STEAM ENGINES

LUDY

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A THOROUGH AND PRACTICAL PRESENTATION OF MODERN STEAM ENGINE PRACTICE

BY

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INTRODUCTION

THE modern steam engine, whether it be the majestic Corliss, which so silently operates the massive electric generators in one of our municipal power plants, or the giant locomotive which pulls the Limited at sixty miles an hour, commands our unstinted admiration. And yet every movement is so free and perfect in its action, every function is performed with such precision and regularity, that we lose sight of the wonderful theoretical and mechanical development which brought these machines to their present state of perfection.

¶ The genius of Watt, the "father" of the steam engine, was so great that his basic conception of this, his greatest invention, and of many of his minor discoveries in connection with it, remain almost as he gave them to the world over a century ago. Yet he was so far in advance of the mechanical development of his time that his workmen could not build engine cylinders nearer true than three-eighths of an inch. Modern builders demand an accuracy of at least two-thousandths of an inch—almost two hundred times greater.

¶ But mechanical skill is not the only particular in which progress has been made. Many minor but important improvements have been brought about by a careful study of the theory of heat engines. The reduction of enormous heat losses, the use of super-heated steam, the idea of compound expansion, the development of valve gears—all have helped to make the steam engine well-nigh mechanically perfect and as efficient as is inherently possible.

¶ The story has been developed from a historical standpoint and along sound theoretical and practical lines. It will be found absorbingly interesting and instructive to the stationary engineer as well as to all who wish to follow modern steam engineering development. If the book should prove of real value in stimulating the interest of the trained man or the layman in the technical developments of the day, the publishers will feel that its mission has been accomplished.

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STEAM ENGINES

PART I

DEVELOPMENT

Early History. In the study of this subject, it is thought advisable to review the historical development of the steam engine in order that a broad conception of it may be obtained. It is not intended, however, to give the history of the steam engine in detail—although it is an exceedingly interesting one, which would be beneficial for any one to review—but rather, a short résumé in order that the student may be prepared for a detailed study of the modern engine.

The first steam engines of which we have any knowledge were described by Hero of Alexandria, in a book written two centuries before Christ. Some of them were very ingenious, but the best were little more than toys. From the time of Hero until the seventeenth century little progress was made. At this time, however, there was a great need of steam pumps to remove water from the coal mines. In 1615, Salomon de Caus devised an arrangement, consisting of a vessel having a pipe leading from the bottom which was filled with water and then closed. When heat was applied to the vessel, steam was formed, which forced the water through the discharge pipe.

Later an engine was constructed in the form of a steam turbine, but was unsuccessful, and the attention of the inventors was again turned to pumps.

Savery. Finally Thomas Savery completed, in 1693, the first commercially successful steam engine. It was very wasteful of steam as compared with engines of today but, as being the first engine to accomplish its task, it was successful. Savery's engine, Fig. 1 consisted of two oval vessels A_1 and A_2, placed side by side and in communication with a boiler B_1. The lower parts were connected by tubes fitted with suitable valves. In operation, steam from the boiler was admitted, say, to the vessel A_2 and the air driven out. The steam was then condensed and a vacuum formed by letting water play over the surface of the vessel. When valve *1* was opened, this vacuum drew water from below until the vessel was full. The valve was then closed and steam again admitted by valve *2*, so that on opening valve *3* the water was forced out through the delivery pipe *C*. The two vessels worked alternately. When one was filling with water, the other was open to the boiler and was being emptied. Of the two boilers B_1 and B_2, one supplied steam to the oval vessels and the other was used for feeding water

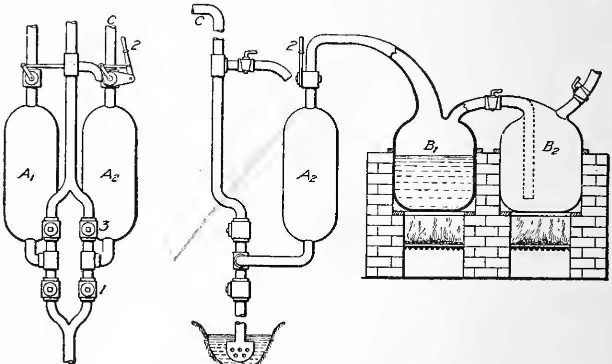


Fig. 1. Early Form of Steam Pumping Engine

to the first boiler. In operation the second boiler was filled while cold, and after a fire had been lighted under it, acted like the vessel used by Salomon de Caus and forced a supply of feed water into the main boiler.

A modification of Savery's engine—the pulsometer shown in Fig. 2—is still found in use in places where an ordinary pump could not be used and where extreme simplicity is of especial advantage. Its valves work automatically and it requires very little attention.

A serious difficulty with Savery's engine resulted from the fact that the height to which water could be raised was limited by the pressure which the vessels could sustain. Where the mine was very deep it was necessary to use several engines, each one raising the water a part of the whole distance. The consumption of coal in proportion to the work done was about twenty times as great as that of a good modern steam engine. This was largely, though not entirely, due to the immense amount of steam which was wasted by condensation when it came in contact with the water in the oval vessels.

Newcomen. The next great step in the development of the steam engine was taken by Newcomen, who in 1705 succeeded in developing a scheme which prevented contact between the steam and the water to be pumped, thus diminishing the amount of steam uselessly condensed. He introduced the first successful engine which used a piston working in a cylinder.

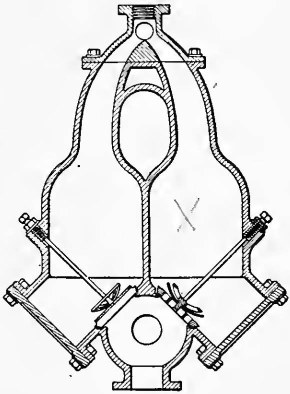


Fig. 2. Pulsometer

In Newcomen's engine, Fig. 3, there was a horizontal lever *A*, pivoted at the center, carrying at one end a long heavy rod *B* which connected with a pump in the mine below. A piston *C* was hung from the other end of the lever and worked up and down in a vertical cylinder *D*, which was open at the top. Steam acting on the lower side of the piston, at atmospheric pressure, was admitted from the boiler to the cylinder, and as the pressure was the same both above and below the piston, the weight of the heavy pump rod raised the piston. A jet of water in the cylinder condensed the steam and formed a vacuum. This left the piston with atmospheric pressure above and very little pressure below (a partial vacuum), so it was forced down and the pump rod raised again. Steam could again be admitted to the cylinder; the pump rod would fall; and the process could be continued indefinitely.

In the days of Newcomen it was very difficult to obtain good workmanship. For this reason it was often necessary to make the cylinders of wood. In order to prevent steam from blowing around the piston, or air from leaking in where steam was being condensed, it was customary to keep a jet of water playing on the top of the piston.

One great trouble with all of these engines was that some one was required to open and close the cocks, and boys were generally employed to do this work. One boy, in order to get time to play, rigged a catch at the end of a cord which was attached to the beam

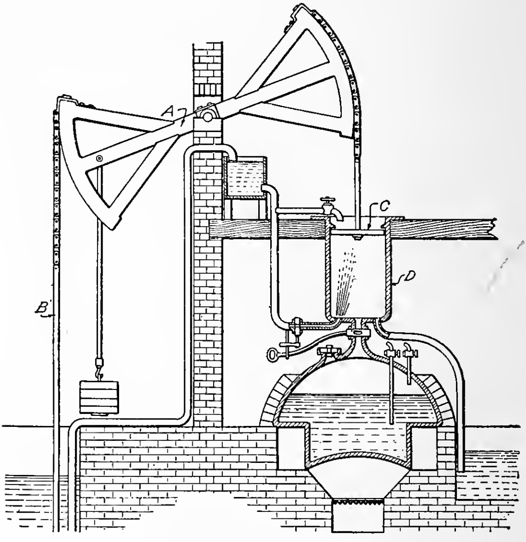


Fig. 3. Newcomen's Steam Pumping Engine

overhead, and this did the work for him. By making the valves in this way automatic, made it possible to dispense with the services of the boy and at the same time greatly increase the speed of the engine.

The Newcomen engine was improved slightly from time to time by different inventors and was very extensively used until the time of Watt, a very few of them still being in existence today. While this engine was a success and a great improvement over its predecessors, it was still very large, wasteful, and heavy in comparison with the work done, and the cylinders, when made of iron, were simply cast and not bored, thus leaving a rough, inner wall.

Watt. In the year 1763, a small model of a Newcomen engine was taken to the shop of an instrument maker in Glasgow, Scotland, to be repaired. This instrument maker, whose name was James Watt, had been studying steam engines for some time and he became very interested in this model. He was a man of great genius, and before he died his inventions had made the steam engine so perfect a machine that there has been but one really great improvement in it since his time, namely, compounding.

He found that to obtain the best results it was necessary, "*first,* that the temperature of the cylinder should always be the same as that of the steam which entered it; and *second,* that when steam was condensed it should be cooled to as low a temperature as possible." All improvements in steam-engine efficiency have been in the direction of a more complete realization of these two conditions.

In order to keep the cylinder nearly as hot as the entering steam, Watt no longer injected water into the cylinder to condense the steam, but used a separate vessel or condenser. He made his piston tight by using greater care in construction, so that it was unnecessary to have a water seal at the top. He then covered the top of the cylinder to prevent air from cooling the piston. When this was done he could use steam above the piston as well as below; this made the engine double acting.

Also, in the effort to keep the cylinder as hot as the entering steam, he enclosed the cylinder in a larger one and filled the space between with steam. This was not often done, however, and only of late years has the steam jacket been of much advantage. Also, the steam was used expansively, that is, the admission of steam was stopped when the piston had made a part of its stroke; the rest of the stroke was completed by the expansion of the steam already admitted. This plan is now used in all engines that are built for economy.

Other inventions made by Watt on his steam engines were: a *parallel motion,* that is, an arrangement of links connecting the end of the piston rod with the beam of the engine in such a way as to guide the rod almost exactly in a straight line; a *throttle valve* for regulating the rate of admission of steam; and a *centrifugal governor*, which controlled the speed of the engine shaft by acting on the throttle valve. Watt's engine as finally developed is shown in Fig. 4.

