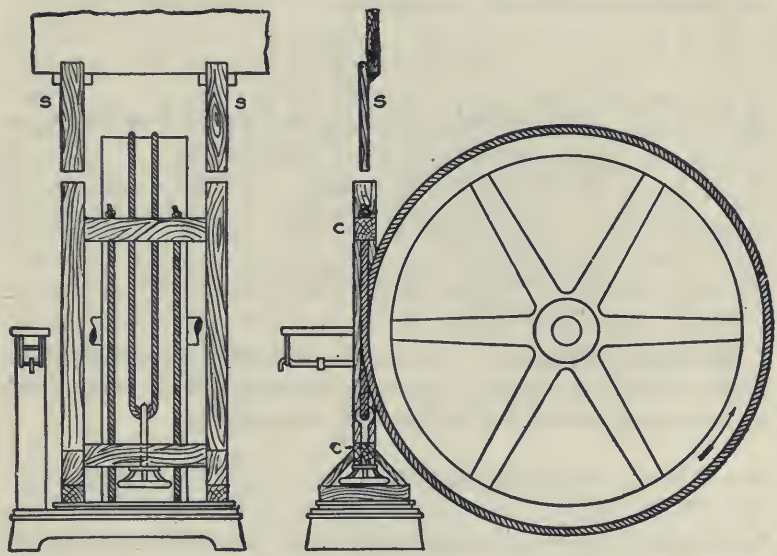


FIG. 70.

Example.—What is the brake horsepower of a steam engine running at 210 r.p.m. when fitted with a Prony brake which measures 8 feet from center of the flywheel to the point of support at the end of the arm, the scale reading 742 lbs., the unbalanced weight of the brake being 13 lbs., and the weight of the standard being 10 lbs.?

Solution

$$W = 742 - 13 - 10 = 729 \text{ lb.}$$
$$n = 210 \text{ r.p.m.}$$
$$l = 8 \text{ ft.}$$

Then

$$\text{B.H.P.} = \frac{2\pi lnW}{33{,}000} = \frac{2 \times 3.1416 \times 8 \times 210 \times 719}{33{,}000} = 230 \text{ horsepower}$$

Another form of brake for measuring power is shown in Fig. 70. This form of brake is called a rope brake because the friction which furnishes the load for the engine is produced by a rope wound around the flywheel. In this brake the ends of the rope are attached to the top crosspiece C of a wooden frame which rests on a platform scales. The rope is looped around the flywheel and the middle attached to a screw which passes through the bottom crosspiece C. This screw passes through a hand wheel which is used to tighten the rope and thus regulate the load on the engine. Instead of a hand wheel a large nut may be used for this purpose.

The brake horsepower, as measured with this form of brake, may be calculated from the formula:

$$\text{B.H.P.} = \frac{2\pi RWn}{33{,}000}$$

in which $R =$ the radius of brake or distance from center of wheel to center of rope, in feet

$W =$ Pull of the engine = weight indicated by scales minus weight of wooden frame, in pounds

$n =$ number of revolutions per minute

$\pi = 3.1416$

Mechanical Efficiency.—The mechanical efficiency of an engine or its efficiency considered simply as a machine, is the ratio of the brake horsepower to the indicated horsepower or

$$\text{Mechanical Efficiency} = \frac{\text{B.H.P.}}{\text{I.H.P.}}$$

This quantity is always less than one, since there is always a loss of power by friction in the engine; that is, the brake horsepower is always less than the indicated horsepower. The mechanical efficiency of steam engines varies from 85 per cent. to 95 per cent.

The difference between the indicated horsepower and the brake horsepower is the amount of power required to overcome the friction. This quantity is sometimes called the *friction horsepower* or

$$\text{Friction horsepower} = \text{I.H.P.} - \text{B.H.P.}$$

The friction horsepower is the indicated horsepower of the engine when it is running without load.

CHAPTER VIII

ACTION OF STEAM IN THE CYLINDER

Cylinder Condensation.—It is a well-known fact that the steam engine is a wasteful machine for developing power because it turns into work only a small part of the heat energy delivered to it. The amount of work obtained from a steam engine is often only 4 or 5 per cent. of the amount of energy delivered to it, and it rarely exceeds 20 per cent. This means that from 80 to 96 per cent. of the heat energy supplied to the steam engine is wasted or at least is not utilized. For example, suppose an engine uses 35 pounds of dry saturated steam per hour for each indicated horsepower developed, the steam having an absolute pressure of 100 lb. per sq. in. The heat delivered to the engine amounts to $35 \times 1186.3 = 41{,}520.5$ B.t.u. per horsepower per hour and from this amount of energy only one horsepower is obtained. One horsepower for an hour is equivalent to 2545 B.t.u., therefore, the part of the energy supplied which is turned into work is only

$$\frac{2545}{41{,}520.5} = .0613 \text{ or } 6.13 \text{ per cent.,}$$

the remaining 93.87 per cent. being lost or wasted. A large part of this loss occurs through the exhaust but another considerable part occurs through the condensation of steam in the cylinder.

Cylinder condensation is caused by the alternate cooling and heating of the cylinder walls as they are alternately in contact with high pressure steam (which has a high temperature) during admission, and to low pressure steam (which has a lower temperature) during exhaust.

The exchanges of heat taking place between the steam and cylinder walls may best be studied by considering the cycle of events occurring in only one end of the cylinder. Exhaust occurs during the greater part of the return stroke of the piston and during this time the cylinder walls, face of the piston, and cylinder head are in contact with steam having a comparatively low temperature, thus cooling these parts of the engine. Compression at the end of the return stroke raises somewhat the temperature of the steam in the clearance space, but the warming effect on the cylinder is small because the temperature of the compressed steam is not so high as the steam admitted from the boiler and the piston is near the end of its stroke, exposing very little of the cylinder walls to the compressed steam. Most of the surface so exposed consists of the face of the piston and the cylinder head. Consequently, when the admission valve opens and a fresh charge of high pressure (and high temperature) steam rushes into the cylinder it meets comparatively cool metal surfaces and a part of it is condensed, collecting in a thin layer of water on these surfaces. As the piston advances on its forward stroke it uncovers more and more of the chilled cylinder walls which condense still more of the steam which is being admitted to the cylinder, with the result that, up to the point of cutoff, from 30 to 50 per cent. of the steam fed into the cylinder during admission is condensed, thus requiring that a greater volume of steam be supplied to the cylinder than if none of it was condensed. The condensation occurring up to the point of cut-off is called *initial condensation.*

After cut-off the piston continues to advance and uncover more of the cooled cylinder walls. Hence, condensation continues after cut-off but at a lessening rate because, after cut-off, the steam in the cylinder is expanding and its temperature falling. The difference in temperature between the steam and the cylinder walls is not so great. Besides the condensation due to contact between the steam and cooler cylinder walls there is now also a certain amount of condensation caused by energy being taken out of the expanding steam to move the piston.

Whatever steam is condensed during the early part of the stroke is deposited in the form of a thin film of water on the cylinder walls, the face of the piston, and the cylinder head. This film of water has a temperature equal to that of the steam from which it was formed, hence it is at or very near the boiling point corresponding to the pressure of the steam. After cut-off the steam in the cylinder begins to expand and its pressure falls. This lowers the boiling point below the temperature of the water already deposited on the inside of the cylinder with the result that this water begins to reevaporate. As expansion proceeds, the boiling point is lowered and the difference between the boiling point and the temperature of the layer of water becomes larger, therefore reevaporation proceeds at a faster and faster rate. For this reason, a point is reached soon after cut-off where the reevaporation balances the condensation and at this point the amount of water in the cylinder is a maximum. From this point on reevaporation occurs faster than condensation and the amount of water in the cylinder grows smaller. When release occurs, there is a sudden drop in pressure, accompanied by a sudden drop in the boiling point, and the layer of water on the cylinder walls reevaporates very fast. During exhaust the pressure remains low and reevaporation continues at a rapid rate, if there is still any water remaining in the cylinder. If the initial condensation has not been very great, however, the water may be all reevaporated at the beginning of exhaust, and the exhaust steam will then be dry.

It might be thought that if the water in the cylinder is all reevaporated no harm would be done It should be remembered, however, that any water reevaporated near the end of expansion is at a lower pressure than when condensed and consequently it cannot be expanded as much as if it had not been condensed but had remained in the form of steam and expanded through the whole range of pressure.

If the steam admitted to the cylinder already contains some water, as would be the case if wet steam were supplied, the amount of water reevaporated during expansion and exhaust may be greater than the condensation during admission. This would cause a much larger quantity of heat to be taken from the cylinder walls and the chilling effects of reevaporation to be greatly increased. It is an advantage therefore to supply only perfectly dry steam to an engine in order to reduce the amount of water in the cylinder.

In the above discussion of cylinder condensation, the events occurring in ony one end of the cylinder have been considered. If we consider these events as occurring in the head end, for instance, then the events occurring in the crank end have some influence upon the condensation and reevaporation in the head end. During the first part of exhaust from the head end, this end of the cylinder is in contact with low temperature steam and may be further cooled by reevaporation, but at the same time admission and expansion are occurring in the crank end and this end of the cylinder is being warmed slightly by contact with high temperature steam. Admission and expansion in the crank end, therefore, reduces slightly the cooling of the head end and hence reduces the amount of condensation that would occur in the head end. The reduction in initial condensation from this cause will depend upon the lateness of the cut-off in the opposite end of the cylinder, the condensation being less for a late cut-off than for an early one because a late cut-off exposes more of the cylinder walls to high temperature steam.

The cooling effects of reevaporation depend upon the reduction in pressure of the steam during expansion, being greater the more fully the steam is expanded. The greatest amount of expansion occurs with an early cut-off, hence, an early cut-off increases reevaporation. It will thus be seen that the loss of steam by condensation and the cooling of the cylinder by reevaporation are both increased by an early cut-off.

The give and take of heat between the steam and cylinder walls does not affect all of the metal in the cylinder because the heat transfers occur so rapidly that their effects do not have time to extend very far into the metal. The outside of the cylinder assumes a temperature between that of the exhaust and the admission steam and this temperature remains practically constant while the engine is running. The inner surfaces of the cylinder and piston, however, experience great changes in temperature, being alternately heated and cooled, but such changes of temperature grow less the greater the distance from the inner surface at which they are measured. It is probable that the depth of metal thus affected does not average more than .02 to .03 inch, and this depth is less for high rates of revolution than for low rates because there is not time, with a high rate of revolution, for the transfer of heat to take place. Other things being equal, the losses from cylinder condensation and reevaporation are less for high speed engines than for low speed ones.

It has been pointed out in a previous paragraph that the amount of heat taken from the cylinder walls by reevaporation exceeds the amount given to the walls by condensation. In some cases the amount of heat taken from the cylinder walls may be only slightly greater than that given to them and, in such cases, it might be thought that the cooling effect would be very small. In this connection, however, it should be remembered that the transfer of heat affects only a thin layer of metal and the weight of metal affected, therefore, is small. Since one B.t.u. will change the temperature of $7\frac{1}{2}$ lbs. of cast iron one degree or one pound of cast iron $7\frac{1}{2}$ degrees it will be seen that the transfer of even a small amount of heat may produce comparatively great changes of temperature in the inner surfaces of the cylinder.

The effects of high speed and late cut-off in reducing the losses from cylinder condensation and reevaporation have been noted. Other remedies that are used for accomplishing this purpose are: the use of superheated steam; the use of a steam jacket surrounding the cylinder; and compounding, or dividing the total range of expansion between two or more cylinders. The effects of compounding will be considered in a later chapter.

The benefits to be derived from the use of superheated steam come from the prevention of initial condensation. Superheated steam contains more heat per pound than saturated steam at the same pressure, and, before it can be condensed, the extra heat which it contains must first be taken out of it, thus reducing its temperature and changing it into saturated steam. Taking more heat from it will then cause condensation. When superheated steam is supplied to an engine the heat needed to warm the cooled cylinder walls may be supplied from the extra store of heat which the steam contains and there will be no initial condensation. In order to fully accomplish this purpose, however, the steam must be superheated enough to secure dry steam at the point of cut-off. If it is not superheated to this degree all the excess heat will be taken from the steam before cut-off and it will then begin to condense and deposit moisture on the cylinder walls.

There is but little advantage in using steam which is superheated to such an extent that it will still be superheated after expansion commences, because reevaporation begins at this point and also because the condensation which occurs before cut-off causes more serious loss than that which occurs after cut-off, and it is, therefore, of more advantage to prevent the initial condensation.

Since initial condensation is usually greatest in the plain slide valve type of engine, which employs a late cut-off, it is to be expected that more benefit may be derived from the use of superheated steam in this type of engine than from those of other types, employing an earlier cut-off, or those making use of compound expansion.

A steam jacket consists of a hollow space surrounding the cylinder, connected to the main steam supply for the engine, and filled with steam at high pressure. The steam jacket is supposed to benefit by keeping the cylinder walls at a uniformly high temperature and preventing the rapid changes of temperature in the walls. It is found in practice, however, that but little benefit is obtained from the steam jacket because the changes of temperature take place in only a thin layer of metal on the inner surfaces of the cylinder and these changes of temperature occur so rapidly that the heat from the jacket does not have time to flow through the walls rapidly enough to prevent them. Cylinder condensation is reduced somewhat, however, by the presence of the jacket because it maintains a higher average temperature of the cylinder walls. On the other hand, it must be remembered that whatever heat is supplied by this means comes from the condensation of steam in the jacket and also that the presence of the jacket makes the outside diameter of the cylinder greater and increases its surface. The greater surface of the cylinder together with its higher temperature increases the amount of heat lost by radiation from the cylinder. Since the advantages of a steam jacket are doubtful and its presence increases the cost of the engine, it is not used as much now as formerly. Instead, the cylinders of the better classes of engines are now simply lagged with a nonconducting substance to reduce the radiation of heat.

The Uniflow Engine.—Within recent years a single expansion engine has been designed with a view to reducing the losses from cylinder condensation. This type of engine is called the *Uniflow Engine.* A section of the cylinder of the uniflow engine is shown in Fig. 71. The cylinder contains no exhaust valves but a ring of exhaust ports are cut in the middle of the cylinder which is uncovered by the piston at the end of its stroke so that, in effect, the piston is the exhaust valve. For this purpose, both the cylinder and the piston are made longer than in the ordinary steam engine. The two admission valves, A, which are of the Corliss type, are located in the cylinder heads and the steam spaces over the valves become steam jackets for the heads. The clearance pocket B is also kept filled with steam so that the head is completely steam jacketed.

After cut-off, the steam expands behind the piston as in the ordinary types of engines, but at the end of expansion the piston uncovers for an instant the exhaust ports and the remaining pressure in the cylinder falls. On the return stroke the steam at exhaust temperature and pressure is caught between the piston and cylinder head, and, as the piston moves back, this steam is compressed so that its temperature is gradually increased and at the end of the stroke the clearance space is filled with steam at admission pressure. The temperature of the steam in the clearance space is increased not only by compression but also by absorbing heat from the head jackets. The result is that the temperature of the steam in the clearance space may be raised even higher than that of the admission, therefore, when the admission valve is opened the incoming steam meets no cold surfaces and initial condensation is reduced. An indicator diagram from a uniflow engine is shown in Fig. 72.

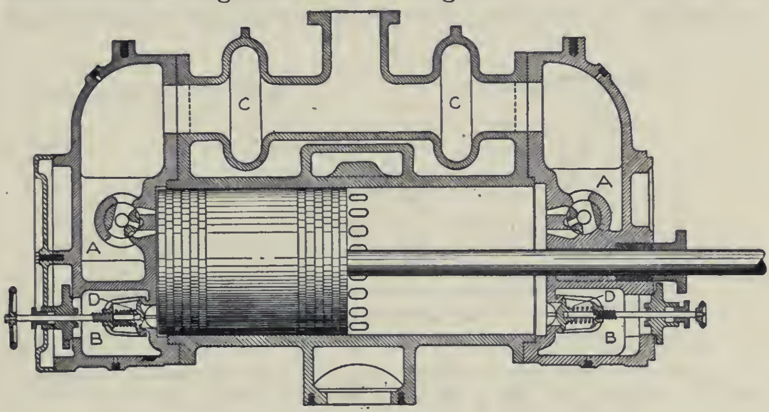


Fig. 71.

By referring to Fig. 71 it will be seen that each end of the cylinder is provided with a large relief valve *D* opening into a pocket *B* in the cylinder head. This valve serves two purposes: First, it is a relief valve of large size which will relieve the engine of any entrained water; second, if, when exhausting into a vacuum, the vacuum should be broken, it is necessary to provide the engine with a larger clearance volume in order to prevent excessive compression. These valves open automatically in case the vacuum is broken, and, if it is then desired to run the engine noncondensing, means are provided to back these valves off their seats, thus increasing the clearance space by the volume of the clearance pockets. The enlargements *C*, *C*, in the steam passages are to provide for expansion of the metal without distorting the cylinder.

That these engines at least partly accomplish their purpose is shown by the fact that they have developed a horsepower with a steam consumption of only 13.2 pounds per hour, which is a very good performance for a single expansion engine even when exhausting into a vacuum of 26 inches.

Measuring Cylinder Condensation.—The quantity of water present in the cylinder at any time between cut-off and release may be found from the indicator diagram and a knowledge of the weight of steam used by the engine. The method of doing this is best shown by means of an actual example. Fig. 73 shows an indicator diagram from the head end of a 12″ × 24″ engine, with a clearance on the head end of 7.9 per cent. The engine was running 100 r.p.m. when the diagram was taken, and, by condensing and weighing the exhaust for an hour, it was found

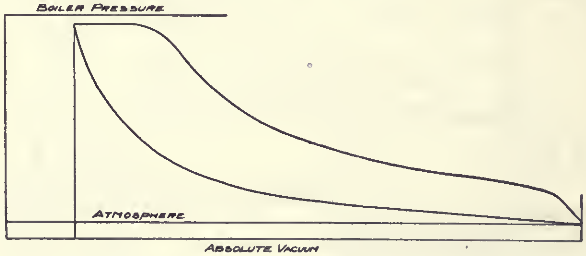


FIG. 72.

that the engine was using 2384.4 pounds of steam per hour. The barometer read 28.52 inches, equivalent to an atmospheric pressure of 14 lbs. per sq. in. The spring in the indicator was No. 100. By drawing on the diagram the "dry steam line" *SS*, as expalined below, the percentage of water in the cylinder, or "quality" of the steam at any time between cut-off and release, may be measured directly from the diagram. Thus in Fig. 73, the quality of the steam at cut-off is found by taking the proportion between the lengths of the lines AB and AC. From the diagram, the length of the line AB is 1.0 inch and the length of AC is 1.43 inches, therefore the quality of the steam at cut-off is

$$\frac{AB}{AC} = \frac{1.0}{1.43} = .70 = 70 \text{ per cent.}$$

or, of the mixture in the cylinder at cut-off, 70 per cent. is dry steam and 30 per cent. is water. In other words, the initial condensation has been 30 per cent. of the steam supplied to the cylinder. In a similar manner the quality of the steam at any other point F in the expansion line is found by measuring the line DE, which is 2.28 inches and the line DF, which is 2.86 inches and taking the ratio

$$\frac{DE}{DF} = \frac{2.28}{2.86} = .797 \text{ or } 79.9 \text{ per cent.}$$

showing that the steam in the cylinder is dryer at the point E than it is at the point B, a result that is to be expected.

In order to draw the dry steam line it is first necessary to locate the line of no pressure, OG, and the line of no volume, OH.

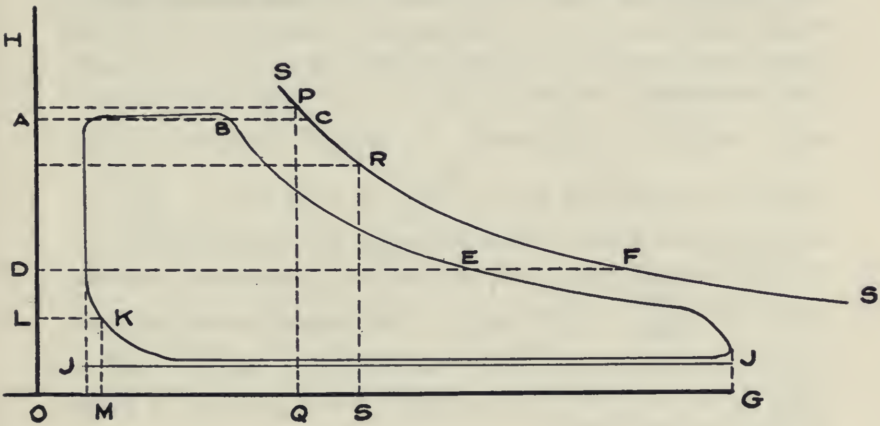


Fig. 73.

The line OG is used as a base line from which to measure absolute pressures and is drawn parallel to the atmospheric line, JJ, and at a distance below it equal to the atmospheric pressure, 14 lb. per sq. in. to the same scale to which the diagram is drawn. Volume of steam in the cylinder is also measured on this line. The line, OH, is used for determining the amount of steam in the clearance volume during compression and also to measure pressures. The line OH is drawn perpendicular to the atmospheric line and at a distance from the end of the diagram equal to the volume of the clearance space, to the same scale to which the diagram is drawn. By drawing limiting lines at the ends of the diagram, extending to the atmospheric line, the length of the diagram is measured on the atmospheric line and is found to be

3.34 in. The piston displacement is $.7854 \times 1^2 \times 2 = 1.5708$ cu. ft. Therefore one inch length on the diagram represents a volume of $\frac{1.5708}{3.34} = .47$ cu. ft. The volume of the clearance is $.079 \times 1.5708 = .124$ cu. ft. Therefore the line OH is laid off from the end of the diagram a distance of $\frac{.124}{.47} = .264$ inch. The weight of steam in the clearance space during compression is found by taking any point such as K on the compression curve after the exhaust valve is closed and measuring the pressure and volume represented by this point. The absolute pressure of the point K is found by measurement to be 40 lb. per sq. in. The volume of one pound of dry steam at this pressure is, from the steam table, 10.3 cu. ft. The volume of steam in the clearance space is represented by the line LK which is .34 inch. It, therefore, represents a volume of $.34 \times .47 = .1598$ cu. ft., and its weight is $\frac{.1598}{10.3} = .0152$ pound. The weight of steam fed to each end of the cylinder per hour is $\frac{2384.4}{2} = 1192.2$ lbs. or $\frac{1192.2}{60} =$ 19.87 lbs. per minute. Since the engine was running 100 r.p.m., the weight of steam fed to the engine while the diagram was being drawn was $\frac{19.87}{100} = .1987$ pound. The weight of steam expanding in the cylinder each time was, therefore, $.0152 + .1987 =$ .2139 pound and it is for this weight of steam that the dry steam line SS must be drawn.

The dry steam line SS is drawn by taking from the steam table the volumes of one pound of dry steam at various pressures and multiplying them by the weight of steam expanding in the cylinder .2139 lb. Thus at an absolute pressure of 150 lb. per sq. n. the volume of one pound of dry steam is, from the steam table, 2.978 cu. ft., therefore, the volume of .2139 lbs. is $.2139 \times 2.978 = .6443$ cu. ft. Measuring from O a distance equal to $\frac{.6443}{.47} = 1.37$ inches the point Q is located, which represents a volume of .6443 cu. ft. Drawing the line PQ perpendicular to OG and of a length equal to 150 lbs. per sq. in., the point P is located, which is one point on the dry steam line. At an absolute pressure of 120 lb. per sq. in., the volume of one pound of dry steam is, from the steam table, 3.726 cu. ft., therefore, the volume of .2139 pounds is $.2139 \times 3.726 = .7970$ cu. ft. Measuring from O a distance equal to $\frac{.7970}{.47} = 1.70$ inches, the point S is located, which represents a volume of .7970 cu. ft. Drawing the line RS perpendicular to OG and of length equal to 120 lb. per sq. in. the point R is located which is another point on the dry steam line. In a similar manner any number of points on the dry steam line may be found, and a smooth curve drawn through these will give the dry steam line SS. The quality of steam in the cylinder may then be measured from this line, as explained before. In order for this method of finding the quality of steam in the cylinder to give accurate results, there must be no leakage in the cylinder, as this would change the shape of the expansion line.

CHAPTER IX

STEAM ENGINE TESTING

Principles.—The usual steam engine test is made to determine the weight of steam or the number of heat units which the engine consumes per hour for each horsepower developed or else to determine the efficiency of the engine. If possible, the steam consumption and efficiency should be based on the brake horsepower of the engine because this is the useful power of the engine. However it is sometimes impossible or impractical to obtain the brake horsepower and, in this case the steam consumption and efficiency are based on the indicated horsepower.

As practically all engines operate under variable loads, it is advisable in testing them, to test at different per cents of the full load in order to determine what performance may be expected under different conditions. For this purpose it is convenient to test an engine under one quarter load, one half load, three quarters load, full load, and one and one quarter of its full load capacity. With this data at hand a curve may be plotted with brake horsepower or indicated horsepower on one axis and steam consumption or efficiency on the other axis, and from the curve so obtained, one may determine what performance to expect from the engine under the load at which it operates most of the time.

If the engine is belted and not very large, its brake horsepower may be measured with any of the forms of friction brakes described in a previous chapter, although these require the use of a special pulley designed to hold water for cooling. If the engine is connected directly to an electric generator, it will be necessary to determine separately the amount of power required to run the generator without load, or the friction load of the generator, so that this may be deducted from the output of the generator in calculating the brake horsepower of the engine. If the brake horsepower cannot be determined by one of these methods, it will be necessary to base the calculations for steam consumption and efficiency upon the indicated horsepower.

Steam Consumption.—The steam consumption is best determined by means of a surface condenser. In this case the exhaust steam from the engine is simply run into a surface condenser where it is condensed and the condensate weighed. In order to secure accurate results by this method the condenser should be free from leaks and the condensate should be cooled to a temperature that will prevent its giving off much vapor, as otherwise the loss of condensate by evaporation will seriously affect the results of the test.

The steam consumption may also be determined by means of one of the commercial forms of steam meters which measures the weight of steam passing through it. If this method of determining the steam consumption is used, the steam meter should first be carefully calibrated to insure its giving accurate results.

If neither of the above methods is available, it may be possible to isolate the boiler or boilers supplying the engine, so that all of the steam generated by the boiler is used in the engine. The feed water supplied to the boiler may then be measured and taken as the steam consumption of the engine. Sometimes it is necessary for the boiler used in this way to supply steam for some auxiliaries such as feed pumps, etc. In such a case it is necessary to determine separately the amount of steam used by the auxiliaries. This may usually be done, by condensing the exhaust steam from them. In using this method of determining the steam consumption of an engine extreme care should be taken to insure that there are no leaks, especially at branches stopped by valves. This is done by closing all valves in branches and the main stop valve at the engine so that the main supply pipe is open from the boiler to the engine valve, but closed everywhere else. With a quiet furnace fire so that there is no active evaporation the level of the water in the boiler is noted from time to time. If the water level falls, leakage is taking place and the leaks should be located and stopped or else the rate of leakage allowed for in the steam consumption.

Steam Consumption from Diagram.—It is sometimes impossible to find the weight of steam used by an engine by condensing the exhaust and weighing it, or by isolating the boiler and weighing the feed water. In such cases the weight of steam used may be found from the indicator diagram, but this method should not be used except when the weight of steam used cannot be found by any other method, because it is subject to serious errors on account of leakage of steam into or out of the cylinder and from one side of the piston to the other.

The method of finding the steam consumption from the indicator diagram may be illustrated by Fig. 73. By this method it is first necessary to draw the no volume line, *OH*, and the no pressure line, *OG*, as described before. If the barometer reading is not known, it is customary to draw the line *OG* at a distance below the atmospheric line equal to 14.7 lb. per sq. in. to the same scale as the spring used in drawing the diagram. A line *DE* is drawn across the diagram parallel to the atmospheric line and at a point near the end of the expansion line. The weight of steam represented by the volume *DE* is the weight which is expanding in the cylinder. This weight minus the weight of steam compressed into the clearance space is the weight of steam fed to the cylinder at each stroke. The length of the line *DE* is 2.66 inches and since a length of one inch on the diagram represents a volume of .47 cu. ft. the volume represented by the line *DE* is $.47 \times 2.26 = 1.0622$ cu. ft. At the point *E* on the expansion line, the steam in the cylinder has an absolute pressure of 65 lb. per sq. in. From the steam table one cubic foot of steam at 65 lbs. absolute pressure weighs .1503 pound, hence the weight of steam expanding in the cylinder is $1.0622 \times .1503 = .1626$ lb.

The weight of steam in the clearance space is found by selecting a point *K* on the compression curve after the exhaust valve has closed and measuring the pressure and volume of steam represented by this point. This was done before in drawing the dry steam line, and it was found that the weight of steam in the clearance space was .0152 pound. Therefore the weight of steam fed to the cylinder at each stroke was $.1626 - .0152 = .1474$ lbs. Since the engine was making 200 strokes per minute, the weight of steam used per hour as shown by the diagram was $.1474 \times 200 \times 60 = 1768.8$ lbs. This weight makes no allowance for the condensation in the cylinder, hence it must be corrected by means of the values given in the following table.

Percentage of strokes completed at cut-off	Part of steam accounted for by the indicator diagram		
	Simple engines	Compound engines H.P. cylinder	Triple expansion engines H.P. cylinder
5	0.58		
10	0.66	0.74	
15	0.71	0.76	0.78
20	0.74	0.78	0.80
30	0.78	0.82	0.84
40	0.82	0.85	0.87
50	0.86	0.88	0.90

By measurement it is found that cut-off in the above example occurs at about 19 per cent. of the stroke, hence the weight of steam from the diagram should be divided by .74 or

$\frac{1768.8}{.74} = 2390$ pounds per hour as the probable weight of steam used by the engine. While this result is close to the actual weight of steam used, 2384 lbs., it must be remembered that this method of finding the weight of steam used is liable to serious error, sometimes amounting to as much as 50 per cent.

It is customary to express the steam consumption per horsepower hours in terms of the "dry steam equivalent," that is, in terms of the number of pounds of dry steam which would contain as many B.t.u. as is contained by the steam of the quality actually supplied to the engine. In order to make this calculation, the quality of steam supplied to the engine during the test is measured and the number of heat units in one pound of this steam calculated. This quantity multiplied by the number of pounds of steam supplied per horsepower gives the total number of heat units supplied to the engine per horsepower. The number of heat units per horsepower is then divided by the number of heat units in one pound of dry steam of the same pressure as that supplied to the engine and the quotient will be the number of pounds of equivalent dry steam supplied per horsepower.

This method of stating the performance of an engine forms a very satisfactory basis for comparing one engine with another, provided the engines are operating under similar conditions, but the quality of the engine cannot be judged by this method of comparison if one engine uses superheated steam and the other one uses saturated steam. Neither does a comparison of efficiencies as calculated in a previous paragraph form a satisfactory basis for comparing the qualities of the engines if one is run condensing and the other noncondensing. The steam consumption of engines varies widely, depending upon the kind of engine and the conditions under which it is operated. Some of the best performances of engines that have been recorded are given below.

	H.P.	Gage pressure lbs.	Vacuum inches	R.P.M.	Super-heat degrees	Lbs. of dry steam per I.H.P. per hour
Westinghouse vertical at Brooklyn, N. Y.	5,400	185	27.3	76	....	11.93
Rockwood-Wheelock at Natick, R. I.	595	159	25.4	76.4	....	13.0
McIntosh & Seymour at Webster, Mass.	1,076	123	27.10	99.6	20	12.76
Rice & Sargent at Brooklyn, N. Y.	627	151	28.6	121	....	12.10
Rice & Sargent at Philadelphia, Pa.	420	142	25.8	102	297	9.56
Horizontal 4-valve	658	150.4	26.4	80	16.4	12.03
Leavitt pumping engine at Chestnut Hill, Mass.	575.7	175.7	27.25	50.6	....	11.20

Duration of Engine Test.—The duration of a test will depend upon the conditions under which the test is conducted and upon the methods used in making the different measurements. If the engine is tested under a brake load which may be kept constant, the test is simplified and the time of conducting the test shortened. During each test at a constant load, the steam used is weighed at uniform intervals of time, say ten or fifteen minutes. When there are six or eight of these which are nearly constant in amount, the run may be discontinued provided the error of starting and stopping is not large. The error of starting and stopping will depend upon the method used in measuring the steam consumption. If the steam consumption is measured from a surface condenser, the error from starting and stopping will be only the difference in the amounts of condensate in the condenser at the beginning and end of the test. This will be a relatively small amount. With a steam meter used for measuring the steam consumption the error of starting and stopping the test will be even smaller than with a surface condenser. But, when the steam consumption is determined by measuring the feed water supplied to a boiler, the error of starting and stopping the test will be large and the test must be conducted for a greater length of time. The error in this case may arise from a difference in level of the water in the boiler at starting and stopping or from a difference in the density of the water due to a different rate of boiling.

The frequency of taking indicator diagrams from the engine will depend on how the load is varying. With a constant brake load indicator diagrams may be taken at ten minute intervals, but with a varying load the intervals should be from three to five minutes. The object in any case is to get a fair average of the mean effective pressure or indicated horsepower.

Efficiency of Steam Engines.—The term efficiency usually means the ratio between the work obtained from a machine and the energy supplied to it. The efficiency of a steam engine may therefore be expressed as:

$$\text{Efficiency} = \frac{\text{Work obtained from the engine}}{\text{Energy supplied to the engine}}$$

On this basis the efficiency of a steam engine is very low on account of the large losses of heat taking place in the engine itself and of the large amount of heat rejected by the engine in the exhaust steam.

In calculating the efficiency of a steam engine either the brake horsepower or the indicated horsepower may be used as the "work obtained from the engine," but it should be stated which of these is used. If the energy supplied to the engine is expressed in B.t.u. per minute, the work obtained from the engine should also be expressed in B.t.u. per minute; this may be done by multiplying the horsepower by 42.42. If the energy supplied to the engine is expressed in B.t.u. per hour, the horsepower should be multiplied by 2545 to obtain the work done by the engine per hour in B.t.u.

The "energy supplied to the engine" should include all of the heat actually supplied to the engine, calculated above the heat of the liquid for the exhaust pressure. It would not be fair to the engine to charge it with the heat of the liquid below the exhaust pressure because the engine could not possibly change this heat into work. Neither should the engine be charged with all of the heat in the steam above 32°F., because the engine would have to exhaust into a vacuum in which the absolute pressure was only .089 lb. per sq. in. in order to make all of this heat available, and it is not possible to produce and maintain this low pressure in a condenser.

The above expression for efficiency of a steam engine now becomes:

$$E = \frac{\text{B.H.P.} \times 42.42}{W(qL + h - h_1)}$$

in which

E is efficiency of the engine

B.H.P. is the brake horsepower of the engine. (Use I.H.P. if necessary.)

W is the weight of steam supplied to the engine per minute

q is the quality of the steam supplied to the engine

L is the latent heat of the steam per pound at admission pressure

h is the heat of the liquid per pound at admission pressure

h_1 is the heat of the liquid per pound at exhaust pressure

Example.—An engine receiving steam at a pressure of 150 lbs. per sq. in. absolute and having a quality of 98 per cent. develops 600 I.H.P. and uses 12,000 lbs. of steam per hour. The exhaust pressure is 16 lb. per sq. in. absolute. What is the efficiency of the engine?

Solution.—The weight of steam used per minute is

$$\frac{12{,}000}{60} = 200 \text{ lbs.}$$

The latent heat of steam at 150 lbs. per sq. in. absolute pressure is 863.2 B.t.u.

The heat of the liquid at 150 lbs. per sq. in. absolute pressure is 320.2 B.t.u.

The heat of the liquid at 16 lb. per sq. in. absolute pressure is 184.4 B.t.u. Therefore

$$\begin{aligned} E &= \frac{\text{I.H.P.} \times 42.42}{W(qL + h - h_1)} \\ &= \frac{600 \times 42.42}{200(.98 \times 863.2 + 330.2 - 184.4)} \\ &= \frac{26{,}452}{200 \times 991.8} = \frac{25{,}452}{198{,}346} \\ &= .1313 \text{ or } 13.13 \text{ per cent.} \end{aligned}$$

The above formula for calculating the efficiency of a steam engine is used only when the engine is supplied with saturated steam. If superheated steam is supplied, the heat supplied to the engine should include the heat required to superheat the steam.

Efficiency of a Perfect Engine.—The efficiency of an imaginary perfect engine may be calculated by the formula

$$E_p = \frac{T_1 - T_2}{T_1}$$

in which E_p = the efficiency of the perfect engine
T_1 = the absolute temperature of the admission steam
T_2 = the absolute temperature of the exhaust steam

By absolute temperature is meant the temperature reckoned from the absolute zero of temperature or the point below which it would be impossible to cool any substance. This point is located at 460° below zero on the Fahrenheit scale of temperatures, hence to change Fahrenheit temperature to absolute temperature it is necessary to add 460° to the Fahrenheit temperature.

If the engine in the above example had been a perfect engine, its efficiency would have been calculated as follows: The temperature of steam at 150 lb. per sq. in. absolute pressure is, from the steam table, 358.5°F. and its absolute temperature is, therefore, $358.2 + 460 = 818.5°$. The temperature of steam at 16 lb. per sq. in. absolute pressure is, from the steam table, 216.3° F and its absolute pressure is, therefore, $216.3° + 460 = 676.3°$. Hence, the efficiency of the perfect engine would be

$$E_p = \frac{818.5 - 676.3}{818.5} = .1734 \text{ or } 17.34 \text{ per cent.}$$

It will be observed that the efficiency of the perfect engine depends only upon the temperature of the admission steam and of the exhaust steam. It follows, therefore, that the efficiency of the perfect engine can never be 100 per cent. because the maximum temperature of the admission steam is limited, and also it is impossible to reduce the temperature of the exhaust steam to the absolute zero.

The efficiency of a perfect engine, as calculated above, is often used as a standard by which to compare the efficiencies of different engines, the admission and exhaust temperature being taken the same for both the perfect and the actual engine. The method of comparison is to divide the efficiency of the actual engine by the efficiency of the perfect engine, the result being called the efficiency ratio or

$$\text{Efficiency Ratio} = \frac{E}{E_p}$$

in which E = the efficiency of the actual engine

E_p = the efficiency of the perfect engine taken between the same limits of temperature

The efficiency ratio for the engine mentioned in the example above is

$$\text{Efficiency Ratio} = \frac{E}{E_p} = \frac{13.13}{17.34} = .7514 \text{ or } 75.14 \text{ per cent.}$$

Computations.—In order to show how the computations for an engine test are made and also to show a form for reporting the results, the data and computed results of an efficiency test of an engine are given below. Both the data taken in making the test and also the results computed from this data are first tabulated, the calculated results being in heavy face type, and following this the method of making the calculations is shown.

The test given below is one of a series of tests made on an automatic high speed engine in the steam laboratory of the University of Wisconsin to determine its economy, thermal efficiency, and mechanical efficiency. During each test the engine carried a practically constant load made by a Prony brake similar to that shown in Fig. 69. The steam consumption was measured by passing the exhaust steam from the engine into a surface condenser operated at atmospheric pressure, where it was condensed and weighed. This method of loading the engine and determining the steam consumption made it possible to secure sufficiently accurate results by means of a test of twenty minutes duration.

Report of Steam Engine Test

Item.		
1.	Date, November 15, 1915.	
2.	Kind of Engine. Weston high speed automatic noncondensing single cylinder and simple valve.	
3.	Dimensions 10″ × 13″ Piston Rod 1¾″.	
4.	Rated horsepower	75 I.H.P.
5.	Horsepower constant, head end	**0.002578**
	Horsepower constant, crank end	**0.00250**
6.	Atmospheric pressure, in mercury	28.768
	Atmospheric pressure, lbs. per sq. in.	**14.12**
7.	Length of brake arm	5 ft.
8.	Brake constant	**0.000952**
9.	Duration of test	20 min.

Item.		
10.	Average R.P.M	243
11.	Average steam line pressure, lbs. gage	116
	Average steam line pressure, lbs. absolute	130.12
12.	Average M.E.P. from indicator diagrams H.E	62.83
	Average M.E.P. from indicator diagrams C.E	61.51
13.	Total weight condensed steam, lbs	1113.5
14.	Weight of condensed steam, lbs. per hour	3340.5
15.	Quality of steam, per cent	98.5
16.	Dry steam supplied per hour, lbs	3303.7
17.	Brake load, net, lbs	306
18.	Brake horsepower	70.78
19.	Indicated horsepower, head end	39.36
20.	Indicated horsepower crank end	37.37
21.	Indicated horsepower, total	76.73
22.	Friction horsepower	5.95
23.	Mechanical efficiency, per cent	92.2
24.	Dry steam per I.H.P. hr., lbs	43.06
25.	Dry steam per B.H.P. hr., lbs	46.8
26.	Exhaust pressure, lbs. absolute from indicator diagrams	18
27.	Thermal efficiency, on I.H.P. per cent	5.916
28.	B.t.u. supplied per I.H.P. hr. above exhaust pressure	43,015

Calculating Results.—*Item* 5.—The formula for calculating the indicated horsepower is

$$\text{I.H.P.} = \frac{Plan}{33{,}000}$$

which gives the indicated horsepower developed on one side of the piston if n in the formula is the revolutions per minute. For any particular engine the length of stroke l is a constant quantity; the area of the piston, a, is constant; and the quantity 33,000 is constant. Therefore $\frac{la}{33{,}000}$ will be a constant quantity and it is this part of the horsepower formula that is called the "horsepower constant" or

$$\text{H.P. constant} = \frac{la}{33{,}000}$$

The reason for calculating the horsepower constant separately instead of calculating the horsepower directly is that it saves considerable time when the horsepower must be calculated a large number of times from indicator diagrams. The mean effective pressure from these diagrams will vary slightly, as will also the number of revolutions per minute. If the horsepower constant is calculated separately, this part of the calculations need be done only once because then the horsepower may be calculated for any M.E.P. and R.P.M. by merely multiplying the engine constant by the M.E.P. and the R.P.M. It is necessary to calculate the horsepower constant for each end of the cylinder because the area of the piston is larger on the head end than on the crank end, since the area of the piston rod cuts off some of the area of the piston.

For this engine the length of stroke is 13 in. or $\frac{13}{12}$ ft. and the area of the piston is $10^2 \times .7854 = 78.54$ sq. in. The head end horsepower constant is therefore

$$\text{H.P. constant, head end} = \frac{la}{33{,}000} = \frac{13 \times 78.54}{12 \times 33{,}000} = .002578$$

In calculating the horsepower constant for the crank end it is necessary to deduct the area of the piston rod from the area of the piston. The area of the piston rod is

$$1.75^2 \times .7854 = 2.405 \text{ sq. in.}$$

As the area of the piston is 78.54 sq. in., the net effective area is

$$78.54 - 2.405 = 76.13 \text{ sq. in.}$$

Therefore the crank end horsepower constant is

$$\text{H.P. constant, crank end} = \frac{la}{33{,}000} = \frac{13 \times 76.13}{12 \times 33{,}000} = .0025$$

Item 6.—The atmospheric pressure is read on a barometer and is expressed in inches of mercury. To express the atmospheric pressure in pounds per square inch it is necessary to multiply by .4908 or

Atmospheric pressure, lbs. per sq. in. $= 28.768 \times .4908 = 14.12$.