Watt saw that by using high-pressure steam he could get more work from it; but as it was not possible to make a very reliable

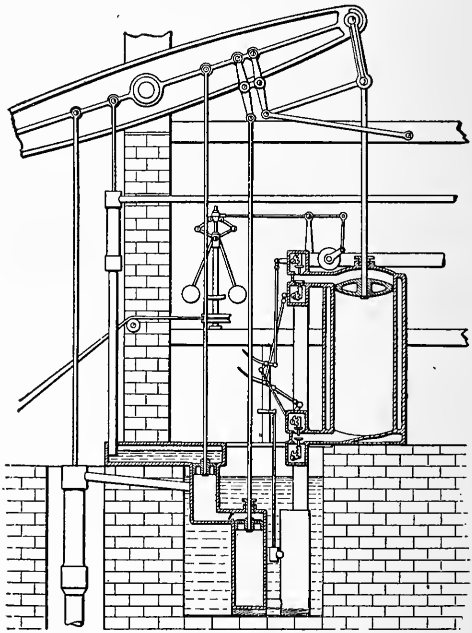


Fig. 4. Final Form of Watt's Steam Pumping Engine

boiler he never used a pressure of more than seven pounds per square inch above the atmosphere. About the year 1800, comparatively high pressures came more into use and the non-condensing engine was introduced. In Watt's engine, and all those preceding his, a vacuum was produced in front of the piston by condensing the steam, and either the atmosphere or steam at atmospheric pressure pushed it through the stroke. In the non-condensing engine, using high-pressure steam, the space in front of the piston could be opened to the atmosphere at exhaust and, although the atmospheric pressure resisted its motion, the pressure of the steam behind the piston was still greater than that of the air. These engines were much more simple than the condensing engines, as they required no condenser.

Compound Pumping Engine. About this time what would now be called a compound engine was introduced by Hornblower and later by Woolf. It had two cylinders of different size, steam being admitted into the smaller one and then passing over into the larger. Only a little expansion occurred in the small cylinder and much more in the larger one.

About the year 1814, Woolf introduced a compound pumping engine in the mines of Cornwall, but a simpler engine was later introduced and Woolf's engine fell into disuse. This later engine became known as the *Cornish pumping engine* and was famous for many years because of its economy. It was the first engine ever built that could compare at all with modern engines in the matter of steam consumption. It consisted of a single cylinder placed under one end of a beam from the other end of which hung a heavy rod which operated a pump at the foot of the shaft. Steam was admitted to the upper side of the piston for a short portion of the stroke and allowed to expand for the remainder of the stroke. This forced the piston down, lifted the heavy pump rod, and filled the pumps with water. Then communication was established between the upper and under side of the piston, exhaust occurred, and the heavy pump rod fell, lifting the piston and forcing the water out of the pumps. Steam was cut off at about three-tenths stroke, and the pump made about seven or eight complete strokes per minute with a short pause at the end of each stroke to allow the valves to close easily and the pumps to fill with water. These engines needed great care and were in charge of competent men, to whom prizes were frequently given for the best efficiency, which doubtless accounts for their wonderful performance.

Parts of Steam Engine. Leaving the historical side of the steam engine let us now turn to the modern simple steam engine and study briefly its construction. Figs. 5, 6, and 7, will serve to illustrate a horizontal, center crank engine, all the more important parts being numbered. The function of the various parts will be considered in detail later in the work.

Referring to the numbers in Figs. 5, 6, and 7, the names of the parts are shown in the following list:

LIST OF PARTS

Sub-base *1*	Flywheels *34*
Frame *2*	Valve pistons *35*
Main bearing caps *3*	Valve rings *36*
Main bearing liners *4*	Valve cages *37*
Cylinder *5*	Valve rod *38*
Cylinder head *6*	Valve rod nuts (valve end) *39*
False head cover *7*	Valve rod nuts (ram end) *40*
Valve chest head (head end) *8*	Valve rod gland *41*
Valve chest head (crank end) *9*	Ram box *42*
Piston *10*	Ram box caps *43*
Piston rings *11*	Ram *44*
Piston rod *12*	Ram pin and nut *45*
Piston rod nut (piston end) *13*	Ram pin cap *46*
Piston rod nut (crosshead end) *14*	Eccentric rod connection *47*
Piston rod stuffing box *15*	Eccentric rod *48*
Piston rod gland *16*	Eccentric rod nut (ram end) *49*
Crosshead *17*	Eccentric rod nut (eccentric end) *50*
Crosshead shoes *18*	Eccentric *51*
Crosshead adjusting screws *19*	Eccentric strap *52*
Crosshead pin *20*	Dash plate *53*
Crosshead pin nut *21*	Dash plate gland *54*
Cross pin washer *22*	Doors *55*
Connecting rod *23*	Door handle *56*
Connecting rod bolts *24*	Door clamps *57*
Connecting rod strap *25*	Oil hood *58*
Crosshead pin box *26*	Oil hood handles *59*
Crosshead pin box wedge *27*	Eccentric oil boat *60*
Adjusting screws *28*	Valve rod oil boat *61*
Crank pin box *29*	Oil vent *62*
Crank pin box wedge *30*	Sheet steel lagging *63*
Adjusting screws *31*	Drain cocks *64*
Crank disks *32*	Shaft governor *65*
Crank shaft *33*	

Sub-Base. The sub-base *1*, Fig. 6, is made of a good grade of cast iron and is usually heavily ribbed and made high enough to permit the wheels to clear the floor. The sub-base is often omitted with engines of large size, the engine being set upon a concrete base.

Frame. The frame *2* is the element or link by which all of the parts of the engine are held in place, so that their relative positions are always maintained to the end that their proper functions may be performed. The frame is a heavy, substantial casting so designed

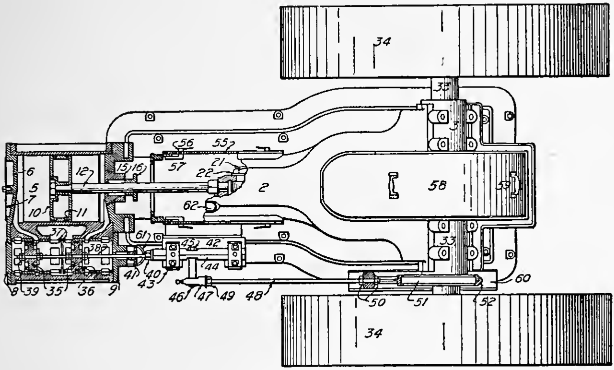


Fig. 5. Plan View of Modern Simple Engine

that it is strong enough to take all the stresses put upon it. The type, size, and details of the frame vary with the type and size of

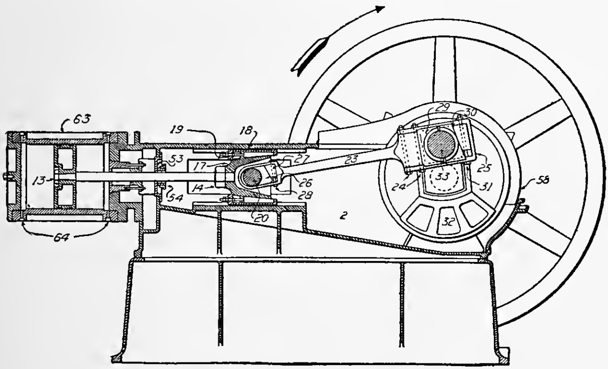


Fig. 6. Side Elevation of Modern Simple Engine

the engine of which it is a part. Usually the lower guide, valve rod guide, and seats for the main bearings are cast integral with it. In small sizes the cylinder is frequently cast integral with the frame. Provision is always made for adjustments necessitated by any wear of the frame or parts attached thereto. It is to be noted in Fig. 6, that the frame crank case *2* is connected with the crosshead guide. It frequently has an opening into the sub-base, thus permitting the oil from the crosshead, guides, and crank to drain into a suitable receptacle in the sub-base, from which it is taken by means of a drain cock conveniently located in the side or end. The crank is enclosed

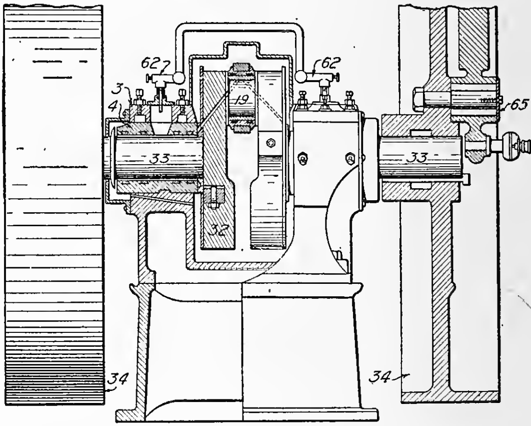


Fig. 7 End Elevation and Part Section of Modern Simple Engine

by a neat, pressed, sheet steel cover, which prevents the oil from being thrown outward on the floor while the engine is running. Quite frequently the crank cover is made of cast iron.

Cylinders. The cylinder *5*, Fig. 5, is one of the most important parts of the steam engine, for it is in the cylinder that the energy of the steam is converted into useful work. The cylinder is circular in section and is attached to the bed by means of a number of bolts. It is made of close grain, gray cast iron. The casting of the cylinder should be done with great care, so as to insure a casting free of blow-holes or other defects.

Fig. 8 illustrates the cylinder in cross section as well as showing its contained parts. The cylinder barrel *1* is accurately bored and fitted. Inside of this barrel the piston *2* is driven back and forth by the steam, which is admitted alternately on one side and then on the other through the ports *13*. The piston is connected to the crosshead through the piston rod *3*. The continuous movement back and forth of the piston causes the surface of the cylinder to wear away, and in order to avoid a shoulder being formed by this action, the cylinder is counterbored at each end by an amount depending on the size of the cylinder. The diameter of the counter-

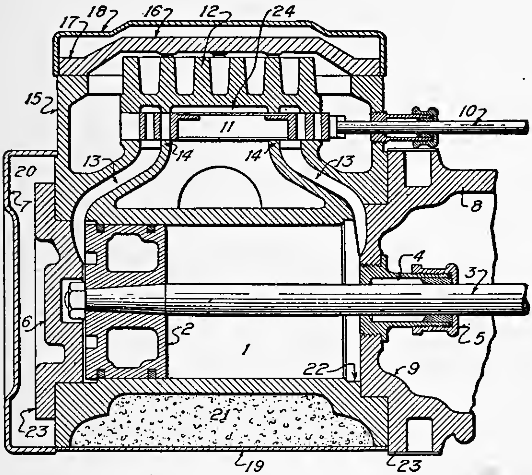


Fig. 8. Cylinder and Valve Mechanism Shown in Section

bore *22* is usually about one-quarter of an inch larger than that of the cylinder proper, depending, however, somewhat on the size of the cylinder. The stroke of the piston is such that the piston moves beyond the wearing surface at each stroke, thus preventing any shoulder being developed in the cylinder wall.

The cylinder is attached to the bed of the engine by a number of bolts which are placed through the flanges *23* of the cylinder and cylinder head. Each end of the cylinder is closed by means of the cylinder heads *6* and *9*. The cylinder head *6* is called the back cylinder head (head end), and *9* is known as the front cylinder head (crank end). In the illustration the front head *9* is a portion of the frame, but in many constructions it is entirely independent of the frame. In order to have a steam-tight cylinder, it is necessary to make a tight joint between the cylinder heads and the cylinder barrel. This is accomplished by turning both surfaces true, then grinding the joints with emery and oil. After the joints are well ground the heads are tightly drawn up against the cylinder by means of bolts suitably arranged. A sheet iron jacket *19* is put around the cylinder, leaving an air space *21* between the cylinder walls and the jacket. This air space retards the cooling off of the cylinder walls, hence initial condensation of the steam in the cylinder is reduced. In some types of engines such as locomotives, this air space is filled with some non-conducting material such as asbestos. This is also

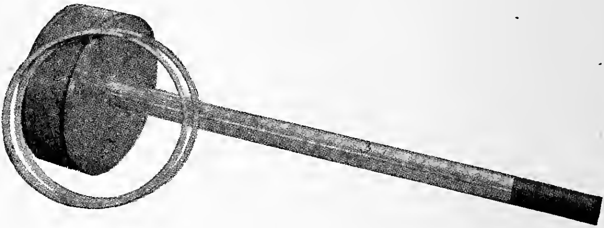


Fig. 9. Piston, Showing Piston Rings for Making Steam-Tight Joints

sometimes done by builders of stationary engines. It should be noted also, that the back cylinder head has an air space for the same reason as that given for the space surrounding the cylinder. Since condensation does take place in the cylinder, some means must be provided for removing the water, hence the drain cocks *64*, Fig. 6, are placed in the bottom of the cylinder at each end. The pipe connection for these cocks enters the cylinder in the counterbore near the wearing surface. Any water that may be in the cylinder will be forced out through these cocks if they are open. Care must be taken that the cylinder is freed of water, for if it is not, on account of the incompressibility of water, the cylinder head may be forced off or other damage result therefrom. Some cylinders are provided with relief valves, which automatically open when the pressure from any cause reaches a certain amount, thus preventing the bursting of a cylinder head.

Piston Rings. Between the piston *2*, Fig. 8, and the walls of the cylinder there must be a steam-tight joint, so that the live steam can not pass around the piston and be exhausted before expanded, otherwise a great waste of power will be incurred. The requirement is fulfilled by having the piston grooved, as shown in Fig. 9, and fitted with packing rings. These packing rings, commonly called *snap rings,* are turned up slightly larger in diameter than the cylinder and being cut, as shown in Fig. 9, they spring out into the cylinder, always pressing against the walls and forming an almost perfect steam joint. The piston of every engine is made with two or more of these packing rings. The cuts in the rings must not be placed directly in line with each other, otherwise the steam would have a better chance to blow through. In order to prevent this the joints are always placed on opposite sides of the piston. Packing rings are always made of cast iron, and are usually turned up to a uniform section. The outside portion and the two sides are carefully machined.

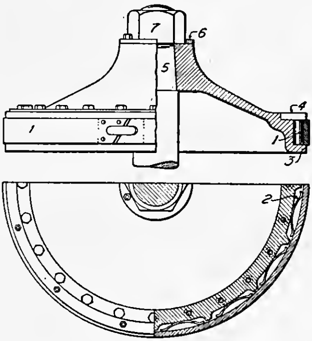


Fig. 10 Part Section Plan and Elevation of Conical Piston

Pistons. The piston *2*, Fig. 8, is usually made of cast iron, but sometimes is made of cast steel. It may be a solid disk grooved, as in Fig. 9, or the central portion may be cored out, as in Fig. 8.

Another type of piston that is largely used in marine and sometimes locomotive service is illustrated in Fig. 10. This is a comparatively light cast steel piston, but at the same time a very strong one, due to its conical construction. It will be noted also that only the one packing ring *1* is used. This packing ring is much wider than the ordinary snap ring and is pressed out against the cylinder wall by a number of single leaf springs being placed between the body of the piston and packing ring, as shown in Fig. 10 at *2*. The piston is made with an **L**-shaped edge *3*, a band *4* being bolted on the open side of the **L**, thus forming a groove or opening for the reception of the small springs and the packing ring. The connection of the piston rod to the piston is also clearly shown. The rod *5* has a tapered end which is forced by hydraulic pressure into a tapered hole in the piston; the nut *7* is then tightened up and locked by placing the plate in position around the nut and fastening it with cap screws. This arrangement insures a lasting connection between the piston and the piston rod.

It is essential that the piston be as light as possible in order to reduce the amount of work absorbed in pulling it to and fro, and also to reduce the wear on the lower portion of the cylinder.

The piston rod *3*, Fig. 9, is fitted into a tapered hole in the piston and secured by means of a lock nut and cotter pin placed on the back end. Oftentimes the tapered fit is made very tight and the piston forced on by hydraulic pressure. An older form of attaching the piston rod to the piston is shown in Fig. 8. In this instance the rod has a tapered end, which is driven into a tapered hole in the piston where it is secured by a nut, no cotter pin being used. The other end of the piston rod is threaded, screwed into the crosshead, as shown in Fig. 6, and secured by means of a lock nut. In some constructions the crosshead end of the piston rod is tapered and secured by a key. Many schemes have been employed by different manufacturers for fastening the piston rod to the crosshead, all of which have their advantages and disadvantages. The piston rod, although usually made of a good quality of open hearth steel, is frequently made of nickel steel, which possesses great strength.

Stuffing Box and Packing. As the piston rod passes through the front cylinder head, some provision must be made for making a steam-tight joint between the piston rod and the cylinder head. This is accomplished by means of the stuffing box *4*, and the gland *5*, shown in Fig. 8. Some form of packing is placed around the piston rod within the stuffing box *4* and the gland is forced in by means of bolts or a secured cap as shown, thus holding the packing in the box and at the same time crowding the packing tightly against the piston rod.

The piston packing may be made of woven strands of hemp or cotton; or asbestos may be used. To insure lubrication of the rod this fibrous packing is soaked in oil before being placed in position. In addition to this form of packing there are different compositions of rubber, graphite, cotton, etc., also various kinds of metallic packing in use. A metallic packing is made of material such as babbitt metal, which is a soft alloy of copper, tin, and antimony. This and other compositions are used for metallic packing, and the metal, being comparatively soft, wears away much more rapidly than that of the piston rod. Fig. 11 illustrates one form of packing, known as the U. S. Metallic packing. The principle of operation is as follows: The babbitt metal rings *2*, consisting of three rings cut in half, provide the packing and are the only parts which come in contact with the rod. These rings are forced into the vibrating cup *6* against the rod, and are fed down as wear takes place by the pressure of the steam itself. The spring behind the follower *3* is merely intended to hold the rings and other parts in place when steam is shut off. A ground joint is made between the flat faces of the vibrating cup *6* and the ball joint *4*. There is also a ground joint between the ball joint *4* and the gland *7*. The combination of the sliding face of the vibrating cup and the ball joint permits the packing to follow the rod freely without any increase in friction should it run out of line for any cause. This is an important feature, since the wear of the crosshead, guides, piston head, and cylinder produces an irregular alignment of the piston rod, which would injure the packing to a marked degree, if it was not flexible. The parts of the packing are held in place by the gland *7*, which is bolted to the cylinder head. A steam-tight joint is made between the gland and the cylinder head by means of a copper gasket. The purpose of the swab cup *5* is to hold in place a swab, which is usually made of waste, candle wicking, or a braided material, soaked in oil and oiled from time to time as a means of keeping the piston rod well

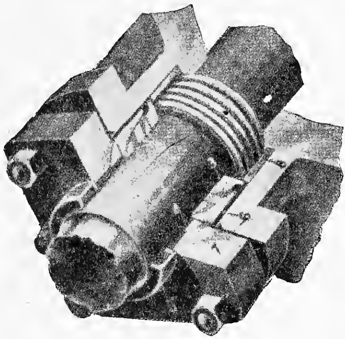


Fig. 11. Stuffing Box Packed with Metallic Packing

lubricated. In addition to this service, the swab catches and retains a considerable amount of dust and grit which would otherwise find its way into the cylinder, where it might do harm. It is to be said in favor of the various so-called rubber packings, that they give very good service. The four different styles of rubber packing illustrated in Fig. 12, are not composed entirely of rubber, but contain other material such as graphite, cotton, etc. These different styles of packing are used both on piston and valve rods.

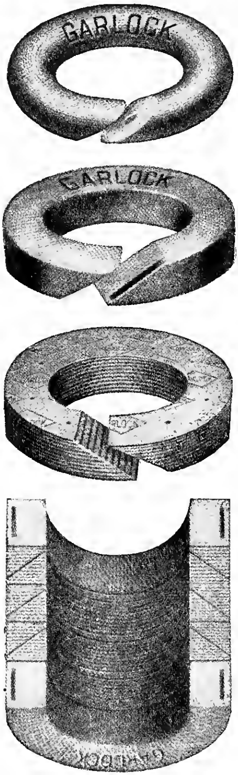


Fig. 12. Types of Rubber Packing

It is to be borne in mind that all which has been said with reference to the piston rod is equally applicable to valve stem packing. The general construction of the valve stem glands, vibrating cups, etc., is identical with those of the piston rod. The same materials are used for the packing medium and the same watchful care is required in order to obtain satisfactory results. Packing is an important subject and one which should be carefully looked after. It can not be said that any one particular kind or style of packing is the proper one to use in every case, for a packing which may give very satisfactory results under one set of conditions may utterly fail under another. For instance, a packing suitable for low steam pressures is not efficient where high steam pressures are used, and a packing that may give satisfaction with high pressures may not in any measure meet the requirements imposed upon it by the use of superheated steam. Each particular installation is, therefore, a different problem and must be solved in a different manner.

Valves. In Fig. 8, the valve *11* is shown in position. It will be noted that the valve rests upon the valve seat *14* and works between the valve seats and the pressure plate *12*. The valve *11* is usually made of cast iron and may be of many different shapes, as will be seen in the study of the various types of engines. There are, however, two general types of valves—one, a plain slide or **D**-valve; and the other some form of piston valve. While there are many modifications and combinations of these two types, yet they are akin to the two types named. The valve in Fig. 8 is of the slide valve type. It is what is known as a double ported valve, that is, steam is admitted to the cylinder by two edges of the valve by reason of the fact that there is an opening through the valve. The pressure plate *12* is used for the purpose of reducing the area of the valve exposed to live steam pressure, it being noted that the portion under the hollow space *24* is not in contact with live steam. This reduc-

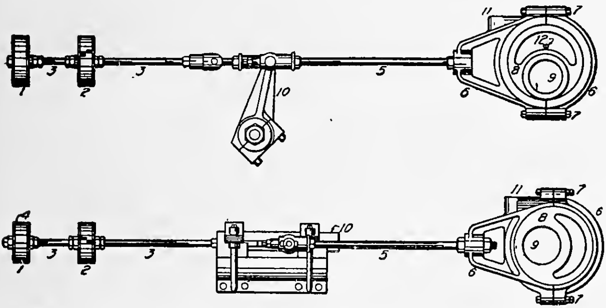


Fig. 13. Eccentric Mechanism Showing Rocker and Ram Methods of Connecting Eccentric Rod with Valve Rod

tion of the exposed area is made in order to reduce the amount of effort required to pull the valve back and forth. When the surface of a valve of medium size is considered and an average steam pressure per square inch of 180 pounds is being exerted upon it, some conception can then be had of the amount of friction that must be overcome every time the valve is moved across its seat. To eliminate a portion of this negative work is the primary object of the pressure plate. Pressure plates are of various shapes and designs depending of course upon the type of the engine and the valve used.