Item 8.—The brake constant is a constant quantity similar to the "horsepower constant" but relating to the Prony brake and it is used for lessening the work of calculating the brake horsepower. The formula used for calculating the brake horsepower with a Prony brake is

$$\text{B.H.P.} = \frac{2\pi R\,WN}{33{,}000}$$

In which

R = the length of the brake arm in feet

W = the net weight in pounds registered by the brake

and

N = the number of revolutions per minute.

For any given engine test the weight registered by the brake and the R.P.M. may vary but the other parts of the formula will remain constant. Therefore

$$\text{Brake Constant} = \frac{2\pi R}{33{,}000} = \frac{2\pi 5}{33{,}000} = .000952.$$

Item 14.—Item 13 gives the weight of condensed steam in 20 minutes, therefore the weight of condensed steam in one hour will be

$$\text{Item 13} \times \frac{60}{20} \text{ or}$$

$$1113.5 \times \frac{60}{20} = 3340.5$$

Item 16.—The number of heat units in one pound of steam supplied to the engine is found by the formula

$$qL + h$$

in which q = the quality of the steam
L = the latent heat per pound
and h = the heat of the liquid per pound.

Substituting in this formula from the steam table the values for 130.12 lb. per sq. in. absolute

$$L = 872.2 \text{ and } h = 319.4$$

or B.t.u. per lb. $= qL + h = .985 \times 872.2 + 319.4 = \mathbf{1178.5}$
The total amount of heat supplied to the engine in one hour equals the weight of steam actually supplied per hour multiplied by 1178.5 or

$$3340.5 \times 1178.5 = 3936779.25 \text{ B.t.u.}$$

If the steam had been perfectly dry, it would have contained $L + h$ heat units per pound, or

$$L + h = 872.2 + 319.4 = 1191.6 \text{ B.t.u. per pound}$$

Therefore the equivalent weight of dry steam supplied per hour is

$$3936779.25 \div 1191.6 = 3303.7 \text{ lbs. per hour.}$$

Item 18.—As explained before the brake horsepower is found by multiplying together the net brake load in pounds (Item 17), the R.P.M. (Item 10), and the brake constant (Item 8) or

$$\text{B.H.P.} = 306 \times 243 \times .000952 = 70.78$$

Item 19.—In a similar manner the indicated horsepower, head end, is the product of the M.E.P., head end (Item 12), the R.P.M. (Item 10) and the head end horsepower constant (Item 5) or

$$\begin{aligned}
&\text{Head end I.H.P.} = 62.83 \times 243 \times .002578 = \underline{39.36}\\
&\text{Crank end I.H.P.} = 61.51 \times 243 \times .0025 = \underline{37.37}\\
&\text{Total I.H.P.} = 39.36 + 37.37 = \underline{76.73}
\end{aligned}$$

Item 22.—The friction horsepower is equal to the difference between the indicated horsepower and the brake horsepower or

$$\text{Item 22} = \text{Item 21} - \text{Item 18}$$
$$\text{Friction H.P.} = 76.73 - 70.78 = \underline{5.95}$$

Item 23.—The mechanical efficiency is equal to the brake horsepower divided by the indicated horsepower or

$$\text{Mechanical efficiency} = \frac{\text{Item 18}}{\text{Item 21}} = \frac{70.78}{76.73} = .922 \text{ or } 92.2 \text{ per cent.}$$

Item 24.—The dry steam per I.H.P. per hour is equal to the dry steam per hour (Item 16) divided by the indicated horsepower (Item 21) or

$$\text{Dry steam per I.H.P. hr.} = \frac{3303.7}{76.73} = \underline{43.06}$$

Item 25.—In a similar manner the dry steam per B.H.P. hr. is equal to the dry steam per hour divided by the B.H.P. or

$$\text{Dry steam per B.H.P. hr.} = \frac{3303.7}{70.59} = \underline{46.8}$$

Item 27.—The thermal efficiency of the engine, based on the I.H.P. may be calculated by the formula

$$\text{Efficiency.} = \frac{\text{I.H.P.} \times 42.42}{W\,(qL + h - h_1)}$$

in which I.H.P. is the indicated horsepower (Item 21)

W is the actual weight of steam supplied to the engine per minute or Item 14 ÷ 60

q is the quality of the steam supplied to the engine, Item 15

L is the latent heat per pound of the steam supplied to the engine

h is the heat of the liquid per pound of the steam supplied to the engine

h_1 is the heat of the liquid per pound of steam at exhaust pressure

In this case I.H.P. = 76.73

$$W = \frac{3340.5}{60} = 55.675$$

$$q = .985$$
$$L = 872.2$$
$$h = 319.4$$
$$h_1 = 190.5$$

Therefore

$$\text{Efficiency} = \frac{76.73 \times 42.42}{55.675\ (.985 \times 872.2 + 319.4 - 190.5)}$$
$$= \frac{3254.88}{55.675 \times 988} = \frac{3254.88}{55006.9} = .05916$$

or 5.916 per cent,

Item 28.—In calculating Item 27 it was seen that the number of heat units in one pound of steam above exhaust pressure as actually supplied was 988. Since there was supplied to the engine 3340.5 lbs. of steam per hour, the number of heat units supplied per hour above exhaust pressure was

$$988 \times 3340.5 = 3{,}300{,}414$$

and as the I.H.P. was 76.73 the number of heat units supplied per I.H.P. per hour above exhaust pressure was

$$\frac{3300414}{76.73} = \underline{43015}$$

Duty of Pumps.—The performance of pumping engines is usually stated in terms of the number of foot-pounds of work performed by the water piston of the pump per thousand pounds of dry steam, or per million B.t.u. consumed by the engine. The performance, stated in this way, is called the duty of the pumping engine; thus,

$$\text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Weight of dry steam used}} \times 1000 \text{ or}$$

$$\text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Weight of dry steam used}} \times 1{,}000{,}000$$

In using the above formulas it is to be understood that the foot-pounds of work and the weight of steam or number of B.t.u. consumed are to be taken for the same periods of time.

Example.—A compound pump uses 80 pounds of steam per I.H.P. per hour and develops 48 I.H.P. The pump receives steam at an absolute pressure of 135 pounds per sq. in. and exhausts against an absolute pressure of 17 lbs. per sq. in. The quality of the steam delivered to the pump is 97 per cent. The capacity of the pump is 400 gallons per minute and pumps against a pressure of 175 pounds per sq. in. Calculate the duty of the pump, on the dry steam and on the heat unit bases.

Solution.—175 lb. per sq. in. pressure is equivalent to

$$175 \times 2.3 = 402.5 \text{ feet head}$$

400 gallons is equivalent to

$$400 \times 8.33 = 3332 \text{ pounds}$$

Work done by pump per minute

$$= 3332 \times 402.5 = 1{,}341{,}130 \text{ foot-pounds}$$

Work done by pump per hour

$$= 1{,}341{,}130 \times 60 = 80{,}467{,}800 \text{ foot-pounds}$$

Heat units in one pound of wet steam above 32°

$$= (.97 \times 869.9 + 321.7) = 1165.5 \text{ B.t.u.}$$

Heat units in one pound of dry steam above 32°

$$= 1191.6 \text{ B.t.u.}$$

Weight of wet steam used per hour

$$= 80 \times 48 = 3840 \text{ pounds}$$

Equivalent weight of dry steam used

$$= 3840 \times \frac{1165.5}{1191.6} = 3755 \text{ pounds}$$

$$\text{Duty} = \frac{80{,}467{,}800}{3755} \times 1000 = 21{,}423{,}800 \text{ foot-pounds}$$

Heat units in one pound of steam above heat of liquid at exhaust pressure

$$= 1165.5 - 187.5 = 978.0.$$

Heat supplied to pump per hour above heat of liquid at exhaust pressure

$$= 978.0 \times 3840 = 3{,}754{,}520 \text{ B.t.u.}$$

$$\text{Duty} = \frac{80{,}467{,}800}{3{,}754{,}520} \times 1{,}000{,}000 = 21{,}432{,}247 \text{ foot-pounds.}$$

CHAPTER X

THE SLIDE VALVE

Steam and Exhaust Lap.—The valves controlling the distribution of steam to the cylinder are the most important parts of a steam engine, because both the smooth running of the engine and the economical use of steam depend largely upon them.

Of the different kinds of valves used on engines, the ordinary slide valve is the most important. The slide valve combines in one valve the office of admission and exhaust for both ends of the cylinder; it is used on a greater variety of engines than any other form of valve; and the principles underlying the operation of the slide valve are also the principles underlying the operation of other types of valves. For these reasons a thorough study of the slide valve will be made before considering other types

The operation of the slide valve is described in Chapter 1, and the valve and its mechanism are illustrated in Figs. 2 and 4. The eccentric which moves the valve backward and forward gives it the same motion as would a crank, and in fact it is equivalent in all respects to a crank having a length equal to the distance from the center of the shaft to the center of the eccentric. This distance is called the eccentricity.

The position of a line connecting the center of the shaft with the center of the eccentric also represents the position of the eccentric or the position of the equivalent crank. The distance which the valve travels in going from one end of its stroke to the other is called the valve travel. If the motion of the eccentric is communicated directly to the valve, the valve travel will equal twice the eccentricity.

When a valve is at the middle point of its travel, in which position the eccentric will be vertical to the center line of the engine, the valve is in mid-position. This position of a valve is used as a reference point from which the parts of the valve and also its different positions are measured. The cross section of a slide valve in its mid-position is shown in Fig. 74. When it is in this position, the length from the outer edges of the ports

P and P to the outer edges A and A of the valve is called the outside lap. The outside lap is shown by the dimensions O and O. It is not necessary that the outside lap at one end of the valve be equal to that at the other end, and, in fact, they are usually unequal, as will be explained later. The distance from the inside edges of the ports P and P to the inner edges B and B of the valve is called the inside lap. The inside lap is shown by the dimensions I and I. The inside laps of a valve are usually unequal. The width of the ports, or the distance f, is usually the same for both ends of the cylinder.

When steam is admitted past the outer edge of the valve, the outside lap is usually called the steam lap and the inside lap the exhaust lap. When steam is admitted from inside the valve

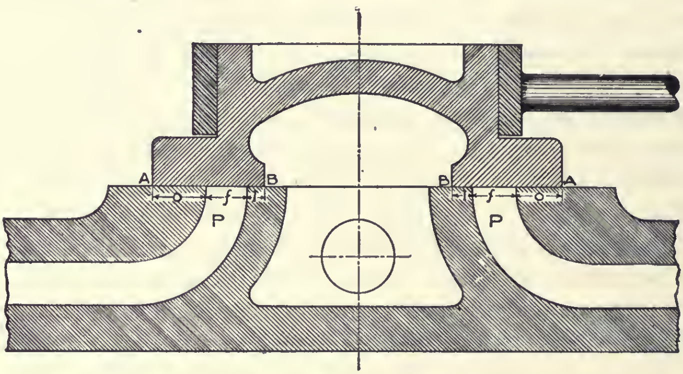


Fig. 74.

and exhausted past the outer edge, as is sometimes done, the inside lap is called the steam lap and the outside lap is called the exhaust lap. The steam lap is usually much greater than the exhaust lap, hence a valve designed for outside admission will not distribute the steam properly, if it is used for inside admission. Unless otherwise mentioned the outside lap will be considered as the steam lap as this is the more usual arrangement.

Valve Without Laps.—A form of slide valve often used on small direct acting steam pumps is shown in Fig. 75. It will be observed that this valve differs from the one shown in Fig. 74 in having neither steam nor exhaust laps, the width of the valve being just equal to the width of the ports. In the position shown in Fig. 75 the valve is in its mid-position and, since it has neither steam nor exhaust lap, it will open the port to admission on one end and to exhaust on the other if the valve moves ever so little to either side of its mid-position. If this kind of slide valve were to be used on a steam engine it would have to be set so as to be in its mid-position when the piston was at the end of its stroke. This would require that the eccentric be placed 90° from the crank as shown at the right of Fig. 75, where *Oc* represents the position of the crank and *Oe* the position of the eccentric. With the valve in its mid-position and the piston at the end of its stroke, steam will begin to be admitted to the cylinder as soon as the piston starts forward and the valve will remain open until the piston has reached the end of its stroke, thus giving admission throughout the entire stroke. At the end of the stroke the valve will close the admission and open the exhaust, and exhaust will occur throughout the entire return stroke of the piston. It

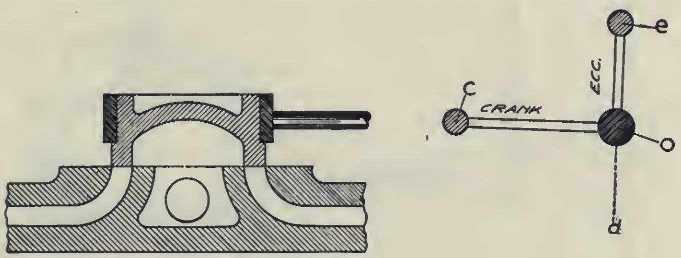


FIG. 75.

should be noted that the eccentric may be set either in the position *Oe*, Fig. 75, or in the position *Od*, and the engine will run either in a clockwise or counterclockwise direction of rotation, depending upon the direction in which it starts. The only requirement in setting the eccentric for a valve with no laps is that it must be placed 90° from the crank.

An indicator diagram taken from an engine fitted with a valve having neither steam nor exhaust lap will be simply a rectangle, as shown in Fig. 76. This diagram shows that steam is admitted to the cylinder during the entire stroke, being released at the end of the stroke without having expanded. Exhaust also occurs during the entire return stroke and none of the steam is compressed near the end of the stroke.

A valve without laps is suitable for small direct acting pumps because the load on these pumps is a constant water pressure and the steam pressure on the piston must therefore be constant throughout the entire stroke in order to overcome the constant water pressure. Moreover, the constant load on the piston serves to bring it to rest at the end of the stroke without shock at the low speeds at which such pumps are usually run, hence no compression is necessary. A valve of this kind would not, however, be suitable for a steam engine because it would be uneconomical in the use of steam since the expansive force of the steam would not be used, and also because the high speed of the engine makes compression necessary if the engine is to run smoothly. It is not necessary to admit steam to the cylinder of an engine throughout the entire stroke because the engine has a flywheel which stores up energy in the first part of the stroke, giving it out again in the last part of the stroke when the

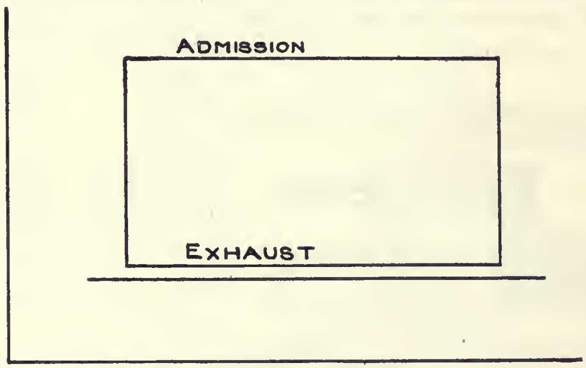


FIG. 76.

steam pressure is small and, by this means, causing the engine shaft to rotate at a uniform speed.

Valves With Lap.—A valve that has steam lap will keep the port closed against the admission of steam until the valve has moved from its mid-position a distance equal to the steam lap. The valve shown in Fig. 77 is an outside admission valve and it has moved to the right of its mid-position a distance equal to the steam lap. In this position the valve is just on the point of admitting steam to the head end of the cylinder. If the valve now moves to the right steam will be admitted to the cylinder and admission will continue until the valve moves to the left and returns to the position shown in Fig. 77 when the port will be closed. The port will then remain closed and the steam will expand until the valve moves far enough to the left to bring the inner edge of the valve in line with the inner edge of the port.

A further movement of the valve to the left uncovers the port for exhaust, which continues until the valve, in moving to the right on its forward stroke, reaches a position where the inner edge of the valve is again in line with the inner edge of the port. As the valve continues its movement to the right the port remains closed and the steam in the cylinder is compressed. Compression continues until the valve, still moving towards the right, reaches a position in which the outer edge of the valve is in line with the outer edge of the port, when admission again occurs. It is thus seen that the purpose in having steam and exhaust laps is to permit the steam to be expanded and compressed.

Position of Crank and Eccentric.—In Fig. 77 the position of the valve is such that steam is about to be admitted to the head end

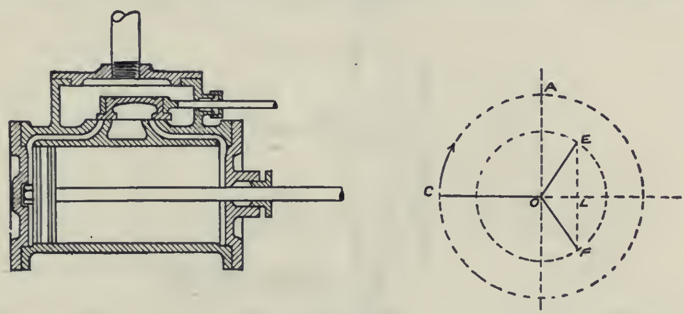


Fig. 77.

of the cylinder and the piston is shown at the beginning of its forward stroke. If the valve is connected to the eccentric without the use of a rocker arm, the corresponding positions of the crank and eccentric will be as shown in the diagram at the right of Fig. 77, in which O represents the center of the shaft, OC the position of the crank, and OE the position of the eccentric. When the valve is in its mid-position the eccentric is vertical, in the position of the line OA, but in the position shown at OE it has been moved around on the shaft in a clockwise direction enough to move the valve through a distance equal to its steam lap, which is equal to the distance OL on the diagram. The distance OL on the diagram represents the displacement of the valve from its mid-position.

With the positions of crank and eccentric as shown in Fig. 77 if the shaft turns in a clockwise direction, as shown by the arrow, the piston will move forward and the valve will move to the right, admitting the steam behind the piston. This will push the piston forward and cause the engine to run. If, however, an attempt is made to start the engine by turning the shaft in a direction opposite to that indicated by the arrow, or in a counterclockwise direction, the valve will be moved to the left and prevent the admission of steam. It is seen then that, with the positions of crank and eccentric as shown, the engine can run only in a clockwise direction. *If there is no rocker arm, the direction of rotation of the engine will be such that the crank follows the eccentric.* If it was desired to have the engine under consideration run in a counterclockwise direction the eccentric would have to be set in the position *OF*, the crank being at *OC*.

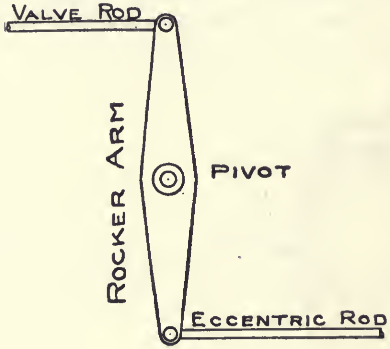


Fig. 78.

Sometimes the shape of the engine is such that the valve and eccentric are not in line with each other and the motion is transmitted from the eccentric to the valve through a rocker arm as shown in Fig. 78. If the valve rod and eccentric rod are both on the same side of the pivot the motion of the valve will be in the same direction as if the valve rod was connected directly to the eccentric rod, but if they are connected on opposite sides of the pivot, as shown in Fig. 78, the valve will move in the opposite direction to that in which it would move if directly connected. Since a rocker arm pivoted between valve and eccentric rods reverses the motion of the valve, the position of the eccentric with respect to that of the crank will be as indicated by the line *OF* of the diagram at the right of Fig. 77 for a clockwise direction of rotation. In other words, the use of such a rocker arm requires that the eccentric be set to follow the crank.

Lead.—In Fig. 77 the valve is set to admit steam to the cylinder just at the beginning of the stroke. If a valve is set in this way there will be considerable drop in steam pressure during admission, or "wire drawing." The effect of this on the indicator diagram is shown in Fig. 79 in which the reduction in the admission pressure is indicated by the drop in the admission line. Since this reduces the area of the diagram it shows that the power of the engine is reduced. It also reduces the efficiency of the engine because the range of pressure through which the steam may be expanded is reduced.

An inspection of the diagram at the right of Fig. 77 will show that, if the shaft rotates with a uniform speed, the valve travels fastest when it reaches mid-position and the piston travels fastest

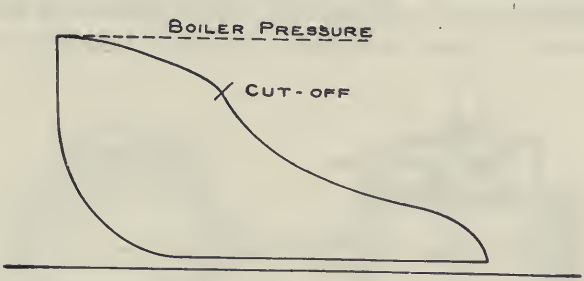


FIG. 79.

when it reaches mid-stroke, the speed of each increasing during the first part of its stroke and decreasing during the last part. This diagram also shows that, at the beginning of the piston stroke, the speed of the valve is decreasing and the speed of the piston is increasing. The result is, that if the valve is set to open just at the beginning of the piston stroke, steam cannot flow into the cylinder through the narrow opening fast enough to maintain full pressure behind the piston, hence the pressure in the cylinder drops.

In order to prevent excessive drop in pressure during admission the valve is set so as to open slightly before the end of the exhaust stroke thus insuring enough port opening to allow free admission when the piston starts forward. The amount which the port is open for admission when the piston is at the end of its stroke is called the *lead* of the valve. The amount of lead which a valve should have depends upon the size of the cylinder and speed of the engine, being larger for high speeds and large cylinders and smaller for slow speeds and small cylinders.

The cushioning of the piston depends to a certain extent upon the lead as a large lead gives an early opening of the valve with an early admission of high pressure steam against which the piston must advance at the end of the exhaust stroke.

A valve set with lead is shown in Fig. 80, the relative positions of the crank and eccentric being shown in the diagram to the right of the figure. In this illustration the piston is at the head end of its stroke and the valve is open an amount l for the admission of steam to the head end. The distance l, in this case is the lead. It will be observed that, in the position shown, the valve has moved to the right of its mid-position a distance equal to the steam lap plus the lead.

In the diagram at the right of Fig. 80 the crank OC is shown on dead center to correspond with the position of the piston. The

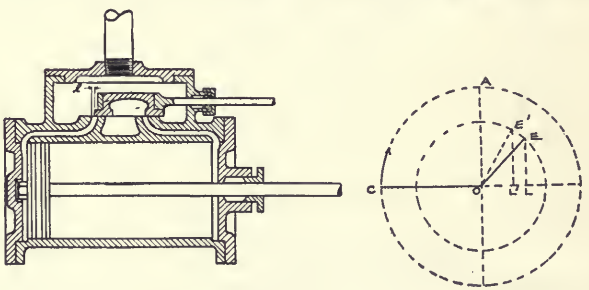


Fig. 80.

line OE shows the position of the eccentric and the distance OL shows the displacement of the valve from its mid-position. If the valve was set without lead the position of the eccentric would be OE', the distance OL' being equal to the steam lap. Since the distance OL is equal to the steam lap plus the lead, the distance $L'L$ is equal to the lead of the valve. It will be observed that when the valve has lead, the eccentric must be moved around on the shaft far enough to displace the valve from its mid-position a distance equal to the steam lap plus the lead when the crank is on center.

Angle of Advance.—For an outside admission valve connected directly to the eccentric, the eccentric must be set so that it leads the crank. When the crank is on center the valve must also be displaced to the right of its mid-position a distance equal to the steam lap plus the lead. These two conditions determine the position of the eccentric with respect to that of the crank.

If the eccentric was in the position OA, Fig. 80, when the crank was on center, the valve would be in its mid-position, hence the eccentric must be moved forward through the angle AOE in order to displace the valve from its mid-position a distance equal to the steam lap plus the lead. This makes the angle between the crank and eccentric, which is called the *crank angle*, greater than 90°. The angle AOE is called the *angle of advance* and it is the angle, in excess of 90°, between the crank and eccentric. The angle of advance is usually about 20° to 30° but its amount will depend upon the steam lap and the lead which it is desirable to give the valve. The angle of advance is important in valve setting, as will be shown later, because it is the only thing about the valve mechanism besides the length of the valve rod which may be adjusted.

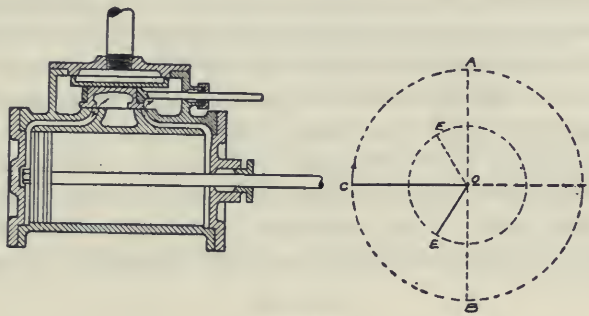


Fig. 81.

Inside Admission Valve.—Many valves, especially of the piston type, are designed to admit steam from the inside and exhaust past the outer edge of the valve. In this case the inside lap is the steam lap and the outside lap is the exhaust lap. A slide valve arranged for inside admission is illustrated in Fig. 81. As shown here the piston is at the head end of its stroke and steam is being admitted to the head end of the cylinder. The valve is therefore displaced to the left of its mid-position a distance equal to the steam lap plus the lead, and the eccentric must be on the left-hand side of the vertical line AB in the diagram to the right of Fig. 81. In order for the engine to run the valve must move to the left when the piston starts on its forward stroke.

If this valve is connected directly to the eccentric, without a rocker arm, and the rotation is to be clockwise the eccentric must be in the position OE when the crank is in the position OC, since in this position the valve will be moved to the left and be opened a greater distance when the crank moves in a clockwise direction. If the crank should move in a counterclockwise direction the valve would close and cut off the supply of steam when the piston started forward. Suppose the eccentric was placed at OE' when the crank is at OC. The valve would then be in the position shown in Fig. 81 but a clockwise rotation of the shaft would move the valve to the right and close the port. However, if the shaft rotates in a counterclockwise direction the valve would move to the left and open the port further when the piston starts forward. It may be stated, then, that *for an inside admission valve without a rocker arm the eccentric should be set to follow the crank.* Since a rocker arm reverses the direction of motion of the valve, the presence of a rocker arm requires that the eccentric be set to lead the crank, that is, in Fig. 81 if there were a rocker arm between the eccentric and the valve, a clockwise direction of rotation would require that the eccentric be set at OE' and a counterclockwise rotation would require that it be set at OE.

In Fig. 81 the angle of advance is BOE and it is negative, since the crank angle is less than 90°. The angle of advance for a valve with inside admission does not differ in amount from that for a valve with outside admission, since its amount depends only upon the steam lap and the lead, but it does differ in position.

CHAPTER XI

THE VALVE DIAGRAM

Valve Displacement.—Since an eccentric is equivalent to a crank, it may be represented as a crank having a length equal to the eccentricity. In Fig. 82 the valve V is connected by the eccentric rod E to the eccentric OC, the eccentric being represented here by a crank having a length OC equal to the distance from the center of the shaft to the center of the eccentric. The circle $ABCF$ represents the path followed by the center of the eccentric as the shaft rotates—and its diameter AB shows the length of the valve travel, which is twice the eccentricity OC.

When the center of the eccentric C is at the point A, the valve is at the extreme left of its travel, when C is at the point B the

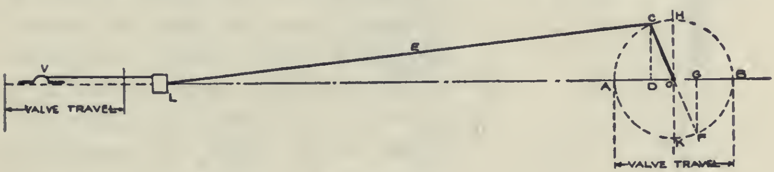


Fig. 82.

valve is at the extreme right of its travel, and when C is at H the valve is in its mid-position. As C passes through the half circle AHB the valve moves through a distance equal to that from A to B, and as C passes through the half circle BKA the valve moves through a distance equal to that from B to A. The diameter AB represents the valve travel to the same scale that OC represents the eccentricity, and the position of the valve at any time during its travel may be located on the diameter AB. It will be observed that the eccentric rod is long in comparison with the valve travel, and that during a revolution of the shaft the eccentric rod never makes a large angle with the center line BL. When this is the case the position of the valve may be located for any position of the eccentric by simply projecting vertically to the diameter AB, the point representing the center of the eccentric. Thus, when the eccentric is in the position

OC the valve will be at the point *D* in its travel, the point *D* being found by drawing *CD* at right angles to *AB*. The distance *OD* is the displacement of the valve from its mid-position. When the eccentric is at *OF*, *G* represents the position of the valve, and *OG* its displacement from mid-position. It will be observed that if *F* is exactly opposite *C*, the valve displacement *OG* when the eccentric is at *OF* is the same as its displacement *OD* when the eccentric is at *OC*.

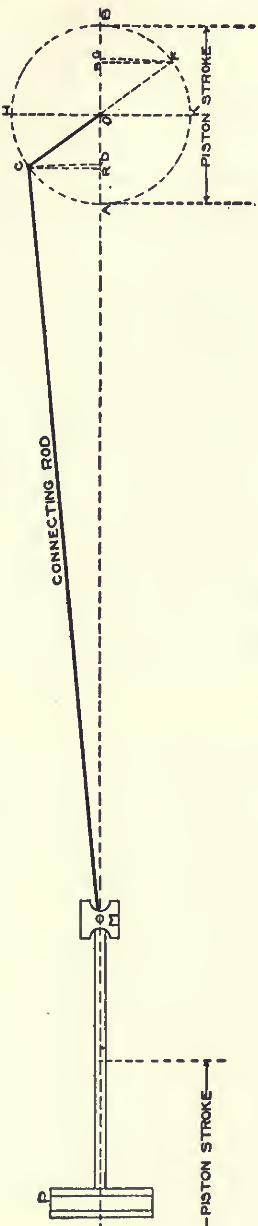


Fig. 83.

Piston Position.—The same kind of diagram as shown in Fig. 82 may be used to represent the travel of the piston, but the method of locating the position of the piston differs on account of the fact that the connecting rod is usually shorter when compared to the length of the piston travel and that at certain parts of the revolution of the crank, the connecting rod makes a considerable angle with the center line of the engine.

In Fig. 83 the circle *AHBK* represents the path followed by the center of the crank pin *C* during a revolution of the shaft whose center is at *O*. *OC* is the length of the crank and the diameter *AB* of the crank circle represents the piston stroke. When the crank is in any position as *OC*, the corresponding position of the piston may be located by taking a radius equal to the length of the connecting rod *MC*, and with *M* as a center drawing an arc *CD* through *C* until it strikes the diameter *AB* at the point *D*. The point *D* will represent the position of the piston for the position *OC* of the crank and the distance *AD* shows how far the piston is from the end of its stroke when the crank is at *OC*. The apparent position of the piston is at R, obtained by projecting the point *C* vertically on the diameter *AB* as was done for the valve position in Fig. 82. The actual position *D* of the piston is displaced from its apparent position *R* by the distance *RD* and this is due to the comparatively large angle which the connecting rod makes with the center line *LB* of the engine. When the crank is in the position *OF*, exactly opposite *OC*, the actual position of the piston is at *G* while its apparent position is at *S*. This effect is due to the *angularity of the connecting rod* and its amount depends upon the length of the connecting rod

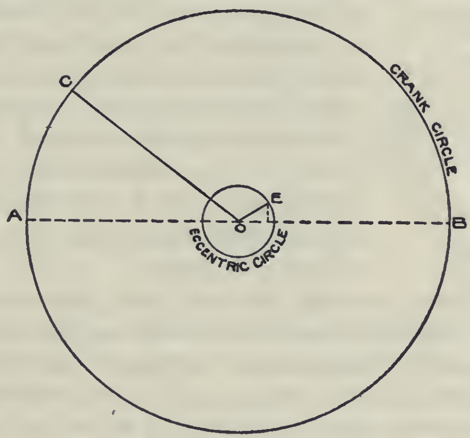


Fig. 84.

as compared with the length of piston stroke. In Corliss engines the length of connecting rod is so great, as compared with the piston stroke, that the angularity of the connecting rod seldom need be taken into account. Other types of engines, on the other hand, have comparatively short connecting rods and the angularity must be considered in locating the position of the piston for cut-off, release, and compression, and sometimes also for admission, which occurs nearer the end of the stroke than any of the other events and hence is not so much affected by the angularity of the connecting rod.

Position of Crank and Eccentric.—In Fig. 82 it is shown that the position of the eccentric and the valve displacement may be represented on a circle having a radius equal to the eccentricity and in Fig. 83 it is shown that the position of the crank and piston may be represented on a circle having a radius equal to the length of the crank. The corresponding positions of crank and eccentric may therefore be represented by two circles drawn about the same center, one for the crank and one for the eccentric, as shown in Fig. 84.

In most engines the eccentricity is small as compared with the length of the crank, hence if both the crank circle and the eccentric circle are drawn to the same scale, one will be large and the other small, and it may happen that the eccentric circle will be so small as to cause difficulty in making measurements upon it. In Fig. 84 the length of the crank OC is 12 inches, giving a stroke of 24 inches, and the eccentricity OE is 2 inches, giving a valve travel of 4 inches, these being common proportions between valve travel and stroke. In order to draw the corresponding positions of crank and eccentric upon these circles, the position of the crank OC is first drawn. Then by laying off the crank angle COE, the eccentric OE may be drawn.

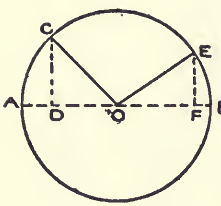


Fig. 85.

It will be seen from Fig. 84 that if both the crank circle and the eccentric circle be drawn to the same scale it will be difficult to measure valve displacements accurately on account of the small size of the eccentric circle. In order to avoid this difficulty the eccentric circle may be drawn the same size as the crank circle, thus making a single circle serve for both crank circle and eccentric circle, as in Fig. 85. If this is done the diameter of the circle will represent the piston stroke to one scale and the valve travel to a different scale. Thus, in Fig. 85 the diameter AB represents a piston stroke of 24 inches and it also represents a valve travel of 4 inches. For the crank position OC the piston is at a distance AD from the end of its stroke and this distance is 3.6 inches on the scale by which the diameter represents 24 inches. The valve displacement corresponding to the crank position OC is shown at OF and this distance is 1.6 inches on the scale by which the diameter represents 4 inches.

Valve Diagram.—A diagram may be drawn which shows the valve displacement for all positions of the crank. Such a diagram is called a *valve diagram.* A valve diagram is useful in setting the valve because it shows at a glance the effects of any changes which may be made in the valve or eccentric and thus tells what changes to make in order to accomplish desired results. The diagram shown in Fig. 85 might be used as a valve diagram since from it could be obtained the valve displacement for any position of the crank, but it is not convenient to use this diagram because the crank angle must be laid off for each new position of the crank at which it is desired to measure the valve displacement.

The most common form of valve diagram is called the Zeuner diagram, after the name of its inventor. The Zeuner valve diagram is drawn as follows: The circle ACB in Fig. 86 is drawn so that its diameter represents the piston stroke to one scale and the valve travel to another scale. Imagine the eccentric to be in the position OB; the crank will then be in the position OC and the angle COB will represent the crank angle. With the eccentric in the position OB the valve will be at the extreme right of its travel and its displacement from mid-position will be a maximum, being equal to the eccentricity. Using the line OC, which represents the crank, as a diameter, draw

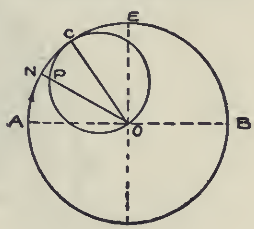


FIG. 86.

the small circle OPC. This is called the valve circle. Any radial line drawn from O to represent any position of the crank, and cutting the valve circle, will show the valve displacement from mid-position by the length which the valve circle cuts off on it. Thus, when the crank is in the position OC, the valve displacement is equal to the length OC. When the crank is in any other position, as at ON the valve displacement is OP, being always the length which the valve circle cuts off on the radial line which represents the crank position.

It is to be observed that in Fig. 86 the angle COE represents the angle of advance since the angle of advance is equal to the crank angle minus 90°. In this case the angle COB is the crank angle and taking away 90°, or the angle EOB, leaves the angle COE as the angle of advance.

In the preceding chapter it was shown that the valve displacement at admission and cut-off, is equal to the steam lap. Therefore, if the arc of a circle be drawn with O, Fig. 87, as a center and a radius OL equal to the steam lap, it will cut the valve circle at the points H and J and a line OD drawn through O and H will represent the position of the crank at admission, when the valve displacement is equal to OH, the steam lap. Also, a line OG drawn through O and J will represent the position of the crank at cut-off when the valve displacement is again equal to the steam lap, OJ. The arc HLJ is called the steam lap circle. Fig. 87 is drawn in the same way as Fig. 86, but is made separate in order to avoid confusion.

The lead of a valve is the amount of port opening when the crank is on dead center. The port opening is equal to the valve displacement minus the steam lap. In Fig. 87, OA represents the position of the crank when on dead center. OM shows the

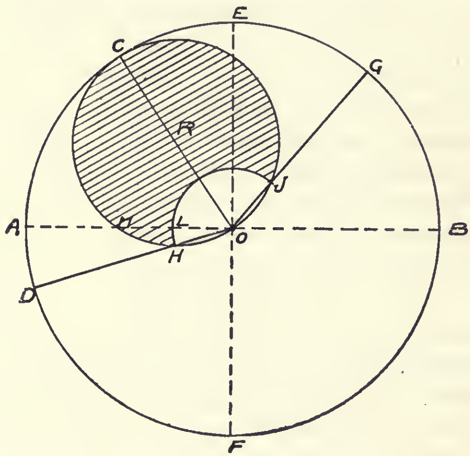
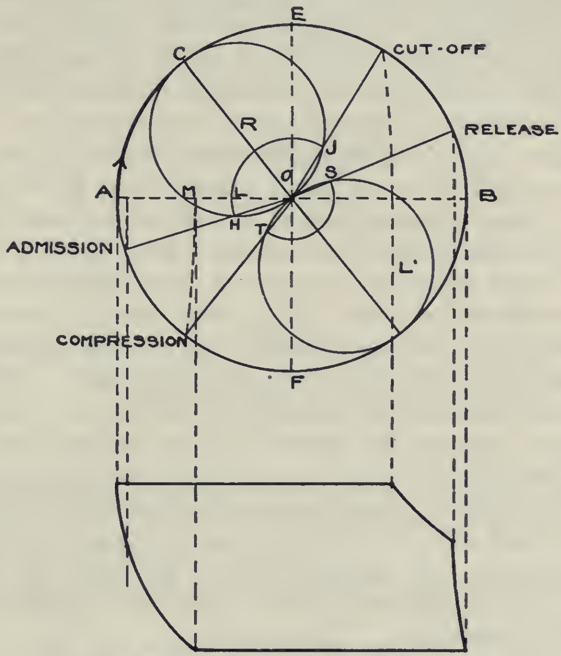


Fig. 87.

valve displacement for this position of the crank and OL the steam lap, hence LM represents the lead of the valve. The amount of port opening is always the distance between the lap circle, and the valve circle, as shown by the shaded area in Fig. 87.

It should be observed that the diagram in Fig. 87 shows only valve displacements to the right of mid-position and for this reason the valve circle is marked R. In order to show valve displacements to the left of mid-position another valve circle L' must be drawn opposite the one marked R, as shown in Fig. 88. The valve circle marked L' for showing displacements to the left must be the same size as the other one and must be drawn on the same line OC extended through the crank circle. Since release and compression occur when the valve displacement is equal to the exhaust lap, the lap circle *TS* must be drawn with a radius equal to the exhaust lap. The four positions of the crank, at admission, cut-off, release, and compression may now be drawn on the diagram as shown in Fig. 88. Since admission and cut-off, as shown here, take place when the valve is displaced to the right, and release and compression take place when the valve is displaced to the left, these events are all for one end of the cylinder only. In Fig. 88 the events shown are for the head end



HEAD-END INDICATOR DIAGRAM

FIG. 88.

of the cylinder, if there is no rocker arm, the direction of rotation being clockwise, as indicated by the arrowhead.

With the crank positions at admission, cut-off, release, and compression as shown in Fig. 88 the approximate shape of the indicator diagram that will be obtained from this valve setting may be shown if the admission and exhaust pressures are assumed. In order to draw this indicator diagram the crank positions are projected to the line *AB* with a radius equal to the length of the connecting rod (to the same scale that *AB* represents the piston stroke). These points are then projected vertically to the admission and exhaust lines below and the indicator diagram sketched in as shown, using smooth curves to connect cut-off and release, and also compression and admission.

The effects of certain changes in the valve setting may readily be observed from the valve diagram in Fig. 88. For a given eccentric the eccentricity is a fixed quantity and cannot be changed. The steam and exhaust laps are parts of the valve and cannot be readily changed although they may be decreased slightly by chipping or filing off the end of the valve. This leaves only the angle of advance that may be changed readily, which may be done by shifting the eccentric around on the shaft either to increase the angle of advance or to decrease it.

The angle *COE* in Fig. 88 represents the angle of advance. It will be observed from the valve diagram that if the angle of advance is made larger admission, cut-off, release, and compression will all occur earlier in the stroke. If the angle of advance is made smaller all of these events will occur later in the stroke. It will be observed also that, other things being left unchanged, a large steam lap gives late admission and early cut-off and a large exhaust lap gives late release and early compression. The steam and exhaust laps might be increased by fastening a block to the outer or inner edges of the valve but this is not often done as it is inconvenient and not often necessary since the valve is designed with the proper laps. A smaller steam lap causes early admission and late cut-off and a small exhaust lap causes early release and late compression. As explained before, the steam and exhaust laps may be made slightly smaller by filing or chipping, but this is not often necessary.

It will be further observed from the valve diagram that the angle which the crank turns through while the steam is being compressed in the cylinder, is the same as the angle turned through while the steam is expanding. This sets an important limitation upon the action of the slide valve because it does not permit so large a degree of expansion of the steam with economy as might otherwise be secured. In order to secure good economy an engine must expand the steam through a large range of pressure. The range of pressure through which steam may be expanded depends upon the point of cut-off; if cut-off is early the range of pressure will be large, but if cut-off is late the range of pressure will be small. If it is attempted to secure an early cut-off with a slide valve the point of compression will also be early. If the cut-off is made earlier than about half stroke, the gain from greater expansion will be more than counterbalanced by the loss from greater compression. This result may be seen from Fig. 89 in which the full line diagram shows cut-off at seven-eighths of the stroke with the corresponding compression, and the dotted lines show the greater expansion for cut-off at half stroke with the corresponding compression. With the cut-off at half stroke the gain in economy from the greater expansion is counterbalanced by the loss of area from the indicator diagram by the earlier compression. The action of the slide valve described above explains why an engine fitted with this type of valve is uneconom-

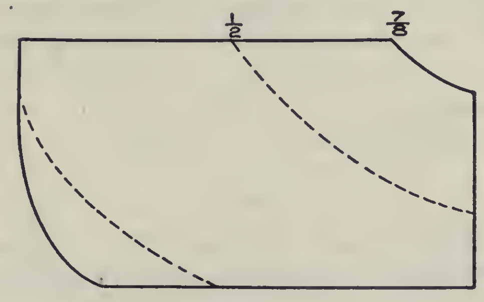


Fig. 89.

ical in the use of steam; that is, it cannot use the full expansive force of the steam as can engines which have separate valves for admission and exhaust.

The valve diagram in Fig. 88 shows the events occurring in only one end of the cylinder, namely, the head end. The same diagram may also be used for showing events occurring in the crank end of the cylinder by drawing the crank end steam lap circle in the left (L') valve circle and the crank end exhaust lap circle in the right (R) valve circle. This has been done in Fig. 90, the steam and exhaust lap circles for the head end being drawn with full lines and the steam and exhaust lap circles for the crank end being dotted. The crank positions for events in the head end are also drawn with full lines and those for the crank end with dotted lines. This makes a complete valve diagram which shows all of the actions occurring in both ends of the cylinder and shows also the effects produced by any changes in the valve or its setting.

All of the valve diagrams shown up to the present time have been for a clockwise direction of rotation and for a direct con-

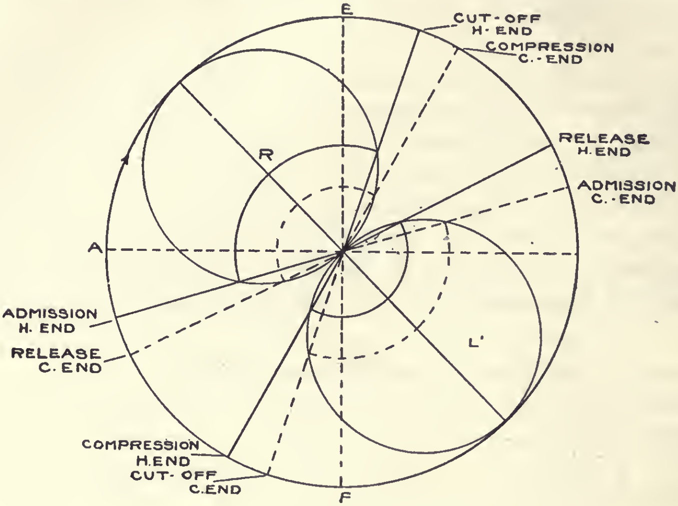


Fig. 90.

nection between valve and eccentric. Fig. 91 shows how the valve diagram should be drawn for a counterclockwise direction

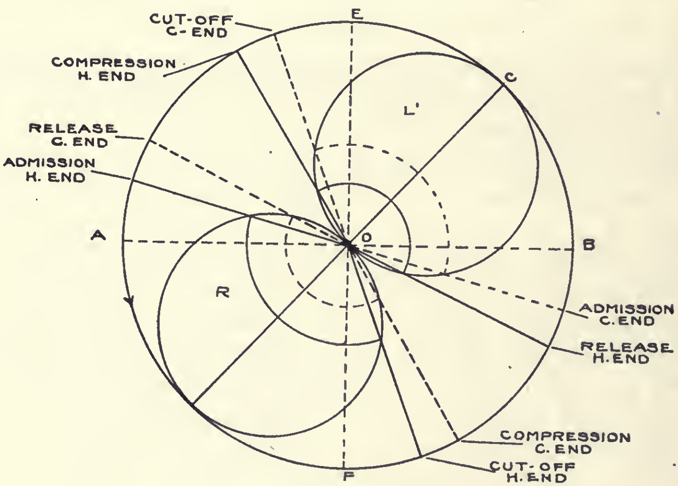


Fig. 91.

of rotation. In this case the angle of advance *COE* is laid off to the right of the vertical line *EF* instead of to the left of it as with clockwise rotation; and in all cases the valve circles are drawn on the line representing the position of the crank when the eccentric is on dead center. In Fig. 91 the crank will be at OC when the eccentric is at OA, hence the valve circles are drawn on the diameter through C and O. Valve displacements to the right of mid-position are then measured on the bottom valve circle R and those to the left are measured on the top valve circle L'. Admission and cut-off for the head end are therefore located by means of the bottom valve circle and release and compression for the head end by means of the top valve circle. Events for the crank end of the cylinder are located by means of the opposite valve circles from those for the head end events.

The late cut-off generally employed with plain slide valve engines makes them uneconomical on account of not using much

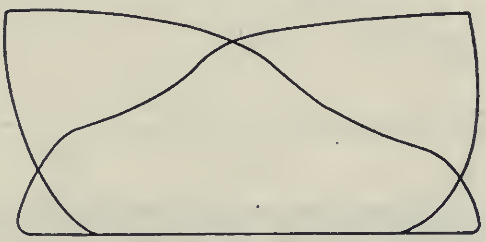


FIG. 91*a*.

of the expansive force of the steam. With a late cut-off the terminal pressure, or pressure at the end of expansion will be high. When the exhaust port is opened, this high pressure remaining in the steam is wasted through the exhaust pipe.

When such an engine is operated under a fairly constant load and not stopped at frequent intervals the steam consumption may sometimes be greatly reduced by redesigning its valve, changing its steam and exhaust taps, and re-setting the eccentric. Fig. 91*a* illustrates a set of indicator diagrams from an engine whose valve was redesigned to give an earlier cut-off. It can be seen from these diagrams that the steam distribution and economy is much better than is obtained with the ordinary slide valve cutting off at about three-quarters stroke. This valve was redesigned so as to give cut-off at about half stroke, release at about 90 per cent. of the stroke, and compression at about 72 per cent. of the exhaust stroke. The angle of advance was also changed to about 50°. Making the cut-off earlier increases the probability that the engine will stop in a position after the valve has closed against admission which makes it impossible to start again until the flywheel has been turned, but if the engine is seldom stopped during the day, this is not likely to become a nuisance.

Defects in the setting or adjustment of valves are readily detected by irregularities in the indicator diagram. A study of the valve diagram will show the causes of such defects in the valve setting and will suggest the remedy that should be applied. One of the most common defects in valve adjustment comes from slipping of the eccentric around on the shaft, the eccentric usually being fastened to the shaft by a single set screw. If the eccentric slips it is likely to do so against the direction of rotation, resulting in a decrease of the angle of advance. A decrease in the angle

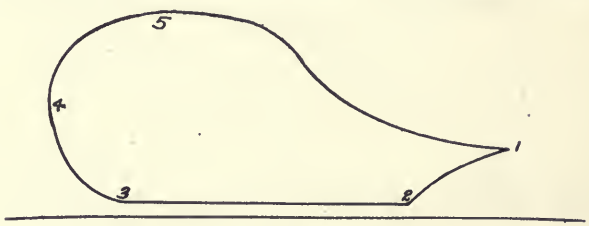


Fig. 92.

of advance causes all of the events (admission, cut-off, release, and compression) to occur later. The effect on the indicator diagram is shown in Fig. 92, which was taken from an engine on which the eccentric had slipped backward. The most noticeable defect due to the slipping backward of the eccentric is seen at release where the diagram will have a beak, as at 1, 2, showing that the valve does not open soon enough to allow the pressure in the cylinder to fall to the exhaust pressure before the piston starts on the return stroke. Slippage of the eccentric cannot be detected readily from the late cut-off because some engines have a much later cut-off than others. With a properly designed slide valve, however, late release is always accompanied by late cut-off. If the valve has proper lead before the eccentric slips backward, it will have too little lead afterwards. The effects of this on the admission line is shown by the rounding from 4 to 5 caused by the admission side of the valve opening too late, thus preventing the pressure in the cylinder from rising quickly and also causing wire-drawing. Small lead is not a good indication of a slipped eccentric because the valve may have been set originally with too little lead. The compression, in this case, is too late for a slide valve and gives another indication that the eccentric has slipped backward.

If, by any means, the eccentric should become turned forward

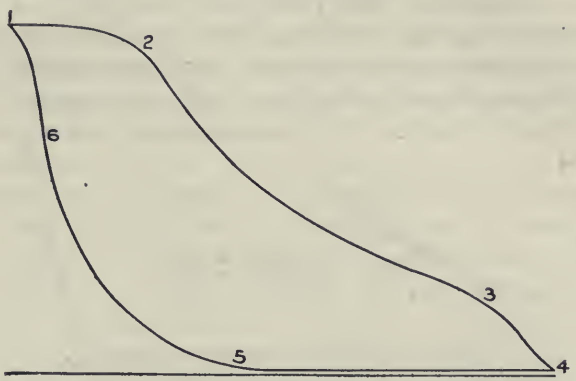


Fig. 93.

on the shaft, the angle of advance will become larger, and all of the events will occur earlier. An indicator diagram showing the results of too great angle of advance is illustrated in Fig. 93. Too early release is shown at 3, 4, by the sharp toe of the diagram, Early admission, the result of too much lead, is indicated at 6.

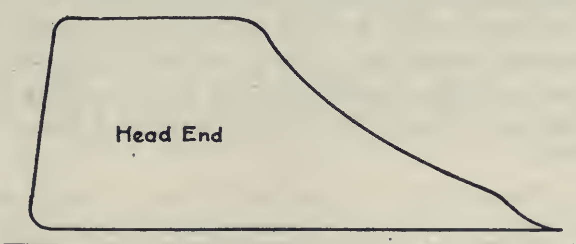


Fig. 93a.

1, by the backward pointing beak. The cut-off, 2, is evidently too early for a slide valve engine, although, in general, the position of the cut-off does not give a good indication of the valve setting. The early compression, 5, agrees with the early cut-off, as it should with a slide valve.

Sometimes the valve becomes displaced on the valve stem, resulting in the valve stem being either too long or too short for the valve setting desired. This will affect the events occurring in the two ends of the cylinder differently since the valve is moved bodily either to the right or to the left. If the valve stem is too long the steam lap on the head end is increased, which delays admission and hastens cut-off on the head end, as illustrated in Fig. 93*a*. The valve stem being too long also decreases the exhaust lap on the head end, which hastens release and delays compression on the head end. The effects produced on the events of the crank end as illustrated in Fig. 93*b* are just opposite

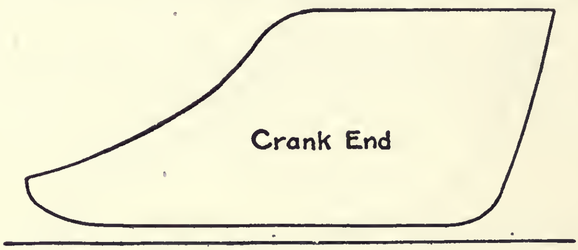


Fig. 93*b*.

from those produced on the head end. The steam lap on the crank end is reduced, which hastens admission and delays cut-off, while the crank end exhaust lap is increased, which delays release and hastens compression.

The changes in events produced by shortening the valve stem will be the same as those produced by lengthening it, except that the changes produced in the head end events by lengthening the valve stem will be produced in crank end events if the valve stem is shortened, and the changes produced in the crank end events by lengthening will be produced in the head end if the valve stem is shortened.

CHAPTER XII

VALVE SETTING

General Considerations.—In setting the valves of an engine the principal requirements are to secure an economical distribution of steam between the two ends of the cylinder and to secure smooth running of the engine. In order to obtain these results it is desirable to divide the work to be performed equally between the two ends of the cylinder. An unequal division of the work between the two ends of the cylinder throws unduly large strains upon the engine, is likely to cause variations in the speed, and causes the steam to be used under more unfavorable conditions in one end of the cylinder than in the other. An equal division of the work will be secured approximately when cut-off occurs at equal points of the admission strokes for the two ends of the cylinder, therefore, in setting engine valves it is desirable to secure equal cut-offs for the two ends of the cylinder.

Vertical engines form an exception to the above statements as in these, the cut-off in the crank end of the cylinder should occur later than that in the head end by enough to lift the weight of the piston, piston rod, crosshead, and connecting rod.

Another requirement towards securing smooth running is that there should be equal leads on the two ends of the cylinder as, by this means, the cushioning effect at each end of the piston stroke is made equal and, if the lead is sufficient for the size and speed of the engine, it will pass the dead centers smoothly and without shock.

It appears from the above discussion that engine valves should be set with respect to the cut-off and lead. An inspection of Fig. 90 will show that both the cut-off and lead depend upon the eccentricity, angle of advance, and steam lap. All of these, except the angle of advance, are fixed dimensions and cannot be readily changed after the engine is built. However, as valve gears are usually constructed, the length of the valve stem may be changed or, what amounts to the same thing, the position of the valve on the stem may be changed Changing the length of the valve stem, which moves the valve bodily along on its seat, has the effect of increasing the steam lap and decreasing the exhaust lap on one end, and decreasing the steam lap and increasing the exhaust lap on the other end. For example, if the length of the valve stem is increased, the steam lap on the head end will be increased, the exhaust lap on this end decreased and the steam lap on the crank end will be decreased and the exhaust lap on this end increased. In setting valves, therefore, the two things that may be changed are the angle of advance and the length of the valve stem.

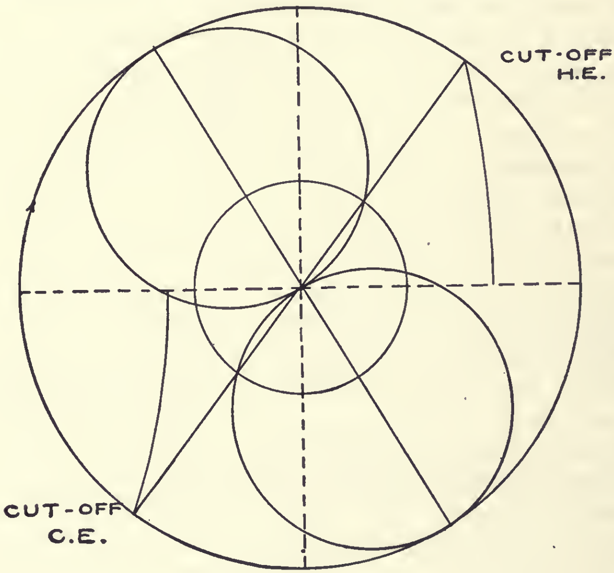


Fig. 94.

An inspection of Fig. 94 will show that if a valve has equal steam laps on head and crank ends, the leads on the two ends will also be equal. If the connecting rod was long enough so that its angularity did not affect the motion of the piston the cut-off on the two ends would also be equal when the steam laps are equal but actually, the angularity of the connecting rod does affect the motion of the piston and causes cut-off to occur later in the stroke from head to crank end than it does in the stroke from crank to head end. It may be stated as a general rule, therefore, that *an ordinary slide valve cannot be set to give both equal cut-offs* *and equal leads,* although it would be desirable if this could be done.

If the valve gear contains a rocker arm which reverses the motion of the valve it is possible to shape the rocker arm so the angularity of the connecting rod may be compensated for and the cut-offs made equal while retaining equal leads, but this can be done for cut-off at only one point of the stroke. If the valve is set to cut off at any other part of the stroke either the cut-offs or the leads will be unequal. Rocker arms of this kind are not made straight, but instead, the part on one side of the pivot is made at an angle to the part on the other side of the pivot. These are sometimes used on automatic high speed engines but are seldom used on plain slide valve engines.

A plain slide valve may be set to give equal leads on the two ends of the cylinder, the cut-offs being unequal, or to give equal cut-offs, the leads being unequal, or a compromise may be made

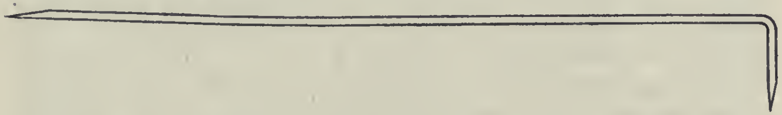


Fig. 95.

and the valve set to give slightly unequal leads and slightly unequal cut-offs.

In setting engine valves it is necessary to place the engine on dead center. A person cannot judge when an engine is exactly on dead center because when near the end of the stroke the crank moves through a considerable angle while the piston and crosshead move very little For this reason it is desirable to use some method for putting the engine exactly on dead center and the following method will be found satisfactory.

Placing an Engine on Center.—An engine is placed on center by means of a *tram,* which is an iron rod pointed at both ends and having one end bent at right angles to the main part of the rod, as shown in Fig. 95. A tram about 30 inches long and made of a rod $\frac{3}{16}$ inch in diameter is a convenient size to use.

In placing an engine on center by the use of a tram the flywheel is turned by hand until the crosshead is within three or four inches of the end of its stroke. In turning the engine by hand it should always be turned in the direction in which it ordinarily runs in order to take up the lost motion or backlash in the various bearings. When the crosshead has been brought within three or four inches of the end of its stroke a scratch mark is made on both crosshead and guide, as shown at *B* Fig. 96, so that by bringing the parts of this mark together at any time the crosshead will be in the same position as before. The straight end of the tram is now placed on a fixed mark on the floor and a mark is

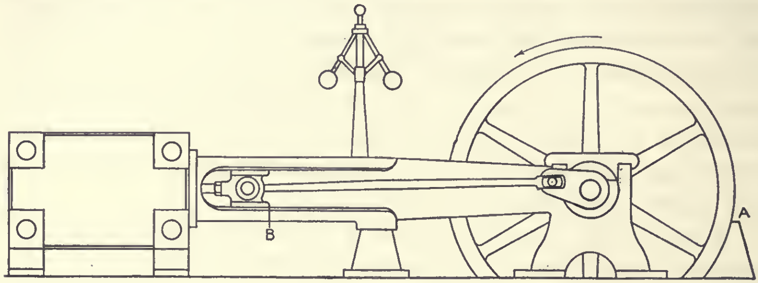


Fig. 96.

made with bent end on the rim of the flywheel as shown at *A* in Fig. 96, the crosshead still being in such position that the marks on crosshead and guide are together. The flywheel is now turned by hand until the crank is past center and the crosshead has been brought to the same position as before, as indicated by the marks on crosshead and guide coinciding. The crank

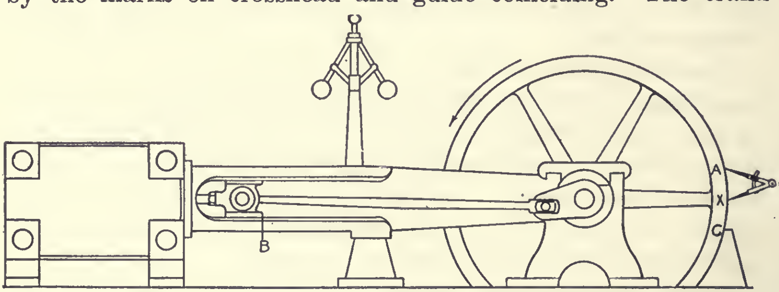


Fig. 97.

will now be as far past center as it was in front of center before. With the engine in this position the tram is placed on the permanent mark on the floor and another mark made on the rim of the flywheel, as shown at *C* in Fig. 97. With a tape measure or pair of dividers find the point *X* midway between *A* and *C*. Make a center punch mark here and turn the engine until the tram, still on the permanent mark on the floor, falls square into the punch mark, when the engine will be exactly on center.

The above method should be used in finding the other dead center position and, with these two positions marked, the engine may be quickly placed on center at any time by turning it until the tram, resting on the permanent mark on the floor, comes to the punch mark representing the dead center position.

To Set Valves With Equal Leads.—In setting the valve, two results must be accomplished. First, the valve must be made to travel equal distances each side of its central position, thus giving the valve *equal* leads on its two sides; second, after making the leads *equal*, they must be adjusted to the desired amount.

In order to accomplish these results, two adjustments are possible; first, the position of the valve on the rod may be changed, or, what amounts to the same thing, the length of the valve rod may be changed; second, the eccentric may be shifted around on the shaft. Changing the length of the valve rod (or the position of the valve on the rod) *increases* the lead at one end of the valve and *decreases* it at the other end. If the rod is lengthened the lead will be increased at the crank end and decreased at the head end. If it is shortened, the lead will be increased at the head end and decreased at the crank end. Shifting the eccentric around on the shaft *increases both leads* or *decreases both leads*, depending upon which direction the eccentric is shifted. If the eccentric is shifted so as to *decrease* the angle of advance, both leads are *shortened* and if the angle of advance is *increased* both leads are lengthened.

In setting a slide valve, proceed as follows: (1) Set the engine on head end dead center and (having removed the cover of the valve chest) measure the lead which the valve has on that end. If the valve covers the port (negative lead) mark the position of the valve and then turn the engine forward until the edge of the valve is in line with the edge of the port, and measure the distance which the port was overlapped by the valve.

2. Turn the engine forward until the crank end dead center is reached and measure the lead in the same manner.

3. If the leads are *equal* and of the *required* amount, no further adjustment is needed.

4. If the leads are equal but not of the required amount, move the eccentric forward to give more lead or backward to give less lead, as required.

5. If the leads are *unequal*, they must first be made equal by changing the length of the valve rod. To do this, take half of

the difference between the leads and change the length of the valve rod by this amount, lengthening it if the head end lead is larger or shortening it if the crank end lead is larger. This will make the leads equal. Then make the leads the required amount by the method indicated in (4) above.

After the valve has been set by measurement, as above, the engine should be run and indicator diagrams taken. The indicator diagrams will show whether or not the valve is set properly, and any slight readjustment that may be necessary may be made after an inspection of the diagrams.

The above method of setting valves requires considerable turning of the engine by hand which, if the engine is large, may be inconvenient. If it is difficult to turn the engine by hand, the following method of setting a plain slide valve may be used and good results obtained. This method, however, is suitable for only plain slide valve engines in which there is no rocker arm for equalizing the cut-off on the two ends of the cylinder.

This method is based on the fact that a slide valve without an equalizing rocker arm will give the same maximum port opening on the two ends of the cylinder when the leads are equal. The valve is therefore first set to give the same maximum port opening on the two ends of the cylinder. This is done by loosening the eccentric on the shaft and turning it around until it gives maximum port opening on first one end and then on the other. If the maximum port openings are not equal they are made so by changing the length of the valve stem by one-half of the difference in the maximum port openings. This operation gives the valve stem its proper length. The engine is now put on dead center and the valve given the proper lead by turning the eccentric on the shaft. This adjusts the angle of advance and will give equal leads at the two ends of the cylinder. The adjustment should now be verified by indicator diagrams as with the preceding method.

Setting Valves for Equal Cut-off.—If there is no equalizing rocker arm the steam laps and leads will be unequal when the valve is set to give equal cut-off on the two ends of the cylinder and, as changing the length of the valve stem is equivalent to making the laps unequal, most of the adjustment is made by the valve stem.

The engine is first placed exactly on its head end dead center, using a tram for this purpose, as previously described. The eccentric is then loosened and turned on the shaft until the valve has the proper lead on the head end. The engine is then moved forward until the valve comes line on line with the edge of the port, which is its position at cut-off. Now measure the displacement of the crosshead from the beginning of its stroke. The engine is then moved forward again until cut-off occurs on the return stroke, and the displacement of the crosshead from the crank end of its stroke is measured. If the cut-off on the head end is earlier than that on the crank end the valve stem is too long, but if the cut-off on the crank end is earlier than that on the head end the valve stem is too short. In either case the length of the valve stem should be changed by an amount which it is estimated will correct the inequality in the cut-offs. Changing the length of the valve stem will, of course, change the lead on the head end; therefore the engine must now be turned to the head end dead center and the lead adjusted to its original amount by moving the eccentric around on the shaft. The cut-offs are again measured for equality and, if necessary, the length of the valve stem again adjusted. In setting valves by this method, a valve diagram will often prove helpful in determining the amount of adjustment to make on the valve stem.

With all the methods of setting valves described above it is necessary to remove the cover of the steam chest. It is convenient sometimes to be able to set the valves without removing the steam chest cover and even when steam is on the engine. This may be the case with a locomotive when out on the road, due to the eccentrics slipping.

In order to be able to set valves without removing the steam chest cover it is first necessary to set the valves properly while the steam chest cover is off and then make reference marks on the valve stem and steam chest.

Preparatory to setting the valves by this method a tram with both ends pointed and bent at right angles should be made. This tram should be about half the length of the valve stem. The steam chest cover is then removed and the valve set with equal leads by the first method described above. After the valve is properly adjusted the engine is placed on the head end dead center as shown in Fig. 98, and a punch mark made on the valve chest. One end of the tram is placed in this mark and a mark made on the valve stem with the other end of the tram. The mark on the valve stem is then made permanent by a punch mark. The engine is then turned to the crank end dead center, as shown in Fig. 99, and with the tram in the mark on the valve chest, another mark which is also made permanent by a punch mark, is made on the valve stem. The valves may now be set at any time without removing the steam chest cover by placing the engine on center and then turning the eccentric on the shaft until the tram reaches from the mark on the valve chest to the mark on the valve stem which corresponds to that dead center.

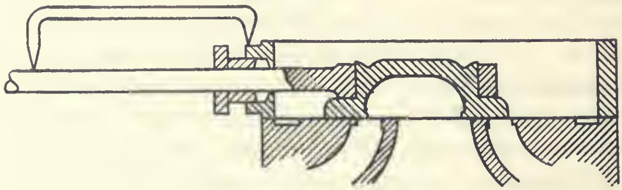


Fig. 98.

Types of Slide Valves.—The form of slide valve which has been illustrated in the previous discussion is known as the *plain* or *D* slide valve, and it has been used in this discussion on account of its simplicity. This form of valve is widely used on the cheaper grades of slide valve engines; but there are certain objections to its use on better grades of engines, the most important of these objections being the wire-drawing, or drop in pressure produced by it during admission, and its friction.

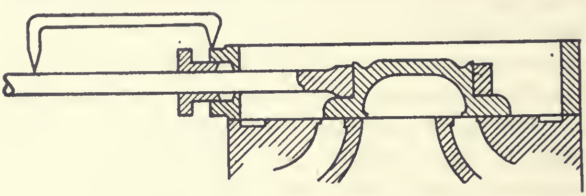


Fig. 99.

Drop in pressure during admission is caused by friction of the steam rushing past the edges of the valve and through the narrow port opening and by the comparatively slow motion of the valve while it is opening and closing. This objection is not so serious as might be first thought because while there is a rather large drop in pressure, there is also some heat produced by friction of the steam which tends to dry the admission steam if it is wet or to superheat it if it is dry. However, on the whole, the drop in pressure during admission is an objection and some engine builders try to avoid it by designing their valves so as to produce a large port opening with a small movement of the valve. In this way the valve opens quicker and wider and there is less friction because of the smaller valve travel.

One of the ways in which these results have been accomplished is by the use of double ported and multiple ported valves.

The double ported valve illustrated in Fig. 100 is designed to give twice as large port opening with the same valve movement as would be obtained with the ordinary *D* valve. Steam surrounds the valve and also fills the hollow chambers *A* and *A* which extend

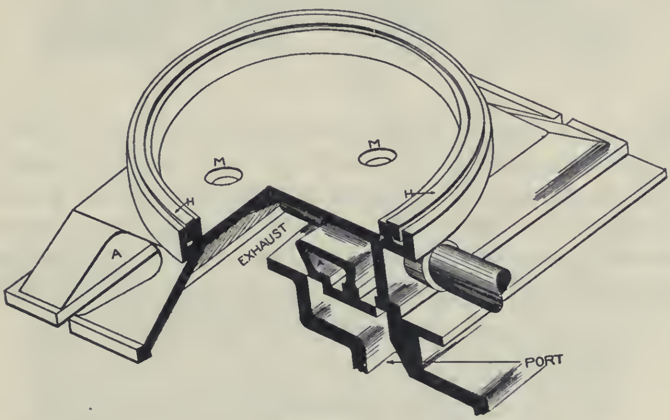


Fig. 100.

entirely through the valve from one side to the other. The bottoms of the chambers *A* and *A* are open so that steam may flow from them into the ports. The ports have two openings so arranged that when the outside edge of the valve uncovers one opening the chamber *A* or *A* uncovers the other opening thus permitting steam to enter the ports at two points. Exhaust occurs past the inside edges of the valve and the inside edges of the chambers *A* and *A*. Passages are cored over the tops of the chambers *A* and *A* so that steam exhausted past the inside edge of the valve may find its way to the central exhaust chamber.

The form of valve shown here is partly balanced by means of the ring *H* which slides on the under side of the steam chest cover. Any leakage of high pressure steam past the ring finds its way into the exhaust chamber through the holes M and M.

This type of valve is widely used on marine engines and it is made both balanced and unbalanced. The smaller sizes of valves are often unbalanced but the larger sizes are invariably balanced.

The Trick valve, illustrated in Fig. 101 and named after its

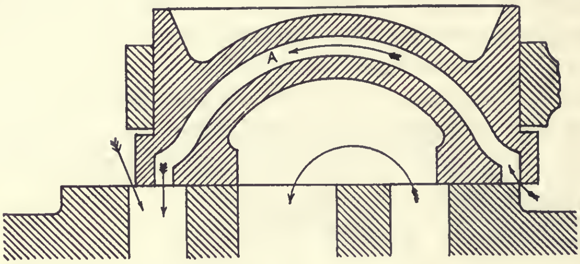


Fig. 101.

inventor, is a form of double admission valve, and is used to a considerable extent on locomotives. Live steam is admitted past the outer edge of this valve and also through the passage A cast in the body of the valve. When the outer edge of the valve uncovers the port for admission the opposite end of the passage A is also uncovered thus giving a double admission of steam, the

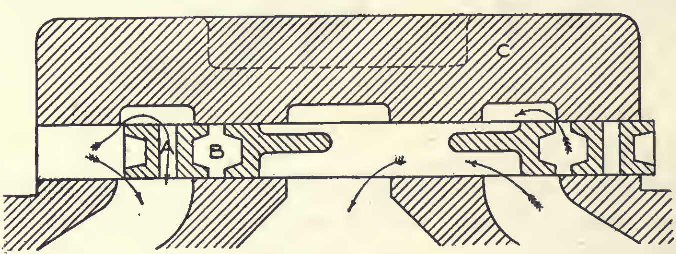


Fig. 102.

same as with a double ported valve. In a similar way, when the outer edge of the valve closes the port for cut-off the opposite end of the passage is also closed. The Trick valve gives double admission but does not give double exhaust since the passage A is not open to exhaust at any time. In this respect the Trick valve differs from a double ported valve.

The Straight-line valve shown in Fig. 102 is both a double admission and a double exhaust valve. This result is secured by means of two ports through each end of the valve, the port A

being for admission, and the port *B* for exhaust. Both sides of the valve are exactly alike and the balance plate *C* has recesses cored in it to correspond with the port openings in the cylinder valve-face. This manner of constructing the pressure plate permits almost perfect balancing of the valve since both sides of the valve are subjected to the same steam pressure.

The passage *A* through the valve gives a wide, quick opening and thus prevents wire drawing during admission. The passage

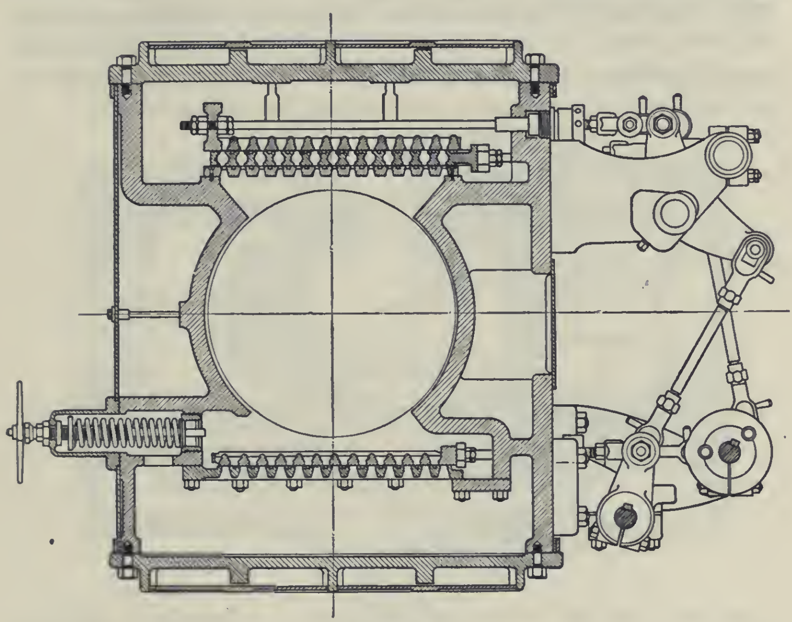


Fig. 103.

serves the same object with the exhaust, preventing wire drawing at release.

The Straight-line valve is used on automatic high speed engines which are controlled by a shaft governor. The shaft governor is connected directly to the valve and changes the lead and angle of advance. The valve is well adapted for this purpose because it is so perfectly balanced that but little power is required of the governor in changing its position.

Multi-ported valves are usually of the gridiron type, consisting of a flat plate with a number of slots in it which slides over a seat with a like number of slots. Valves of this type are illustrated in Fig. 103. With this type of valve a large port opening is obtained.

with but small valve travel. For example, a valve of this type having eight slots and a valve travel of ¼ in. would have a port opening of two inches, while an ordinary valve would have a travel of two inches to secure the same amount of port opening. The smaller travel of the multi-ported valve reduces its friction and the amount of work necessary to move it, and at the same time, makes effective lubrication easier.

Multi-ported valves are usually placed across the cylinder, as shown in Fig. 103, instead of lengthwise of it, and there are usually four valves, one for admission to each end of the cylinder and one for exhaust from each end. This method of placing the valves permits shorter ports and reduces the clearance.

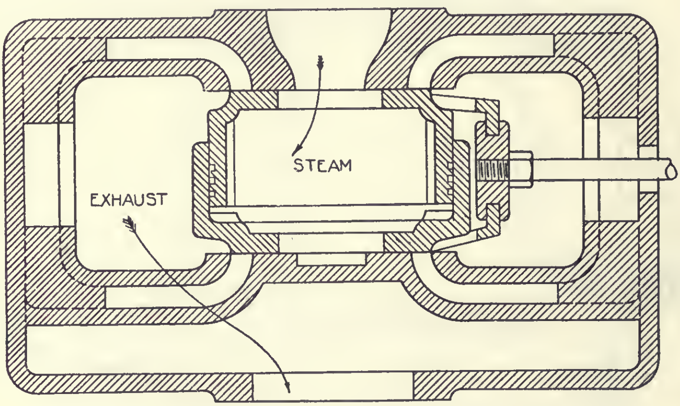


Fig. 104.

Some of the slide valves which have been described and illustrated are balanced or partly balanced. Most balanced valves, except piston valves, are balanced by means of a balance plate over which the valve slides, or by means of a balance ring recessed into the back of the valve and sliding on the inner surface of the steam chest cover plate.

The excessive unbalanced pressure on the common *D*-valve which causes friction and cutting is mainly due to the large exhaust cavity which is filled with low pressure steam while high pressure steam surrounds the outside of the valve. If the valve is arranged so that live steam may be admitted to the cylinder from the inside of the valve while the outside is subjected to exhaust steam the unbalanced pressure is greatly reduced.

The Ball telescopic valve, illustrated in Fig. 104, is designed on this principle. This valve consists of two parts, one of which telescopes into the other. Each part consists of a rectangular frame which slides over the ports, and on the back of which is a short hollow cylinder. The cylindrical parts telescope and the inner one is provided with packing rings to prevent leakage of the steam from the inside to the outside of the valve.

Steam is admitted to the inside of the valve and the exhaust escapes past the outside edges. Both sides of the valve have working edges which make unnecessary double ports and large clearance volume.

Piston valves are used on a great many automatic high speed

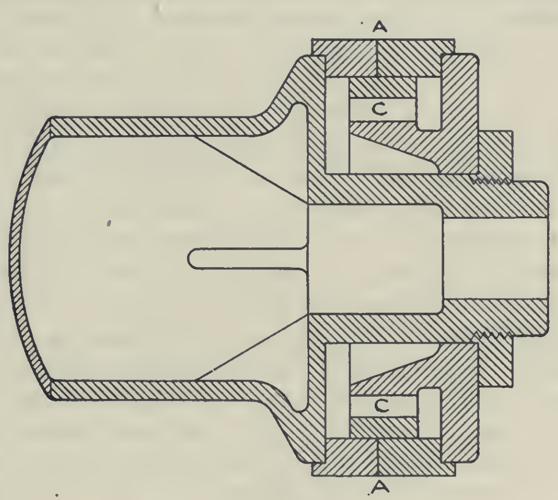


Fig. 105.

engines. This type of engine requires a balanced valve because the governor is attached directly to the valve and governs the speed by changing the position of the valve. If a large amount of power is required to move the valve a sensitive governor cannot be used with it.

Piston valves are cylindrical valves moving in the direction of their axis. They may be made either for inside or for outside admission. The steam ports consist of annular spaces surrounding the valve, and the admission and exhaust edges extend all around the circumference of the valve, hence a large port opening is secured with a small diameter of valve.

Packing rings to prevent leakage of steam are used on the larger sizes of piston valves but the smaller sizes have none. The

Ideal piston valve illustrated in Fig. 105 shows one method of using packing rings on piston valves. Only one end of the valve is shown here, the other end being exactly like this one. Two rings *A* are used on each end of the valve and they are made the exact size of the cylinder bore. These are not "spring" rings, as they would cut the cylinder, but they are split and are slightly adjustable by means of the four shoes *C*. Thus the rings may be adjusted to take up wear and keep the valve steam tight at all times.

CHAPTER XIII

SHIFTING ECCENTRIC AND MEYER VALVE

Shifting Eccentric.—In the plain slide valve engine the eccentric is fastened to the shaft by means of a set screw or key, hence, the cut-off occurs at a fixed point in the stroke and cannot be changed unless the engine is stopped and the eccentric moved around on the shaft. This would change the angle of advance and consequently the point of cut-off.

An inspection of the valve diagram, Fig. 88, will show that the point of cut-off with a plain slide valve may be changed by changing either the angle of advance or the eccentricity. Cut-off may be made earlier either by increasing the angle of advance or decreasing the eccentricity and it may be made later either by decreasing the angle of advance or increasing the eccentricity.

An eccentric constructed in such manner that its eccentricity and angle of advance may be changed without stopping the engine is used on the automatic high speed type of engine. This type of eccentric, which is called a shifting eccentric, is attached to the governor in such manner that the position of the governor controls the eccentricity and angle of advance and by this means regulates the speed of the engine by changing the point of cut-off to suit the load carried by the engine. A device of this kind makes it possible to use the simple slide valve and, at the same time, secure a variable cut-off in governing the speed of the engine.

The principle of the shifting eccentric is illustrated in Fig. 106. The eccentric is not fastened directly to the shaft but to the side of a plate, C, with a projecting arm. The projecting arm is pivoted at A to a point on one of the spokes of the flywheel. The governing mechanism is contained in the flywheel and is attached to the eccentric on the side opposite the projecting arm, C, hence, the eccentric turns with the flywheel. The shaft passes through a slot cut in the eccentric, the width of the slot being a little greater than the diameter of the shaft. The slot is curved to a radius OA equal to the distance from the pivot A

to the center of the shaft, O, so that the eccentric is free to swing about the pivot, A, without touching the shaft.

The governor, which is attached to the eccentric, changes the position of the eccentric with respect to the shaft and thus changes the eccentricity and angle of advance. For example, when the eccentric and shaft occupy the positions shown by the full lines in Fig. 106 the eccentricity, which is the distance from the center of the shaft to the center of the eccentric, is OE and the angle of advance is the angle BOE. If the eccentric is shifted

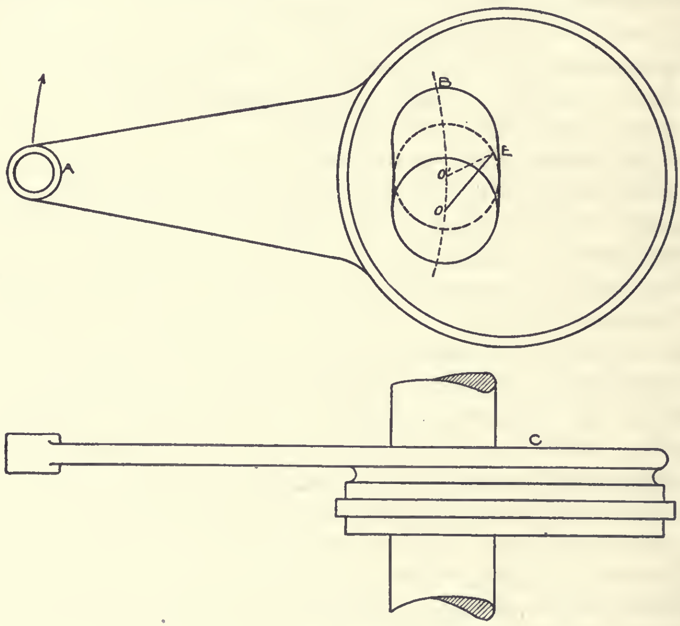


Fig. 106.

until the shaft occupies the position indicated by the dotted circle, with its center at O', the eccentricity will be $O'E$ and the angle of advance will be the angle $BO'E$. It will be observed that in shifting from the first to the second position the eccentricity has been decreased and the angle of advance increased. Both of these changes have the effect of making the point of cut-off occur earlier in the stroke, as shown by the valve diagram in Fig. 107. In this valve diagram the full lines represent the first position of the eccentric, with cut-off occurring at C, and the dotted lines represent the second position of the eccentric with the shorter eccentricity and greater angle of advance, and the cut-off occurring at C'.

The manner in which a swinging eccentric on an automatic high speed engine is operated by the governor is illustrated in Fig. 108. The eccentric with the slot and shaft, a, is shown at R. This is fastened to a plate T which is pivoted to the flywheel at S so that it may turn about the point S and thus change the eccentricity and angle of advance. The governor consists of the weight C, the link H, and the spring E. As the flywheel turns,

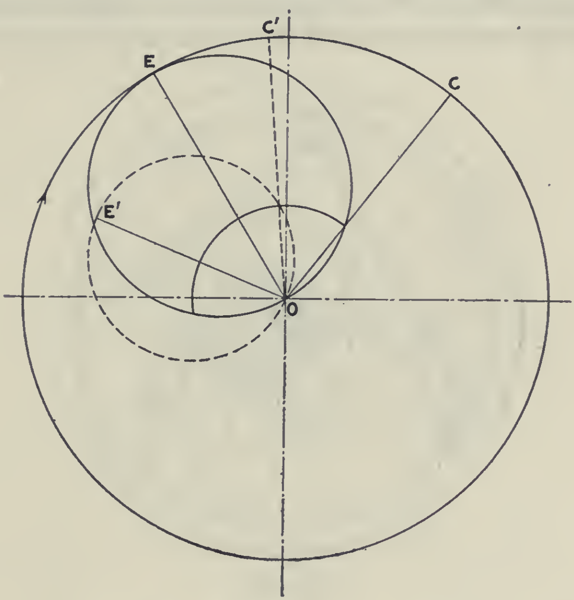


Fig. 107.

the weight, C, is acted upon by centrifugal force which causes it to move outward from the center of the flywheel. The outward movement of the weight is resisted by the spring E so that for any particular speed the weight C will move outward until the resisting force of the spring just balances the centrifugal force acting upon the weight. The weight, C, is pivoted to the arm of the flywheel at O, and the link H is pivoted to the eccentric at T. When the weight, C, moves outward the pivot T is moved in an opposite direction and the center of the eccentric is moved nearer to the center of the shaft. This decreases the eccentricity and increases the angle of advance, which causes cut-off to occur earlier in the stroke and reduces the volume of steam supplied to the cylinder of the engine. A movement of the weight, *C*, towards the center moves the eccentric in the opposite direction, lengthening the cut-off and admitting a greater volume of steam to the cylinder.