One of the advantages the piston valve has over the slide valve is that it is almost perfectly balanced by reason of the fact that steam surrounds it on all sides, hence there is no excessive pressure on any part of the valve. It is comparatively light and therefore easily driven and lubricated. The general form of construction of piston valves is illustrated in plan in Figs. 21 and 22.

Eccentric. The valve is driven by its connection to the shaft by means of the valve stem, eccentric rod, and the eccentric. The relation of these parts is well illustrated in Fig. 13. The valve shown is an ordinary piston valve with flexible snap packing rings *4* similar to those previously described for the piston packing rings. In fact, the piston valve, as the name implies, behaves very much like the steam engine piston. The two piston ends *1* and *2* are held together by the valve rod *3*. The valve rod has nuts so placed that the pistons are held the proper distance apart. The valve rods, or stems as they are often called, extend beyond the valve box some distance and connect with the eccentric rod *5*. The manner of making the connection between the valve rod and the eccentric rod varies widely, this connection being governed largely by the type of engine and the exigencies of the case. Fig. 13 shows two methods of making this connection, one being accomplished by making use of a rocker arm and the other by using a ram. The way in which the rocker arm *10* is used, is obvious from the figure. The ram *10* is a square block, working in a bearing and so constructed that the valve and eccentric rod can be attached to it. When the ram is used, the motion is transmitted to the valve in a straight line, hence there is less strain upon the connecting parts than if a rocker arm was employed.

The eccentric rod *5*, in both cases, is attached at one end to the eccentric strap *6* and at the other end to the ram or rocker arm. Nuts suitably arranged make the rod secure and at the same time provide a means for lengthening or shortening the rod as needs demand. The valve and eccentric rod are usually made of mild steel turned true and polished

The eccentric strap *6* is made of gray cast iron, lined with good babbitt metal for a wearing surface upon the eccentric. The strap is held on the eccentric by means of the bolts *7*. By removing liners or shims from between the two sections of the strap, adjustments for wear can be made. There are several patented straps on the market that possess particular features, but the essential elements of all eccentric straps are about the same. Provision is made for lubrication by having an oil cup *11* cast with the strap.

The eccentric *8* is mounted on the main shaft *9* and is held secure in the position desired by means of the set screw *12*. Eccentrics for large engines are held by means of one or more set screws and a key. For a discussion of the function of the eccentric, the student is referred to the instruction book on "Valve Gears."

Steam Chest. The box *15*, Fig. 8, containing the valve and its parts, is known as the steam chest. The steam chest cover *16* is held in place by studs which pass through the flanges *17* into the box. The steam chest is connected to the steam supply by suitable pipe connections, steam being turned on or off as desired by means of the throttle valve. When the throttle valve is opened, steam passes into the chest through the valve, into the cylinder, where it is expanded and then ejected through the exhaust opening. The energy of the steam is transmitted through the piston and piston rod to the crosshead *17*, Fig. 6, thence to the connecting rod *23*, crank pin *33*, to the main shaft. In order that these parts may properly perform the function of transmitting this energy, a correct design is highly essential; therefore, a discussion of their construction is deemed necessary.

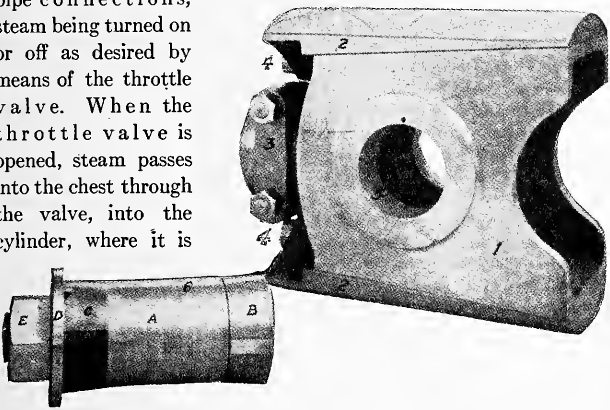


Fig. 14. Typical Crosshead and Pin for Large Size Engine

Crosshead and Connecting Rod. The crosshead is usually made of steel which forms a connecting link between the piston rod and the connecting rod. It is made in various shapes and patterns. One type is illustrated in Fig. 6. In engines of larger size, the prevailing form of the crosshead used is similar to that illustrated in Fig. 14. This crosshead consists of a steel casting *1* and two wedges, or shoes, *2*, which fit over a projection on the outside surface of *1*. These wedges are either cast or forged and serve as a retainer for a layer of babbitt metal on the outside. It will be noted that there are oil grooves cut on the surface of *2* in order to facilitate the oiling of the crosshead guides. These wedges are provided with a nut and bolt *4*, whereby adjustment for wear can be made as necessary. Usually there is a slight amount of clearance between the crosshead and the guides, but it should not be in any case excessive. The piston rod is fitted into the end *3*, as already described. The connecting rod is attached to the crosshead pin *6*, which fits into the hole *5* and is held in place by a nut. The crosshead pin *6* is made of a good grade of steel and has a portion *B* which fits into the

Fig. 15. Solid Forged Steel Connecting Rod

back side of the crosshead, as viewed, the other portion *C* fitting into the outside part. When the pin is in place, the collar *D* is adjusted and the nut *E* tightly drawn. The straight portion *A* goes between the sides of the crosshead, and upon it the connecting rod brasses bear.

There are two general types of connecting rods in use, usually classified as marine and locomotive. Connecting rods of the marine type are as a rule used on engines of comparatively short stroke, while those of the locomotive type are employed on engines having a long stroke.

These rods are forged from open-hearth steel, with solid forged ends for the crosshead end, and a square end for the crank end in case of the marine type; and a solid forged or forked end for the crank end in case of the locomotive type.

The connecting rod, Fig. 15, is a solid forged steel rod having the ends machined out to receive the brasses. The crank end *1* is fitted with a brass or bronze box lined with a good quality of babbitt metal. The crosshead end *2* is usually, but not always, fitted in a similar manner to that of the crank end. Adjustment for wear is made by means of wedges at each end, as shown at *3*. These rods are usually of rectangular cross section, although round shapes sometimes are used, especially on small engines.

The marine type of connecting rod is illustrated in Fig. 16. The body of the rod is forged similar to the locomotive type, as is also the small, or crosshead, end, but the distinguishing difference is in the way in which the large, or crank, end is formed. The end of the rod is enlarged and finished square, and the box containing the crank bearing which is lined with a good wearing material, is fastened to the rod proper by means of the bolts. Adjustment for wear is made by tightening up the nuts on the bolts.

It will be seen in Fig. 6 that the connecting rod is the connecting link between the crosshead and the crank *33*. The length of the

Fig. 16. Marine Type of Connecting Rod

connecting rod bears a definite relation to the length of the crank radius. The ratio of the length of the connecting rod to that of the crank radius varies in practice from four to eight. Occasionally conditions demand a greater ratio than eight, but it is seldom less than four.

Fig. 17 illustrates the connection of the piston, crosshead, connecting rod, and crank shaft. The function and construction of the piston, crosshead, and connecting rod have been previously discussed. However, the figure is valuable in that it shows quite clearly the relation of the various parts to each other. The crank shaft used on center-crank engines is frequently a solid steel forging, which includes the crank pin *2*.

Miscellaneous Parts. In order to compensate for the weight of the connecting rod and brasses it is necessary to put counterweights on the shaft as shown at *4*, Fig. 17. These counterweights are usually heavy castings, machined to slip over projections on the crank shaft, and securely fastened thereto by bolts or set screws. The portion of the shaft marked *1*, Fig. 17, fits into the bearings provided for the main shaft or crank shaft, the length of this bearing portion being the distance between the counterweights and the collars *5*. It will be noted that on one end of the shaft is located a disk *3*. Sometimes this disk is forged as a part of the shaft and at other times it is made separate and forced on by hydraulic pressure. The purpose of this disk is usually intended to provide a ready

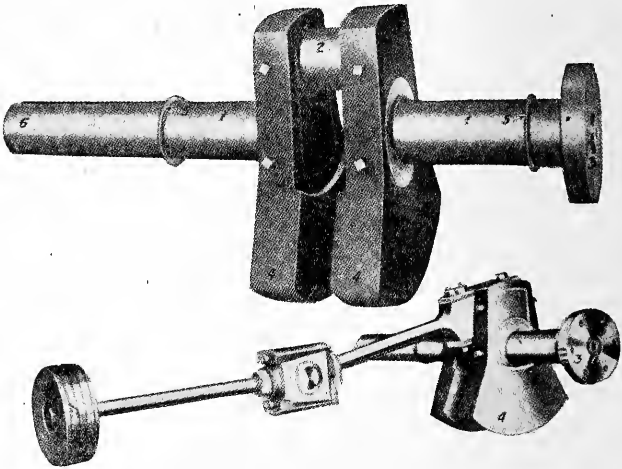


Fig. 17 Connection of Piston, Crosshead, Connecting Rod, and Crank Shaft

means of attaching the shaft of an electric generator when a direct connected plant is feasible or desired. It may be said here that a direct connected plant offers many advantages over a belt-driven system. It simplifies the plant, reduces friction, gives greater reliability, and makes possible more power in a given space.

The projection *6* on the other end of the shaft is the axis upon which the flywheel is forced and held secure by means of a key. The crank pin *2* should be of such ample proportions as to be safe against breakage and the heating of the pin or brasses placed upon it.

All engines are not of the center-crank type, but many have a side crank, the crank being a disk or a crank arm fastened on the end of the main shaft very much in the same manner as the disk *3*, Fig. 17. In this kind of construction the crank pin is usually a piece separate from the crank arm or crank disk, and is connected to it by being forced on and then riveted over, or by nuts put on and cottered. In either the side crank or center crank construction, the distance from the center of the axle to the center of the crank pin is equal to one-half the stroke of the engine, as for instance, an 18×24 engine has a crank arm of 12 inches in length, which is just one-half of the length of the stroke. In speaking of the size of the engine it is customary to mention the diameter of the cylinder first, that is, in speaking of an 18×24 engine is meant a cylinder 18 inches in diameter and a stroke 24 inches.

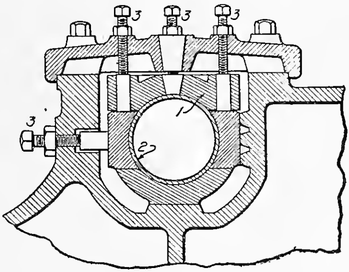


Fig. 18. Section of Babbitt Lined, Quarter-Boxed Main Bearing

The main bearing *4*, Fig. 7, should be designed with great care, having liberal proportions and lined with anti-friction metal, hammered in place and accurately bored and scraped to fit the shaft. On small engines the lower half of the main bearings are usually made of a part of the frame, the upper half being a removable cap. Between the upper and the lower portion of the bearing, metal liners are placed, which afford ready means for making any necessary adjustments.