The amount of centrifugal force acting on the weight, *C*, increases with an increase in speed and decreases with a decrease in speed of the engine. If, on account of an increase in load on the engine, the speed should be decreased the centrifugal force acting on *C* would become smaller and the spring would pull *C* towards the center of the shaft and move the center of the eccen-

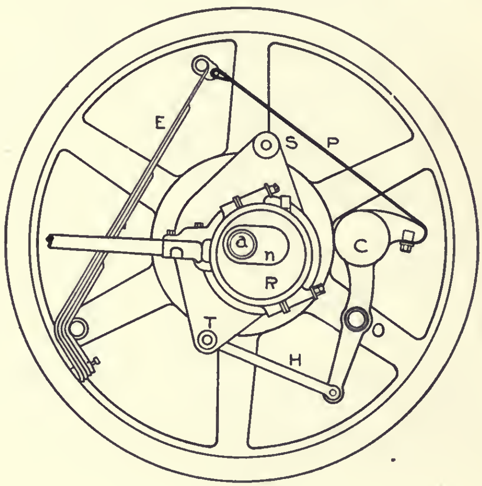


FIG. 108.

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

The spring, *E*, acts as a controlling force upon the weight, *C*, and regulates its position for any given speed, therefore the stiffness of the spring controls the speed at which the engine will run.

The stiffness of the spring may be changed by adjusting the length of the connection, P, between the spring and the weight C, which is provided with a turn buckle for that purpose.

The effects of different positions of the swinging eccentric upon the distribution of steam to the cylinder may be studied from a valve diagram such as that shown in Fig. 109. The swinging eccentric gives its greatest eccentricity when the engine is at rest or when the load is the greatest. In the valve diagram, Fig. 109, the line OE represents this eccentricity and the valve circle shows the cut-off occurring at A_1. The arc ED is drawn with a radius equal to the length of the arm on which the eccentric

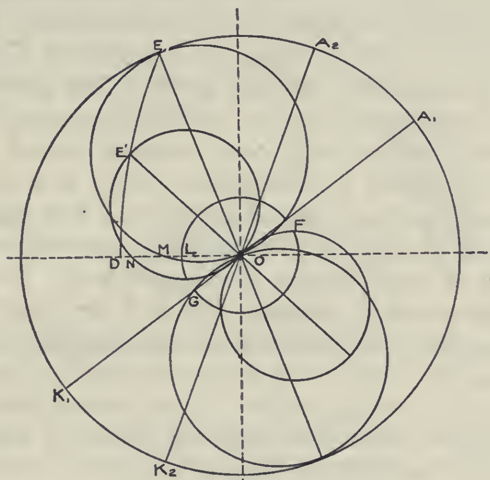


FIG. 109.

swings, or AO in Fig. 106. As the cut-off is shortened by the governor, the eccentricity becomes smaller and for any position of the governor the eccentricity will be the distance from the center O of Fig. 109 to the arc ED. Thus, when the eccentric has been moved one-half of its total swing, its eccentricity will be OE', the point E' being located halfway between E and D, and the corresponding cut-off will occur when the crank is in the position OA_2. A valve circle drawn on OE' will give, by its intersection with the lap circle, the point of cut-off A_2 for the new position of the eccentric, since the lap circle is the same for all positions of the eccentric. In a similar manner, the cut-off for any position of the eccentric may be found by drawing a line from O to a point on the arc ED which represents the position of the eccentric, and then drawing the valve circle on this line.

The events occurring past the exhaust side of the valve may be located by extending the valve circle diameters through the center O and drawing valve circles on them of the same size as those used in locating the point of cut-off. In this way, Fig. 109 shows that the point of compression occurs at K_1 when the cut-off occurs at A_1 and that the point of compression occurs at K_2 when the cut-off occurs at A_2.

It will be observed from the valve diagram that the lead increases as the cut-off is shortened or the load becomes lighter, and that the amount by which the lead changes depends upon the length of the projecting arm to which the eccentric is fastened. When the cut-off occurs at A_1 the amount of lead is LM and when the cut-off occurs at A_2 the lead is LN. The automatic high speed engine is designed to run at practically constant speed and to change the point of cut-off to suit the load carried. The lead has a cushioning effect upon the piston, hence, at constant speed the lead should be practically constant or if it changes at all it should increase when the point of cut-off occurs later in the stroke, as the engine is then carrying its heaviest load. The fact that the swinging eccentric increases the lead for short cut-offs is sometimes urged as an objection to this method of governing. The lead may be made more nearly constant by pivoting the eccentric further from the center of the flywheel since this gives a flatter arc ED on the valve diagram, but there will always be some change in lead with this kind of eccentric.

A study of the valve diagram shows also that the period of compression is increased as the cut-off becomes shorter. Compression cushions the piston, hence it would be desirable to have a constant amount of compression on a constant speed engine. However, the changes in compression have but little effect upon the smoothness of running of the automatic high speed engine on account of the large clearance volume which the engines have. The large clearance volume, made necessary by the high speed of the engine, flattens the compression curve on the indicator diagram, showing that the cushioning effect upon the piston is applied gradually, hence a change in the point of compression does not affect the cushioning effect as much as it would on an engine with smaller clearance.

The action of the flywheel governor and swinging eccentric may be seen from Fig. 110 which shows a series of indicator diagrams drawn for different cut-offs. These diagrams were drawn when the engine was running at the same speed, but with different loads. The changes in lead are particularly noticeable on these diagrams, the one with the latest cut-off showing almost no lead and the others showing increasing lead as the cut-off is shortened. The manner in which the point of compression is changed with the cut-off is also very noticeable. With the latest cut-off the point of compression occurs almost at the end of the exhaust stroke, but, with no load, when cut-off is earliest, the point of compression occurs before mid-stroke.

The valves used on automatic high speed engines are always balanced valves, balancing being secured either by a cover plate on the back of the valve or by the use of a cylindrical or piston valve. It is very necessary, in these engines, that the power

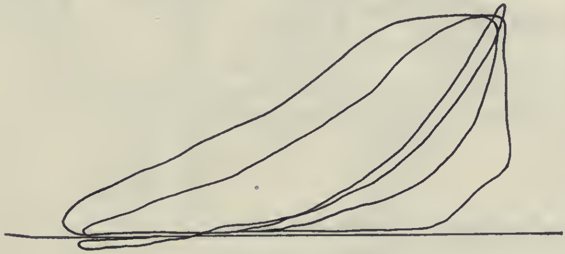


Fig. 110.

required to move the valve back and forth be reduced to a minimum because the valve is moved by the governor and, if much power is required to move it, the sensitiveness of the governor will be reduced. For this reason the valves on automatic high speed engines are balanced and friction is reduced as much as possible by good workmanship in making the valves.

An automatic high speed engine is built to run at a certain speed and the governor is designed and adjusted by the manufacturer for this speed. The speed may, however, be changed to a certain extent either by changing the tension of the springs or by decreasing the weights. The speed will be increased by increasing the tension of the springs or by decreasing the weights as, in either case, a greater speed will be required to overcome the tension of the springs.

Effects Produced by Slide Valve.—One of the principal objections to the plain slide valve engine is its unfavorable steam distribution. All of the events, admission, cut-off, release, and compression are controlled by a single valve and if one of the events is changed all of the others are changed in proportion. For example, if cut-off is made to occur earlier in the stroke all of the other events, release, compression, and admission are also made to occur earlier.

The construction of a plain slide valve is such that the angle turned through by the flywheel during expansion is always equal to the angle turned through during the period of compression, as an inspection of the valve diagram for this type of valve will show. Ordinarily a long period of expansion is desirable as a greater expansion of the steam is then secured and more work obtained from the steam, but this can be obtained, with the plain

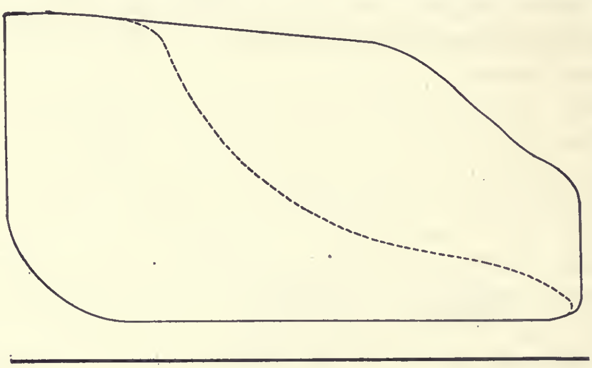


Fig. 111.

slide valve, only by having a long period of compression, which reduces the amount of work secured from the steam. With these opposing conditions the best results will be obtained from the engine by locating the point of cut-off early enough so there will be some expansion of the steam but not enough to result in excessive compression. For this reason the point of cut-off for a plain slide valve engine is usually made to occur between one-half and seven-eighths of the stroke. If it occurs earlier than mid-stroke the compression will be excessive, and if it occurs later than at seven-eighths of the stroke there will be but little expansion of the steam. Under the latter condition the pressure of the steam at release will be almost the original pressure during admission and when the exhaust valve is opened this pressure will be wasted.

Meyer Valve.—A study of the plain slide valve shows that if the point of cut-off could be changed without changing the other events, a much more economical steam distribution could be obtained by cutting off the steam earlier in the stroke and allowing it to expand a greater number of times. This is made plain by considering Fig. 111 in which the full line indicator diagram is from a plain slide valve engine and the dotted line shows the shape of diagram that would be obtained if the point of cut-off was made earlier without disturbing any of the other events. These diagrams show that more than half as much work is done with the early cut-off as with the late one, but that less

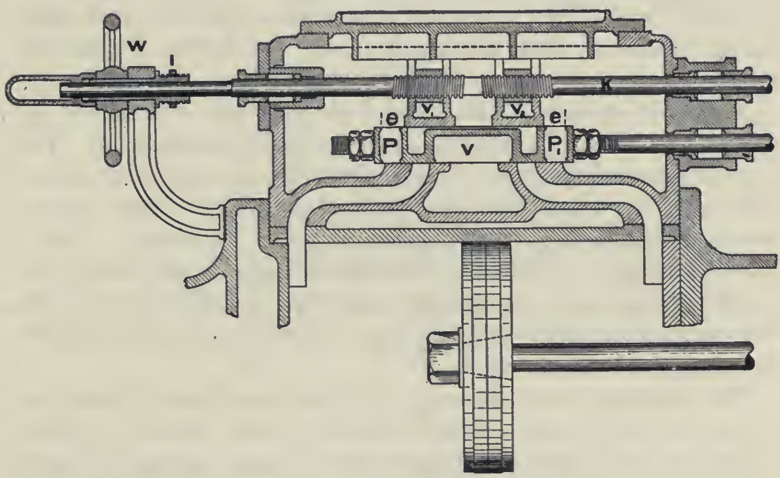


FIG. 112.

than half as much steam is admitted to the cylinder. Therefore, more work is obtained per pound of steam with the early cut-off than with the late one. This result is to be expected since very little of the expansion force of the steam is used when a late cut-off is employed.

An early cut-off by a slide valve may be obtained by means of a device called a Meyer valve. The Meyer valve consists of an auxiliary valve sliding on the back of the regular or main slide valve, as illustrated in Fig. 112. The main valve is like a plain slide valve except that instead of admitting steam past the outer edge, the valve is extended and has two ports, P and P_1 through it and the admission steam flows through these ports. The value is constructed in this way in order to provide a proper surface on which the auxiliary or riding valve may slide.

The auxiliary valve consists of two blocks, V_1 and V_2, carried on the valve rod K which is operated by a separate eccentric. The auxiliary valve slides on the back of the main valve and, at the proper instant, covers the ports, P and P_1 through the main valve thus stopping the flow of steam through them and cutting off the steam from the cylinder. The auxiliary valve will close the ports P and P_1 through the main valve and cause cut-off when it has moved with respect to the main valve a distance equal to e, in Fig. 112, which is called the *clearance*. The clearance may be adjusted by turning the handwheel, W, which turns the valve rod. The valve rod is provided with left- and right-hand threads so that turning it moves the blocks V_1 and V_2 farther apart or closer together and thus changes the clearance e.

The main valve is set the same as an ordinary slide valve but with a late cut-off in order to secure a short compression. The eccentric which operates the auxiliary valve is then set, usually at 180° from the crank, and the blocks placed on the valve stem in such position as to give equal cut-offs on the two ends. This is done by turning the blocks around separately on the valve stem until the clearance, e, is adjusted properly to equalize the cut-offs. The point of cut-off is then adjusted for both ends of the cylinder by turning the valve stem.

The auxiliary valve controls only the cut-off and it does this by closing the ports through the main valve before the main valve itself cuts off in the regular way. The main valve controls all the other events, release, compression, and admission. By setting the main valve to give a short compression and setting the Meyer valve to give an early cut-off, an indicator diagram may be obtained which resembles very closely the indicator diagram obtained from a Corliss engine, as shown by the dotted diagram in Fig. 111.

The action of the Meyer valve may best be examined by means of a valve diagram such as that shown in Fig. 113. In this diagram the cricle A is for the main valve and the circle B is for the auxiliary valve. The diameter of A is located so that the angle JOF is equal to the angle between the crank and main eccentric but laid out in a direction opposite to the rotation as indicated by the arrow. Likewise the diameter of the auxiliary valve circle, OG, is laid out so that the angle JOG is equal to the whole angle between the crank and auxiliary eccentric but in a direction opposite to the rotation. It will be observed that while the auxiliary eccentric is usually placed directly opposite the crank, in this case it is placed at a somewhat greater angle than 180° in order to make the diagram more general. Both the main and auxiliary eccentrics in this case have the same eccentricity as they usually have in practice but this is not at all necessary.

Since both valves are moved by eccentrics, their relative motion with respect to each other may be represented by a third valve circle C which is drawn with its diameter OK equal and parallel to GF. The position of the crank at cut-off for any amount of clearance may then be located on the circle C by

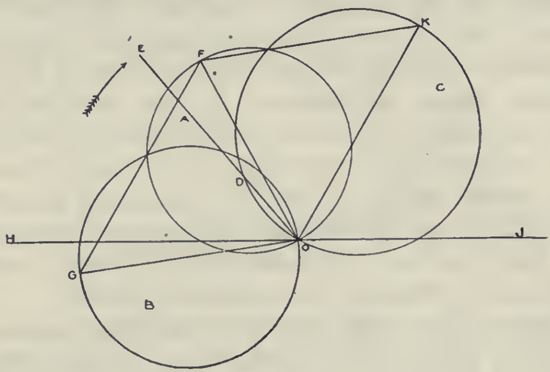


Fig. 113.

drawing a line OE through O so that the distance OD is equal to the clearance. Cut-off will then occur when the crank is in the position OE.

The circle C can be used only for finding the point of cut-off on one end of the cylinder. In order to find the cut-off on the other end the line KO must be extended and another circle of the same size as C drawn on it. The cut-off for the other end of the cylinder may then be located on it in the same manner as just described. Since admission, release, and compression are controlled entirely by the main valve they must be located by means of the main valve circle. Admission is located by drawing the lap circle in A as with other valve diagrams that have been described before. Compression and release must be located by extending OF and drawing on it another valve circle of the same size as A. By drawing the exhaust lap arc in this circle, both release and compression may be located the same as on other valve diagrams.

The clearance of the Meyer valve may be adjusted to cut off the supply of steam at any point between the beginning of the stroke and the point at which the main valve cuts off, the only limitations being that the valve must give sufficient port opening at the beginning of the stroke and must not re-open before the main valve closes, a matter which depends upon the design of the valve.

Engines which have Meyer valves are usually governed by means of a throttling governor which reduces the pressure in the steam chest. Attempts have been made to operate the auxiliary valve from a shifting eccentric connected to a flywheel governor as is done in the automatic high speed engine but this method is not successful with the Meyer valve because it gives a very unfavorable steam distribution. If the swinging eccentric lengthens the cut-off on one end of the cylinder it shortens it on the other because such a governing device moves the auxiliary valve as a whole, whereas, in order to change the cut-off equally on both ends of the cylinder, the blocks must be separated or brought closer together. This can only be done by turning the valve rod, and a swinging eccentric cannot do this.

The Meyer valve finds its most successful use on engines which carry a fairly uniform load such as on those running air compressors and similar machines. For this class of service the point of cut-off may be adjusted by hand to the most favorable part of the stroke, which will be as early as the load will permit, and then the small fluctuations in the load may be taken care of by the throttling governor, and a constant speed maintained. The point of cut-off may be changed while the engine is running, since the handwheel is outside the valve chest, hence, if the load changes enough to require a different point of cut-off, it may be changed to suit the new conditions. The handwheel is usually provided with a pointer and scale marked in the fractions of the stroke at which cut-off occurs, so that the clearance may be adjusted to any point of cut-off by simply turning the handwheel until the pointer is at the fraction at which it is desired that cut-off shall occur.

CHAPTER XIV

REVERSING MECHANISMS

Reversing Gears.—Many kinds of steam engines require a form of valve gear by means of which the direction of rotation may be readily reversed. Some of the kinds of engines which require a reversing valve gear are locomotives, marine engines, hoisting and winding engines, traction engines, and rolling mill engines.

A study of the valve diagram shows that any slide valve

Fig. 114.

engine may be reversed by simply shifting the eccentric around on the shaft. For an outside admission valve and no reversing rocker arm between the eccentric and the valve, the eccentric is set ahead of the crank by an angle somewhat greater than 90°. If such an engine is to run in a clockwise direction, the relative

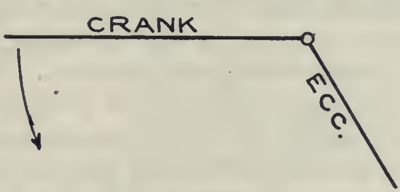


Fig. 115.

positions of the crank and eccentric are as shown in Fig. 114. If it is desired to have this engine run in a counterclockwise direction, the eccentric must be loosened and turned to the position shown in Fig. 115, when the eccentric will be ahead of the crank but in a counterclockwise direction. This method of reversing an engine could not be used with any engine which required a quick reversal because of the time required to change the position of the eccentric and also because this method makes it necessary first to stop the engine.

Many forms of valve gears have been invented which will quickly reverse the engine without having to change the position of the eccentric. Some of these mechanisms make use of two eccentrics, one for each direction of rotation; some of them use only one eccentric; and some derive their motion from a pin on the end of the shaft placed with its center a short distance from the center of the shaft so as to give the same motion as a short crank. This arrangement is equivalent to a single eccentric. Some of the more common forms of reversing mechanism are the Stephenson link motion, the Walschaert valve gear, and the Woolf reversing gear.

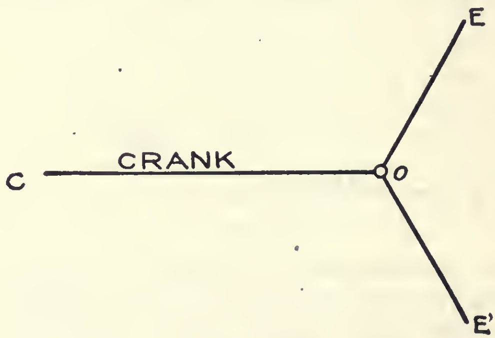


Fig. 116.

Stephenson Link Motion.—The Stephenson link motion is a reversing mechanism made up of two eccentrics, two eccentric rods connected by a link and a single valve stem. The two eccentrics are keyed or fastened to the shaft in the positions shown in Fig. 116. The eccentric OE is in the proper position for producing clockwise rotation and the eccentric OE' is in the proper position for producing rotation in a counterclockwise direction. The mechanism is arranged so that either of these eccentrics may be made to operate the valve, and rotation in either direction produced at will.

A Stephenson link reversing gear is illustrated in Fig. 117 showing the relations of the different parts. The two eccentrics are shown at A and B and the crank at C. It will be observed that the positions of the two eccentrics in relation to the crank

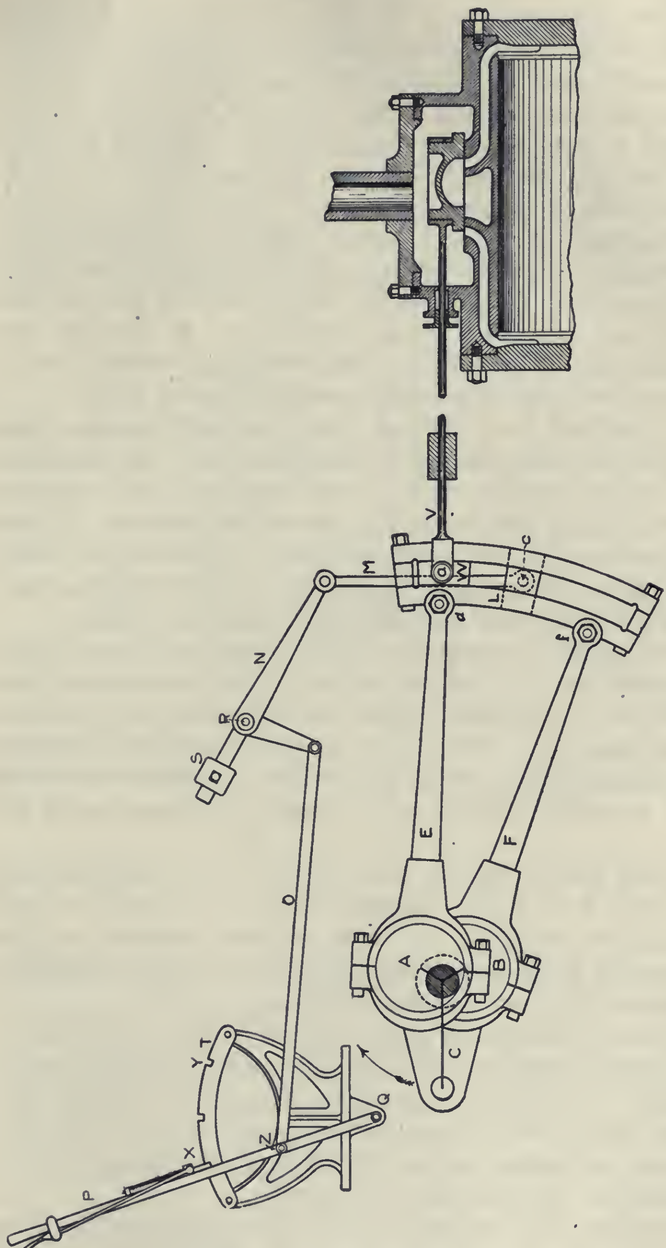


Fig. 117.

is the same as those shown in Fig. 116. The eccentric A is in the proper position for producing rotation in a clockwise direction, and B is in the proper position for producing rotation in a counterclockwise direction. Each eccentric has its own eccentric rod connected to one end of the slotted link L. The link L is suspended by M from the bell crank N which is pivoted at R. The bell crank is operated by the lever P so that by moving P the link L may be raised or lowered. The end of the valve stem V is bolted to a block W which fits in the slot of the link but is free to move in it, so that, as the link is raised or lowered, the block W and valve stem remain stationary. In this way the valve stem may be brought into line with either eccentric rod or it may be given any intermediate position between them.

When the block is at the end of the link, and the valve stem in line with the eccentric rod E, as shown in Fig. 117, the valve has the same motion as if connected directly to the eccentric A and the engine will rotate in a clockwise direction. In this case the eccentric B has no effect upon the motion of the valve. Its only effect is to swing the bottom of the link about the point a as a pivot in the same manner that a pendulum swings.

When the link is raised so that the valve rod is in line with the eccentric rod F, all of the motion of the valve comes from the eccentric B and the engine therefore runs in a counterclockwise direction. In this case the only effect of the eccentric A is to swing the upper end of the link like a pendulum about the block W as a pivot (the block now being at the lower end of the link).

When the link is raised so that the block is mid-way between the two ends of the link, the valve is acted upon equally by both eccentrics, one tending to produce clockwise rotation and the other tending to produce counterclockwise rotation; therefore, the engine will not run in either direction.

The valve mechanism is said to be in "mid-gear" when the block is in the middle of the link and to be in "full gear" when the block is at the end of the link. There are two "full gear" positions, one called "full gear foward" and the other "full gear backward," to indicate the position of the link which will cause the engine to move forward or backward.

The effects upon the valve motion of placing the block in different positions in the link and also the effects upon the steam distribution to the cylinder may be studied by means of a valve diagram. A valve diagram for a Stephenson link motion is shown in Fig. 118. In drawing this diagram it must be remembered that when the valve mechanism is in "full gear," the motion of the valve is the same as if it were connected to one of the eccentrics directly and the other eccentric were not present; therefore the line *OE* represents the eccentricity of one of the eccentrics, and the angle *COE* represents the angle of advance of this eccentric. The valve circle *OAE* is then drawn on the line *OE* and the steam lap circle *AHJB* is drawn about *O* as a center. The line *AF* then gives the position of the crank at

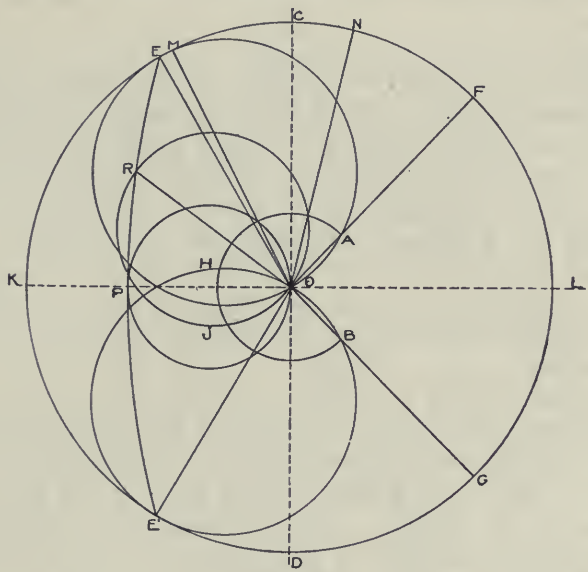


Fig. 118.

cut-off when the valve mechanism is in full gear and the engine is to rotate in a clockwise direction.

Both eccentrics of a Stephenson link motion have the same eccentricity and the same angle of advance, hence the location of the crank for full gear cut-off when the engine is rotating in a counterclockwise direction is found on the valve circle *OHE'B*. The diameter of this valve circle is the same as *OE* and the angle of advance *DOE'* is the same as the angle *COE*. Since the valve has the same steam lap for any position of the link, cut-off will occur when the crank is in the position *OG*.

When the block is at any position in the link intermediate between its two ends, the motion of the valve is derived from both eccentrics and this motion might be produced by a single imaginary eccentric having a smaller eccentricity and a greater angle of advance than either of the actual eccentrics.

On the valve diagram the center of the eccentric is at E for one full gear position and at E' for the other full gear position. As the link is shifted, the center of the imaginary equivalent eccentric moves along a curved path EPE' which is approximately the arc of a circle. The arc EPE' may be drawn as follows: Take a radius equal to

$$R = \frac{sl}{2k}$$

in which s is the distance from one eccentric center to the other (the distance from E to E' on the valve diagram), l is the length of the eccentric rods, and k is the length across the link measured from the center of the block in one full gear position to the center of the block in the other full gear position. With a center on the line KL and a radius as calculated above draw an arc passing through the points E and E', cutting KL at the point P. A valve circle drawn with OP as a diameter gives the location of the crank at cut-off OM for the mid-gear position of the valve mechanism.

For any position of the link intermediate between full gear and mid-gear the point of cut-off may be located by drawing a valve circle for that position of the link. This is done by dividing the arc EPE' in the same proportion that the link is divided by the position of the block. For a position of the block half way between the mid-gear and the full gear positions, the valve circle is drawn on OR as a diameter, R being half way between E and P, and it is seen that cut-off occurs when the crank is in the position ON.

It will be observed from Fig. 118 that the latest cut-off is obtained with the link in its full gear position and that as the link is brought towards the mid-gear position the cut-off becomes earlier, being at ON for one-quarter gear, and at OM for mid-gear. The Stephenson link motion may therefore be used to a certain extent as a governor since the point of cut-off may be regulated by it. On locomotives, where this form of valve gear is commonly used, the speed is regulated by both the throttle valve and the valve gear. In starting a locomotive pulling a load the link motion is put in full gear where the latest cut-off is obtained. The full steam pressure then acts upon the piston for nearly the entire stroke. As the locomotive comes up to speed, the link motion is gradually notched up towards mid-gear which shortens the cut-off and allows more of the expansive force of the steam to be used.

The valve diagram shown in Fig. 118 is not complete, only enough of it being shown to illustrate the method of drawing it. The remainder of the diagram is omitted in order not to complicate the figure. For the full gear position the line OE would be extended to the other side of the center E and another valve circle drawn on it with the exhaust lap to locate the positions of the crank at release and compression, in the same manner as for an ordinary slide valve diagram. These two valve circles will give all of the full gear events for one end of the cylinder. The events for the other end of the cylinder would be found by drawing an exhaust lap circle in the valve circle EO and a steam lap circle in the other valve circle. The same process would be followed in drawing a complete diagram for any other position of the link.

The link motion illustrated in Fig. 117 is known as the *open rod* construction and the valve diagram shown in Fig. 118 is for this form of link motion. There is another form of link motion called the *crossed rod* construction in which the eccentric rod E, Fig. 117, is attached to the end f of the link, and the eccentric rod F is attached to the end d of the link. With either the open or crossed rod construction the eccentric rods are alternately open and crossed every half revolution, but these two constructions may be distinguished by means of the following rule: For an outside admission valve and no reversing rocker arm the link motion is open rod construction if the rods are open when the crank is in the dead center position on the side of the shaft away from the valve. Under the same conditions the valve gear is of the crossed rod construction if the rods are crossed when the crank is in dead center position on the side away from the valve gear.

The motion of the valve for crossed rods is very different from its motion with open rods. For this reason, in dismounting a Stephenson link motion it is very necessary to know whether the link motion is open or crossed rod construction and, in reassembling, not to change from one construction to the other. A center-line diagram of a crossed rod link motion is illustrated in Fig. 119.

The valve diagram for the crossed rod construction is the same as that for the open rod construction except that the arc

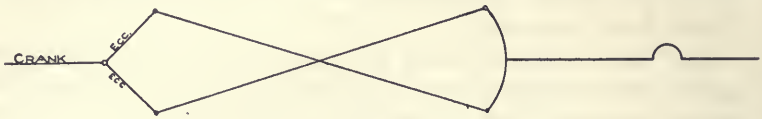


FIG. 119.

ERPE' Fig. 118 is drawn so as to curve in the opposite direction. This is done by placing the center of the arc on the opposite side of the center of the diagram. The method of finding the length of radius used for drawing the arc with both the open and crossed

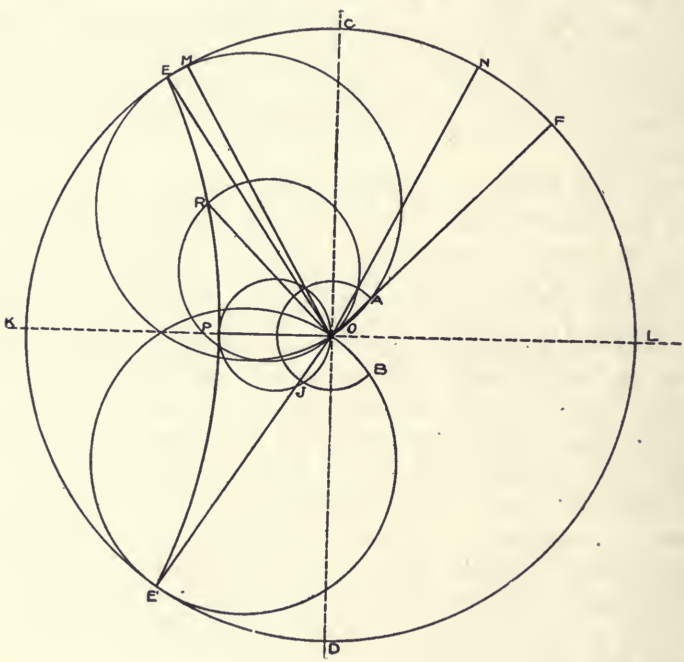


FIG. 120.

rod constructions is the same. The valve diagram for crossed rods is illustrated by Fig. 120.

It will be observed from Figs. 118 and 120 that with the open rod construction the lead increases as the link is moved from full gear to mid-gear while with the crossed rod construction the lead decreases as the link is moved from full gear to mid-gear. The fact that the lead changes for different positions of the link is sometimes urged as a disadvantage of the Stephenson link motion, and some of the later types of reversing mechanism are designed to give constant lead. For use on locomotives, if the lead must change, it is desirable to have it increase from full gear to mid-gear in order to give more cushioning effect as the engine speeds up.

In setting the valve of a Stephenson link motion the link is placed in full gear and the same method is then used as in setting the ordinary slide valve. The precaution must be taken however to see that the eccentric rods are of the same length; otherwise the steam distribution will not be the same for forward running as for backward running.

Walschaert Valve Gear.—While the Stephenson link motion is used on most American locomotives at the present time, it is

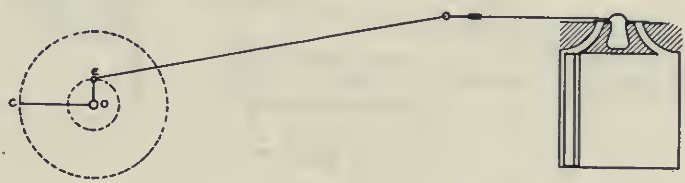


Fig. 121.

being displaced on the later types of locomotives by a form of reversing mechanism called the *Walschaert valve gear*.

The Stephenson link motion and other types of link motions have two eccentrics, one for forward motion and the other for backward motion. The Walschaert reversing mechanism, and others of the same type have but one eccentric or, in some cases, only a pin on the end of the crank shaft which acts as a crank of small throw. Reversing mechanisms having but one eccentric are called radial valve gears.

With the Walschaert valve gear, the valve takes part of its motion from the crank shaft and part from the crosshead, these two motions being combined into the actual motion of the valve. The part of the motion that comes from the crank shaft is produced by connecting the valve to a pin set at 90° to the crank. A valve connected directly to a pin placed at 90° to the main crank is shown in Fig. 121 in which OC is the crank and E is the pin which operates the valve. The small crank OE may be spoken of as an eccentric since it serves all the purposes of an eccentric.

An engine fitted with a mechanism like that shown in Fig. 121 will not run if the valve has steam lap, because, when the piston is at the end of its stroke the valve is in its mid-position and covers the steam ports a distance equal to the steam lap. In order to allow the engine to run the valve must be displaced a distance equal to the steam lap when the crank is in the position shown in Fig. 121. The valve will then be on the point of opening when the piston reaches the end of its stroke. For proper operation the valve should also be given some lead; that is, when the piston is at the end of its stroke, as shown in Fig. 121, the valve should be displaced from its mid-position a distance equal to the steam lap plus the lead.

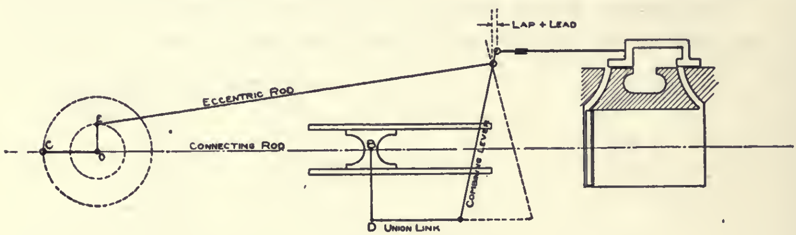


Fig. 122.

The method of securing lead with the Walschaert valve gear is shown in Fig. 122. The valve stem is not connected directly to the eccentric rod but to a combining lever which is connected to the crosshead. The eccentric rod is also attached to the combining lever, so that its swing is due both to the motion of the crosshead and to the eccentric crank OE. The crosshead carries a drop bar BD fastened rigidly to it and the combining lever is connected to it by a union link. The use of the union link is made necessary by the fact that the crosshead travels in a straight line while the lower end of the combining lever swings in the arc of a circle. The rise and fall of the lower end of the combining lever is therefore taken up by a slight swing of the union link about the point D, thus permitting the valve stem to travel in a straight line through its stuffing box.

The combining lever is supported at the point where the eccentric rod is attached to it and this point forms the fulcrum of the lever. The motion of the crosshead then displaces the valve independently of the motion produced by the eccentric crank. The combining lever is in a vertical position when the crosshead is at the middle point of its travel; therefore, when the piston is at either end of its travel the valve is displaced from its mid-position a distance depending upon the distance from the end of the combining lever to the point at which the eccentric rod is attached. This distance is made great enough to displace the valve a distance equal to the steam lap plus the lead, as shown in Fig. 122. When the crank is on the back dead center, as shown in Fig. 122, the combining lever displaces the valve to the right of its mid-position and when the crank is on the other dead

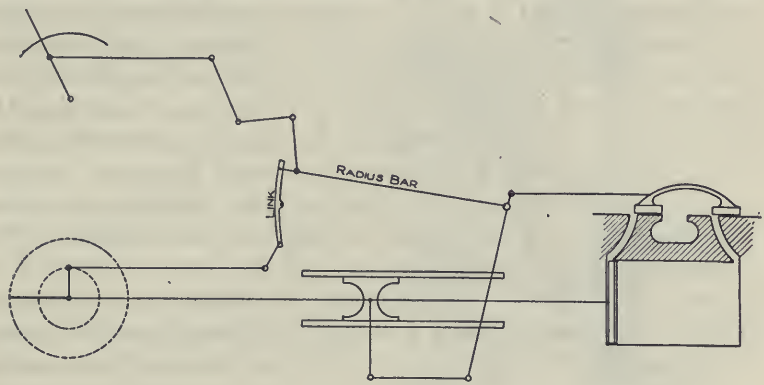


Fig. 123.

center, the valve is displaced to the left of its mid-position, thus giving lead at both ends of the piston stroke.

By erecting the mechanism so that the combining lever stands in a vertical position when the piston is at the middle point of its stroke, the displacement of the valve is made equal for each dead center position of the crank. If the valve has the same steam lap on both sides, the leads will then be equal. Moreover the lead will be constant since it depends only upon the proportions of the combining lever, which are fixed.

While the valve gear illustrated in Fig. 122 gives the valve the proper lead and will run the engine correctly, the direction of rotation of the engine cannot be reversed nor can the point of cut-off be changed. Both of these objects are accomplished in the Walschaert valve gear in the manner illustrated in Fig. 123. Instead of a single eccentric rod extending from the eccentric crank to the combining lever, the eccentric rod is divided into two parts and a link interposed between them. The link is supported at its center, about which it oscillates. The part of the eccentric rod which extends from the link to the combining lever (called a radius bar) has a block on its end which may be moved up or down in the link. Moving the block from one end to the other of the link reverses the movement of the valve and therefore reverses the direction of rotation of the engine. The point of cut-off is changed as the block is moved from the center of the link towards the end. The latest cut-off occurs when the block is at the end of the link for then the valve travel is greatest and the valve uncovers the steam ports the greatest amount. As the block is moved towards the center of the link, the valve uncovers the ports a smaller amount and the cut-off is shortened.

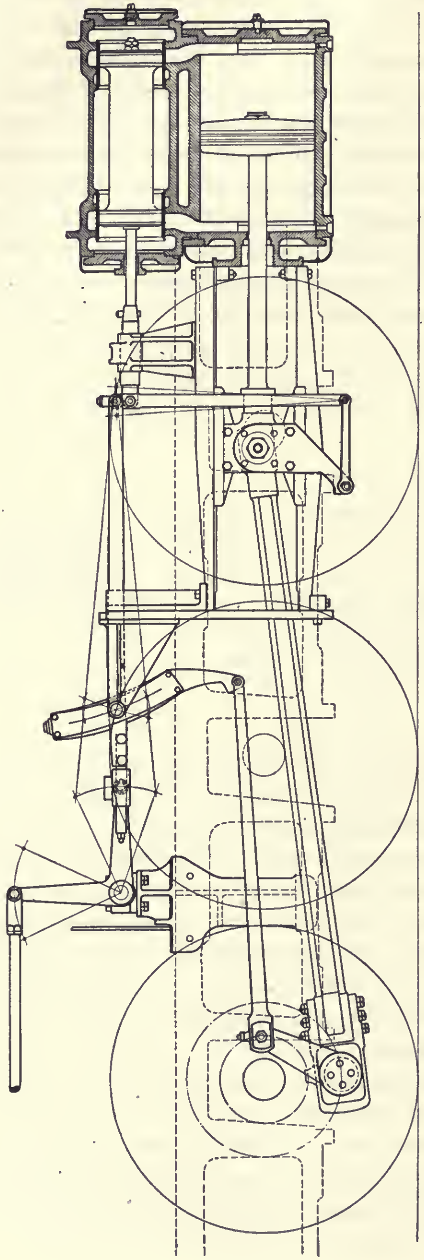


FIG. 124.

By curving the link to a radius equal to the length of the radius bar the lead is kept constant for all positions of the block in the link. This is one of the points of difference between the Stephenson link motion and the Walschaert valve gear. The former varies the lead for different points of cut-off and the latter keeps the lead constant. A valve gear may be tested for constant lead by shifting from one full gear position to the opposite and watching the valve or valve rod. With constant lead the valve will not move as the mechanism is shifted from one full gear position to the other.

An example of the Walschaert valve gear as applied to a locomotive is illustrated in Fig. 124. It will be observed that the eccentric rod is not connected directly to the lower end of the link but rather to an arm which projects from the lower end of the link and which inclines backward. This is made necessary by the fact that the eccentric rod is not horizontal but inclines at a considerable angle. This angularity of the eccentric rod distorts the motion of the valve. The projecting arm on the link is added to bring the eccentric rod as near the horizontal as possible; and the arm is shaped so as to correct the distortion in the valve motion produced by the angularity of the eccentric rod.

The Walschaert valve gear is not a new device since, it was invented in the year 1848 and has been in more or less extensive use on European locomotives since then. It is only within recent years, however, that it has been used to any extent in the United States. Its many advantages over the Stephenson link motion is now causing its rapid adoption on locomotives in this country.

The Walschaert gear is much lighter in weight than the Stephenson link motion. The Stephenson link motion has two heavy eccentrics with their straps and eccentric rods while the Walschaert gear is made up entirely of comparatively light rods. On some of the largest locomotives the Stephenson link motion weighs as much as two tons. Two reversals of this large weight in each revolution with a fast running locomotive throws a great strain upon the engine.

It is desirable that any valve gear keep its adjustment. In this respect the Walschaert is superior to the Stephenson gear. The joints in its moving parts are all pin joints with hardened bushings which reduce wear to a minimum and which may be easily renewed should wear occur. In the Stephenson link motion there is a certain amount of sliding between the block and the link called the "slip." The two eccentrics are also subject to considerable wear. Both of these parts are under a locomotive and exposed to the dust and grit from the road bed, which causes them to wear and cut very fast so that the Stephenson motion soon loses its adjustment.

The Stephenson link motion is usually placed between the drive wheels of a locomotive. It is therefore not so accessible as the Walschaert gear which is placed outside the drive wheels. The more accessible location of the valve gear insures its receiving better attention from the engineer, especially in the matter of lubrication. It might be thought that the more exposed location of the Walschaert gear renders it more liable to damage, but experience shows that it is no more liable to injury than the Stephenson link motion.

The amount of space occupied by the Stephenson link motion under a locomotive has, in some cases, interfered seriously with the proper bracing of the frame. The location of the

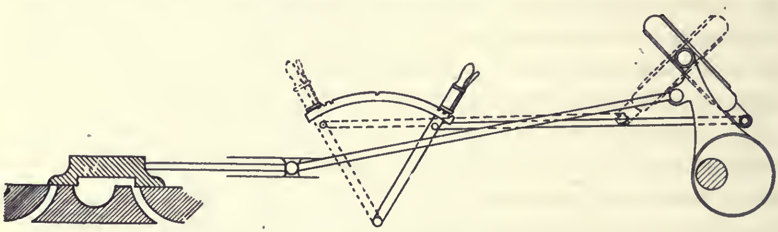


FIG. 125.

Walschaert gear outside the drive wheels leaves the space under the locomotive free for proper bracing, and it does not interfere with the design of any other part of the locomotive.

The varying lead of the Stephenson link motion is urged as an objection by some engineers who prefer the constant lead of the Walschaert gear.

Woolf Reversing Gear.—This is a simple form of single eccentric valve gear used on a great many traction and similar engines. This form of valve mechanism is illustrated in Fig. 125. The eccentric is set 180° from the crank. The eccentric strap carries a projecting arm which extends upward a little in front of and inclined to a vertical line through the center of the shaft. The outer end of this arm carries a block which slides in a straight but inclined guide. The eccentric rod is attached to the eccentric arm a short distance below the block, and extends in a direction almost at right angles to the eccentric arm. The other end of the eccentric rod is connected to a rocker arm to which the valve stem is attached.

The action of the valve gear may be seen more clearly from the center-line diagram shown in Fig. 126. The block Q on the upper end of the eccentric arm moves in a straight inclined line in the guides while the lower end moves in a circle. This makes the point P at which the eccentric rod is attached move in an ellipse. The engine is reversed by throwing the guide over to the dotted position. The point P then follows the path of the dotted ellipse. The point of cut-off may be varied by changing

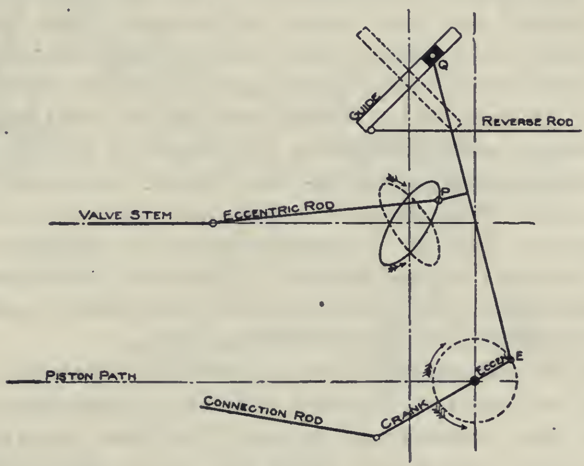


Fig. 126.

the inclination of the guide, and the reverse lever is supplied with a notched sector for this purpose.

The Woolf reversing gear gives a constant lead for all degrees of cut-off but the leads for the two ends of the cylinder are not necessarily equal The steam distribution is not the same for both directions but the mechanism is usually designed to give equal cut-off for both ends of the cylinder. Placing the center of the guides a little forward of the center of the shaft makes the steam distribution more nearly alike for both directions of rotation.

The simplicity of this reversing gear, together with the fact that it gives a fairly good steam distribution, has made it very popular for traction and thrashing engines where the engineer is usually not an expert.

CHAPTER XV

CORLISS VALVE GEARS

Advantages of the Corliss Valve.—The Corliss valve mechanism has been described in a previous chapter and the two admission and two exhaust valves shown. The use of four valves on a steam engine has three distinct advantages. They may be located close to the cylinder bore and long ports thus avoided. This reduces the clearance volume and the surfaces upon which steam may be condensed. When a single valve is used to control both admission and exhaust, it is first heated by the live steam passing it and then cooled by the exhaust steam passing it. This promotes condensation of steam upon the surfaces of the cooled valve. The use of separate valves for admission and exhaust reduces this condensation. The use of four valves has the further advantage that each one may be adjusted separately, which simplifies the operation of setting them.

Besides the advantages which arise from the separation of the admission and exhaust functions the Corliss mechanism gives an almost ideal motion to the valves. The valves move quickly while they are opening and closing thus giving sharp and well-defined events of admission, cut-off, release, and compression and also avoiding wire-drawing during admission. The valves have but little motion after they are closed, when motion is not needed, and the motion at this time is slow. This reduces the friction of the valve and makes good lubrication possible.

The slow motion of the valves after they are closed is due to the angle at which the valve rods are connected to the wrist plate. This angle is such that the valves are given a rapid motion when the wrist plate is near its central position and a slower motion as the wrist plate approaches either extreme position. Fig. 127 will make this clear. The heavy line CC' shows the position of the steam valve rod when the wrist plate is in its extreme position towards the left and the steam hook picks up the valve. As the wrist plate moves to the right through the angle from C to B the steam arm moves through the angle between C' and B'. The valve remains closed during this period. A further movement of the wrist plate through the angle between B and A moves the steam arm through the angle between B' and A' and the valve is opened during this period. It will be observed that although the wrist plate moves through a smaller angle while the valve is open than it does while the valve is closed that the steam arm (and also the valve) moves through a *larger* angle while the valve is open than while it is closed. Hence the valve

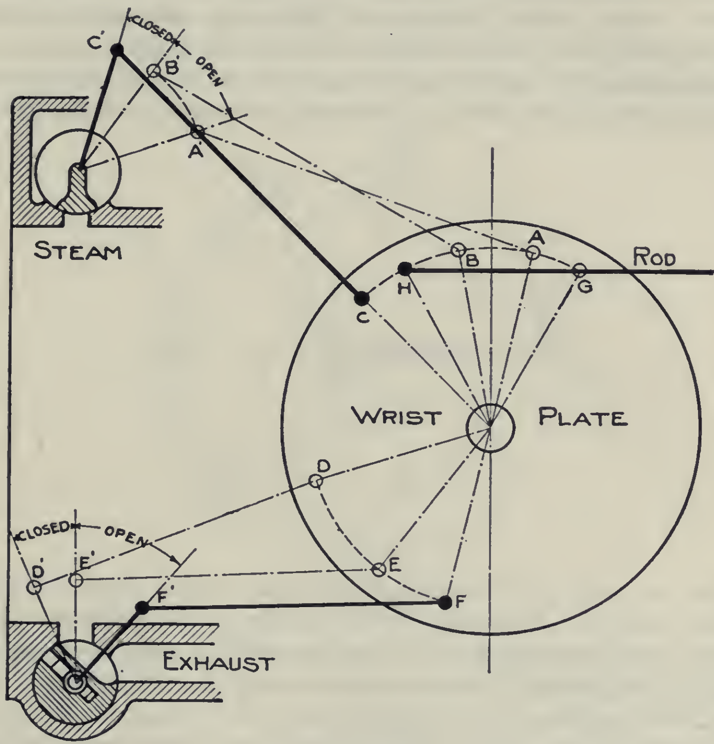


FIG. 127.

has a slow motion while it is closed and a fast motion while it is open. This effect is even more pronounced since the wrist plate moves slower when near its extreme position (when the valve is closed) than it does when near its central position.

The above remarks apply only to the opening of the steam valve, but the same effect is produced by the dashpot in closing the valve. As soon as the steam valve is released the dashpot moves rapidly and closes the valve. When the dashpot piston is near the end of its downward stroke, it moves more slowly in seating gradually and the valve, which is closed by this time, also moves slowly.

The exhaust valves are connected to the wrist plate at all times but they also have a slower motion when closed than when opening. It will be observed that the wrist plate moves through the small angle between F and E while the exhaust valve arm moves through the large angle between F' and E'. The exhaust valve is open during this period. While the wrist plate moves through the large angle between E and D the exhaust valve arm moves through the small angle between E' and D', the valve being closed during this period. Moreover, the motion of the wrist

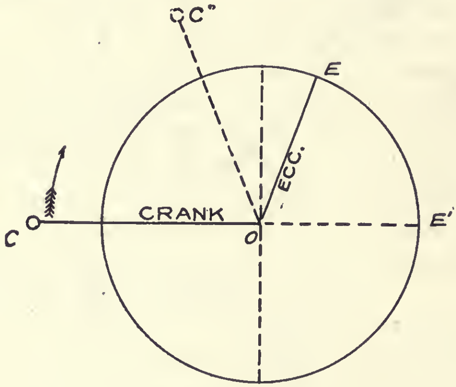


Fig. 128.

plate is slower while moving from E and D than while moving from F and E. Each valve rod is usually connected to the wrist plate in such position that a line through its axis, if extended, would pass through the center of the wrist plate when it is in its extreme position.

Single Eccentric Valve Gear.—The valve gear of some Corliss engines is operated by a single eccentric while the valve gear of others is operated by two eccentrics. With a single eccentric the cut-off may occur anywhere between the beginning and 35 per cent. to 40 per cent. of the stroke. With two eccentrics, one to operate the admission valves and the other to operate the exhaust valves, the range of cut-off may be extended considerably beyond mid-stroke.

With a single eccentric valve gear the admission valve must be unhooked for cut-off before the eccentric reaches the end of its throw. When the eccentric reaches the end of its throw it begins to move in the opposite direction. If the steam valve, which is hooked up at the beginning of the piston stroke, has not been released by this time it will not be released at all but will be closed by the gradual motion of the eccentric on its return stroke. Since the eccentric is set a little more than 90° ahead of the crank it will reach the end of its stroke before the crank reaches a vertical position, representing mid-stroke of the piston. This will be made clear by referring to Fig. 128.

The admission valve is hooked up when the eccentric is in the position OC, at the beginning of its stroke. When the crank reaches its dead center position the eccentric is in the position OE

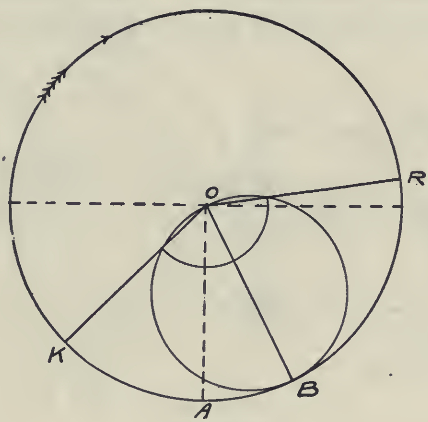


Fig. 129.

and the valve has been opened an amount equal to the lead. The valve may be tripped by the knock-off cam at any time while the eccentric is moving through the angle between E and E'. During this time the crank is moving from the position OC to the position OC'. This is somewhat less than half the piston stroke. If the steam valve is not tripped by the time the eccentric reaches the position OE' (and the crank reaches the position OC') it will not be tripped at all, but will remain hooked up and it will then be closed by gradual motion of the eccentric on its return stroke.

By referring to Fig. 128 it will be seen that if the angle of advance is increased the range of cut-off will be decreased, hence it might be thought that a greater range of cut-off could be secured by decreasing the angle of advance. The proper working of the exhaust valves which are connected with the eccentric at all times, however, sets a limit to the decrease in the angle of advance.

The exhaust valves, being connected with the eccentric at all times, have the same motion as a plain slide valve and this motion may be shown by a Zeuner valve diagram. Fig. 129 shows a Zeuner valve diagram for the head end exhaust valve of a Corliss engine, the eccentric being set with the angle of advance AOB. Release occurs when the crank is in the position OR, the piston being near the end of its stroke from left to right. Compression occurs when the crank is in the position OK, the piston being near the end of its stroke from right to left. The Zeuner valve diagram does not show the position of the cut-off for a Corliss valve gear

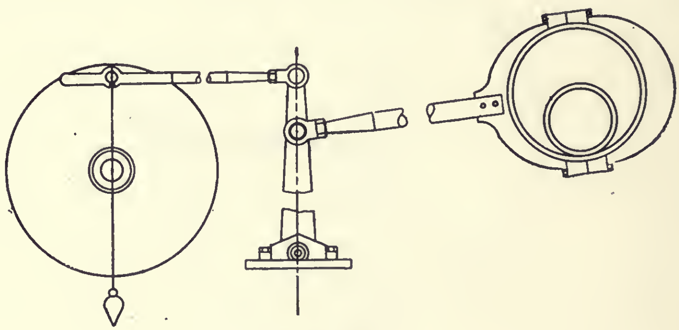


Fig. 130.

hence it is of but little use in setting these valves. Corliss valves are usually set by measurement and the setting then checked from an indicator diagram.

Setting Corliss Valves.—When the various parts of a Corliss valve gear are in proper adjustment the reach rod and the eccentric rod should be of such length that both the rocker arm and the wrist plate will be plumb when the eccentric is vertical, as shown in Fig. 130.

Since it is difficult to judge by the eye when an eccentric is vertical the following method should be used for finding exactly its vertical position. A tram shaped as shown at A in Fig. 131 is made of sheet iron and a hole bored in it large enough to receive a scratch awl or pointed nail. With the crotch of the tram placed against the shaft and with a scratch awl in the hole in the tram the arcs *BC* and *EF* are drawn on the eccentric ending at the points *B* and *E* at the edge of the eccentric. It is convenient to rub chalk on the eccentric and the arcs drawn in the chalk in order to make them more easily seen. With a pair of dividers draw arcs from *B* and *E* which meet exactly on the edge of the eccentric as at *G*. By a similar method the point *I* exactly opposite *G* is located. The point *G* is furthest away from and the point *I* is nearest the center of the shaft, therefore the line *IG* represents the position of the eccentric.

The joint in the eccentric strap is usually at right angles to the eccentric rod, hence the eccentric may be placed in a vertical position by turning it until the point *G* comes to the

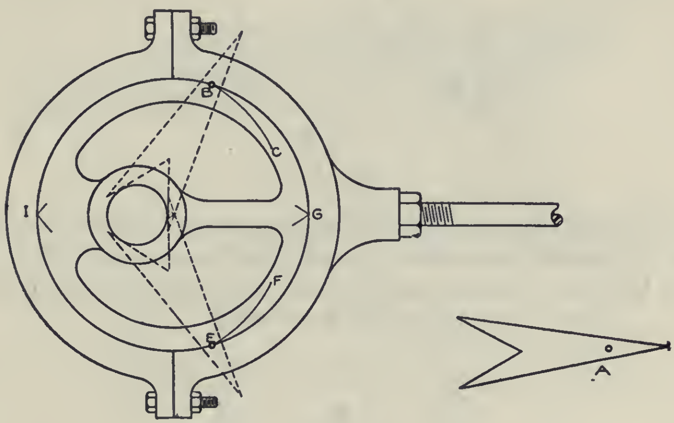


Fig. 131.

joint. In case the joint in the eccentric strap is not at right angles to the eccentric rod, the following method of placing the eccentric in a vertical position may be used.

Another tram, shaped as shown at *H*, Fig. 132, is made of steel wire. A punch mark *J* is made on the side of the eccentric rod and on its center line. With one end of the tram in the mark *J* arcs are drawn on the eccentric strap ending at the points *K* and *L* on the edge of the strap. With a pair of dividers draw arcs *M* from *K* and *L* which meet exactly on the edge of the strap. When the point *M* coincides with either the points *G* or *I* the eccentric is exactly on center.

With the dividers set to the length *KM*, strike arcs *P* and *R* from *I* which end at the edge of the strap. Draw arcs *S* from *P* and *K* which meet at the edge of the strap and arcs *T* from *R* and *L* which meet at the edge of the strap. When the point *G* coincides with either the points *S* or *T* the eccentric is exactly vertical.

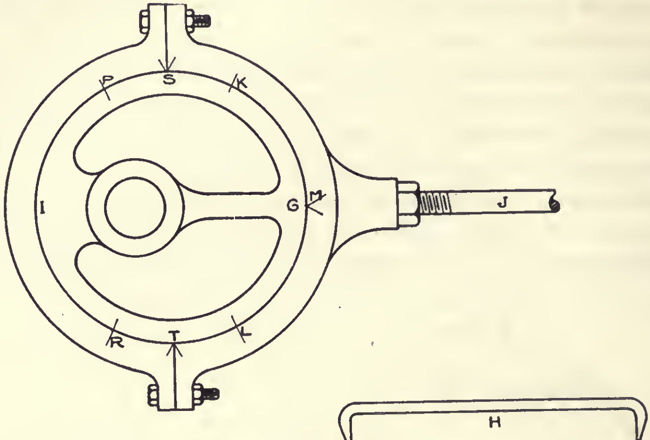


Fig. 132.

The manufacturers of Corliss engines usually place a mark *A*, Fig. 133, on the hub of the wrist plate and three marks *D, B,* and *C* on the wrist plate stud. When *A* coincides with *B* the wrist

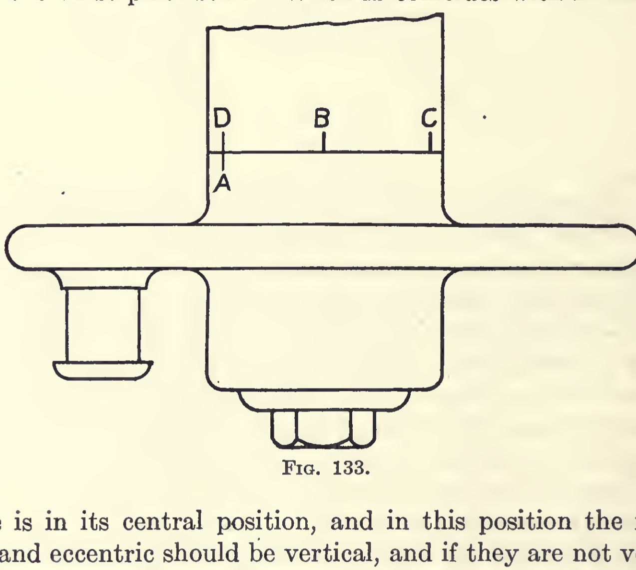


Fig. 133.

plate is in its central position, and in this position the rocker arm and eccentric should be vertical, and if they are not vertical the length of the eccentric and reach rods should be adjusted until they are vertical. When A coincides with D the wrist plate is at one extreme of its travel and the eccentric is on dead center; and when A coincides with C the wrist plate is at the other extreme of its travel and the eccentric is on the other dead center. If the marks A, B, C, and D are not on the wrist plate and stud they should be placed on with a chisel, A and B being marked when the eccentric is vertical and the rocker arm and wrist plate are plumb, and D and C being marked opposite A when the eccentric is on its dead centers.

The reach rod is now disconnected from the wrist plate and the wrist plate placed on its central position (so that A coincides with B, Fig. 133). With the wrist plate in its central position and both

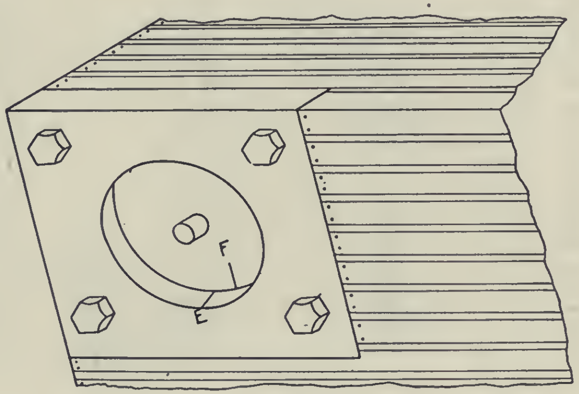


Fig. 134.

steam valves hooked on, the valves should have the proper lap. The amount of the lap may be measured by removing the steam bonnets on the side of the cylinder opposite the wrist plate and inspecting the marks on the valve and cylinder. These marks, as shown in Fig. 134, are placed on by the manufacturer of the engine, the mark F on the valve being opposite its working edge, and E, on the cylinder being opposite the edge of the port. When F coincides with E the edge of the valve is exactly in line with the edge of the port. By removing the valve it can be determined which edge of the port the mark E on the cylinder is opposite and therefore it will be known on which side of E the mark F should be for the valve to have lap. The lap is measured with a pair of dividers, being the distance between E and F,

Fig. 134. With the wrist plate in its central position the laps on the two ends of the cylinder should be equal. The laps may be adjusted by lengthening or shortening the radial rods, which are provided with left- and right-hand thread connections for this purpose. The proper amount of lap to give the steam valves depends upon the size of cylinder and may be determined from the following table:

Table for Setting Corliss Valves

Diam. of cylinder, inches	Lap of steam valves, inches	Lap of exhaust valves, inches	Lead of steam valves
8	3/16	1/16	1/32
10	3/16	1/16	1/32
12	3/16	1/16	1/32
14	1/4	1/8	1/32
16	1/4	1/8	1/32
18	1/4	1/8	1/32
20	1/4	1/8	1/32
22	5/16	3/16	3/64
24	5/16	3/16	3/64
26	5/16	3/16	3/64
28	5/16	3/16	3/64
30	5/16	3/16	3/64
32	3/8	1/4	1/16
34	3/8	1/4	1/16
36	3/8	1/4	1/16

The exhaust valves are now given equal laps by adjusting the length of their radial rods in the same manner as was done with the steam valves. Marks are placed by the manufacturer on the cylinder and on exhaust valves, similar to those on the steam valves, and the exhaust laps may be set by measurement between these marks. The amount of the exhaust lap is determined from the above table.

The wrist plate is now turned to its head end extreme position and the length of the dashpot rod adjusted so there will be equal clearance around the catch block as shown in Fig. 135, at *G* and *H*. It is very important to adjust the length of the dashpot rods properly because if they are made too short the valves will not hook on and if they are too long the valve stem is liable to be bent or the steam bracket broken, or both. Now turn the wrist plate to its crank end extreme position and adjust the length of the crank end dashpot rod as was done for the head end dashpot rod.

The steam valves may now be given their proper lead, as indicated by the above table. To do this the engine is placed on its head end dead center, using a tram to locate the dead center exactly. The eccentric is then loosened on the shaft and the reach rod hooked to the wrist plate. The wrist plate is then moved over to its head end extreme position in order to hook up the head end steam valve. The eccentric is then turned around on the shaft until the port is open the amount of the

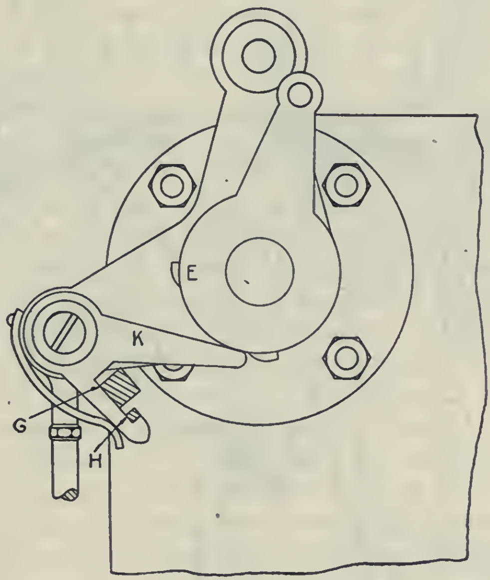


FIG. 135.

desired lead. The eccentric is then fastened in this position. The port opening may be measured by the lines on the valve and back side of the cylinder. The engine should now be turned to the crank end dead center, the crank end steam valves hooked up, and the lead measured to see if it is equal to that on the head end. If it is not, any slight adjustment that may be required can be made by moving the eccentric.

The governor and governor rods should next be adjusted if they require it. Fig. 136 shows a common form of Corliss engine governor with its connections to the valves, parts of the governor being cut away to show its construction. As shown here the parts of the governor are in the position they will occupy when

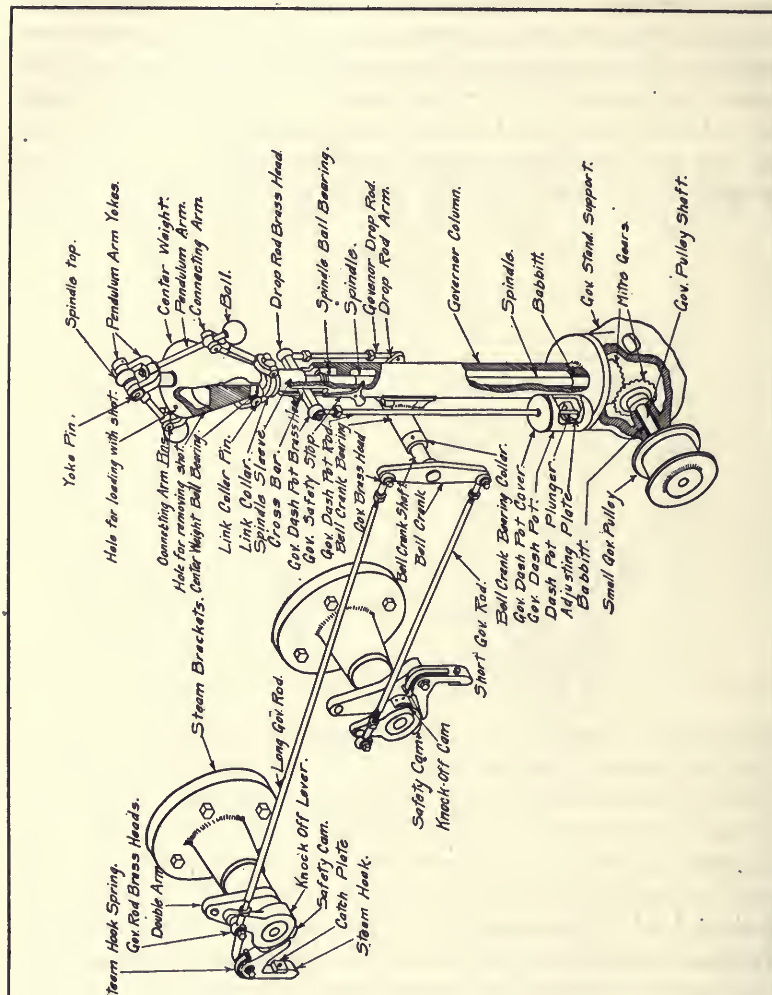


Fig. 136.

the engine is running at its normal speed. If the speed rises above normal, centrifugal force throws the governor balls farther from the center. This raises the cross bar and with it the drop rod arm and throws the knock-off levers (on the valve stems) around so that the knock-off cams strike the steam hook earlier and thus cause an earlier cut-off. If the speed falls below normal the knock-off cams are moved in the opposite direction and cut-off occurs later.

If the belt which runs the governor should break, the cross bar would drop to its lowest position and this would make cut-off come at the latest possible point in the stroke or the steam hooks would not be disengaged at all and the cylinder would take steam for the entire stroke. This would cause the engine to run away.

In order to prevent the engine from running away if the governor belt breaks or any other accident happens which would throw the governor to its lowest position, safety cams are placed on the knock-off levers. When the governor falls to its lowest position the knock-off levers are thrown around far enough to bring the safety cams under the steam hooks, thus preventing the admission of any steam to the cylinder. If the safety cams are allowed to come into action as described above the engine could not be started, after it was shut down, until the governor was raised far enough to prevent the safety cams from coming in contact with the steam hooks. It would be a nuisance to have to raise the governor every time the engine is started, hence a governor safety stop is placed on the governor to prevent it from falling to its lowest position and bringing the safety cams into action when the engine is shut down by hand.

When the engine is to be shut down by hand the governor safety stop is raised to the position shown in Fig. 136. The cross bar will then rest on the safety stop, which is high enough above its lowest position to prevent the safety cams from coming into operation. In this position of the cross bar the valves will hook up and open when steam is turned on again. As soon as the engine is started again the safety stop falls to one side. If, then, the governor belt should break the cross bar would fall to its lowest position and bring the safety cams into operation.

When the engine is running at normal speed the cross bar on the governor is about halfway between the upper limit of its travel and the end of the safety stop (in its raised position), hence in adjusting the governor it should be blocked up to this position. The length of the governor drop rod is then adjusted until the drop rod arm is horizontal and the bell crank stands vertically. The governor is then unblocked, the engine started slowly, and the length of the governor rods adjusted so that cut-off is equal on both ends of the cylinder. The governor rods are provided with right- and left-hand screws so their length may be changed without stopping the engine. An indicator diagram should be used to determine when the cut-off is equal on the two ends, as well as for all other valve adjustments.

CHAPTER XVI

GOVERNING

Governing.—The work that most steam engines do requires a constant or practically constant speed of rotation. This requirement is more difficult to meet than might at first appear, and much thought has been expended on this problem in order to solve it satisfactorily.

As mentioned in a previous chapter changes of speed occur in two entirely different ways. First, the speed may change during a single revolution or cycle of the engine due to a variation in pressure against the piston and to the angle at which the force of the steam pressure is transmitted to the crank. Second, the speed may change due to a change of load or to varying boiler pressure, such a change extending over a period of more than one revolution. The first of these kinds of speed variation is taken care of by the flywheel which stores up energy during one part of a revolution and gives it out again during another part, as explained in a previous chapter. The second kind of speed variation must, however, be corrected by some kind of controlling device, or governor.

If the load on an engine is increased or if the boiler pressure of the steam becomes less the speed of the engine will decrease. On the other hand, if the load decreases or the boiler pressure increases the speed of the engine will increase. The governor is for the purpose of regulating the supply of steam to the engine so that its speed will remain constant or practically constant. In order to do this a steam engine governor either operates on a throttle valve placed between the engine and boiler to change the pressure of the steam which is being admitted to the engine; or it alters the volume of steam admitted to the engine by changing the point of cut-off.

Whatever method of controlling the speed is used, no governor can control the speed perfectly because the governor is run by the engine itself and some change in speed must occur before the governor can operate. That is, the governor operates on account of a change of speed, hence the governor cannot keep the speed of the engine absolutely constant. Also, any change of speed which occurs after the steam has passed beyond the influence of the governor cannot be controlled until the next stroke of the piston. For example, if the load changes after cut-off has occurred this may affect the speed of the engine and the governor cannot have any effect because it can only operate on the next admission of steam or during the next stroke of the piston. This is especially true of compound engines, where the governor controls the supply of steam to the high pressure cylinder only. If a change of load occurs after cut-off in the high pressure cylinder, the steam in this cylinder expands and does work in the cylinder and then passes into the low pressure cylinder and again expands and does work, all outside of the control of the governor, which can only act upon the next admission to the high pressure cylinder. But for all of the difficulties in the way of securing close speed regulation, a good steam engine governor will control the speed within 2 per cent. of its normal speed.

Pendulum Governor.—Nearly all steam engine governors operate through centrifugal force. They usually consist of a pair of weights revolving about a spindle which is driven from the engine shaft. The centrifugal force of the revolving weights is resisted by some controlling force, such as gravity, the tension of a spring, or both. When the engine (and governor) is running at constant speed the weights take up a fixed position at which the controlling force just balances the centrifugal force. When an increase of speed occurs the additional centrifugal force causes the weights to move outward to a new position, and in moving outward they act upon the throttle valve or some form of automatic gear by which the cut-off is varied so that the speed is reduced.

The most common forms of centrifugal governors are those known as pendulum governors. In these governors the spindle is vertical and there are two weights, each of which is placed at the end of an arm. The two arms are suspended from the top of the spindle and pivoted at or near it. When the spindle is rotated the weights move outward and upward and their upward motion is resisted by the force of gravity. When the engine is running at constant speed the weights take up a position in which the force of gravity, or other controlling force, just balances the centrifugal force. If the speed of the engine is increased or decreased the weights take up a new position in which the controlling force balances the centrifugal force developed by the revolving weights.

A form of governor commonly used with plain slide valve engines and operating on the above principles is illustrated in

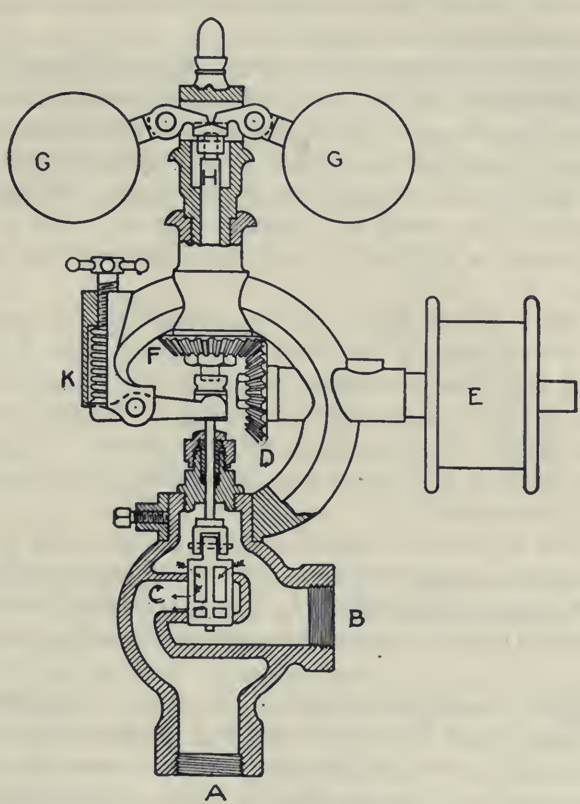


Fig. 137.

Fig. 137. This is a throttling governor since it acts upon a throttle valve and thus controls the pressure of the steam admitted to the cylinder. In the governor illustrated here the opening A is connected to the steam chest and the opening B connected to the steam pipe leading from the boiler, so that the steam passes through the valve C before entering the cylinder. The governor is run by a belt from the engine shaft to the pulley

E. This transmits motion through the gear wheels *D* and *F* to the vertical spindle to which the weights *G* and *G* are attached. As the speed of the engine increases the weights move outward and upward and press downward upon the valve stem *H* thus partly closing the valve *C*. In the same way, a decrease in the speed of the engine causes the weights to assume a lower position and the valve stem *H* rises and admits more steam to the cylinder, thus causing an increase of speed.

The upward movement of the weights is resisted partly by the force of gravity and partly by the tension of the spring *K*, so that for any particular speed of the engine the weights will take a position at which their centrifugal force is just balanced by the force of gravity and the tension of the spring. If the engine departs from this speed the weights will rise or fall until the steam pressure is adjusted to suit the load which the engine is carrying.

The speed of the engine can be changed by changing the tension of the spring *K* by means of adjusting screw *T*. If the tension of the spring is increased, the weights will have to revolve faster in order to secure a given opening of the throttle valve, and thus the speed of the engine will be increased. In the same way, the speed of the engine may be reduced by decreasing the tension of the spring.

The form of governor described above was invented by James Watt, one of the early inventors of the steam engine, and for this reason it is sometimes called the Watt governor. It is one of the simplest forms of steam engine governors used at the present time.

Stability.—A governor is said to be stable when there is a definite position of the weights for any definite speed; that is, if the speed of the engine changes by any amount the weights move up or down to a new position which corresponds to that particular speed, and then remain in this new position until there is another change of speed. From the preceding description of the Watt governor it will be seen that the new position of the weights gives a larger or smaller opening of the throttle valve and this serves to bring the speed back to normal. The speed is thus automatically maintained at or near the number of r.p.m. for which the governor is set.

If a governor was unstable it would have no definite position for a given speed and its movements would be irregular and uncertain. For this reason it could not maintain a constant speed of the engine and would therefore not be suitable for governing a steam engine from which a constant speed was desired.

It is evident from the above discussion that stability is a desirable and even necessary quality of a steam engine governor because a stable governor is always in equilibrium and exercises a positive control over the speed.

In order for a governor to be stable the controlling force, or the force acting against the rise of the weights, must increase at a faster rate than the radius of the circle about which the weights are revolving, and the larger this ratio between the controlling force and the radius of the circle about which the weights are revolving, the greater will be the stability of the governor.

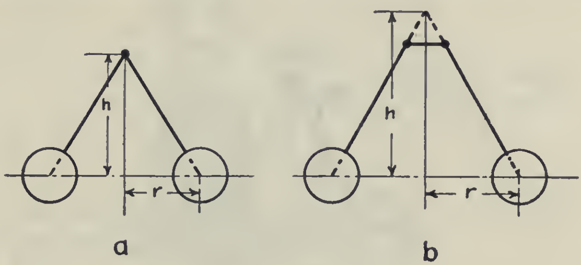


Fig. 138.

A change in the speed of a governor causes a change in the position of the weights, and if the governor is stable there is only one position of the weights which correspond with a given speed. Since the steam supply depends upon the position of the weights, a stable governor cannot maintain a strictly constant speed, because if the boiler pressure or load changes a certain displacement of the weights s necessary to admit more or less steam and the weights can maintain this new position only by turning faster or slower. However, the variaton from a constant speed can be reduced by reducing the stability of the governor.

The ordinary forms of pendulum governors such as illustrated in Fig. 138 are stable and also the crossed arm form of pendulum governor illustrated in Fig. 139 provided the points of support are located close to the central column.

Sensibility.—The movement of the weights of a governor from their lowest to their highest position can produce only a certain movement of the regulating mechanism, whether it is a throttle valve or a cut-off attachment. This will produce the greatest change of speed for which the governor is responsible. If an engine is overloaded or if the steam pressure is too low, the speed may drop even after the governor has done all that it can do to admit steam freely, but the variation in speed for which the governor is responsible is only that which will cause the weights to move from the position of no steam to the position of full steam. When a small variation of speed is sufficient to do this the governor is said to be sensitive.

It is evident from the above discussion that the more sensitive a governor is the less stable it must be. As both of these features can be controlled in the design of the governing mechanism, the

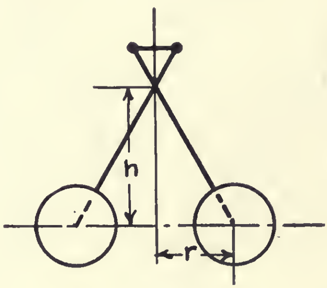


Fig. 139.

designer aims at securing a governor which is stable and which at the same time is sensitive enough to control the speed within the limits needed for the kind of work the engine is to do.

It is absolutely necessary that a steam engine governor be stable and it is highly desirable that it be sensitive. However, it should not be too sensitive as this causes the engine to *hunt* or over-govern. Hunting is brought about by the conditions mentioned below.

When an alteration of speed begins, the governor does not act immediately because the governor can only operate after a change of speed has occurred. Moreover, a change in position of the governor does not affect the speed of the engine immediately both on account of the inertia of the moving parts of the engine which has the effect of resisting a change of speed, and also because of the energy contained in the steam which has passed the control of the governor. If the governor is of the throttling type, the steam chest is filled with steam which has passed the control of the governor at the time the change of speed begins, and if the governor acts upon the cut-off its opportunity for controlling the speed has passed if cut-off has occurred. Hence, there is a certain time lag between the governor and the engine speed. The consequence of this is that, if the governor is too sensitive, by the time the change in engine speed has had

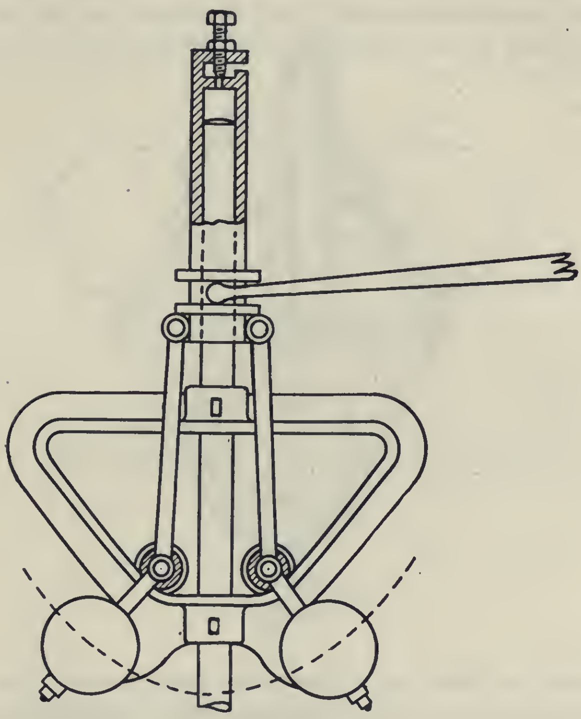


Fig. 140.

full effect upon the governor, it is forced into a position of over-control or a position beyond that which is necessary to bring the engine speed back to normal. The speed of the engine then begins to change in the opposite direction and, for the same reasons, the governor is forced into a position of over-control in the opposite direction. Thus a state of forced oscillation is set up which causes the speed to be first too high and then too low, a condition known as *hunting*.

Hunting is avoided by allowing a certain margin of stability in the governor, that is, by not making it too sensitive, and also by the use of dashpots attached to the governor in such way as to dampen its motion in case it is too sensitive.

Some governors, on account of their form, are much more sensitive than others. It has been found that if the form of governor is such that the weights, in rising, follow a parabola instead of a circle the governor will be extremely sensitive. A governor constructed in this way, illustrated in Fig. 140, is so sensitive that an air cylinder and piston is placed at the top of

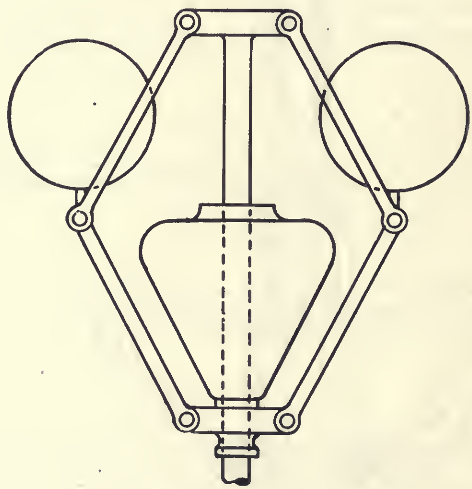


Fig. 141.

the column to check the movement of the weights. Other forms of sensitive governors are the Pröll governor illustrated in Fig. 141 and the Hartnell governor illustrated in Fig. 142. In the Hartnell governor, the weights move in a practically horizontal path and the controlling force is furnished by a coil spring shown at the top of the central column, and the sensitiveness of the governor can be adjusted by means of this spring.. This form of governor is more suitable for high speed engines than for low speed ones on account of the small size of the weights.