Large engines have a babbitt lined, quarter-boxed main bearing of ample size, Fig. 18. To provide for both vertical and lateral adjustments it consists of four parts carefully machined on all sides and scraped to fit accurately. This bearing is so constructed that the bottom piece can be removed by slightly raising the shaft. The other three parts are removed after taking off the cap. By use of the adjusting screws *3*, the side *2* and the top *1* may be properly adjusted by the sense of feeling when the engine is in motion.

There are many other types of main bearings besides those mentioned, but they differ only from those already described in some of the minor details. The value of these details varies through wide limits, each builder contending for his own particular design.

A side-crank engine needs but one heavy bearing, such as that shown in Fig. 18, as the flywheel end of the shaft, being subjected to forces acting in but one direction only, requires a much smaller bearing. This outer bearing, Fig. 19, is called an out-board bearing and is smaller and simpler in construction than the main bearing. It is supported by a special casting, which has a hollow recess into which lubricating oil is poured. The shaft carries one or more small chains or rings which fit loosely on the shaft and dip into the

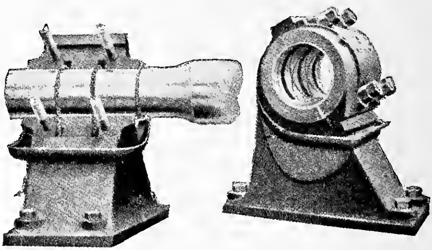


Fig. 19. Out-Board Bearing of Simpler Construction than Main Bearing

oil. Thus it is seen that oil is constantly brought in contact with the bearing of the shaft. This same scheme of lubrication is also used for the main bearing. As the different types of engines are considered, the several types of bearings will be noted and discussed.

The belt wheels *34*, Fig. 5, serve a two-fold purpose—one as a governing device, the value of which will be discussed later, and the other as a means of storing up energy while the piston is in mid-stroke, where the crank effort is greater than the resistance to be overcome. The belt wheels act as a flywheel and give up this energy at the ends of the stroke, thus enabling the engine to run over the dead centers. The design of the belt or flywheel is an important item in the proper proportioning of a steam engine. Its weight and dimensions must be very acurately determined. The belt wheel, or flywheel, whichever is employed, is made of cast iron of various sizes, some being cast solid in one piece, others being cast in two or more sections. In any case the wheel is forced on the shaft and securely fastened thereto by means of a key and set screws.

TYPES AND CONSTRUCTION

Classification. Thus far an effort has been made to give the student some idea of the development of the steam engine and enable him to become familiar with the various parts and their functions. The natural sequence to the above study is to make a detailed study of the several types in use. No hard and fast rule can be given for classifying steam engines as they overlap in so many instances. That is to say, a simple engine may be either condensing or non-condensing; it may be high speed or low speed, etc. According to well-known authorities, various piston engines may be grouped under the following classes:

- I. Number of cylinders
  - Single cylinder
  - Multiple cylinder
- II. Construction of cylinders
  - Fixed cylinder
    - Vertical
    - Horizontal
    - Inclined
  - Movable cylinder
    - Oscillating
    - Rotary
- III. Action of steam
  - Single acting
  - Double acting
- IV. Transmission of steam power
  - Direct acting
  - Indirect acting
    - with balance lever or beam
    - without balance lever or beam

Professor R. H. Thurston in his book entitled "A Manual of the Steam Engine" classifies steam engines according to their purpose and use, as follows:

- I. Stationary mill engines
  - Moderate speed
  - High speed
- II. Agriculture engines
- III. Portable and semi-portable engines
- IV. Road locomotives
- V. Railway locomotives
- VI. Pumping engines
  - Crank and flywheel
  - Direct acting
- VII. Marine engines
  - Paddle engines
  - Screw engines
- VIII. Special types

The same authority further classifies engines according to their structure, as follows:

- I. Expansion
  - Simple
  - Compound
- II. Position of cylinder
  - Direct acting
  - Beam
  - Vertical
  - Inverted
  - Horizontal
  - Inclined
- III. Steam
  - Condensing
  - Non-condensing
- IV. Pressure
  - High pressure
  - Low pressure
- V. Piston action
  - Reciprocating
  - Vibrating
- VI. Steam turbines
- VII. Rotary
- VIII. Connection
  - Direct connected
  - Geared
- IX. Condensation
  - Jet condensing
  - Surface condensing

They are frequently designated by the name of the inventor, designer, or constructor, as the Watt, the Corliss, or the Porter engine.

From the last two groupings it is evident there is no sharp line of demarcation, for in many instances engines of one class have essential parts similar to those of another type. In this work the classification outlined in the last group will be taken as a basis for study.

Simple Engines. The simplest type of engine is the single expansion. It has one cylinder and admits steam for a part of the stroke, expands it during the remainder, and exhausts either into the atmosphere or into a condenser. Simple engines, Figs. 5 and 6, are now used only for comparatively small powers, say 200 h.p. or less, and although more extravagant in the use of fuel than the others, may still be the most economical financially, if low first cost is an important item; if they are not run continuously; or if the load fluctuates widely.

Compound Engines. Compound engines have two cylinders known as the high pressure and low pressure, Figs. 20 and 21. It will be noted that two different types of compounds are represented, the one in Fig. 20 being known as a cross-compound, the two cylinders being parallel, and the one in Fig. 21, a tandem-compound engine, the cylinders being in line with each other.

Steam enters the smaller or high pressure cylinder, and then expands until release, when it is exhausted into the larger cylinder, where it expands further. The cylinders should be so proportioned that approximately the same amount of work can be done in each, which may be accomplished by making the high pressure cylinder enough smaller than the low so that when the steam leaves the high at a lower pressure than when it entered it, the increased volume of the steam may be taken care of and at the same time the increased area of the low pressure piston may compensate for the drop in steam pressure.

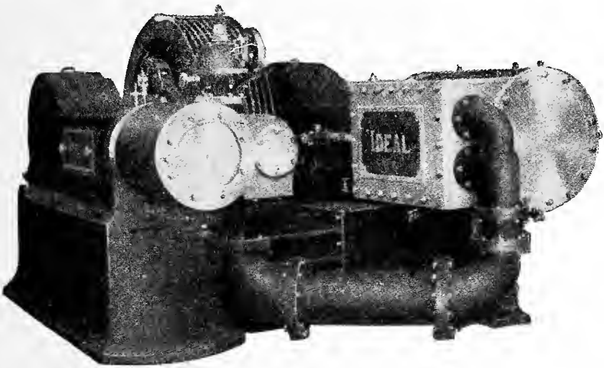


Fig. 20. Typical Cross-Compound Engine

Besides being economical, the cross-compound has a distinct mechanical advantage. The two cranks may be set at right angles so that when one is on dead center, the other is at a position of nearly its greatest effort. This makes a dead center impossible, and gives a more uniform turning moment. Then the individual parts may be made lighter and are thus more easily handled.

When the cranks of the cross-compound engine are at 90 degrees with each other the low pressure piston is not ready to receive the steam when the high pressure exhausts; therefore, there must be a receiver to hold the steam until admission occurs in the low. Such engines are called cross-compound, because steam crosses over from one side to the other. Sometimes instead of having the cranks at 90 degrees, they are placed together or opposite. Then the strokes begin and end together, and the high can exhaust directly into the low without a receiver.

A tandem-compound engine, Fig. 21, has both pistons on one rod, the high pressure piston rod forming the low pressure tail rod. Such engines are less expensive because there is but one set of reciprocating parts instead of two, but like simple engines they have the disadvantage of dead points.

Triple Expansion Engines. Triple expansion engines expand the steam in three stages instead of two. There are usually three

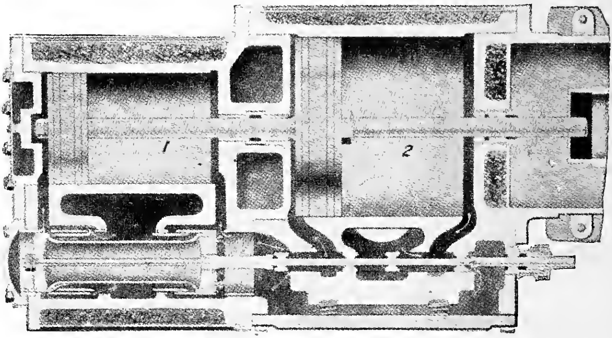


Fig. 21. Section of Cylinder and Valves of a Tandem-Compound Engine

cylinders, viz, the high, the intermediate, and the low, arranged with cranks 120 degrees apart. This gives a more uniform turning moment than a compound. Sometimes there are four cylinders on the triple expansion engine, viz, one high, one intermediate, and two low. This arrangement gives better balance and is often used in marine work.

For triple engines there must be a receiver between each two cylinders. Fig. 22 shows the essential features of a triple expansion engine.

Quadruple Engines. Quadruple engines expand their steam in four stages instead of three. Multiple expansion engines are nearly always condensing.

Cylinder Ratios. There are several considerations to be remembered when proportioning the cylinders of the multiple expansion engines. The ratio of the cylinders should be such that each develops nearly the same power, and the drop in pressure between the cylinders and receivers should be as small as possible.

There are many formulas in use, some simple, others more complex involving mathematical calculation. A common rule for compound engines is to make the ratio of the cylinders equal to the square root of the total ratio of expansion. Thus, if the steam has an expansion ratio of 9, the ratio of the cylinder volumes will be $\sqrt{9}$, or 3; that is, the low pressure cylinder will have a volume three times as great as the high pressure cylinder. If the cylinder ratio is 3 and the length of the stroke is the same for both, the diameter of the low pressure cylinder will be 1.75 times that of the high pressure cylinder.

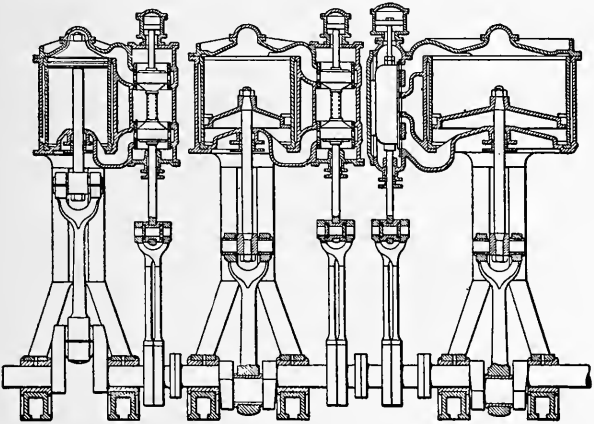


Fig. 22. Section of Essential Features of Triple Expansion Engine

Another rule is to make the cylinder ratio equal to the total ratio of expansion multiplied by the fractional part of the stroke completed when cut-off occurs in the high pressure cylinder.

Suppose the ratio of expansion is 9, as above, and that cut-off occurs at one-third of the stroke in the high pressure cylinder, the ratio of cylinder volumes will be $9\times\frac{1}{3}$, or 3. If cut-off occurs at one-half of the stroke, the ratio will be $9\times\frac{1}{2}$, or 4.5.

For triple expansion engines the low pressure cylinder is made large enough to develop the full power if steam at boiler pressure is used.

The intermediate cylinder is made approximately a mean between the high and low. The area of the intermediate piston is found by dividing the area of the low by one and one-tenth times the square root of the ratio of the low to the high.

The above may be written thus:

$$\text{Area of high pressure cylinder} = \frac{\text{Area of low pressure cylinder}}{\text{Cut-off of high pressure} \times \text{ratio of exp.}}$$

$$\text{Area of inter. cyl.} = \frac{\text{Area of low pressure cylinder}}{1.1 \times \sqrt{\text{ratio of low to high}}}$$

In general, for triple expansion the ratios of the volume of the three cylinders are about as follows:

$$V_1 : V_2 : V_3 :: 1 : 2.25 \text{ to } 2.75 : 5 \text{ to } 8$$

For quadruple expansion engines, the ratios are as follows:

$$V_1 : V_2 : V_3 : V_4 :: 1 : 2 \text{ to } 2.33 : 4 \text{ to } 5 : 7 \text{ to } 12$$

It is self-evident that the compound engines illustrated are of the multiple cylinder class. They also have fixed cylinders and are double and direct acting. That is, steam acts on both sides of the piston, and the power is delivered directly from the piston to the shaft or flywheel without the intervention of a walking beam or some other transmitting medium. The engines illustrated in Figs. 20 and 21, are horizontal, whereas the one shown in Fig. 22 is vertical. A horizontal engine is, therefore, an engine whose cylinder is parallel to the ground, and a vertical engine is one which has its cylinder or cylinders perpendicular to the ground. These engines may also be operated either condensing or non-condensing.

From the foregoing it must be obvious that it is not possible to classify an engine within narrow limits, so it appears to be more logical to classify them according to the service for which they are to be used, as in the second grouping.

Selection of Type. In the selection and design of an engine there are a great many factors to be considered. The engine must be as light as possible, and yet must be strong enough to do the work likely to be imposed upon it. The bearings should be large and ample in number. Lubrication must be given especial attention if high speeds are to be used. Lightness of design tends towards small first cost, which is important, but durability and efficiency should not be entirely sacrificed for low first cost. In the course of time the more expensive engine may prove to be the cheaper as maintenance and repairs may amount to considerable on a poorly designed and built engine. For some classes of service, however, the cheap engine is the one best adapted. For instance, in saw mills, cotton mills, and for similar class of service, a low first cost simple engine is the one best suited for the work, because the labor employed to operate it is often inexperienced and ignorant. In such cases the protection and care that can be given the engine is poor, hence the lower the value of the property exposed, the less will be the loss resulting from the depreciation. On the other hand, if one is selecting an engine for a lighting plant in a city, he would more than likely select one of the most improved types of high speed, condensing machines. In the latter case the first cost would be considerably more than the one selected for the saw mill, but the increased efficiency of operation, the slight depreciation, and the reduction in maintenance would more than compensate for this.

From the foregoing it is evident that there are many factors to be taken into consideration when selecting a steam engine for any given service. In the further study of the several different types the class of service for which each is best suited will be indicated in so far as it is possible to do so. There are, however, some general features every engine should possess independent of its class. It should be simple in construction, having compactness combined with great strength and durability. It should be well balanced and free from severe vibration. Accessibility of parts is also an important consideration.

STATIONARY ENGINES

Simple Side-Crank Type. Stationary engines for ordinary mill service, such as machine shops, small power plants, and various manufacturing concerns, are generally simple engines operating at moderate speed, having either plain slide valves or piston valves. There are, however, some cases where compound engines of moderate size have been installed in similar plants in more recent years. The demand for electric generators has also largely affected the design of steam engines for small electric power plants. In speaking of small plants, in this connection, it may be taken as meaning from 25 to 500 horsepower.

A simple slide valve engine of the side-crank type, which has been largely used in plants where a cheap, efficient engine was the requirement, is illustrated in Fig. 23. This engine has one slide valve, an automatic or shaft governor, and a heavy flywheel which is used as a belt pulley. It is built in sizes varying from 9 inches × 14 inches to 22 inches × 28 inches, and develops a horsepower of

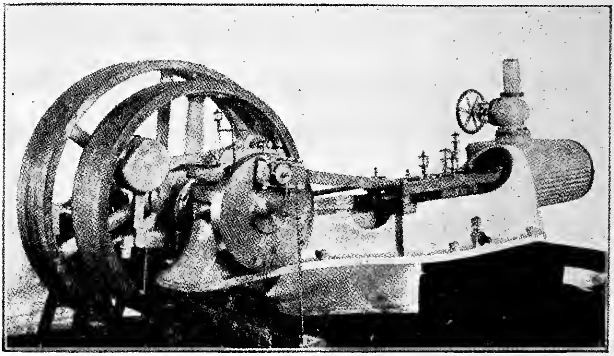


Fig. 23. Simple Slide Valve Engine of Side-Crank Type

about 45 to 300 according to size of cylinders, steam pressure used, and the speed at which the engine is operated.

Lubrication of the cylinders is secured by the use of a sight feed lubrication attached to the steam pipe. The main and crosshead bearings are lubricated by oil cups.

This type of engine has been extensively used in cotton gins and saw mills and in small machine shops throughout the country. There are, however, several grades on the market, and it may be purchased for a comparatively low figure where the work to be done does not demand a machine of high grade. This engine has a concrete foundation, is well proportioned, and makes a neat appearance.

Simple Vertical Type. A simple vertical high speed engine that is particularly well adapted for isolated lighting plants in factories, stores, mines, and aboard ships, is illustrated in cross section in Fig. 24. It requires little attention, occupies small floor space, and is not extravagant in the use of steam. The engine is neatly and well designed. It has a large base, which insures stability and rigidity. All of the working parts are enclosed but readily accessible for inspection and repairs. The frame, cylinders, valves, pistons, etc., are carefully made and adjusted, and the same general types of these various parts conform to the general practice of high speed engines. It will be noted from the illustration that it has a center-crank, automatic governor, and a piston valve. The lubrication of the moving parts is accomplished by means of a geared pump located in the interior of the base of the frame. This pump forces the oil through pipes and grooves to the various bearings. This type of engine is furnished by the makers in sizes from $3\frac{1}{4}$ inches $\times$ 3 inches up to 9 inches $\times$ 7 inches for general service. Much larger vertical engines may be obtained, but are made as a special order. The engine illustrated is so designed to operate at speeds from 250 to 500 revolutions per minute, depending on the size, and uses steam pressure from 60 to 150 pounds. Its commercial rating is from $1\frac{1}{2}$ to 60 horsepower, according to the size of cylinders, steam pressure, and speed of operation.

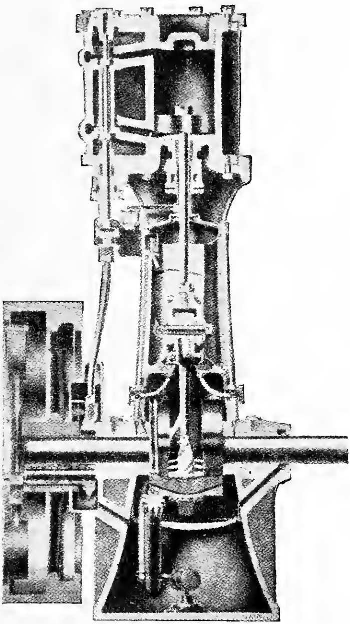


Fig. 24. Vertical High Speed Engine

Advantages of Vertical over Horizontal Type. While the discussion given above has had to do with a vertical engine of rather small dimensions and power, yet it must be borne in mind that vertical engines in very large units are built and successfully operated. This leads to a discussion of the relative merits of horizontal and vertical engines. At the present time the most common type of engine is the horizontal direct-acting, that is, an engine whose cylinder is horizontal and whose piston acts on the crank through a piston rod and a connecting rod. In small engines the whole is often on one bed plate. Such engines are said to be self-contained. The cylinder is either bolted to the back of the bed plate or rests directly on it.

In marine work vertical engines are used in almost every case, on account of the *saving of floor space,* which is so important in a vessel. This saving of space is also a very important factor in many other cases, such as in crowded engine rooms in cities where land is expensive.

A second advantage of the vertical over the horizontal engine is the *reduction of the cylinder friction and unequal wear* in the cylinder of the latter. In the horizontal engine the piston is generally supported by resting on the cylinder, which is gradually worn until it is no longer round, causing leakage of steam from one side to the other. This is entirely avoided in the vertical engine.

Still another advantage of the vertical engine is the *greater ease of balancing the moving parts* so that there shall be no jarring or shaking. It is impossible to perfectly balance a steam engine of one or two cylinders. If it is balanced so there is no tendency to shake sidewise it will shake endwise; and if it is balanced endwise it will shake sidewise. The jarring is due to the back and forth motion of the reciprocating parts and the centrifugal force of the crank and the connecting rod. The crank can be readily balanced by making it extend as far on one side of the shaft as it does on the other, but the piston and the connecting rod are more difficult to balance. The effect of jarring can be greatly reduced, if the crank be balanced and the endwise throw made to come in line with the foundation, which should be heavy enough to absorb the vibration transmitted. In a horizontal engine this endwise throw not being in line with the foundation will cause vibration in the engine itself.

In machines that can be anchored down to a massive foundation, a state of defective balance only results in straining the parts and causing needless wear and friction at the crank-shaft bearings and elsewhere, and in communicating some tremor to the ground. The problem of balancing is much more of consequence in locomotive and marine engines.

To sum up the general advantages of the vertical engines: they

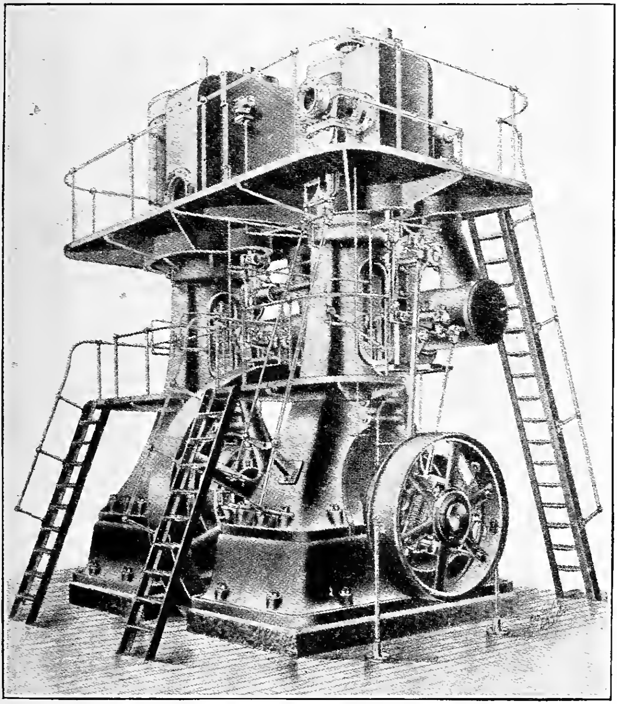


Fig. 25. Buckeye Vertical Cross-Compound Engine

have less cylinder wear, they take up less floor space, and they can be better balanced. In addition to these there are certain advantages which vertical engines have for certain kinds of work.