The Pröll governor illustrated in Fig. 141 is better adapted to slow speed engines, such as the Corliss engine. This type of governor is known as a *loaded* governor on account of the heavy weight placed around the central column. This weight revolves with the governor and has the effect of increasing the controlling force without adding to the centrifugal force, as would be the case if the additional weight was placed at the ends of the rotating arms.

The advantages of a loaded governor are that it is more powerful than an unloaded one, that the increase in power is gained without a corresponding loss of sensitiveness, and that it

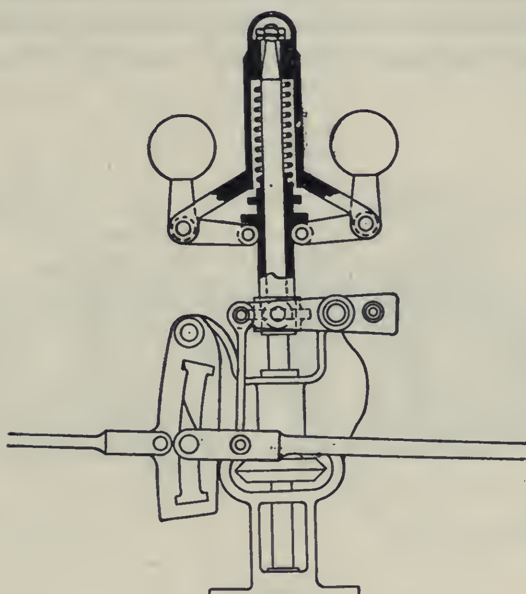


FIG. 142.

may be run at a lower speed than an unloaded governor having balls of equal size.

A powerful governor is necessary in order to overcome the friction of the moving parts of the governor and controlling mechanism. Friction in a governor and its connected mechanism has the effect of increasing the controlling force and thus reducing the sensitiveness of the governor. If the controlling force of a governor, neglecting friction, is represented by F and the force necessary to overcome friction is represented by f then for an increase of speed the centrifugal force acting on the weights must be increased to $F + f$ in order to change the position of the governor and also the centrifugal force must be decreased to $F - f$ in order to change the position of the governor for a decreasing speed.

A loaded governor of the type shown in Fig. 141 or Fig. 136, that is, a governor having four arms, has the further advantage that the vertical movement of the collar, or central column, is twice as great as the movement of the weights at the ends of the rotating arms. That is, for a given change of speed a governor of the types shown in Figs. 136 and 141 produces twice as much motion in the controlling mechanism as in the plain pendulum governor such as illustrated in Fig. 138*a*. The result of

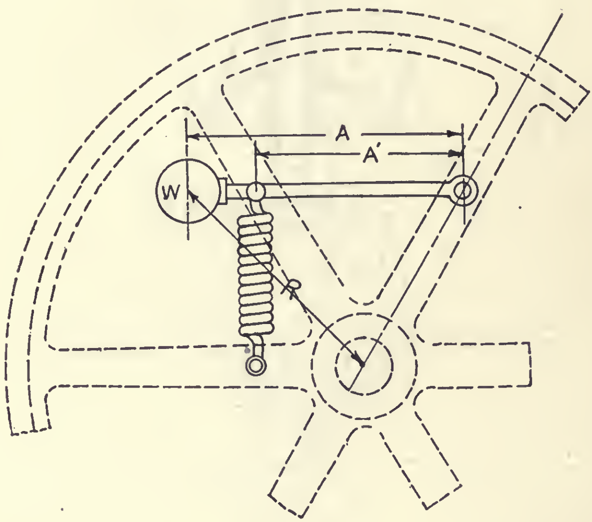
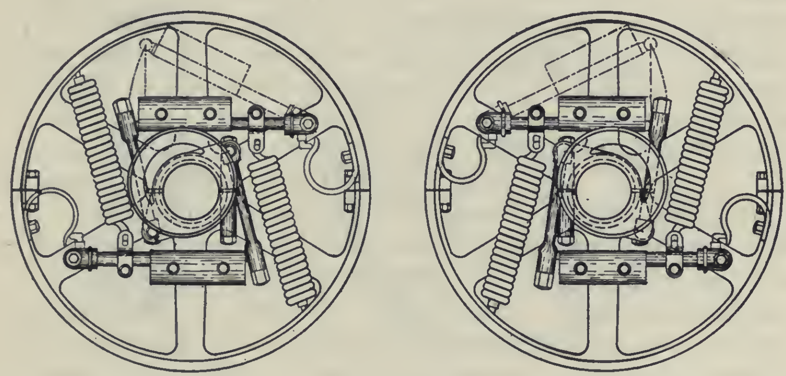


Fig. 143.

this is that these governors may be run at a less speed than the Watt governor, hence their general use on low speed engines of the Corliss type.

Shaft Governors.—Governors which are located in the flywheel and which turn with the flywheel are commonly called shaft governors. These are usually attached to the spokes of the flywheel and operate by shifting the position of the eccentric and thus changing the point of cut-off so the amount of steam admitted to the cylinder is in proportion to the load which the engine carries.

One example of shaft governor operating on this principle is described in Chapter 13 and illustrated in Fig. 108. The principles upon which a governor of this type operates are the same as those upon which the centrifugal pendulum governor operates. The similarity of operation of these two kinds of governors may be seen by a study of Fig. 143. An arm is pivoted at some point on the flywheel and to the end of the arm is attached a weight W, the length of the arm being A. At a distance A' from the pivot a spring is attached to the arm and is arranged so as to act at practically right angles to the arm. The centrifugal force of the weight will be balanced by the pull of the spring in the same manner as gravity balances the centrifugal force in a pendulum governor.



A *B*

Fig. 144.

With a shaft governor, the speed at which the engine will run increases as the tension in the spring is increased, or as the distance between the pivot and the point of attachment of the spring is increased, or as the weight at the end of the arm is made less, because in these cases the governor must rotate faster to maintain the same position. As it is usually impractical to change the weight of the arms, the speed is usually changed by changing the tension of the spring. For this purpose the spring is provided with a turnbuckle or some other arrangement by which its tension may be readily changed.

It is sometimes desired to reverse the direction of rotation of an engine which is fitted with a shaft governor. This can be done with some engines but with others it cannot be done as the manufacturer has not made provision for it.

In order to reverse the direction of rotation of a slide valve engine the eccentric must be turned through an angle of 180 degrees on the shaft. In a shaft governor the eccentric is connected directly to the arms of the governor, consequently the arms must be turned around so as to swing in the opposite direction and the attachment of the springs to the rim or spokes of the flywheel reversed by attaching them to other holes, if these have been provided by the manufacturer. The changes required in order to reverse the direction of rotation of an engine supplied with a shaft governor are illustrated in Fig. 144. In this illustration (*a*) shows the arrangement of the parts for one

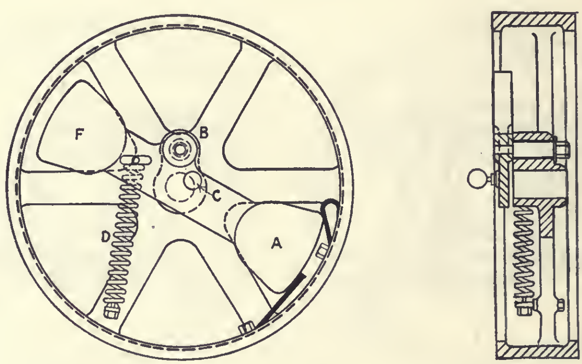


FIG. 145.

direction of rotation and (*b*) shows the arrangement of the parts for the opposite direction of rotation. As each manufacturer of high speed engines has his own arrangement of shaft governor it is impossible to give definite directions for the proper arrangement of the governor in order to reverse the direction of rotation, consequently, if this is desired, it is best to write to the manufacturer before attempting to make any changes in the governor.

Inertia Governor.—A form of shaft governor invented by F. M. Rites and known as the Rites inertia governor is used on several makes of automatic high speed engine. In this governor, which is illustrated in Fig. 145, a heavy bar on the flywheel, carrying two weights *F* and *A*, swings about the pin *B*. The eccentric, which is usually only a pin located near the center of the engine shaft is carried by the arm. As the arm swings about the pin *B*, the eccentric pin *C* swings closer to or further away from the center of the shaft and thus changes the eccentricity. The controlling force is furnished by the coil spring *D*, which has one end fastened to the arm of the governor and the other end fastened to a spoke of the flywheel.

This governor operates by the force of inertia, or the tendency of the weights to keep on moving at a constant speed, when the speed of the flywheel changes. If the engine is running at a constant speed the flywheel and governor weights will be turning at the same rate. Referring to Fig. 145, suppose a load is suddenly put on the engine. This slackens the speed of the flywheel, but the inertia of the governor weights causes them to move forward at the same rate as before. This moves the eccentric pin further away from the center of the shaft, which increases the eccentricity and causes a later cut-off. Also, if the load should be decreased, the speed of the flywheel will increase, causing the governor weights to lag behind and reduce the eccentricity. This causes cut-off to occur earlier and bring the speed back to its normal value.

The inertia governor described above is extremely simple but in securing this simplicity some things have been sacrificed. One of these is that the governor does not give a constant lead, which is desirable for a constant speed engine. It will be seen also that the governor is unbalanced, since it is pivoted away from the center of the shaft. This causes the arm to tend to fall forward during one-half of the revolution and to fall backward during the other half of the revolution. If the speed is high, say over 250 revolutions per minute, this effect is not noticeable, but for lower speeds it will affect the cut-off. If the speed is reduced much below 200 revolutions per minute, this unbalancing effect becomes noticeable as a jerk in the governor action which may send the governor arm through its whole range every second or third revolution.

To prevent this action, a drag or brake spring is attached to the rim of the flywheel in such manner as to bear against one of the weights with enough force to prevent sudden swinging but not enough to prevent the governor from swinging when there is a change in load. In addition to this dampening spring there is also a spring bumper fastened to the rim of the flywheel to prevent the arm from swinging too far and damaging the valve.

In some forms of inertia governor a second arm is placed parallel with the one carrying the weights and arranged so the whole governor will be balanced. This makes the governor more complicated but makes it suitable for running at low speed and it gives the same sensitiveness as the unbalanced governor at high speeds.

CHAPTER XVII

COMPOUND ENGINES

Compounding.—The low efficiency of the steam engine shows that a large part of the heat energy supplied to it is not turned into useful work, but is lost or wasted. Even the best engines utilize only about 20 per cent. of the heat supplied to them, leaving about 80 per cent. to be accounted for by the various losses incident to the operation of the engine. Radiation of heat from the engine and the friction of the moving parts account for only a small part of the loss. A much larger part is accounted for by the heat contained in the exhaust steam. The loss from this source may be reduced considerably by the use of a condenser, which lowers the exhaust pressure and makes a larger proportion of the total supply of heat available for useful work; but, even with the use of a condenser, the loss of heat in the exhaust is considerable.

For a long time after the steam engine was invented, the three sources of loss mentioned above, namely, radiation, friction, and loss of heat in the exhaust were thought to be the only ones. It was discovered later, however, that another serious source of loss comes from the interchange of heat between the steam and the cylinder walls, which results in condensation of steam inside the cylinder. The manner in which cylinder condensation produces a loss has been fully discussed in Chapter 8 and the student should review this chapter at this time in order to understand more fully the principles underlying the compound engine.

Since cylinder condensation produces such large losses in the operation of steam engines, it becomes a matter of considerable importance to understand the causes of cylinder condensation and the means employed for reducing it. The principal cause of cylinder condensation is the large range of temperature to which the walls of the cylinder (including head and piston) are subjected during each revolution of the engine. This range of temperature is due to the expansion of the steam in the cylinder from the high pressure of admission to the relatively low pressure of the exhaust.

The variation of pressure, and therefore the range of temperature, in an engine cylinder depends upon the cut-off. With a fixed exhaust pressure an early cut-off will produce a large variation of pressure during expansion and a late cut-off will produce a small variation of pressure. It is evident that an early cut-off is necessary to the economical use of the steam because this permits the expansive force of the steam to be utilized more fully than with a late cut-off and small expansion. The engine designer is therefore confronted with two opposing conditions. On the one hand, an early cut-off increases the losses from cylinder condensation, and on the other hand, an early cut-off is necessary if the steam is to be expanded through its full range and utilized efficiently.

One of the means most commonly employed for reducing the losses from cylinder condensation is to divide the total expansion of the steam into two or more parts and to perform each part of the expansion in a separate cylinder, thereby reducing the range of temperature in each cylinder. This is called *compounding*, and engines in which the total expansion of the steam is divided between two cylinders are called *compound engines*.

Since the losses from cylinder condensation increase as the total range of pressure through which the steam is expanded increases, the number of parts into which the total expansion should be divided, in compounding, depends upon the pressure of the steam supplied to the engine. It also depends, to a certain extent, upon the kind of work for which the engine is intended. In marine work, where compounding is more generally practised than in stationary work, the number of parts into which the total expansion is divided for different boiler or admission pressures is about as follows:

Simple engines	30 to 70 lb. per sq. in. gage
Compound	80 to 120 lb. per sq. in. gage
Triple expansion	140 to 180 lb. per sq. in. gage
Quadruple expansion	200 to 250 lb. per sq. in. gage

In stationary work there is a tendency to divide the total expansion of the steam into a fewer number of parts and to use higher pressures. Compound condensing engines are often run with pressures of 120 to 150 lb. per sq. in. gage, while the compound locomotive, which is not used with a condenser, is sometimes supplied with steam having a pressure of 200 to 225 lb. per sq. in. gage.

Expansion of Steam.—An ideal expansion line for steam expanding from 120 lb. per sq. in. absolute pressure to an exhaust pressure of 1.6 lb. per sq. in. absolute pressure is shown in Fig. 146. The diagram *ABCDEFG* represents an ideal indicator diagram if the total expansion of the steam occurred in a single cylinder having no clearance. The line *AB* represents the admission line, which is very short compared with the total length of the stroke, which is represented by the line *GF*, and which

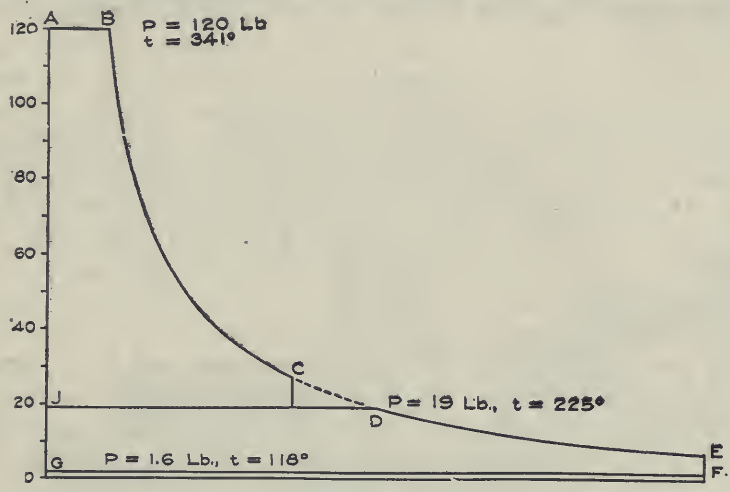


Fig. 146.

also represents the volume of the steam after expansion. The line *GF* therefore also represents the volume of the cylinder. The short admission line is necessary if the steam is to be expanded through the entire range of pressure in one cylinder. It will be observed that if the entire expansion occurred in a single cylinder, this cylinder would have to be large enough to accommodate the entire volume of steam *GF*, and would have to be strong enough to withstand the full pressure of 120 lb. per sq. in. The objection to this would be that the cost of such a cylinder would be excessive; there would be a waste of power in overcoming friction; and the large size of the cylinder would expose so much surface to the cooling action of the exhaust steam that condensation would be excessive.

Now, suppose that a line *JD* be drawn across the diagram at such a height that the area of the diagram will be divided into two approximately equal parts. If the two parts into which the expansion of the steam is divided are performed in separate cylinders, the first one will have a volume JC and would be built to withstand the full steam pressure of 120 lb. per sq. in. This cylinder would admit a volume of steam AB and would expand it to the volume JC. The expansion would not be carried further than the point C because it is desirable to have enough pressure in the cylinder at release to force the steam out of the cylinder rapidly, and also because the extra amount of work obtained by complete expansion to the point D would not be enough to balance the work lost in friction while the piston was moving through this part of the stroke. The exhaust from the first cylinder would form the supply for the second cylinder. This cylinder would have a volume GF, the same as would a cylinder designed for the entire expansion, but, as the supply of steam for the second cylinder has a pressure of only 19 lb. per sq. in. it would not have to be so strong as a single cylinder designed for 120 lb. pressure, hence would be cheaper to construct.

The second cylinder would admit the volume of steam JD at a pressure of 19 lb. per sq. in. and would expand it to the volume GF, when its pressure would be 1.6 lb. per sq. in. If the total expansion occurred in a single cylinder, this cylinder would be subjected to the full range of temperature, 223°, and, since its wall surface would be large, condensation would be excessive. By dividing the expansion into two parts, each cylinder experiences a range of temperature of only about 112°, that is, the range of temperature has been cut in half and the cylinder surface has not been doubled, hence the condensation and reëvaporation in the two cylinders would be less than in a single cylinder subjected to the entire range of temperature. This decreases materially the large loss of heat that would otherwise occur through condensation and reëvaporation; but, on the other hand, the engine would be more complicated and therefore more expensive, and the friction loss would be increased by the greater number of moving parts.

Compound Engines.—Compound engines are divided into two classes, based upon the arrangement of cylinders. These are called *tandem-compound,* in which one cylinder is placed behind the other, and *cross-compound,* in which the cylinders are placed parallel with each other.

The tandem engine, as illustrated in Fig. 147, has only one piston rod, connecting rod, and crank. The piston rod extends from one cylinder through the other and has both pistons

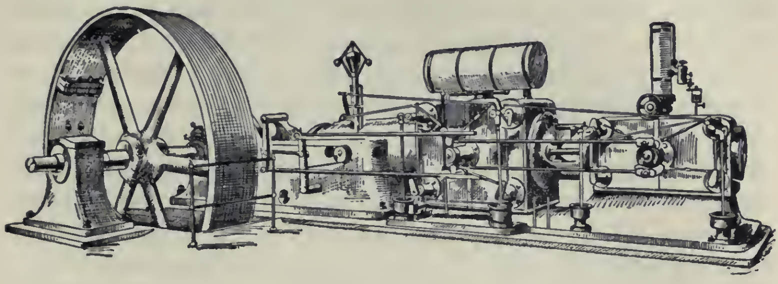


FIG. 147.

attached to it. The exhaust pipe from the high pressure cylinder passes directly to the low pressure cylinder, and, as this pipe is short, it has but little storage capacity; therefore it may be considered that the high pressure cylinder exhausts directly into

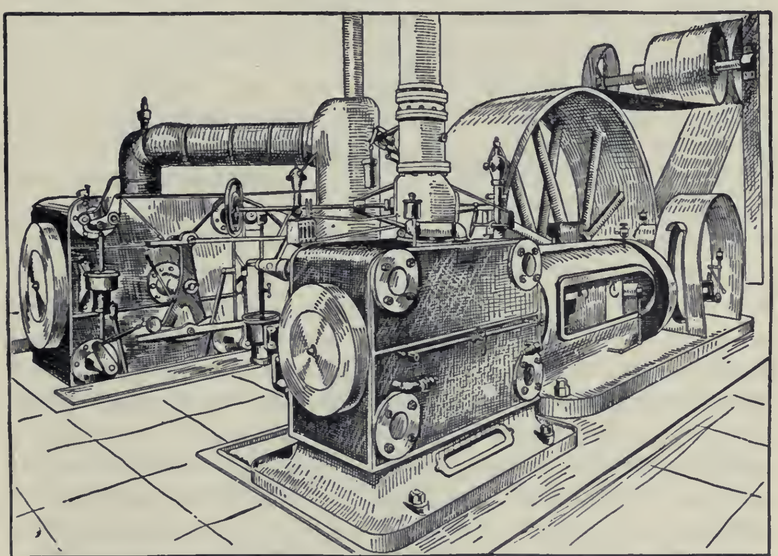


FIG. 148.

the low pressure cylinder. The tandem-compound engine is simple in construction, but the parts must be made large in order to carry the heavy stresses.

The cross-compound engine, illustrated in Fig. 148, has two pistons, piston rods, connecting rods, and cranks, hence it is similar to two simple engines placed parallel with each other and connected to the same shaft. The cranks are usually placed 90° apart, which gives a more uniform turning effort on the shaft. Since each side of the engine transmits only one-half of the power, the parts of the engine are made smaller, but the larger number of parts makes this type of engine more expensive than the tandem-compound. The exhaust pipe from the high-pressure cylinder extends across to the low-pressure cylinder and contains a receiver or vessel in which steam may be stored. This is made necessary by the cranks being placed 90° apart, as explained in a later paragraph.

The action of the steam in the two classes of engines mentioned

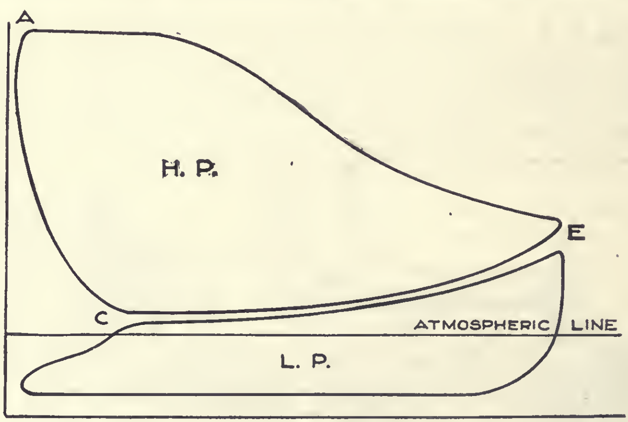


Fig. 149.

above is quite different. In the tandem engine the pistons have the same length of stroke, and move in unison with each other, beginning a stroke at the same time and ending it at the same time. For this reason the steam exhausted from the high-pressure cylinder may be passed directly into the low-pressure cylinder without having any valves on the latter cylinder, and without any storage space or receiver between the cylinders. In this case the valves and governor on the high-pressure cylinder control the action of the steam and the amount of work performed in both cylinders.

Cross-Compound Engines.—The action of the steam in both cylinders of a cross-compound engine with cranks set 180° apart and without valves on the low-pressure cylinder is very similar to that in a tandem-compound engine, and may be studied best by considering the indicator diagrams shown in Fig. 149. This illustration shows the diagram from the high-pressure cylinder, marked H.P. and that from the low-pressure cylinder, marked L.P., placed in their correct relative positions, that is, so that the exhaust stroke for the high-pressure cylinder is the admission stroke for the low-pressure cylinder. These diagrams do not, however, show correctly the division of work between the cylinders, because, being drawn to the same scale of pressure and stroke, they do not take into account the different diameters of the cylinders.

After the supply of steam is cut off from the high-pressure cylinder, the steam expands in this cylinder until released. During exhaust from the high-pressure cylinder, the steam flows directly into the low-pressure cylinder. Since the diameter of the low-pressure cylinder is larger than that of the high-pressure cylinder and both pistons move at the same speed, the volume displaced in the low-pressure cylinder is greater than that displaced in the high-pressure cylinder. The result of this is that each cubic foot of exhaust steam pushed out of the high-pressure cylinder flows into a larger volume than one cubic foot in the low-pressure, and its pressure therefore falls. This is why the exhaust from the high-pressure cylinder and the admission to the low-pressure cylinder show a continually falling pressure. When the point of compression in the high-pressure cylinder is reached, the supply of steam for the low-pressure cylinder is stopped and the steam then in the low-pressure cylinder expands with a rapidly falling pressure, since no new steam is being supplied.

It will be observed from Fig. 149 that the range in temperature in the high-pressure cylinder is that represented by the change in pressure from A to C, which is greater than it would have been if there was less drop in pressure during exhaust. Also the range in temperature in the low-pressure cylinder is that due to the difference in pressure between E and the exhaust pressure from the low-pressure cylinder. Since the pressure at E is greater than at C, the range in temperature is greater in both cylinders than would be indicated by a division of the work into two equal parts.

The above analysis of the action of steam in the cylinders of a cross-compound engine applies only to those engines which have no valves on the low-pressure cylinder or to those engines which have only one valve for both cylinders and this valve so arranged that cut-off in the low-pressure cylinder occurs at the same time as compression in the high-pressure cylinder. This type of engine is not used to a large extent and is made only in comparatively small sizes. A more common arrangement, either in tandem-compound engines or in cross-compound engines with cranks placed 90° apart, is to have separate valves on each cylinder which may be adjusted independently of each other. Engines of this kind must necessarily be supplied with a receiver or storage space in which the exhaust steam from the high-pressure cylinder may be stored if cut-off in the low-pressure cylinder

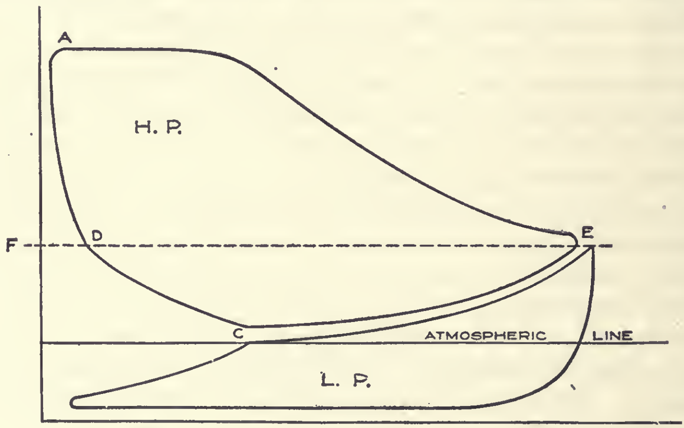


Fig. 150.

does not occur at the same time as compression in the high-pressure cylinder. If the cylinders are placed near each other so that the connecting passages are short, the receiver is usually in the form of a separate vessel connected in the passage between the two cylinders; but when the cylinders are some distance apart, the passage connecting the two cylinders has enough volume to act as a receiver, and no separate vessel is necessary.

Tandem-Compound Engines.—The presence of a receiver modifies somewhat the action of the steam in the cylinders from that described above and illustrated in Fig. 149. For a tandem-compound engine in which the connecting passage acts as a receiver, or for a cross-compound engine with cranks 180° apart and supplied with a receiver, the action of the steam in the cylinders may be shown by the diagrams in Fig. 150, which are similar to those shown in Fig. 149 except that cut-off in the low-pressure cylinder does not occur at the same time that compression occurs in the high-pressure cylinder.

In this case it will be observed from Fig. 150 that cut-off in the low-pressure cylinder occurs a little after half stroke and considerably before compression (marked D) occurs in the high-pressure cylinder. When cut-off occurs in the low-pressure cylinder, the steam then in that cylinder expands in the usual manner. The high-pressure cylinder, however, has not finished exhausting at this time; hence the remainder of the exhaust from the high-pressure cylinder is stored in the receiver. Since no steam is being drawn from the receiver at this time, the pressure in it, which is also the exhaust pressure of the high-pressure cylinder, increases as shown by the line CD in Fig. 150. At D compression occurs in the high-pressure cylinder and the exhaust valve closes communication with the receiver.

The point of cut-off in the low-pressure cylinder controls the increase of pressure in the receiver, from C to D, the increase of pressure being greater with an early cut-off and smaller with a later cut-off. The cut-off in the low-pressure cylinder must be so timed that the pressure in the receiver will be the same at D as at E, the point where the exhaust valve on the high-pressure cylinder opens. If the pressure at D is not so high as at E, the pressure at the end of expansion in the high-pressure cylinder will be greater than that in the receiver and there will be a drop of pressure the next time the exhaust valve on the high-pressure cylinder opens. This would cause a waste of pressure and a loss of work, which is to be avoided if possible.

Cross-Compound with Receiver.—The action of steam in the cylinders of a cross-compound engine with cranks set 90° apart presents another interesting case. An engine of this kind must necessarily be supplied with a receiver because one piston is at mid-stroke when the other is at the end of its stroke; hence, exhaust from the high-pressure cylinder progresses for one-half of a stroke when no steam is being admitted to the low-pressure cylinder, and it is necessary to have a receiver in which to store this steam.

The diagrams from the high- and low-pressure cylinders of an engine of this type are shown in Fig. 151. These diagrams are not drawn in the usual manner, but instead, the low-pressure diagram is displaced one-half stroke from the high-pressure diagram in order to show the relative pressures in the cylinders at any instant.

It will be observed from Fig. 151 that the exhaust pressure in the high-pressure cylinder increases gradually from the beginning to the middle of the exhaust stroke. The reason for this is that during this part of the stroke the high-pressure cylinder is exhausting into the receiver and the low-pressure cylinder is not taking any steam from it; hence the exhaust pressure in the high-pressure cylinder, which is also the receiver pressure, increases. When the high-pressure piston reaches mid-stroke, the low-pressure cylinder begins to admit steam, since the cranks are 90° apart, and the receiver pressure is reduced. Thus, the

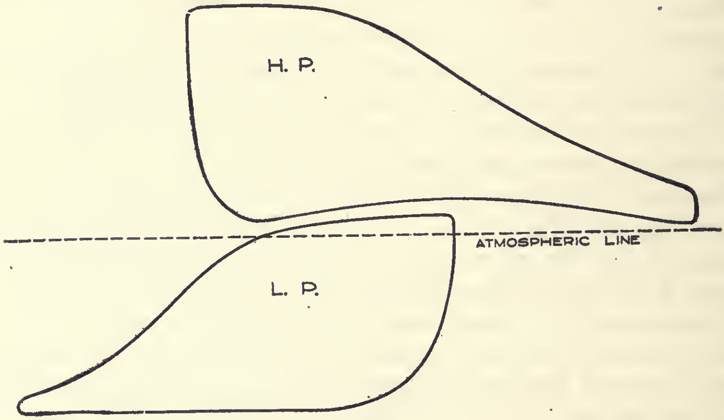


Fig. 151.

high-pressure exhaust line rises from beginning to mid-stroke and falls from mid-stroke to the point of compression.

The admission line for the low-pressure cylinder follows the shape of the last half of the exhaust line of the high-pressure cylinder; hence it shows a decreasing pressure. In the low-pressure diagram shown in Fig. 151 cut-off occurs at or before mid-stroke, or before the high-pressure piston has completed its stroke. If cut-off in the low-pressure cylinder occurs after half stroke the high-pressure piston will have started on its return stroke and exhaust will have commenced from the other end of the cylinder; hence the pressure in the receiver will again begin to increase and this will produce a corresponding increase in the admission pressure for the low-pressure cylinder. The effect of the second admission of steam into the receiver before the low-pressure cut-off is illustrated in Fig. 152, where the admission pressure for the low-pressure cylinder is shown decreasing up to mid-stroke and increasing from mid-stroke to the point of cut-off. This second increase in pressure is called "second admission," and is to be found only when cut-off in the low-pressure cylinder occurs after mid-stroke.

One of the advantages of the cross-compound engine with cranks 90° apart is illustrated by Fig. 151 which shows that the range of temperature in it is less than in the cylinders of a cross-compound with cranks set 180° apart (Fig. 149), or a tandem-compound (Fig. 150), because the exhaust from the high-pressure cylinder shows a more uniform pressure. The variations in pressure illustrated in Figs. 149, 150, 151, and 152 will not show to such a marked degree on the actual indicator diagrams because

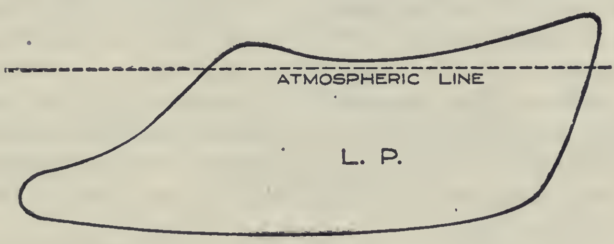


FIG. 152.

the high-pressure diagram is drawn with a stiff indicator spring. The variations in pressure in the low-pressure admission may be detected easily on the actual diagram, however, because this diagram is drawn with a weak spring.

Power of a Compound Engine.—The power developed by any steam engine, whether simple or compound, depends upon the number of times the steam is expanded, that is, upon its ratio of expansion. It evidently does not matter, then, as far as the power of a compound engine is concerned, whether the total expansion of the steam occurs in one cylinder or is divided between two cylinders, provided only that the steam is expanded the same number of times in each case.

In a compound engine, the total expansion is divided between two cylinders for the purpose of reducing cylinder condensation, and not for the purpose of increasing the power of the engine. The total power developed in both cylinders of a compound engine could be developed in the low-pressure cylinder alone by having cut-off in the low-pressure cylinder occur early enough to secure as many expansions of the steam in this cylinder as was secured in both the high- and low-pressure cylinders. For example, if cut-off in the high-pressure cylinder occurs at one-quarter stroke the steam will expand approximately four times in this cylinder. If the volume of the low-pressure cylinder is three times that of the high-pressure cylinder then the total expansion of the steam will be

$$4 \times 3 = 12$$

This number of expansions could have been secured in the low-pressure cylinder alone by admitting the steam directly to that cylinder and having cut-off occur at approximately $\frac{1}{12}$ stroke.

In any case the approximate ratio of expansion in a multiple expansion engine may be found by multiplying the ratio of expansion in the high-pressure cylinder by the ratio of the volume of the low-pressure to the high-pressure cylinder, or by the ratio of the square of their diameters. In order to find the ratio of expansion more accurately, the clearance volumes would have to be considered, but this is not necessary for ordinary purposes, as the ratio of expansion changes with the cut-off which, in turn, varies with the load.

From the above discussion it will be seen that the total power developed by a compound engine depends upon the ratio of the cylinder volumes and upon the point of cut-off in the high-pressure cylinder. For any given engine the ratio of the cylinder volumes is a fixed quantity, therefore we may say as a general proposition that the power developed *by a compound engine depends only upon the point of cut-off in the high-pressure cylinder.*

The point of cut-off in the low-pressure cylinder has no effect whatever upon the total amount of work done by a compound engine. *The point of cut-off in the low-pressure cylinder does, however, control the distribution of work between the two cylinders.* It does this by affecting the exhaust pressure of the high-pressure cylinder. If cut-off in the low-pressure cylinder occurs early in the stroke, the exhaust pressure of the high-pressure cylinder will be high and the work performed in this cylinder will be a smaller portion of the total work and the work performed in the low-pressure cylinder will be a larger portion of the total work. On the other hand, if cut-off in the low-pressure cylinder occurs late, the exhaust pressure of the high-pressure cylinder will be lower and a larger proportion of the total work will be performed in the high-pressure cylinder.

The low-pressure cut-off should be adjusted so as to secure an approximately equal division of work between the high- and low-pressure cylinders, and, also, so there will be a small drop in pressure at the end of expansion in the high-pressure cylinder when the engine is working under load. The object in having a small drop of pressure at the end of expansion is that there will be no gain in carrying the expansion completely down to exhaust pressure and, moreover, a little drop in pressure into the receiver is needed to secure a quick flow of steam out of the high-pressure

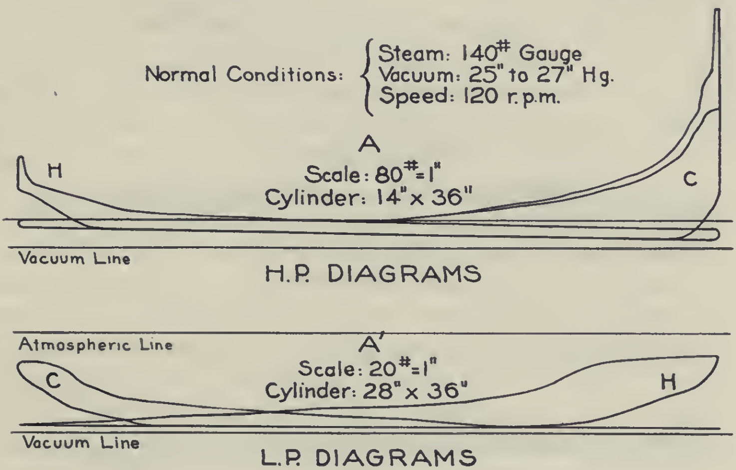


Fig. 152*a*.

cylinder. However, when the engine is working under no load or only a small load there should be no drop of pressure into the receiver, but instead, the receiver pressure should be higher than the pressure at the end of expansion in the high-pressure cylinder. The importance of this should not be overlooked, because, if the receiver pressure becomes too low, a condition may be produced under which the engine will run-away.

Such a condition as this is illustrated in Figs. 152*a* and 152*b*. These illustrations show the indicator diagrams from a cross-compound engine in which the condition mentioned above existed. Fig. 152*a* shows the indicator diagrams from the high-pressure cylinder at A and those from the low-pressure cylinder at A', both being taken while the engine was running under no load. It will be noted by examining these diagrams that the receiver pressure is too low, as indicated by the exhaust line of the high-pressure diagrams and also by the admission lines of the low-pressure diagrams. The exhaust pressure of the high-pressure cylinder is so low that it is impossible for negative work to be done in the high-pressure cylinder, and, even though the governor is causing cut-off at the earliest possible point, the expansion of steam in the high-pressure cylinder is doing more work than needed to carry the friction load at normal speed. Consequently the speed increases. When the speed had reached 140 R.p.m. the

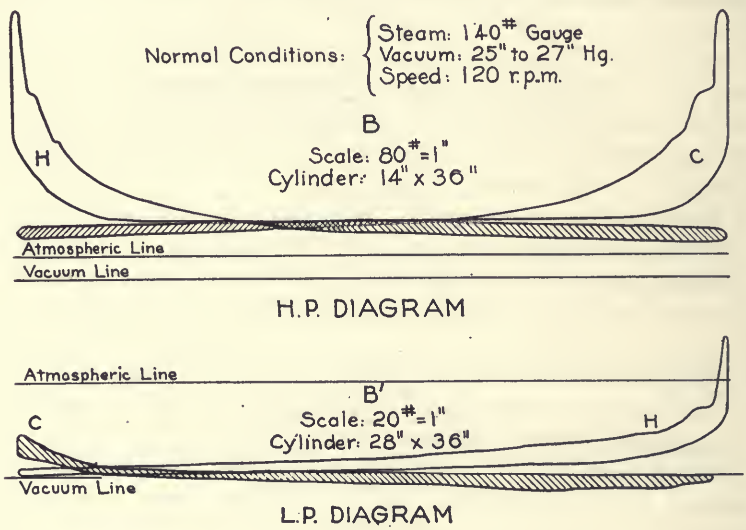


Fig. 152*b*.

engine was stopped by closing the throttle, but if this had not been done the speed would have continued to increase and the engine would have run away. If the cut-off in the low pressure had occurred earlier in the stroke, the receiver pressure, which is the exhaust pressure of the high-pressure cylinder, would have been higher and negative work would have been done in the high-pressure cylinder. This would have put sufficient load on the high-pressure cylinder to prevent the speed from increasing above its normal value.

Fig. 152*b* shows diagrams taken from the engine when running under no load but with the cut-off in the low-pressure cylinder adjusted to occur earlier. It will be observed that the receiver pressure is now $7\frac{1}{2}$ to 8 lbs. above atmospheric pressure and the expansion in the high-pressure cylinder carries the pressure below receiver pressure so that negative work (indicated by the cross-hatched lines) is done by the high-pressure piston during its exhaust stroke. This negative work is sufficient to hold the speed down almost to normal.

Advantages and Disadvantages.—The principal advantage derived from compounding is the reduction of losses resulting from cylinder condensation and reëvaporation. With a simple engine these losses increase with high-steam pressures and with a large number of expansions of the steam. Hence, a simple engine is not well adapted to the use of high pressures nor for an early cut-off, both of which are necessary for the economical use of the steam. It will now be understood why a compound engine is much better adapted for the use of high-pressure steam and for expanding the steam a large number of times, hence the general use of this type of engine for producing large amounts of power, when efficiency is a very important factor.

Most of the disadvantages of the compound engine are of a mechanical nature and arise from the greater complication of this type. The greater number of parts make them more expensive in first cost and also make them more expensive to maintain than a simple engine, on account of more repairs being necessary. The greater number of moving parts also adds to the cost of lubrication and increases the loss of power in friction. In these respects triple expansion and quadruple expansion engines are at even greater disadvantage than compound engines, with the result that quadruple expansion engines have dropped out of use for stationary purposes and the use of triple expansion engines is confined almost entirely to large pumping plants.

CHAPTER XVIII

CONDENSING APPARATUS

Purpose of the Condenser.—When an engine exhausts into the atmosphere, the exhaust stroke of the piston is made against the atmospheric pressure, which acts upon the entire face of the piston. This pressure acts in a direction opposite to that in which the piston is moving and tends to retard its motion. The piston must overcome not only the atmospheric pressure but also the friction of the exhaust steam in passing through the ports and exhaust pipe on its way from the cylinder to the atmosphere. The atmospheric pressure (14.7 lb. per sq. in.) added to the friction of the exhaust passages makes a total pressure of between 15 and 20 lbs. per sq. in. which the piston must move against. When it is considered that this back pressure acts upon the piston during almost the entire exhaust stroke, and that the piston must do work in moving against this pressure, it will be realized that the engine could do considerably more useful work if this back pressure were removed.

Removing or reducing the back pressure on an engine increases its mean effective pressure. When it is remembered that the mean effective pressure of an engine is directly proportional to its indicated horsepower, it will be seen that anything which increases the mean effective pressure will also increase the indicated horsepower developed by the engine. In order to gain some idea of the increase in horsepower by lowering the back pressure consider an engine taking steam at an absolute pressure of 100 lb. per sq. in., cutting off at ¼ stroke, and exhausting into the atmosphere against a back pressure of 16 lb. per sq. in. The theoretical mean effective pressure under these conditions will be 43.7 lb. per sq. in. If the back pressure was reduced to 1.7 lb. per sq. in. (26 in. vacuum) the mean effective pressure would be increased to

$$43.7 + (16 - 1.7) = 58 \text{ lb. per sq. in.}$$

which would result in an increase of power of

$$100 \frac{58 - 43.7}{43.7} = 32.8 \text{ per cent.}$$

The actual gain would be somewhat less than this depending upon the type of engine and the conditions of operation, but in any case it would be considerable.

The back pressure on an engine is reduced by leading the exhaust steam into a closed vessel and condensing it into water, instead of permitting the engine to exhaust directly into the atmosphere. Such a closed vessel is called a condenser. As the exhaust steam enters the condenser, it either meets a spray of cold water or comes in contact with tubes through which cold water is flowing. In either case, the water extracts heat from the exhaust steam and condenses it into water. Since the water occupies only about $\frac{1}{1700}$ of the space occupied by the exhaust steam, the pressure in the condenser is reduced by the condensation of the steam. In order to maintain the low pressure in the condenser it is necessary to condense the exhaust steam as fast as it enters and to constantly remove the water and any air which may come in with the exhaust steam.