Disadvantages of Vertical Type. The *pressure on the crank pin* is greater during the down stroke than during the up stroke, because during the down stroke the weight of the reciprocating parts is added to the steam pressure, and during the up stroke this weight is subtracted.

Another difficulty is that in large engines the various parts are on such different levels that they require considerable climbing. This requires *more attendants* and is sometimes the cause for neglect of the engine. The *foundations* for vertical engines need to be deeper than those for horizontal engines, yet they do not need to be as broad.

Buckeye Vertical Cross-Compound Type. The development of electrical machinery and the increased demand for power in con-

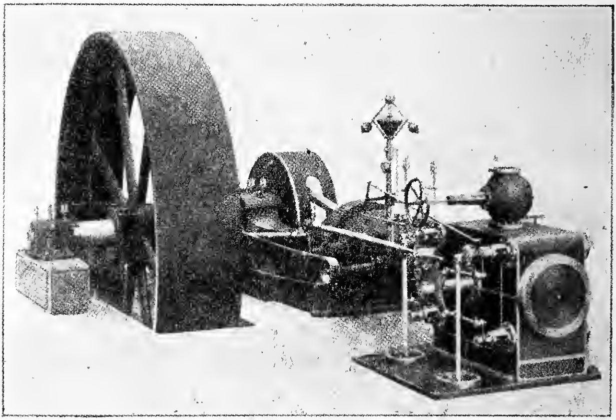


Fig. 26. Simple Corliss Engine, Showing Valve Mechanism

gested city locations, where land is very expensive and buildings costly because of their great height, has been the primary cause of the development of large vertical steam engines of various types. The engine, Fig. 25, represents a vertical cross-compound engine as built by the Buckeye Engine Company, which is especially well adapted to electric railway and power and lighting plants, when floor space is limited. The engine may be obtained either as a side or a center crank design. This engine and simple horizontal engines of the same make are typical representatives of economical, high speed engines. They are high priced, but the economy of operation and maintenance make them very desirable. The vertical engine illustrated may be obtained in sizes developing from 75 to 3,000 horsepower. A discussion of the valve gear used on Buckeye engines is to be found in the instruction paper on "Valve Gears." It is a double valve, giving automatic cut-off as distinguished from throttling cut-off regulation.

Corliss Type. A general utility engine of the highest type both from the standpoint of design and economy of operation and maintenance is the Corliss engine. It is to be found in electric railway power stations; in large and small pumping stations; in blast furnaces and rolling mills; in textile and flour mills; in machine shops and office buildings; in technical schools and colleges; and in nearly all kinds of industrial plants in this country and abroad. The Corliss engine is built in various types, styles, and patterns of any designed capacity up to 10,000 horsepower.

Fig. 26 shows the valve connection and manner of operation of a simple Corliss engine. It will be noted that the engine is governed by a fly-ball governor which is driven by a belt connection to the main shaft. This governor is connected to the steam valves by reach rods. The speed is automatically governed by variation of the point of cut-off. This engine, as well as most engines of this type, has large well-proportioned frames, cylinders, etc. Good workmanship and material enter into its construction, hence it is known as a high-priced engine; but, on the other hand, it is perhaps the most economical in the use of steam.

Valve Mechanism. The distinguishing feature of the Corliss engine is its valve mechanism, a good view of which may be seen in Fig. 27. The gear has four valves, the two top ones being the admission or steam valves and the two lower ones the exhaust valves. There is a connecting rod *7*, Fig. 27, which is connected to the eccentric through a rocker arm, and another rod, as may be seen in Fig. 26. As the shaft revolves, the rod *7*, due to its connection to the eccentric, moves back and forth, and, by reason of its connection through the clamp *8* to the wrist plate *6*, the latter is made to oscillate. The wrist plate *6* is attached to the frame by a pivot projection. The rods *9* have a right and left screw adjustment on each end and transmit motion from the pins *14* on the wrist plate *6* to the steam and exhaust valve bell cranks *10* and *15*, respectively. These valves receive motion in such a manner as to open and close the ports rapidly

The steam valve bell crank *10* is free to rotate on projections of the bonnet and carries at the end of the lever shown nearly horizontal the brass hook *3* which engages with the catch block. This catch block is rigidly attached to the valve lever *13*, which is keyed to the end of the valve stem, the latter transmitting motion to the valve. Attached to the valve lever *13* is the dashpot piston rod *4*. The hook is so made that it may be automatically tripped when the back part of the hook comes in contact with a cam which is operated by the

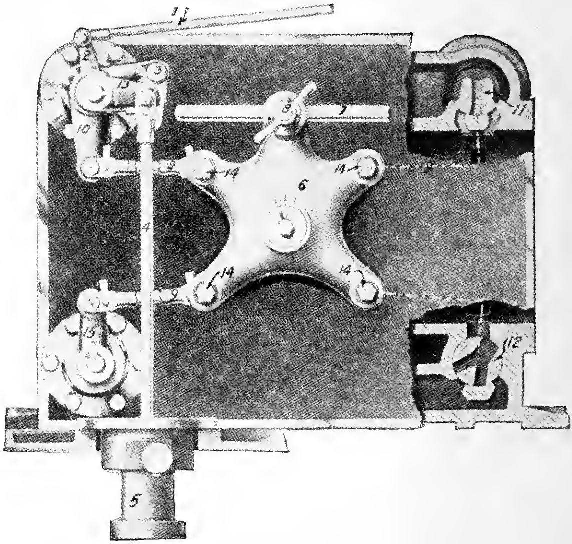


Fig. 27. Corliss Valve Mechanism in Detail

arm *2* connected to the governor by the reach rod *1*. The operation of the mechanism is such that the hook may be disengaged at any point of its travel by means of the cam coming in contact with the tripping leg of the hook *3* and causing it to rotate on the pin and move the steel catch out of engagement with the catch block.

The slowing down of the engine, in consequence of reduced steam pressure or an increased load, causes the catch to hold its contact longer and the steam to be admitted longer. In the event that the speed be increased in consequence of increased steam pressure or diminished load, the hook would be tripped by the cam and the admission valve would be quickly closed by the vacuum dashpot *5*. It must be evident from the foregoing that the regulation obtained by this device must be very sensitive to any change of speed or load. The dashpot *5* closes the steam valve when the hook is tripped by the cam.

The cylinders have four cylindrical holes accurately bored at the four corners, as is shown at *11* and *12* in Fig. 27. Into these openings the valves are placed with their stems and proper packing devices. The seats of the valves are circular. The portion of the valve marked *2* and *1*, Fig. 28, is circular, whereas the remaining portion may have any shape, depending upon the requirements of the design. The valve stem *5-4-6* is also irregular in shape. The portion *4* fits into the slot *3* of the valve and round portions *5* and *6* serve as bearings and as means for attaching the driving mechanism.

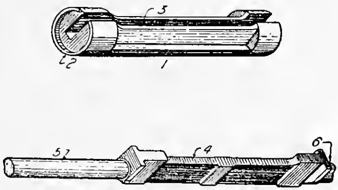


Fig. 28. Corliss Valve and Valve Stem

Advantages and Disadvantages of Corliss Type. Perhaps one of the chief disadvantages of the Corliss engine is the large amount of floor space required, a factor which often precludes its use. It possesses many advantages, however, chief among which may be mentioned the rapid and wide opening of the steam and exhaust ports; shortness and directness of ports, which results in small clearance; the adaptation of the steam valve to the functions of cut-off valves; and the location of the exhaust ports at the bottom side of the cylinder, thus draining the cylinders perfectly. Each of these various factors contribute to good engine performance, and their combination has resulted in making the Corliss engine one of the most economical engines manufactured. It will operate upon from sixteen to eighteen pounds of steam per indicated horsepower per hour.

Angle-Compound Type. As an outgrowth of the demand for an engine of high speed and one that will occupy a small space, but which, at the same time, will be economical in the use of steam, there has been developed the angle-compound engine shown in Fig. 29.

Balancing. In an ordinary high speed steam engine, the inertia of the reciprocating parts—namely, the crosshead, piston, and piston rod—and the crosshead end of the connecting rod, is considerable. If a steam engine is to be installed in office buildings, apartment houses, or in other houses where freedom from vibration is a prime requisite, it becomes almost a necessity for the engine to be perfectly balanced. On an ordinary reciprocating engine it is almost impossible to obtain perfect balancing for two reasons:

First, because of the angularity of the connecting rod, which causes the rate of acceleration of the reciprocating parts to be much faster at one end than the other, therefore, the counterweight which exactly

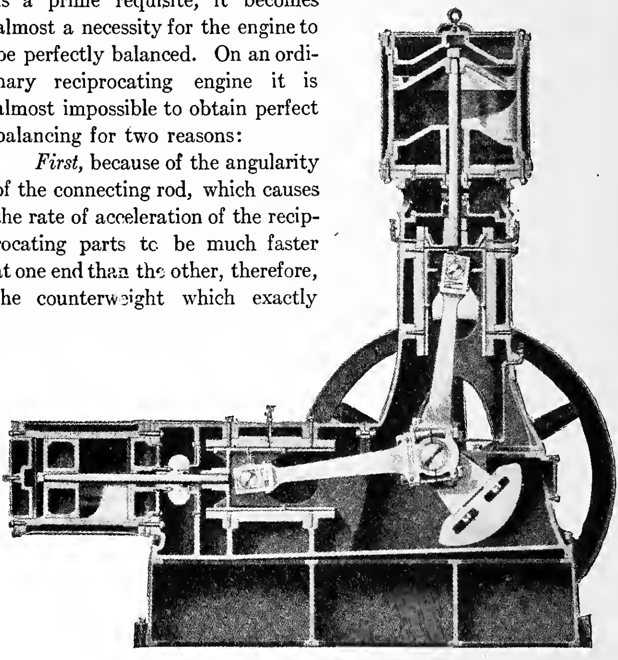


Fig. 29. Section of Angle-Compound Engine

balances the forces at one end would be either too light or too heavy at the other end.