The purpose of installing a condenser may be either to increase the efficiency of the engine or to increase the power of the engine. A condensing engine will be more efficient than a noncondensing one for the reason that cut-off in the condensing engine may be shorter than in the noncondensing engine when the same amount of power is developed in both, on accountof the great number of times the steam is expanded in the condensing engine. For example, an engine running noncondensing may cut off at $\frac{1}{4}$ stroke and develop a certain amount of power. The same engine connected to a condenser may cut off at $\frac{1}{6}$ stroke and develop the same amount of power. Since the amount of steam used by the engine is in proportion to the cut-off, the engine will use $\frac{1}{4} - \frac{1}{6} = \frac{1}{2}$ less steam when running condensing than when running noncondensing. The amount of steam which can be produced from a pound of coal is ordinarily from 7 to 10 pounds, but the amount of power obtained from the steam depends upon how the steam is utilized. Since, by running an engine condensing rather than noncondensing the steam is utilized more efficiently, power plant engines are almost invariably run condensing unless the exhaust steam is used for heating.

A Corliss engine running noncondensing will use from 25 to 30 pounds of steam per indicated horsepower per hour but if run condensing, it will use only about 20 pounds. For compound engines, the amount used will be about 25 pounds noncondensing and about 15 pounds condensing. A triple expansion engine running condensing will produce an indicated horsepower from as little as 10 pounds of steam.

While a condensing engine will require from 20 to 30 per cent. less steam than a noncondensing one, this apparent decrease in steam consumption does not represent a net gain. The steam used by the condenser pumps must be added to that consumed by the engine unless the exhaust from the pumps is used for heating the feed water in which case only the difference between the heat entering and leaving the pumps should be charged to the engine.

Condensation of Steam.—The condensation of steam is just the reverse of the process by which steam is formed, and the amounts of heat involved are the same; the only difference being that heat must be added to water to change it into steam and that heat must be taken away from the steam to condense it into water. Moreover, for the same conditions of pressure, quality, and weight of steam exactly the same amount of heat must be transferred in either case, being transferred *into* the steam in one case and *out* of it in the other.

A pound of steam at any pressure contains a definite amount of latent heat of evaporation, as may be seen by reference to the steam table in Chapter 5. If this amount of heat is taken out of the steam, a pound of it condenses into water and the water will have the same temperature as the steam from which it was condensed. If only one-half of the latent heat in a pound of steam is extracted, then only one-half of a pound of steam will be condensed and the resulting water will be at the same temperature as the steam. The same is true for any amount of heat taken from the steam. The weight of steam condensed will be the number of heat units extracted divided by the latent heat of one pound of steam at the pressure of condensation. If the exhaust steam is wet, that is, contains moisture suspended in it, this moisture contains no latent heat, therefore only the latent heat actually contained in the steam must be extracted in order to condense it. While steam may not be entirely dry at the end of expansion, the drop in pressure at release usually completes the drying, so that in calculations relating to condensers it is usually assumed that the exhaust steam is dry.

In order to condense steam it is necessary to bring it into contact with something which has a lower temperature because heat will only pass into a substance at lower temperature. For this reason the condensing water used in a condenser must have a lower temperature than the exhaust steam that is to be condensed. When the condensing water absorbs heat from the exhaust steam, the steam is condensed and the temperature of the condensing water is increased. It is evident that the steam cannot be condensed unless its temperature is higher than the final temperature of the condensing water.

By continually condensing the steam in the condenser a low pressure is maintained in it. The steam is expanded in the cylinder almost to this pressure, and when the exhaust valve opens, the steam pressure drops to the same pressure as that in the condenser. At the same time, its temperature drops to that shown by the steam table to correspond to its pressure. For example, suppose the absolute pressure in the condenser is maintained at 2 lb. per sq. in. then the exhaust steam entering it will have this pressure and it will have a temperature of 126.15° F. as will be seen by referring to the steam table in Chapter 5. The exhaust steam at this pressure and temperature has a latent heat of 1021 B.t.u. per pound, and in condensing it gives up this heat to the condensing water. The condensate, or water resulting from the condensation of the steam, will also have a temperature of 126.15°F.

Measuring Vacuum.—Strictly speaking, a *vacuum* means a space in which there is *no* pressure, or in which the absolute pressure is zero. However, in steam engineering work the word vacuum refers to any space in which the pressure is less than atmospheric pressure. For this reason, the reduced pressure in a condenser is called a *vacuum*.

The vacuum in a condenser may be measured by means of a mercury column or by means of a gage constructed somewhat like a pressure gage but marked to read pressures less than that of the atmosphere. The mercury column is the more accurate method and is generally used where the pressure in the condenser is very low.

A device for measuring vacuum by means of a mercury column is illustrated in Fig. 153. In this device a glass tube about 80 inches long is bent into a U-shape, and is filled about half full of mercury. One branch of the glass U-tube is connected to the space in which the vacuum is to be measured, the other branch being left open so that it is under atmospheric pressure. As the pressure is reduced in the space into which the U-tube is connected (in this case, a condenser), the mercury will rise in that branch to a height A, corresponding to the *difference* in pressure on the surfaces of the mercury in the two branches of the U-tube.

The amount of the vacuum is usually expressed in inches of mercury, or simply "inches," and is the difference in height of mercury in the two branches of the U-tube. Thus, if the height A in Fig. 153 is 20 inches, the vacuum amounts to 20 inches of mercury, or is said to be "20 inches." It should be remembered that the height of the mercury column indicates the *reduction of pressure,* and not the *actual pressure existing in the condenser.* A

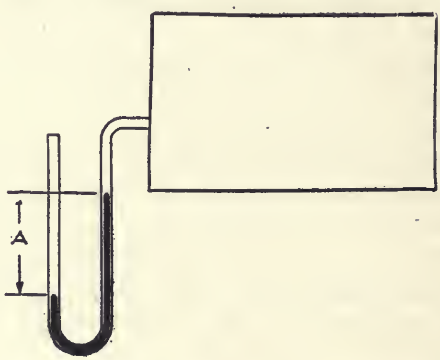


FIG. 153.

vacuum of 20 inches means that the pressure has been *reduced* enough to support a column of mercury 20 inches high. Since a column of mercury 1 inch high is equivalent to a pressure of .49 lb. per sq. in., 20 inches corresponds to a reduction of pressure of $20 \times .49 = 9.8$ lb. per sq. in. below atmospheric pressure. Before the pressure still existing in the condenser can be found, it is necessary to know the pressure of the atmosphere. If the atmospheric pressure is 14.7 lb. per sq. in., a vacuum of 20 inches leaves a pressure of $14.7 - 9.8 = 4.9$ lb. per sq. in. If the barometer, which measures the atmospheric pressure, reads 28 inches, then 20 inches of vacuum leaves a pressure of $28 - 20 = 8$ inches of mercury, or $8 \times .49 = 3.92$ lb. per sq. in.

It is seen from the above discussion that a statement to the effect that the vacuum carried by a condenser is a certain number of inches does not always mean the same thing, because of variation in the atmospheric pressure. Thus, a vacuum of 22.5 inches at a place 5280 feet above sea level is as near a perfect vacuum as 28 inches at New York, which is at sea level. In the first mentioned place 24.5 inches would be a perfect vacuum, while at sea level 30 inches would be a perfect vacuum. Vacuum gages of all kinds are practically always marked to read in inches of mercury to correspond with the U-tube described above.

In the operation of a condenser there will always be some pressure in the condenser due to the presence of water vapor and air, both of which exert a pressure. The amount of this pressure will depend upon the temperature of the condensate and the temperature and quality of the air present in the exhaust steam. Air enters the boiler in the feed water and passes into the piping system with the steam. It also leaks into the low-pressure cylinder of the engine around the piston rod. Leaks in the condenser itself and exhaust piping, both of which are under a pressure less than that of the atmosphere, also account for the presence of some air in the condenser. Whatever air is present adds its pressure to that of the water vapor, so that the total pressure in the condenser is equal to the sum of the vapor pressure and the air pressure.

The vapor pressure depends upon the temperature of the condensate and is the same pressure as that shown in the steam table as corresponding to the various temperatures. For example, if the temperature of the condensate is 101.83°F., the vapor pressure in the condenser is 1 lb. per sq. in. which corresponds to $\frac{1}{.49}$ or 2.04 inches of mercury or to a vacuum of $30 - 2.04 = 27.96$ in. (atmospheric pressure being 14.7 lb. per sq. in.). If the temperature of the condensate was 126.15°F., the vapor pressure would be 2 lb. per sq. in. corresponding to a vacuum of $30 - \frac{2}{.49}$ or 25.92 in. It must be remembered that the pressure of the air in the condenser acts in addition to the vapor pressure, so that if, for example, there were enough air present in the condenser, in the last case mentioned above, to create a pressure of 1 lb. per sq. in. the total pressure in the condenser would be 3 lb. instead of 2 lb. and the vacuum would be $30 - \frac{3}{.49}$ or 23.87 in.

It is necessary continually to remove the air and condensate as they would soon accumulate and destroy the vacuum. But no matter how perfect the condensing and pumping apparatus there will always be some vapor pressure due to the temperature of the condensate, and there will always be more or less air entering the condenser with the exhaust steam. The more perfect

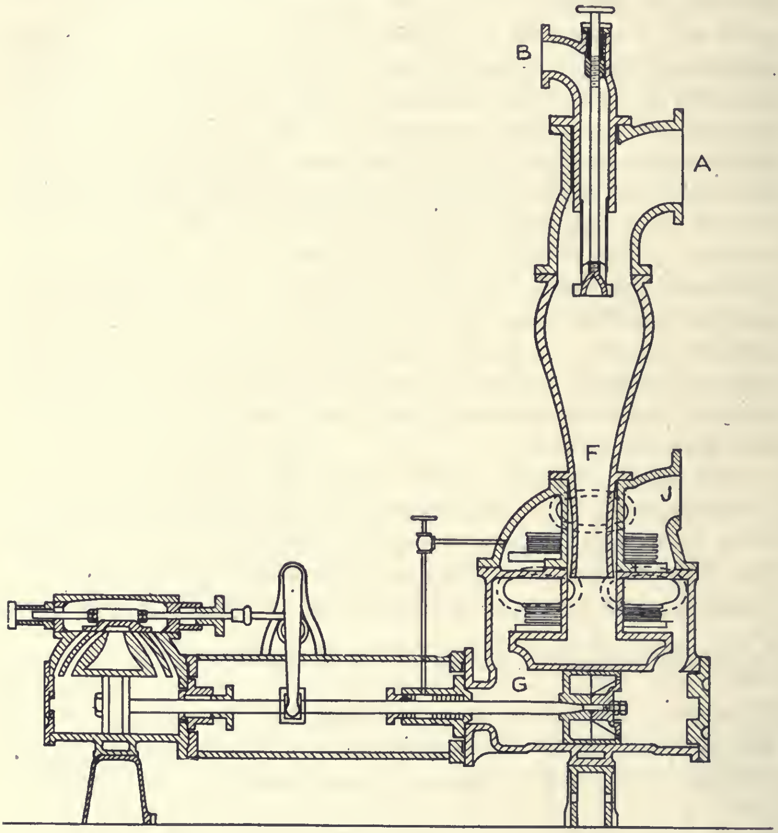


Fig. 154.

the apparatus for removing these, and the colder the condensing water, the higher will be the degree of vacuum that may be carried.

Forms of Condensing Apparatus.—Condensers may be divided into two different classes or types; namely, those in which the steam is condensed in direct contact with water, the condensate and condensing water mixing and leaving the condenser at the same temperature; and those in which the steam is condensed in contact with tubes through which or around which the condensing water flows. In the latter class the condensate and condensing water are kept separate and the condensing water leaves the condenser at a somewhat lower temperature than the condensate. Practically all condensers fall into one or the other of these two classes although there are many varieties in each class.

Jet Condenser.—One of the simplest forms of condensers is the jet condenser illustrated in Fig. 154. In this condenser, the condensing water enters the injection pipe at *B*. At the end of the injection pipe is an adjustable cone-shaped spray head which breaks the water into a fine spray. The exhaust steam entering at *A* meets the spray of water, condenses, and falls to the bottom of the condensing chamber *F*. From the bottom of the condensing chamber the mixture of condensing water, condensate, and air is taken into the steam-driven wet air pump *G* and forced out through *J* into the hot well. If there should be an excess of condensing water, due either to the improper regulation of the spray head or to the failure of the pump to remove the water as fast as it collects in the bottom of the condensing chamber, the water will rise in the condensing chamber until it covers the spray head and the condensation of steam will be practically stopped. The vacuum will then be greatly reduced, or broken, and the pressure of the exhaust steam acting upon the surface of the water will force it through the valves of the pump and into the hot well. There would then be no danger of water flooding back into the engine cylinder and wrecking it.

With this type of condenser it is not necessary to have the supply of condensing water under pressure as the vacuum in the condensing chamber will draw the condensing water through the injection pipe unless the supply is more than about 15 feet below the condenser. However, if the condenser draws in its own supply of condensing water there must be some means provided for creating a vacuum to start the condenser. This may be done by starting the pump or by providing an auxiliary injection of water under pressure so that the first steam to come from the engine may be condensed and thus create a vacuum for starting the water through the spray head.

The amount of vacuum that may be maintained in a jet condenser depends upon the temperature of the water in the bottom of the condenser, the amount of air carried into the condenser by the condensing water and steam and upon the tightness of the valves and joints.

Siphon Condensers.—A type of jet condenser known as the siphon condenser is illustrated in Fig. 155. In this condenser the cooling water enters at the side through the pipe *A* and overflows the edges of the cone-shaped nozzle *H*. This causes the water to form a hollow cone which becomes a solid stream in passing through the throat of the nozzle at *E*. The amount of water passing through the condenser is regulated by means of the valve *D*. The exhaust steam enters the condenser at *B* and is given a downward direction by means of the goose neck *C*. It then comes in contact with the hollow cone of water and is condensed. The mixture of condensed steam and condensing water, together with any entrained air, passes through the contracted neck of the cone at *E* with a high velocity and is discharged into the tail pipe at *F*, the lower end of which is below the surface of the water in the hot well.

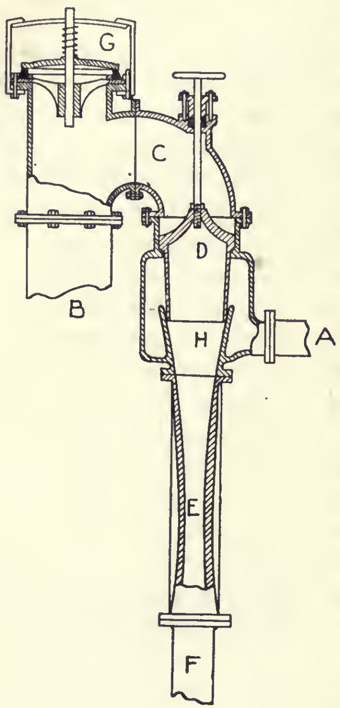


Fig. 155.

When in operation, with a vacuum in the condenser, the tail pipe filled with water, and the end of the tail pipe sealed by the water in the hot well, which is under atmospheric pressure, the condenser acts like a barometer. The atmospheric pressure on the surface of the water in the hot well would therefore force the water in the tail pipe to a height of 34 feet which corresponds to atmospheric pressure. For this reason, it is necessary that the tail pipe be at least 34 feet long in order to prevent the water from backing through the condenser and into the cylinder of the engine. The force of the water passing through the contracted neck of the nozzle will balance several pounds of atmospheric pressure so that the length of the tail pipe might be made a little less than 34 feet but, to guard against any possibility of an accident, the tail pipe is made at least 34 feet long.

The condensing water may be pumped into the condenser under pressure or the vacuum in the condenser may draw it in if it does not have to lift it more than about 15 feet. It is possible to use very muddy or dirty water in this type of condenser as it is not

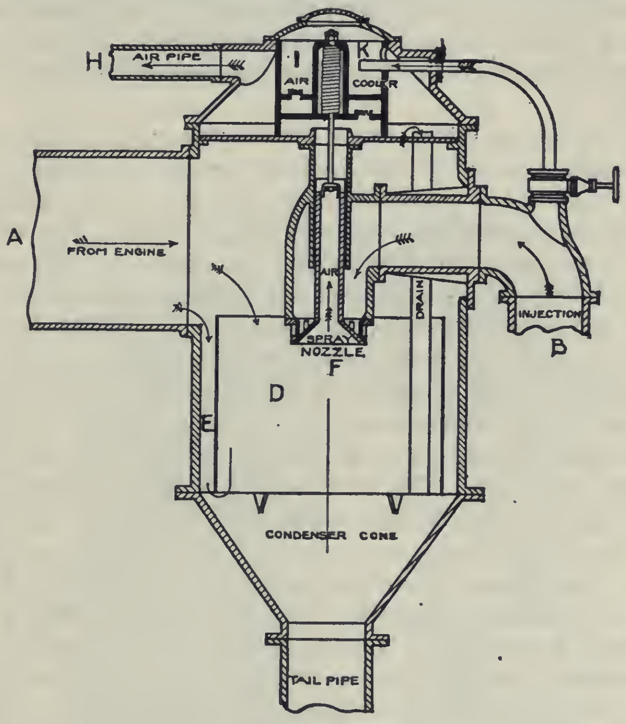


FIG. 156.

very likely to become stopped up. However, should a stoppage occur or should the condenser fail to operate for any reason, pressure will accumulate in the condenser until it opens the automatic relief valve shown at *G* in Fig. 155, and the engine will then exhaust into the atmosphere.

Barometric Condenser.—Although the siphon condenser just described might be classed as a barometric condenser inasmuch as there is barometric action in the tail pipe, the name barometric condenser is usually reserved for that class of condensers in which there is no injector or ejector action through a contracted tube.

Such a condenser as this is illustrated in Fig. 156, the tail pipe being left off to permit a larger illustration being used.

In this type of condenser the exhaust steam is brought into contact with a spray of cold water and is condensed. The mixture of condensed steam and water then flows by gravity through a tail pipe, the lower end of which is sealed by the water in the hot well. In operation, the tail pipe of this condenser acts as a true barometric tube with the water standing in it at a height corresponding to the vacuum carried in the condenser. As more water is added to the column at the top, a like amount flows out at the bottom.

The condensing water enters the injection pipe at *B* and is broken into a spray by means of the cone *F*. This cone is suspended from a coil spring above so that the nozzle opening is automatically adjusted by the water pressure. The exhaust steam enters at *A* and divides into two parts, one part passing directly into the condensing chamber *D* where it meets the spray of cold water and the other part passing downward through the annular space *E* and then upward into the condensing chamber. Any air that is not entrained in the water passing into the tail pipe and also any uncondensed vapor pass up through the center of the spray nozzle, which is hollow, and into the air cooling chamber at the top. The air is cooled and the vapor condensed by an injection of cold water through the pipe *K*. The air is then drawn off through the air pipe *H* by means of a dry air pump. Since the air and water move in opposite directions, this type of condenser is known as a countercurrent condenser.

Surface Condensers.—This class of condensers includes all those in which the steam and condensing water are kept separate and in which the steam is condensed on a metal surface. These condensers usually consist of a shell which contains a large number of small tubes, with the condensing water on the inside of the tubes and the steam on the outside, although in a number of makes of surface condensers the condensing water is on the outside and the steam on the inside of the tubes. The tubes are usually made of brass because this metal conducts heat better than iron.

A typical example of a surface condenser is illustrated in Fig. 157 which shows a Wheeler condenser with wet air pump and circulating pump for forcing the condensing water through the condenser. This condenser, which has a rectangular cross section,

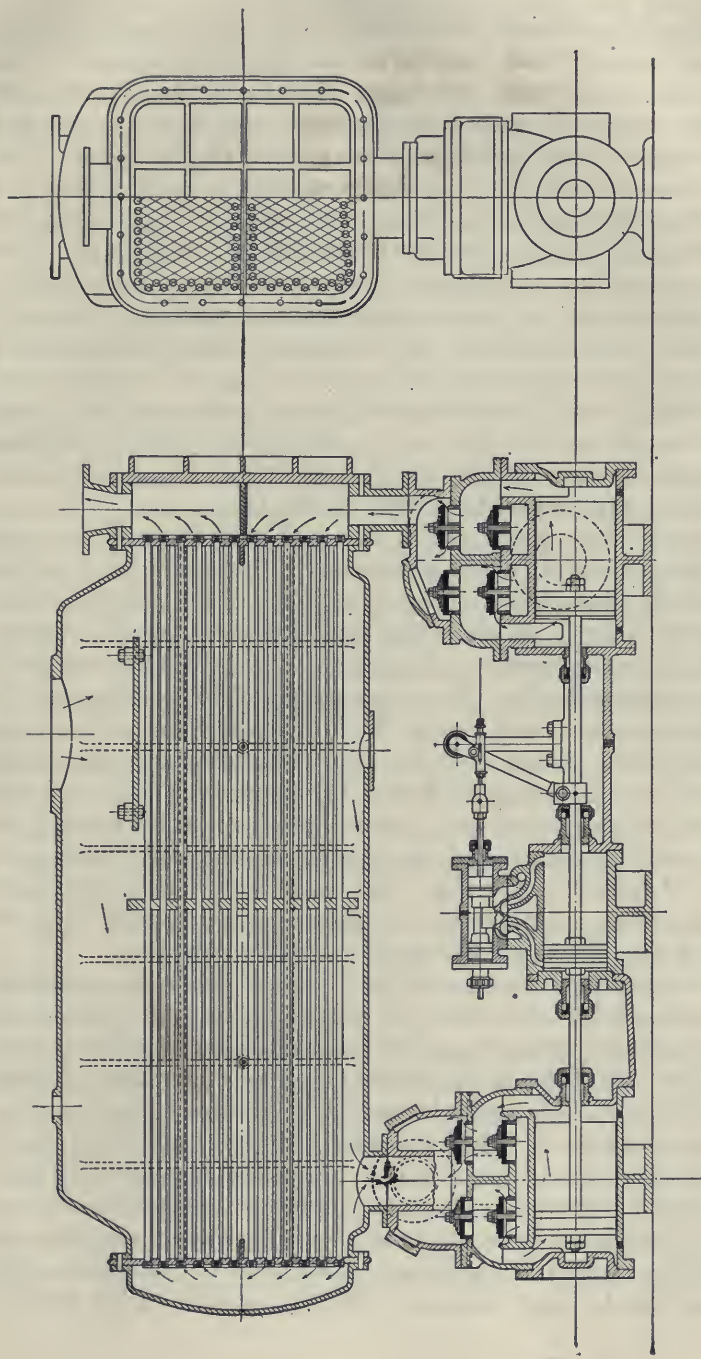


Fig. 157.

consists of a cast-iron shell or case containing a large number of closely spaced small seamless drawn brass tubes through which the condensing water circulates. The tubes have considerable length compared with their diameter and they are therefore supported at their middle points to prevent their sagging. They are fastened into the tube sheets at their ends by means of a stuffing box made of a brass ferrule with packing. This construction is used so that the tubes will be free to expand and contract without leaking and so they may be easily removed.

The space in the shell between the tube sheets and the heads of the condenser forms the condensing water compartments. The water compartment at the right of Fig. 157 is divided by a baffle plate which causes the water to flow in one direction through the bottom set of tubes and to flow in the opposite direction through the top set, the water being forced through the tubes by the pump shown at the right of Fig. 157.

The exhaust steam enters the condenser through the large opening shown at the top of the shell. Upon entering, it strikes the baffle plate shown just over the tubes, which prevents it from striking directly on the tubes and distributes it over a greater area of cooling surface. In this way the entire tube surface is made more effective. The condensation of steam begins on the surface of the upper row of tubes and continues as the steam passes down among the tubes, the condensate dropping to the bottom of the condenser shell. The condensing water entering at the bottom, increases in temperature as it flows towards the top, until its temperature upon leaving is practically the same as that of the exhaust steam. The condensate, together with any entrained air, is drawn into the wet vacuum pump shown at the left of Fig. 157 and is pumped out of the condenser.

In operating a condenser of the kind described above particular care should be used to see that exhaust steam is not turned into the condenser unless there is water in the tubes as the tube packings are liable to be destroyed and the shell or tube sheets cracked, particularly if cold water is admitted. Sudden or large changes of temperature should be avoided as they are apt to injure the tubes or to cause them to leak. The water chambers should be examined frequently to see that no foreign matter is collecting and stopping the tubes as this will lower the vacuum and increase the duty of the tubes and circulating pump. The outside of the tubes can be kept clean and efficient by introducing about a gallon of kerosene with the exhaust steam once a week and just before shutting down. This will free the tubes from oil carried over in the exhaust steam.

High Vacuum Condensers.—Any of the condensing systems described above are well adapted for use with steam engines because a steam engine operates at its best commercial economy with a vacuum of about 26 in. (referred to a 30-in. barometer) and any of these condensers will produce a vacuum of from 24 to 26 in. If a steam engine is operated at a higher vacuum than about 26 in., cylinder condensation becomes excessive because of the large size of the cylinder made necessary by the largely increased volume of the steam at very low pressures. The increased size of the piston also increases the friction so that the loss from these two sources more than balances the gain from increased expansion of the steam.

A steam turbine, on the other hand, will show a considerable increase in efficiency from the use of vacua above 26 in. This is largely because the steam in passing through the turbine experiences a gradual drop in temperature and the parts of the turbine are not subjected to a great range of temperature as in a steam engine, hence there is not much condensation of steam in the turbine. For the reasons just mentioned condenser manufacturers have made considerable improvement in condensing systems since the introduction of the steam turbine. The principal improvements in securing higher vacua were the use of a separate dry vacuum pump for removing the air and noncondensable vapors, and also the providing of means for cooling the air before it enters the air pump so that the volume to be handled will not be so large.

Choice of a Condenser.—The choice of a condenser for a steam engine depends largely upon the kind and cost of the available supply of condensing water. Where there is a plentiful and cheap supply of good condensing water which is also suitable for feeding into the boiler, some good type of jet condenser will generally be found most desirable. If there is sufficient overhead room, a siphon or barometric condenser will be found most desirable and least expensive. In this connection, it must be remembered that these condensers may often be located to advantage outside the building. A condenser of the siphon or ejector type will also use to advantage a very dirty or muddy condensing water which would not be at all suitable for feeding into the boiler. If this is done, however, the feed water must be heated in some other way, but this can usually be done by means of the exhaust from the steam-driven pumps and other auxiliaries around the power plants.

When the supply of condensing water is not suitable for feeding into the boiler, a surface condenser should be used because then the same feed water may be used over and over and only enough additional feed water need be supplied to make up the losses by leakage and other sources of waste. However, the condensing water used in a surface condenser should not be so dirty as to cause stoppage of the tubes nor should it contain enough mineral matter to give trouble from incrustation as the tubes of a surface condenser are so small and so numerous that cleaning them is difficult and expensive. On the other hand, not much trouble from incrustation is to be expected because the condensing water does not reach a high enough temperature to cause mineral substances to deposit very fast and also because the velocity of the water through the tubes is high enough to retard the deposit of mineral matter.

CHAPTER XIX

LUBRICATION

Friction.—Every bearing in every piece of machinery in a power plant produces friction. Friction in bearings generates heat, which is one form of energy, and, as there is no way of utilizing this heat, it is lost. For this reason, friction in bearings represents a direct loss. The continual overcoming of friction requires power, and it is for this reason that some power is required to run a steam engine even when it is carrying no load except the load caused by the friction.

The power lost in overcoming friction in a power plant depends upon the number, kind, and condition of the bearings. It is seldom less than 5 per cent. of the total power generated and it is sometimes as much as 30 per cent. The power wasted by friction in a steam engine will amount to from 5 to 20 per cent. of the total power of the engine, with about 10 per cent. representing the average for an engine with good bearings well lubricated.

The losses due to friction are not only the loss of power but they include also the repairs and depreciation due to wear on the bearings, guides, piston rod and packing, piston, and other rubbing surfaces.

The losses mentioned above may be reduced considerably by using a sufficient quantity of the proper kind of lubricant. The selection of the kind of lubricant is, therefore, a very important problem and a change in the kind of lubricant used may often result in the large increase in the economy of operation of the engine. The choosing of the proper lubricant to use in any particular bearing is not a very simple matter, as there is such a large variety of lubricants on the market. The selection of a lubricant for any particular purpose should, therefore, be undertaken only after the fundamental principles of lubrication and the necessary qualities of the lubricant are thoroughly understood.

Lubrication.—If you should look through a microscope at the surface of a polished shaft it would appear to be rough, even though as far as the naked eye can see, the shaft is perfectly smooth. After looking at such a surface through a microscope and seeing its roughness, it will be realized that when two such surfaces are brought together the elevations and depressions of one surface will interlock with those of the other surface, as shown magnified in Fig. 158, and that considerable force will be required to move one surface over the other. In other words,

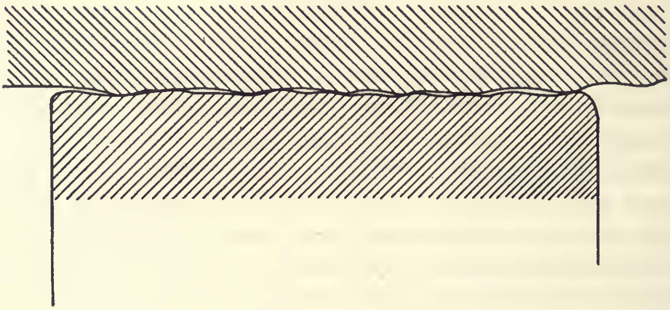


Fig. 158.

there will be considerable *friction* between the two surfaces. The friction in this case would arise from two sources: first, from moving the irregularities on the surface of the journal into and out of the irregularities on the bearing surface, that is, the friction due to the unevenness of the metal surfaces; second, the friction due to the cutting action of the metal surfaces upon one another.

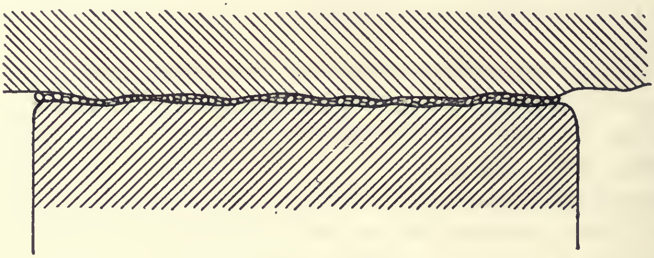


Fig. 159.

Even though the irregularities in the metal surfaces are invisible to the naked eye, it will be realized that the amount of friction between the surfaces is enormous, especially if the bearing supports considerable weight or runs at high speed.

Principles of Lubrication.—The object of lubrication is to place a thin film of oil between the metal surfaces so they will not come in contact with each other while running. If this object is accomplished the metal surfaces will not touch but the journal will "float on oil" and the friction will be enormously reduced. This condition is illustrated in Fig. 159 which shows the two surfaces of Fig. 158 but with a film of oil between them.

In order to secure a film of oil between bearing surfaces some provision must be made so that the surfaces will not scrape off the film of oil but rather that the moving surface will be made to ride upon the particles of oil. When the surfaces to be lubricated are

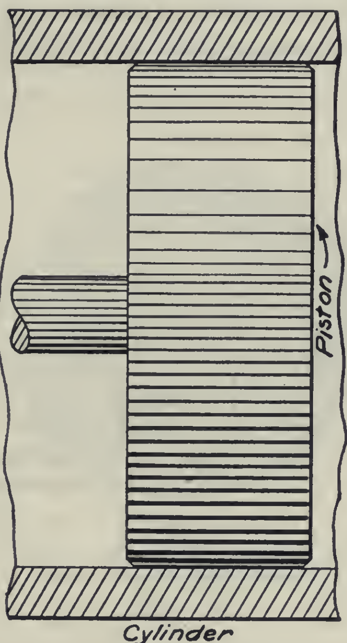


Fig. 160.

flat, as is the case with the crosshead or piston of an engine, the ends of the crosshead or piston should be slightly beveled so that there are no sharp edges. If the edges are left sharp the film of oil will be scraped off and there will be metallic contact between the bearing surfaces. This is illustrated in Fig. 160 which shows the end of a piston slightly beveled so that oil may collect under it and keep the surfaces apart. For the same reason, the edges of piston rings should be slightly beveled instead of left with sharp edges.

With a round shaft which turns in a bearing, such as the main bearing of an engine, the same result is automatically secured through the fact that the bearing which surrounds the shaft always has a slightly larger diameter than the shaft. This leaves a small clearance at the top and sides of the bearing, as illustrated in Fig. 161, which is somewhat exaggerated in order to show the clearance more plainly. The clearance at the sides of the bearing permits the oil to be drawn in between the journal and bearing and thus form a film between them. The oil adhering to the surface of the journal not only causes it to be drawn into the bearing but also causes it to be drawn out at the other side. With large bearings which are supplied with oil at the

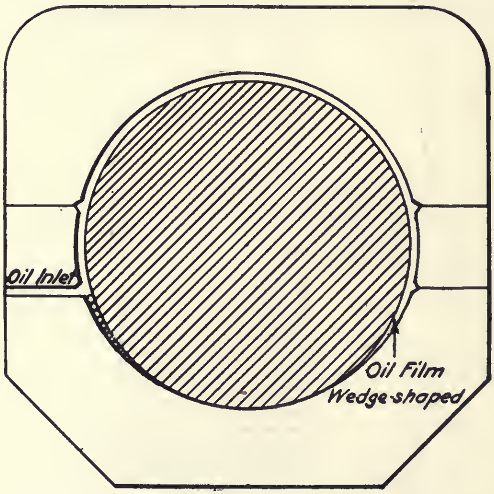


Fig. 161.

center of the top it is necessary to provide diagonal oil grooves in the bearing surface so that the oil may be spread to all parts of it.

There will always be some friction, even with a well-lubricated bearing in which a film of oil is maintained, but in this case the friction is largely a fluid friction instead of the friction of metallic surfaces in contact. When the oil has been carried into the bearing it comes in contact with the bearing surface and adheres or sticks to it. Since the bearing is stationary, the particles of oil next to its surface will also be stationary and the particles of oil which are next to the revolving journal will be in motion due to these particles clinging to the journal. The velocity of the particles of oil in the film, therefore, varies all the way from no velocity on one side of the film to the velocity of the journal on the other side of the film. It is this relative motion of the particles of oil among themselves that is largely the cause of friction in a well-lubricated bearing, and this friction is more of a fluid friction than a friction between metal surfaces.

Characteristics of Oil.—A journal picks up a film of oil and carries it into the bearing by reason of the adhesiveness of the oil, that is, its ability to adhere to the surface of the journal. The quantity of oil that will be drawn into the bearing in this way depends upon the *viscosity* of the oil. A homely illustration of this action is the following: If you stick your finger into a cup of thick molasses and then withdraw it, a large quantity of the molasses will adhere to your finger and be withdrawn with it. If you try the same with a cup of water, only a very small quantity of water will adhere to your finger and be withdrawn with it, because the water is so much thinner and more fluid than the molasses. When an oil is thick or does not flow readily it is said to be viscous, or its viscosity is high. If an oil is thin or flows readily its viscosity is low. In other words, the viscosity of oil refers to its "body."

Since the work which a lubricating oil is called upon to do is keeping the metal surfaces of the bearing apart by means of a film of oil, the conditions under which the oil is to be used determine the proper viscosity or body which the oil should have. These conditions are the speed of the moving parts, the weight on the bearing and its running temperature.

If an oil is too low in viscosity it will permit metallic contact between the surfaces to be lubricated, with a consequent loss of power, and wear on the bearing. On the other hand, if an oil too high in viscosity is used, there will be unnecessary fluid friction, with a consequent loss of power. The more viscous the oil, the greater the pressure which can be sustained without metallic contact. The viscosity of the oil should, therefore, be in proportion to the pressure on the bearing. Heavy, slow-moving bearings require an oil of high viscosity; light, swift-moving bearings require a thin oil, or an oil of low viscosity.

The viscosity of an oil is different at different temperatures; therefore, in selecting an oil to be used, the temperature at which the bearing runs must be considered. Most of the friction in a well-lubricated bearing is internal or fluid friction, as explained before, and this friction causes the temperature of the bearing to be somewhat higher than the room temperature. Since a heavy oil has more internal friction than a light one, the viscosity should be only high enough to maintain the film of oil at the temperature at which the bearing runs.

Testing Oils.—From the above discussion of lubrication it is apparent that one of the most important qualities of a lubricating oil is its viscosity. The measurement of viscosity of lubricating oils is in a certain sense unsatisfactory because the results obtained with the different instruments which are available for this purpose do not agree among themselves. For this reason the make of instrument used in making the test should be stated in quoting the viscosity.

One of the most common instruments used in testing viscosity is the Saybolt viscosimeter. It consists of a tall pipette of small diameter surrounded with a jacket which may be used for maintaining the oil at any desired temperature during the test. The test is made by filling the pipette to a certain point with the oil whose viscosity is to be measured and noting the time in seconds which the oil takes to run out of the pipette. If it is desired to find the specific viscosity, that is, the viscosity of the oil compared with that of water, this may be done by dividing the time which the oil takes to run out by the time it takes an equal volume of water to run out. It should be particularly noted that the viscosity of oil varies greatly with its temperature. The temperature at which the test is made should, therefore, be stated and, to be of practical use, the temperature during the test should be as near as possible to the temperature at which the oil is to be used.

Gumming Test.—A gumming test is of importance because it indicates the extent to which the oil has been refined and also the extent to which the oil may be expected to change, due to oxidation, when in use.

The gumming test may be made by putting a small quantity of the oil to be tested in a small glass vessel, such as a cordial glass, and then mixing with it an equal quantity of nitrosulphuric acid. A properly refined oil will show little if any change, but a poorly refined oil will show the separation of large quantities of material of dark color, due to the oxidation of tarry matter contained in the lubricant. Oils which contain a large percentage of tar absorb the most oxygen and are, therefore, mildly drying.

Flash and Fire Tests.—These tests have nothing to do with the lubricating qualities of an oil but they give an indication of its safety and for this reason they are important.

The flash test is made by heating the oil slowly in a vessel surrounded by a proper bath and noting the temperature at which a flame passed over the surface of the oil will ignite the vapors arising from the oil. The fire test is made by continuing to heat the oil slowly and noting the temperature at which the vapors, when ignited, will continue to burn.

Acid Test.—During the process of refining, oil is agitated with sulphuric acid for the purpose of removing the tarry matter, and this acid must be practically all removed before the oil is put on the market as any free acid in it is apt to corrode the bearings in which it is used.

Acid may be readily detected by thoroughly mixing a small quantity of the oil with an equal amount of pure warm water. The mixture should then be tested with neutral litmus paper. If acid is present, the paper will immediately turn red. The sensitiveness of the paper may be greatly increased by introducing it into the fumes of nitric or hydrochloric acid until it becomes partly red. After it is dried and part of its surface is immersed in the oil to be tested the very smallest amount of acid will turn the part of the paper immersed a shade darker than the rest.

It should be appreciated by the practical man that the tests of lubricating oils give only an approximate idea of the qualities of the oil. In fact, no rigid directions can be given for the choice of an oil for a specific purpose. It is best to try various lubricants which can be purchased for any given lubricating problem until one is found which gives satisfactory results under actual working conditions. This should then be completely tested and the results used in buying oil for the same purpose in the future.

Steam-Engine Lubrication.—The lubrication of a steam engine presents two distinct and separate problems. First, there is the problem of lubricating such bearings as the main bearings, crank and wrist pins, and other external bearings. The principles governing the lubrication of these bearings have been discussed in the first part of this chapter. Second, there is the problem of lubricating the internal bearings, such as piston, valves, piston rod, and valve rod.

The problem of lubricating the internal bearings, especially the piston and valves, is far more difficult than that of lubricating the external bearings. The external bearings are more easily inspected to determine if they are being properly lubricated. If they receive too little oil the temperature of the bearing becomes too high; if they receive too much oil this can be told by the overflow and waste of oil from the bearing. The bearing surfaces of the piston and valves are not so easily inspected, nor is the fact easily detected that too little or too much oil is being supplied or that the oil is being properly distributed. In general, a light oil or oil of low viscosity is used in the external bearings, while the internal bearings require a heavy oil or oil of high viscosity. Cylinder oil should also have a high flash and burning point in order to prevent it from being carbonized by the high temperature in the cylinder. This is especially true when the engine is supplied with superheated steam.

The best method of carrying oil into a steam-engine cylinder for internal lubrication is to introduce it into the steam line supplying the engine. This is accomplished by connecting the feed pipe from the lubricator to the steam line, allowing the feed pipe to extend into the center of the steam line. By doing this, the cylinder oil, coming in contact with the column of steam flowing through the steam line at its point of greatest velocity, is broken up into very small particles and carried along with the steam into the cylinder. Since the steam comes in contact with all internal surfaces requiring lubrication, this method insures a supply of oil to all rubbing surfaces. The point at which the feed pipe enters the steam line should be on the boiler side of the main stop or throttle valve so the spindle of the valve will be lubricated, thus making easy the operation of the valve. However, the feed pipe should not enter at such a point that the spraying or atomization of the oil will be affected by a steam separator or angles in the pipe between the point of introduction and the steam chest, and the oil should be introduced at a point not further than about twelve feet back of the throttle valve.

Superheated steam, which is very dry and hot, is not a good carrying medium for cylinder oil, and for this reason the oil should be introduced just back of the throttle valve when superheated steam is used.

Steam is sometimes used for jacketing the cylinder and heads of the engine before entering the cylinder. Under these conditions some of the oil will be deposited on the walls of the passages if it has been introduced into the steam line in front of the throttle valve. To overcome this difficulty oil is introduced directly into the valve chambers when the current of steam passing through the valves will break up the oil into fine particles and distribute it.

Lubricators.—Cylinder oil is forced into the steam line by means of a lubricator. These are of two kinds: hydrostatic lubricators and mechanically operated lubricators.

A hydrostatic lubricator is illustrated in Fig. 162, being shown partly cut away so that its internal construction and operation may be understood. The body of the lubricator contains cylin-

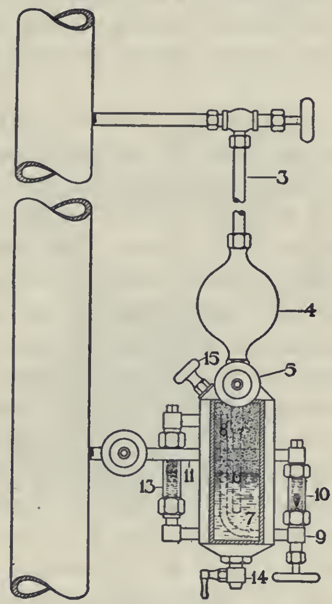


FIG. 162.

der oil and water, the oil remaining on top of the water because it is lighter. Water is introduced into the lubricator from the condensation of steam in the pipe 3 which is connected to the steam line and the bulb 4. It then passes into the bottom of the lubricator through the tube 6, where it exerts an upward pressure on the oil. This pressure forces the oil out of the lubricator through the tube 8 which ends near the top of the lubricator where it will be supplied with oil as long as there is any in the lubricator. The oil passes through the tube 8 to the regulating valve 9 where its rate of flow to the cylinder is controlled. It forms into drops as it passes through the regulating valve and these drops rise through the water in the sight feed glass 10 and through the pipe 11 into the steam line. As the oil is fed into the steam line, its place is taken by more water which enters through tube 6; thus as the amount of oil in the lubricator decreases, the level of the water rises. The amount of oil remaining in the lubricator is indicated by the gage glass 13. Since the lubricator is connected to the steam line at two points, the steam pressure at these two points balance each other, and the pressure which forces the oil out comes from the weight of the column of water in the tube 6 and the pipe 3. The length of the pipe 3 should be at least 18 inches so as to provide sufficient condensing surface and a column of water high enough to force the oil out of the lubricator.

When the level of the water in the lubricator reaches a point near the top of the gage glass 13, the lubricator should be refilled with oil. This is done by first closing the valve 5 and the valve on the feed pipe 11 in order to cut the steam pressure off of the lubricator at both top and bottom. The filling plug 15 of the lubricator may then be opened, but it must never be opened while there is steam pressure on the lubricator, as the hot oil will then be blown out and may cause serious injury to persons standing near; for this reason every precaution must be taken to close valve 5 and the valve on the feed pipe 11, before the filling plug 15 is removed. The drain cock 14 may now be opened and enough water drained out of the lubrictaor until its level is near the bottom of the gage glass 13. The oil may then be poured in through the filling hole at the top until the lubricator is full, after which the filling plug 15 is replaced.

In order to start the lubricator operating again it is only necessary to open the valve 5 and the valve on the feed line 11, and then regulate the flow of oil by means of the regulating valve 9.

The hydrostatic lubricator, described above, is not automatic in its action, since it must be started and stopped by hand. It is difficult to maintain a constant feed, especially at a slow rate, because the feed is affected by the changes in viscosity of the oil due to changes in temperature. This is particularly true just after the lubricator has been refilled.

These disadvantages are overcome in the mechanically operated lubricator, one form of which is illustrated in Fig. 163. This type of lubricator is constructed with single and multiple feed delivery pipes, the number of feeds being limited to the number of points of application demanded in each case.

In Fig. 163 the oil reservoir is shown at 1, the level of the oil being indicated by a glass gage which is not shown in the illustration. The oil reservoir is provided with a strainer so that fresh oil poured into it will be strained and any small particles of solid matter which might clog the small oil passages strained out. The oil plunger 2, draws oil into the pump on the suction stroke and discharges the oil through the nozzle 3 on the delivery

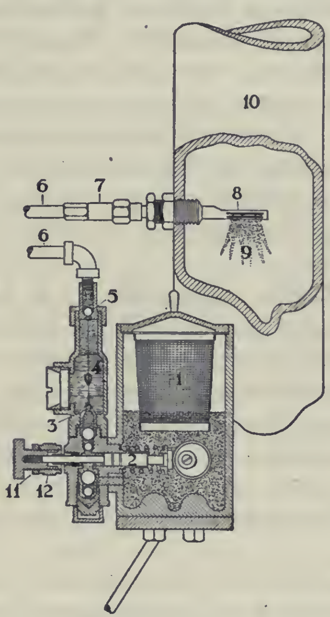


Fig. 163.

stroke. The oil drops form around a guide wire 4 and rise through water in the sight feed. They then pass a nonreturn valve 5, and are forced through the check valve 7, at the extreme end of the oil pipe 6, into the atomizer 8, located in the steam pipe 10.

By means of the adjusting nuts 11 and 12, which change the stroke of the pump plunger, the oil supply can be varied from the smallest amount, say one drop in ten minutes, to practically the full capacity of the pump stroke.

An atomizer, such as shown at 8 in Fig. 163, is the most efficient way of spraying oil into and thoroughly mixing it with the steam. The atomizer is spoon-shaped on its upper side and has slots extending through it. The steam strikes against the spoon-shaped upper surfaces and forces the oil through the slots with great velocity, breaking it up into very fine particles which are thoroughly mixed with the steam flowing through the pipe.

The atomized oil is distributed in the form of a uniform coating or film over the cylinder walls, valve seats, and piston rod, and lubricates these surfaces in a most economical manner.

Mechanically operated lubricators have several advantages over the hydrostatic lubricators. The oil is fed only when the engine is running, as the lubricator pump is operated from the engine and starts and stops with the engine. This is more positive and reliable and a more uniform feed is maintained.

Certain types of mechanical lubricators, however, are not suitable for compound condensing engines operating on a light load, unless the check valves on the discharge side of the pump are spring loaded, as a vacuum acting on the oil pipe with atmospheric pressure on the reservoir will syphon the oil from the reservoir.

Lubrication of Valves. Slide Valve.—As a slide valve is positively operated, it can theoretically be operated at any speed; but in actual service, due to its unbalanced construction, it is not operated at as high speed as the piston valve.

The flat surface of the slide valve which rubs against the valve seat is difficult to lubricate, particularly when the slide valve is large. In some extreme cases, oil grooves are cut in the valve or in the valve seat to assist in spreading the oil all over the frictional surfaces.

The use of the slide valve is restricted to a maximum steam pressure of about 120 lb. or a maximum steam temperature of about 450° F. This is due to the large, flat, frictional surfaces of the slide valve and its seat, and the difficulty of introducing the oil thoroughly between them, owing to the great pressure on the valve. Excessive steam pressure prevents the formation and maintenance of the oil film, resulting in metallic contact of the rubbing surfaces, with excessive friction and wear. Excessive steam temperature will result in unequal expansion and distortion of the valve and valve seat. Steam will leak past the valve seat, causing cutting of the surfaces. As the oil film is thinned out, due to the high temperature, it will be unable to resist the pressure between the frictional surfaces, and metallic contact and wear will follow.

Upon removing the cover from the steam chest for inspection, excessive friction of the slide valve is always indicated by a dryness of the rubbing surfaces, which will show wear and bright streaks of cutting. Wear on the valve seat will be reduced if the cast iron of the valve seat is slightly harder than that of the valve.

Improper lubrication results in abrasion and cutting; excessive leakage of steam takes place and wipes away the lubricating oil from the valve seat, making necessary an increased consumption of oil. Friction of the slide valve often produces groaning during operation, and the excessive resistance in moving the valve causes the eccentric rod to vibrate. With efficient lubrication the valve operates without noise; the eccentric rod works smoothly; and, when inspected, the friction surfaces of the valve and seat have a polished, dull, glossy appearance.

Experience has shown that when the oil is introduced into the steam line and atomized, it is most thoroughly distributed. In many cases, atomizing the oil has furnished the lubrication which has overcome groaning and other troubles with slide valves where direct methods of application have given insufficient results.

Corliss Valves.—Although the Corliss valves have a rotating motion and the rubbing surfaces are cylindrical, conditions of steam pressure and temperature affect their lubrication in the same manner as with slide valves. There is this difference, however, that insufficient lubrication will cause the sluggish closing of the admission valves, since they are not connected directly to the operating mechanism during the closing period. Otherwise, excessive friction is indicated by groaning and vibration of the valve and operating mechanism, as in the case of slide valves.

Corliss valves should be lubricated by introducing the oil into the steam line and atomizing it, as it will then be quickly carried to the rubbing surfaces. Introducing the oil directly over the valves is wasteful or inefficient or both.

Piston Valves.—On account of the symmetrical shape of the piston valve and sleeve there is but little pressure between them. This shape also permits them to expand uniformly under high steam temperature. This type of valve is, therefore, well adapted to running at very high speeds. The large surface moved over by the piston valve at high speeds demands effective lubrication, which is best secured by the atomization method.

Poppet Valves.—Poppet valves have no sliding motion and, therefore, do not require lubrication, but the valve stems which move up and down in guides require a small amount of lubricant. External lubrication of the valve stems is likely to cause them to stick, owing to the fact that the clearance between guide and valve stem is very small. For this reason, atomizing the oil and using it sparingly will give best results.

Piston and Cylinders.—In the lubrication of steam-engine cylinders we have to consider two types of engines, vertical and horizontal. In vertical engines the pressure between the piston and cylinder is moderate, being due almost entirely to the springing action of the rings. For this reason less oil is required for lubrication than with horizontal engines. In horizontal engines, the lower part of the cylinder carries the weight of the piston in addition to the pressure of the rings. Some large horizontal engines have tail rods which, together with the piston rod, carry the major portion of the weight of the piston. The duty of the piston rings, then, is simply to prevent leakage of steam. The pressure of the piston has an important bearing on cylinder lubrication because the oil film is at a high temperature and excessive pressure may easily destroy it, resulting in friction and wear.

The inside of a properly lubricated steam-engine cylinder will have a dull appearance due to the presence of a film of oil. The presence of the oil can be detected by wiping the surface with a piece of white paper, the stain left by the oil having a brownish color. If the stain is black it is an indication that the oil is being carbonized. When the film of oil has been wiped away the surface underneath should appear dull and glossy. If wear has taken place it will be indicated by the surface being bright and there will usually be streaks or scratches also.

Piston and Valve Rods.—Piston and valve rods are always provided with stuffing boxes to prevent leakage of steam out of the cylinder or valve chest, and, in the case of low-pressure cylinders, to prevent the leakage of air into the cylinders when the engine is operated condensing.

Stuffing boxes are packed with either soft or metallic packing. Full and efficient lubrication of the packing is essential as a perfect seal can only be obtained by the presence of a complete oil film on the rods.

Soft packings are used only under moderate steam conditions. The friction between the packing and the rod is always comparatively high, and if the packing is screwed up too tight, it causes grooving or scoring of the rod, and it is then difficult to prevent leakage of steam.

In reversing engines, the engine is reversed by changing the movement of the valves with relation to the position of the pistons. This is done by hand with small engines and by a special reversing engine in the case of large engines. In either case the pull required to reverse the engine depends upon the friction of the valves moving over their seats and by the friction of the rod moving through the stuffing box. Where the valve rods have been lubricated externally by the direct method, which is wasteful and inefficient, a change to the atomization method effects a marked improvement. The valve rod receives lubrication when inside the valve chest and furnishes efficient lubrication to the packing. As a result, external lubrication of the valve rods can be dispensed with.

Metallic packing is much superior to soft packing and it should be used where the steam temperature is high. The friction of metallic packing is less than with soft packing as it exerts only a slight pressure, and there is less danger of scoring the piston rod with it. With highly superheated steam it is usually necessary to supply a small amount of direct lubrication to metallic packing in addition to that supplied by the atomization method, but oil used in this way should be used sparingly as an excess of oil may clog the packing or become carbonized. With only moderate superheat, direct lubrication can usually be dispensed with.

Influence of Operating Conditions.—Stationary steam engines usually operate at constant speed, the changes of load being taken care of by changing the volume of steam admitted to the cylinder or by changing its pressure. Either will reduce the velocity of steam flow through the steam main at reduced loads. As this will affect the atomization of cylinder oil it should be considered in selecting the oil. With full velocity of the steam, heavy oils will be readily atomized, but a low velocity of steam requires a lighter oil in order to secure thorough atomization and good distribution.

The speed ot the engine also affects the lubrication. High-speed engines, making short quick strokes, take in only a small volume of steam at each stroke, so that only a light-bodied, quick-acting oil will give efficient lubrication.

If the supply of steam to the engine is wet it makes efficient lubrication more difficult because the wet steam has a tendency to wash the film of oil off the surfaces of the cylinder. Oil to be used with wet steam should have great endurance and must be of such quality that it will readily combine with moisture and cling to the cylinder walls.

With compound and triple-expansion engines, even when the supply of steam is dry, the drop of pressure causes condensation so that the supply of steam to the following cylinders will be wet. It is, therefore, necessary sometimes to use one grade of cylinder oil for the high-pressure cylinder and a different grade for the low-pressure cylinder in order to secure efficient lubrication.

The use of highly superheated steam requires the highest quality of oil as the friction and high temperature will carbonize and decompose poor quality oils. If the engine operates under fairly heavy load a heavier-bodied oil may be used than when the engine operates at comparatively light loads. Under the latter condition a medium-bodied oil is best.

Low-grade mineral and compounded cylinder oils are difficult to separate from the exhaust steam. The best grades of mineral oils separate more easily from exhaust steam and feed water than do compounded oils; it will, however, be found that more oil will be required to give effective lubrication when using a mineral oil than when using a compounded oil.

CHAPTER XX

STEAM TURBINES

General Principles.—The principles of the steam turbine and the operation of steam in it are entirely different from the principles of the steam engine and the operation of steam in it. In a steam engine some of the moving parts have a reciprocating motion and the action of the steam is intermittent, while a steam turbine is an apparatus in which the moving parts have only a rotating motion and the steam which passes through it acts in such manner as to produce a constant angular velocity.

In order to illustrate in a homely way the difference between the action of a steam engine and that of a steam turbine, consider a large wheel supported in a horizontal plane on a vertical shaft. The wheel may be rotated by grasping the rim and walking continuously around the shaft. In a similar manner the pressure of the steam acts upon the piston of a steam engine and pushes it forward. The wheel may also be turned by standing in one spot and grasping the rim and moving it with first one hand and then the other in a similar manner to opening or closing a large valve by hand. In this case the wheel is not turned by exerting a pressure at one point on its rim, as in the previous case, but by exerting a pressure at first one point on the rim and then another so that all points on the rim are used successively. This is the manner in which steam acts in a steam turbine.

In order to make the wheel turn while standing in one spot we must have a firm place to stand on and a good grip on the floor in order to be able to exert the required pressure on the wheel. This means that in every turbine there must be certain stationary parts which are firmly fixed to the casing in order that the steam may have a good grip on the moving or revolving parts which are fastened to the power-transmitting shaft.

Although we can readily understand how we may turn the wheel by standing in one point, it is not so easy to understand how steam, which is a flexible medium, can grip the wheel of a turbine in such manner as to turn it.

There are two methods by which this may be done and these will be illustrated by the following examples in which water is used instead of steam, remembering that the actions taking place with steam will be the same as those taking place with water.

If a hose nozzle be directed against a board arranged as shown in Fig. 164, the water flowing from the nozzle will exert a pressure on the board as indicated by the scale at the left. The pressure exerted upon the board will depend upon the velocity of the water and upon the angle at which the water strikes the board. If the stream of water is directed almost parallel with the board it will exert only a small pressure but if it is directed so that it strikes

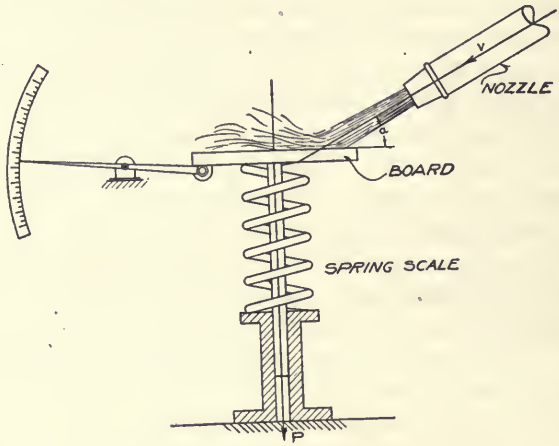


FIG. 164.

the board almost perpendicular to its surface it will exert a much larger pressure.

With the arrangement of the nozzle and board shown in Fig. 164 it will be appreciated that the stream of water will break and cause it to splash. This reduces the pressure which the stream of water can exert. If, however, the board is curved, as shown in Fig. 165, the water will pass over its surface in a smooth stream without splashing and at the same time the nozzle may be directed so as to produce the maximum pressure. The pressure exerted by the stream of water is due entirely to changing the direction in which the stream is flowing. The amount or intensity of the pressure will depend upon the velocity of the stream, upon the weight of the water (or steam) and upon the angle through which it is turned.

The velocity of the stream when it leaves the board will be practically the same as its entering velocity, the only loss of velocity being that due to the friction of the water passing over the surface of the board; this is small if the board is smooth. Moreover, if the water leaves the board at the same angle at which it strikes, it, that is, if the curvature of the board is uniform, the pressure will be directed along the axis of the board as indicated by the arrow, and there will be no side thrust.

The above discussion has shown how a stream of water directed along a curved surface can produce a pressure, but in the case considered, no work was done because the curved board does not move. In order for work to be done, a force must act or move

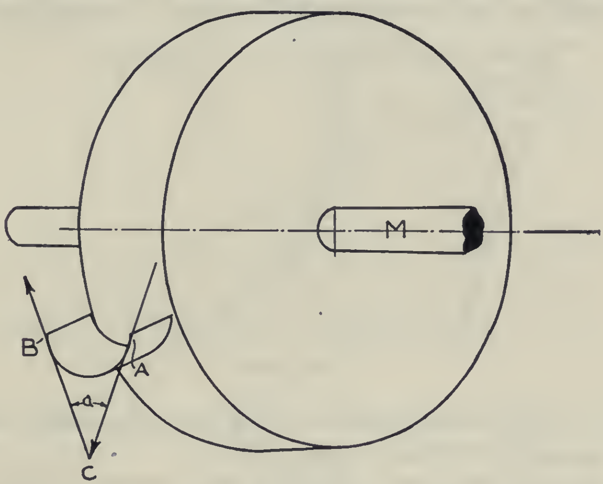


Fig. 165.

through a distance. The product of the force and the distance will then be the foot-pounds of work done. If the curved board is fastened to the rim of a wheel, as shown in Fig. 165, and the wheel is fastened to an axle, then the pressure on the board due to the stream of water will cause the wheel to turn and to do work. If, instead of only one of the curved boards, or blades as we will now call them, a number of blades are placed around the circumference of the wheel so that as soon as one blade has moved out of the stream of water another will move into it, a continual pressure will be exerted and the wheel will revolve uniformly. Also, instead of only one nozzle and stream of water, there may be several nozzles spaced at intervals around the circumference so that instead of the vanes receiving pressure from one nozzle they may receive pressure from several nozzles and thus increase the total pressure acting upon the rim of the wheel. This constitutes a turbine similar to one type which is in common use, a type known as the impulse turbine.

In discussing the action of the stream of water on the curved vanes it has been assumed that the vanes were standing still. It remains now to discuss the effects produced by the movement of the vanes, because this is the condition that will exist when the turbine is running. This is important because the pressure on the vane depends upon the velocity with which the water is moving over the surface of the vane and this velocity is affected very much by the movement of the vane. In other words, the

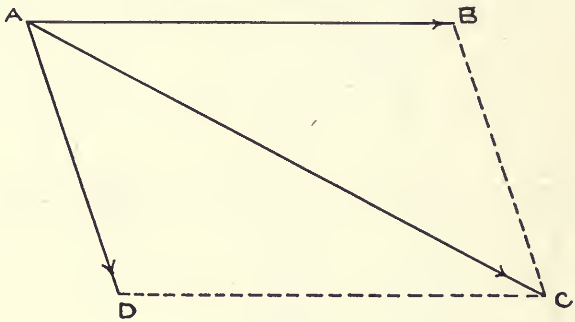


FIG. 166.

velocity of the stream relative to the vanes is entirely different from the absolute velocity of the stream.

To illustrate this and to show how the velocity of the stream relative to the vane, when the vane is moving, may be determined, consider the case of a person walking across the floor of a railway coach when the coach is moving. Referring to Fig. 166, suppose the train is moving forward with a velocity represented by the length of the line AB and that the direction of this line also represents the direction in which the point A on the floor of the coach is moving. Now suppose a person standing at A walks across the coach in the direction AD and walks a distance, measured on the floor of the coach, represented by the line AD, in the same length of time that the train covers the distance AB. Then the actual motion of the person with respect to the ground is represented in direction and length by the line AC which is the diagonal of the figure $ABCD$. That is, the line AB represents in length and direction the absolute velocity of the train with respect to the ground, the line AC represents the absolute velocity of the person with respect to the ground, and the line AD represents the velocity of the person relative to the train.

Applying the same kind of diagram to the case of a stream of steam passing over the surface of a vane of a steam turbine we would have a figure similar to the one shown in Fig. 167. In this diagram the curved line AG represents the curved surface of the vane. The line AD represents the velocity of the vane as the wheel turns, the line AE represents the velocity of the steam as it leaves the nozzle and strikes the vane, and the line AF represents

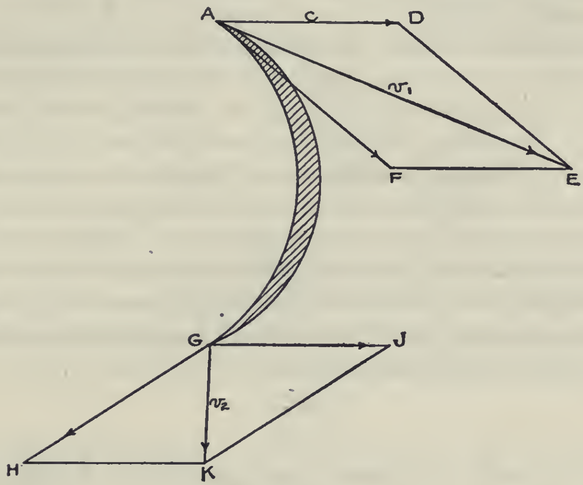


Fig. 167.

the velocity of the steam with respect to the surface of the vane. That is, instead of the entire velocity of the steam AE being effective in creating pressure against the vane, there is only the velocity AF to produce this pressure.

Consider next what happens to the velocity as the stream leaves the vane. This is shown by a similar velocity diagram drawn about G, the point at which the steam leaves the vane. As the steam passes over the surface of the vane, its direction is changed so that if the vane were standing still it would leave in the direction GH, tangent to the vane. The line GH therefore represents the velocity of the steam with respect to the vane. The line GH will have practically the same length as the line AF because the relative velocity of the steam with respect to the vane will be practically the same at the outlet end of the vane as at the inlet, the only loss in this velocity being that due to the friction of the steam on the surface of the vane. This is small because the vanes are made as smooth as possible. The line GJ represents by its length and direction the velocity with which the vane is moving, and it is, of course, the same as the line AD. The diagonal of the diagram, or the line GK therefore represents by its length the absolute velocity with which the steam leaves the moving vane and its direction shows the direction in which the steam leaves the vane. That is, the steam enters the moving vane with an absolute velocity of AE and leaves it with an absolute velocity of GK.

Having considered the velocities involved in the operation of a steam turbine, we are in a position to study the amount of energy obtained from these velocities. If we determine the amount of energy in the steam entering the vane and subtract from this the amount of energy in the steam as it leaves the vane, the difference must evidently be the amount of energy given to the vane. This represents the energy developed by the turbine wheel, if we neglect the small loss of energy that takes place in the vanes.

If a weight of G pounds of working fluid (steam or water) enters the vane per second with a velocity of v_1 feet per second, its kinetic energy or energy of motion is

$$\frac{Gv_1^2}{2g}$$

in which formula g represents the acceleration due to gravity, 32.2 ft. per sec. per sec.

Also the kinetic energy of the working fluid as it leaves the vane is

$$\frac{Gv_2^2}{2g}$$

The difference between these two quantities is the energy given to the wheel, W, and the loss L, or

$$\frac{Gv_1^2}{2g} - \frac{Gv_2^2}{2g} = W + L$$

or

$$\frac{G}{2g}(v_1^2 - v_2^2) = W + L$$

This equation means that we get out of the steam more useful energy W, the greater G is, the smaller the losses, and especially the larger $v_1^2 - v_2^2$ is. With a given inlet velocity the expression $v_1^2 - v_2^2$ will be larger, the smaller the absolute outlet velocity v_2 is as compared with the absolute inlet velocity v_1. It is evident from Fig. 167 that the absolute outlet velocity v_2 will be smallest when the line GK is at right angles to GJ; that is, when the steam leaves the vanes at right angles to the direction in which the vanes are moving. This, therefore, is one of the conditions for maximum efficiency in a steam turbine and the designer chooses the angles and velocities so as to secure this result as nearly as possible.

It can be shown mathematically that another condition for maximum efficiency is that the circumferential speed of the wheel, c, should be approximately one-half of the absolute inlet velocity of the steam, v_1, or

$$c = \frac{v_1}{2}$$

It will thus be seen that the inlet velocity is a very important quantity. Let us see, therefore, what we may expect this velocity to be.

In water turbines the absolute inlet velocity of a simple impulse turbine depends merely upon the height or head of the column of water above the level of the turbine wheel. Suppose this head to be "h" feet, then the theoretical absolute inlet velocity "v" is

$$v = \sqrt{2gh}$$

This means, for instance, that a particle of water flowing down from a height of 100 feet, reaches a velocity of

$$\begin{aligned} v &= \sqrt{2 \times 32.2 \times 100} \\ &= 80 \text{ feet per second, approximately.} \end{aligned}$$

Taking one pound of water in a height of 100 feet above turbine level, we may say that the energy stored up in the pound of water is 100 foot-pounds. In falling through the height of 100 feet this stored-up energy is changed into energy of motion, so that the energy of motion is

$$E = \frac{Gv^2}{2g} = \frac{1 \times 80^2}{2 \times 32.2} = 100 \text{ foot-pounds, approximately.}$$ In

other words, the stored up energy at the top of the column of water is equal to the energy of motion at the foot of the column (100 foot-pounds) 100 feet below, in accordance with the law of constant energy. How is this with a steam turbine. It is exactly the same, but until recently no one thought of comparing steam under pressure in a boiler with water under a very high head, because such a comparison was not necessary in steam-engine practice.

Suppose we consider one pound of steam in a boiler under a pressure of "p" pounds per square inch. What is the amount of energy stored up in this pound of steam expressed in foot-pounds? If we allow this pound of steam to expand to its absolute pressure "p_1," with a volume of V_1 (whereas the volume at the pressure "p" may be V) then the amount of stored up energy L in foot-pounds is

$$L = \frac{pV}{n-1}\left\{1 - \left(\frac{V}{V_1}\right)^{n-1}\right\}.$$

and during this expansion the volume increases according to the law pV^n equals a constant quantity, n being equal to 1.135. Under average boiler and condenser conditions this L is 234,000 foot-pounds. (Steam pressure 150 pounds, vacuum 28 inches.)

To understand clearly this figure, 234,000 foot-pounds, we must bear in mind that the amount of stored up energy in one pound of water with a head equal to that of Niagara Falls is only 150 foot-pounds. The head of one pound of steam in a boiler under 150 pounds pressure against 28 inches of vacuum is therefore $\frac{234{,}000}{150} = 1560$ times as high as that of Niagara Falls, and for such enormous heads steam turbines are designed.

The equation

$$v = \sqrt{2gh}$$

gives us some idea of the velocity this working fluid reaches when it flows down this head

$$\begin{aligned} v &= \sqrt{2 \times 32.2 \times 234{,}000} \\ &= 3880 \text{ feet per second approximately.} \end{aligned}$$

This means that the absolute inlet velocity of a working fluid expanding from 150 pounds per square inch down to 28 inches vacuum would be 3880 feet per second. If the circumferential velocity of the running wheel is one-half of this it would be

$$c = \frac{3880}{2} = 1940 \text{ feet per second}$$

or 116,400 feet per minute
or 1318 miles per hour

Up to the present time no material, not even the best nickel steel, can withstand the enormous strains due to such high centrifugal force.

Unless we allow a considerably greater ratio of inlet velocity to circumferential velocity than two to one we are unable to build a turbine in this way with only one wheel. As soon, however, as we increase this ratio, we have to expect an uneconomically working turbine. Therefore, as it is absolutely necessary to build a turbine so it will be as economical as possible, we have to seek other ways to reach practical limits of circumferential velocity. The best way to do this is shown in the practice of water turbines, as for instance, in Massachusetts the manufacturing industries are using the water power of the Connecticut River

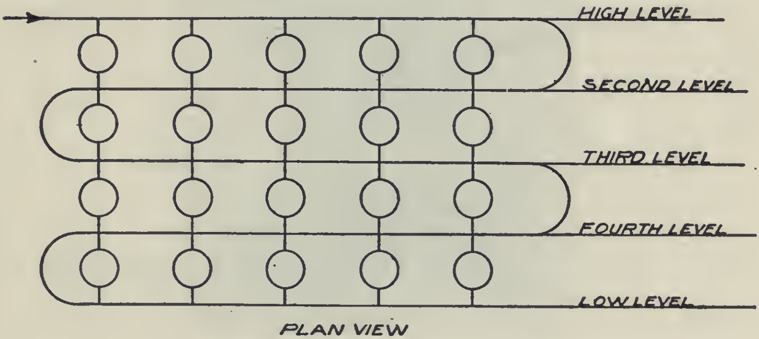


FIG. 168.

in such a way that they lead the power canal in loops as shown in Fig. 168. The circles here represent the water turbines. It is plain that the first row of turbines uses the head between high level and second level; the second row of turbines uses the head between second level and third level, and so on. In short, the whole head between high level and low level is split into a number of fractional heads between intermediate levels.

This arrangement of water turbines indicates plainly the way to use the enormous heads of steam turbines with practical means, and yet follow the fundamental laws as given above. In other words, in order to get an economically working steam turbine with moderate circumferential speeds of the running wheel, we must use a number of running wheels, all mounted upon one shaft, each of these running wheels to be driven by steam flowing through a corresponding number of stationary blades where the steam partly expands and accelerates to a velocity approximately twice the circumferential velocity.

One of the first steam turbines that came into extensive use in this country makes use of the first principle stated above; that is, it has only one running wheel and develops all of its power in this single wheel. This turbine is called the De Laval turbine. It is made in small and medium sizes up to about 350 horsepower but it is not well adapted for large units. Having only one running wheel in which the energy of the steam is utilized, this wheel must necessarily run at a very high circumferential velocity (about 1200 feet per second or 72,000 feet per minute).

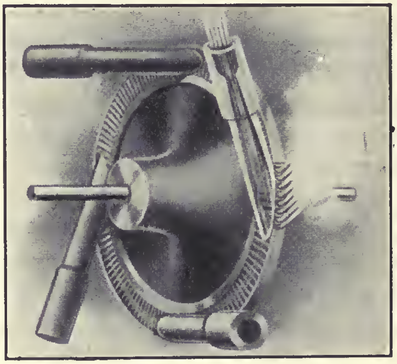


Fig. 169.

An illustration of a De Laval turbine wheel and nozzles is shown in Fig. 169. This illustration shows the casing of the turbine removed so that the construction of the nozzles and wheel may be seen more clearly. Instead of only one nozzle, it will be seen that four nozzles are used to direct the steam into the vanes, as this brings into operation a larger portion of the circumference of the wheel and permits four times as much power to be developed as if only one nozzle was used.

The shape of the nozzles is clearly shown in Fig. 169. This shape is such that the steam is expanded in the nozzle and its pressure energy changed into velocity energy, at the same time causing very little loss of energy in passing through the nozzle. When the steam enters the nozzle, it has a high pressure and small volume. This is changed in the nozzle so that as the steam leaves it, it has only a low pressure but occupies a large volume, and consequently it leaves the nozzle and enters the vanes with a very high velocity. The vanes are spaced uniformly around the circumference of the wheel and they are, of course, deep enough to accommodate the required volume of steam.

A study of the diagrammatic sketch shown in Fig. 170 will give a good idea of the changes of pressure and velocity that take place in a De Laval turbine. The upper part of the sketch

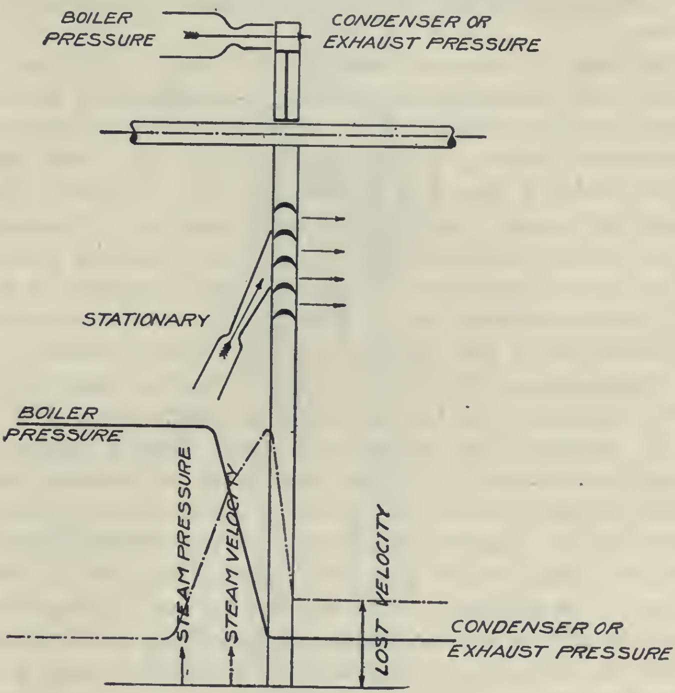


Fig. 170.

shows an elevation of the turbine shaft, the nozzle, and one-half of the wheel. Just below this is shown a plan view of the wheel and nozzle. At the bottom are shown lines representing the steam pressure and velocity in different parts of the turbine, the height of these lines above the base line being proportional to the pressure and velocity. It will be observed that the steam pressure decreases within the nozzle from boiler pressure to exhaust or condenser pressure, that the running wheel revolves in this low pressure, having the least possible resistance, and further, that the steam velocity reaches its maximum at the outlet of the nozzle and decreases within the vanes of the running wheel, thus transmitting the energy of the steam to the turbine shaft, and that it leaves the vanes with a certain lost or unused velocity. This unused velocity is necessary in any turbine to carry the exhaust steam away from the running wheel. The above diagram shows further that the pressure is the same on both sides of the running wheel, and therefore, the axial thrust due to steam pressure is theoretically zero in this turbine.

That class of steam turbines which, instead of utilizing the energy of the steam in a single running wheel, divide it among several running wheels and thereby permit a much lower circumferential velocity, is well represented by the Curtis steam turbine, which is used very extensively in this country. These turbines are made in all sizes from the smallest to the largest and on account of their smaller number of revolutions per minute they can be connected directly to electrical machinery without it being necessary to use a set of reduction gears between the turbine shaft and the shaft of the driven machinery.

A diagrammatic sketch of the Curtis turbine arranged similar to the one for the De Laval turbine, is shown in Fig. 171. It will be observed that the running wheels in this turbine are divided into two sets. Each set consists of two running wheels, marked *R*, and a stationary wheel or set of vanes, marked *S*, between them. Steam is directed against the first set of running vanes by nozzles similar in shape to those used in the De Laval turbine. The stationary vanes between the two running wheels are merely for the purpose of changing the direction of the steam so that the steam may strike each set of running vanes in the same direction.

In the first nozzles one part of the pressure energy is converted into velocity energy. With the velocity corresponding to this partial expansion of the steam the first running wheel is impinged and one fraction of the whole steam velocity is made available in this wheel. Leaving this wheel, the steam is led through intermediate or stationary vanes to the next running wheel, where the remaining part of the velocity due to the first expansion is made available.

The diagram shows the decrease of the pressure from boiler pressure to a medium pressure in the first nozzles, the steam pressure remaining the same in the whole first part of the turbine no further expansion taking place in it.

After having left the last running wheel of the first part of the turbine, the steam is compelled to pass through nozzles in which it expands down to condenser or exhaust pressure thereby increasing its velocity again up to a certain maximum. This

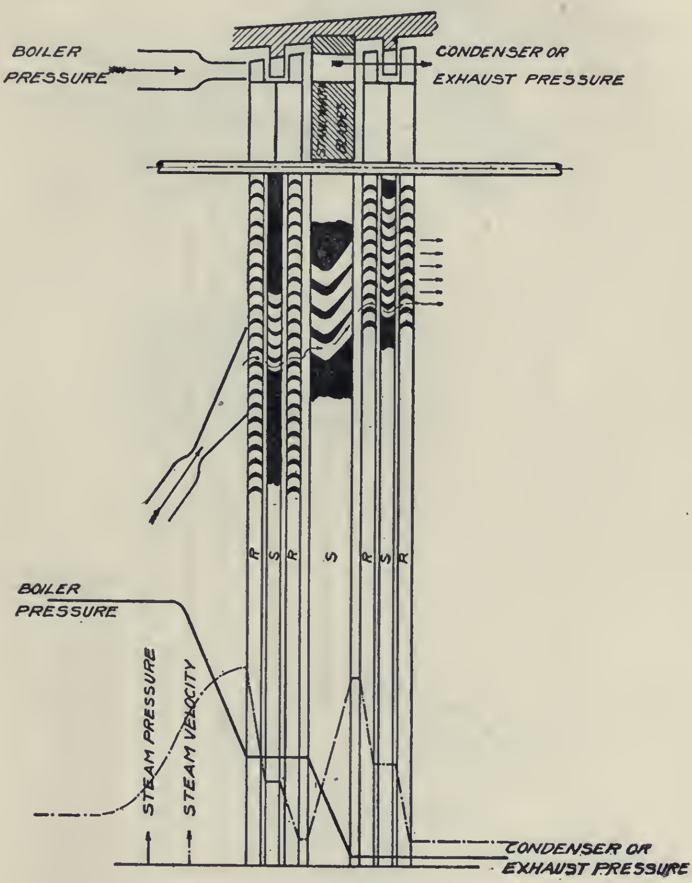


FIG. 171.

steam jet impinges the first running wheel of the second part of the turbine, makes one fraction of its velocity available in it, leaves the wheel with a certain velocity, is led with this velocity through the following stationary vanes and makes the rest of the velocity available in the following running wheel.

The diagram shows the changing steam pressure and steam velocity very plainly. Of course, the number of running wheels and the number of sets into which they are divided depends upon the conditions under which the turbine is to run. In the above figure we merely take schematically two sets of wheels with two wheels in each set to illustrate the principles of the turbine.

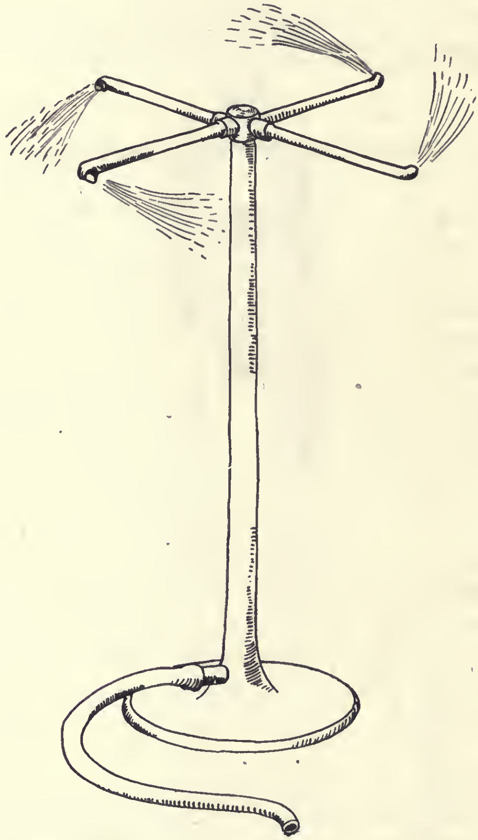


FIG. 172.

The intermediate blades in each stage and the stationary blade separating one stage from another need openings only over a certain part of the circumference. The openings in the intermediate vanes increase in radial height according to the decreased velocity if there is more than one intermediate row in one stage, and the openings of the stationary blades increase n radial height or in the part covering the circumference according to the increasing volume.

In the Curtis turbine, as in the De Laval, the steam pressure is the same on both sides of each running wheel, hence there is no axial thrust along the shaft.

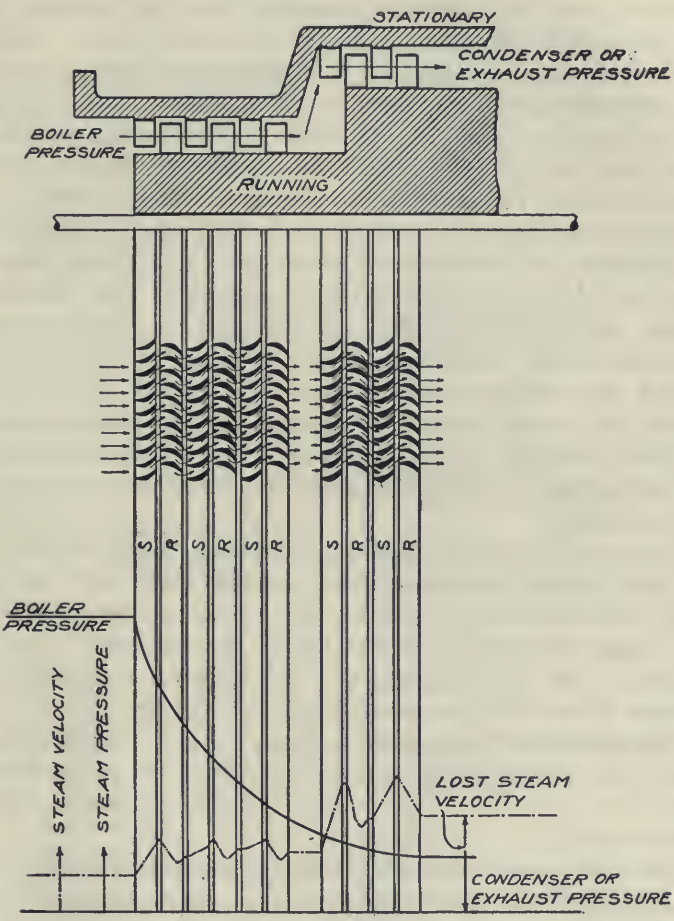


Fig. 173.

The two kinds of turbines described above belong to the general type known as impulse turbines. Another well-known type of turbine operates by both the impulse of the steam against the vanes and by the reaction of the steam as it leaves the vanes.

An ordinary lawn sprinkler such as that shown in Fig. 172 is a good illustration of the force of reaction exerted by a stream of flowing fluid. In this case the streams of water flowing from the ends of the arms of the sprinkler exert a backward push or "reaction" which is sufficient to rotate the arms. In the same way, steam flowing at high velocity from a properly shaped nozzle or vane will produce a reaction. It is this reactive force that is used in the type of turbines mentioned above. One of the best known of this type is the Westinghouse-Parsons turbine, which is illustrated schematically in Fig. 173.

No nozzles are used in this turbine, but the rows of stationary vanes take the place of nozzles and guide the steam into the adjacent rows of running vanes. The outlet opening between the running vanes, being smaller than the inlet opening, the steam is compelled to expand and accelerate within these running vanes exerting thus a back pressure upon them when the steam leaves. In this turbine there are therefore two ways in which the steam moves one vane. First, the impact at the inlet and, second, the reaction at the outlet. On account of the larger angles that must be used in shaping the vanes there cannot be made available in each row such a large fraction of energy as can be made available by using pure impact and considerably smaller angles.

The steam leaving the first row of running vanes flows through the next row of stationary ones, expands there, and impinges with the new velocity the second row of running vanes, and so on, the vanes increasing in radial height in conformity with the expansion line of the steam and the diameter increasing, too, in order to reach higher circumferential velocities.

The schematical diagram shows plainly that the pressure line is nearly one continuous curve from the inlet to the outlet without any offsets or stops. The curve of the absolute steam velocity explains itself.

The next consequence of the shape of the pressure line is that the running wheels have a higher steam pressure on the inlet end than on the outlet end, that is, the whole drum with all the running vanes is thrust axially in the direction of the steam flow. To balance this thrust, balancing pistons are used, keyseated to the turbine shaft and of the same diameters as the different running rows, and being connected by steam channels with the space in which the corresponding running rows revolve, thus balance each running row.

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