Second, the counterweight at all positions in the revolution of the shaft exerts a radial force and when the counterweight is above or below the center of the shaft, there are no reciprocating parts developing a counteracting force, hence the centrifugal force of the counterweight exerts a powerful unbalanced vertical force. (This has been observed a number of times in locomotive practice where the rails have been bent by the extremely heavy blows of the unbalanced forces.)

In tests at Purdue University on their locomotive testing plant, it was clearly demonstrated that the unbalanced vertical forces are so great at high speeds that the locomotive driver is at times lifted clear off the track. It is obvious from the foregoing that the question of balancing is a serious one, and one that should be carefully considered. A thorough study of the question would involve considerable time and space and the use of higher mathematics.

The several engine builders who put the angle-compound engine upon the market claim for it an elimination of the balancing difficulty. As will be seen from the illustration, the angle-compound consists in combining two engines in such a manner that one crank pin serves both. The high pressure and the low pressure cylinders are placed at 90 degrees from each other in the plane of rotation of the crank. The horizontal engine is arranged so that it is perfectly balanced along its horizontal axis, but is, of course, badly out of balance vertically. On the other hand, the vertical engine is perfectly balanced along its vertical axis, but is out of balance in a horizontal direction. The above statements are true only when we consider each engine separately. When the engines are placed together, the unbalanced effect on one tends to neutralize that of the other. Their relation is such that the same counterbalance serves for both engines. It is claimed for this arrangement that there are four points in the revolution where a perfect balance exists and the resultant effect is to give almost a perfect balance. Another point of interest with these engines is that there are no dead centers; hence by employing a by-pass connecting the two cylinders, the engine can be easily started from any position of the crank.

Summary of Advantages. This type of engine, therefore, possesses the advantage of good balancing; it occupies about one-half of the floor space of a simple engine of the same power; and the compounding reduces its steam consumption considerably below that of a plain slide valve engine.

Uniflow Steam Engine. A type of engine which has a very extensive use in Europe, and which is just beginning to be manufactured in this country is the Uniflow engine. Mr. L. J. Todd, in 1886, took out patents in England covering the principle of the Uniflow engine, but Professor Stumpf of Charlottenburg, Germany, deserves credit for developing the engine and for making it a practical success. In Europe the high cost of fuel makes even small economies in the use of steam of considerable value. The Uniflow engine was designed to secure better economy in the use of steam than is possible with other steam engines of equal power. As used in Europe, the poppet type of valve is almost universally employed, because of the better results it gives with superheated steam, which is used practically to the exclusion of saturated steam. The

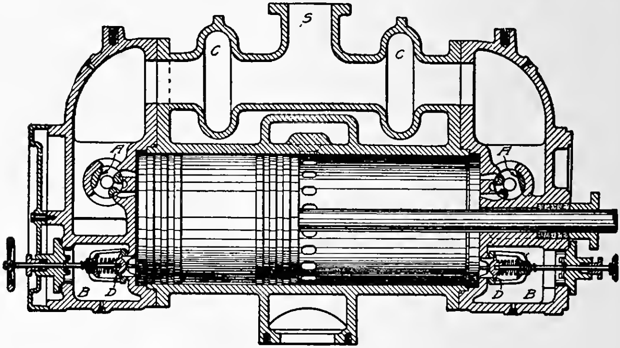


Fig. 30 Sectional View of Uniflow Engine
Courtesy of Nordberg Manufacturing Company, Milwaukee, Wisconsin

Uniflow engine is now manufactured in the United States by a few concerns, the Nordberg Company being one of the first to develop and build a successful engine of this type in this country.

Method of Action of Nordberg Engine. A cross-section view of the Uniflow engine as manufactured by the Nordberg Manufacturing Company is shown in Fig. 30. Steam comes to the engine through the inlet S and is led through the passages shown to either admission valve A. These valves are of the Corliss type, not only to conform with American practice, but because, with saturated steam, they give as good results as the poppet type and are less expensive to construct. The steam is exhausted through a ring of ports cast in the middle of the cylinder and is conducted away by the exhaust pipe shown below.

Thus it is seen that the piston performs the duty of an exhaust valve by uncovering and covering these exhaust ports. *D* is a relief valve of large size, communicating with chamber *B*, which is separated by a bridge wall in the cylinder head from the live steam space above. This relief valve serves two purposes: *first*, it relieves the cylinder of any water that may get into it; and *second*, it opens automatically in case the vacuum is lost and prevents the engine from compressing above line pressure. Also, if it is desired to run the engine non-condensing instead of condensing, the relief valve *D* may be backed off of its seat, thus giving the chambers *BB* as the additional clearance volume which is required for non-condensing operation. The drums *CC* on each side of *S* relieve the cylinder and cylinder heads from the strains caused by the expansion of the inlet pipe.

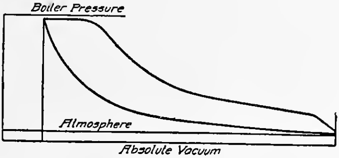


Fig. 31. Indicator card for Uniflow Engine Operating Condensing

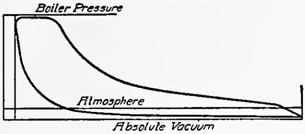


Fig. 32. Indicator Card for Uniflow Engine Operating Non-Condensing

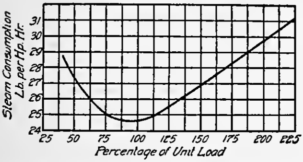


Fig. 33. Economy Curve for Uniflow Engine Operating Condensing

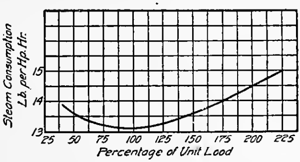


Fig. 34. Economy Curve for Uniflow Engine Operating Non-Condensing

Typical Indicator Cards. Typical indicator cards for a Uniflow engine with condensing and non-condensing operation are shown in Figs. 31 and 32, respectively. Figs. 33 and 34 show economy curves when operating, condensing and non-condensing. The cards show the effect of the large exhaust area by the rapid falling off of the pressure as soon as the piston has uncovered the exhaust ports and also the gradual and high compression which is obtained. The economy curves show that an overload of 100 per cent requires only 10 per cent more steam than for full-load operation when the engine runs condensing, and but 12 per cent more when operating non-condensing.

Chief Factor in High Economy. The chief factor in the high economy of the Uniflow engine is the great reduction of initial condensation. In the ordinary steam engine the piston head and the cylinder head in particular are exposed to the low temperature of the exhaust steam, which cools them considerably and leaves them cooler than the incoming steam. This causes the great loss known as *initial condensation.* In the Uniflow engine the exhaust steam does not pass out near the head of the cylinder, and so does not leave the

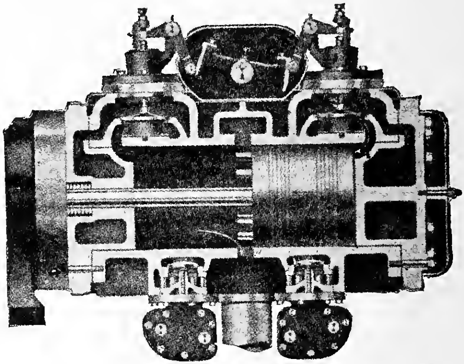


Fig. 35. Cylinder and Valve Arrangement in Skinner Unaflow Engine. Piston on Head-End Dead Center and Exhaust Taking Place Through Central Ports

cylinder and piston heads cool; and in addition, the compression is carried to line pressure, so that when a fresh supply of steam enters the cylinder it meets surfaces of practically the same temperature as its own. Furthermore, the walls of the cylinder in the Uniflow engine are exposed at each successive point in the stroke to temperatures which are more nearly the same than they are in the usual counter-flow engine; this also helps the engine economy.

Cylinder and Valve Arrangement in Skinner Engine. The form of cylinder and valve arrangement used in the Unaflow engine, built by the Skinner Engine Company, Erie, Pennsylvania, is illustrated in Figs. 35 and 36. It will be noticed that the steam valves are of the poppet type and are located on the top of the cylinder. Exhaust takes place through central ports in the usual way and also through the auxiliary exhaust valves shown on the bottom side. Fig. 35 shows the piston on head-end dead center with the steam valve at admission and exhaust taking place through the central ports. Fig. 36 shows the central exhaust ports closed, the steam valve on the head end closed, and exhaust taking place through the auxiliary exhaust valve on the crank end.

When the engine is operating non-condensing, the auxiliary exhaust valve for the end in question is opened by the valve-gear mechanism at the point when the central exhaust ports are just

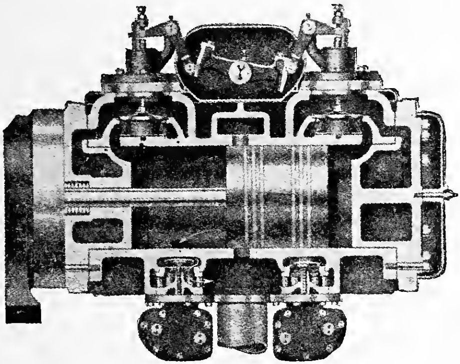


Fig. 36. Section of Unaflow Engine Showing Central Exhaust Ports Closed and Exhaust Taking Place Through Auxiliary Exhaust Valve on Crank End

closing, and compression commences at about 35 per cent of the stroke. When operating condensing, the valve-gear mechanism controlling the auxiliary exhaust is automatically disengaged when the vacuum reaches a predetermined amount. Under these conditions the auxiliary exhaust valves remain closed and compression begins at about 90 per cent of the stroke. The construction is such that if when operating condensing the vacuum should fail, the auxiliary exhaust valves are automatically thrown into operation.

American Locomobile. The locomobile is already highly developed in Europe, particularly in Germany, but in this country developments have just recently begun. On account of its high efficiency, a description of its construction and operation seems desirable.

This apparatus is really a complete power plant all contained in a single unit. It consists of a steam boiler with furnace, superheater, and reheater; a compound steam engine with condenser and vacuum pump; a feed-water heater; and a boiler feed pump. It is now being constructed by different American manufacturers, but that built by the Buckeye Engine Company will be taken as typical, since they were the first builders of locomobiles in the United States.

Details of Power Plant. Referring to Fig. 37, the path of the gases and steam can be traced through the plant, and some of the mechanical features can be seen. Consider first the boiler, furnace, superheater, and reheater. The former is an internally fired, fire-tube boiler, the combustion chamber being at *A* and the fire tubes at *B*. Beyond these tubes is the circular pipe coil *C*, which is the superheater, and still further on is the reheater *D*, consisting of loops of pipe expanded into two headers, as shown. The furnace gases from *A* pass directly through *B*, *C*, and *D*, and then out to the stack through the connection shown in the floor. The boiler is supported on cradle blocks *X*, and the superheater and reheater piping is hung from horizontal beams, as shown.

The steam is led from the dome *E* to the rear end of the superheater and leaves it at the front end, going straight up to the high-pressure cylinder of the engine. Leaving the high-pressure cylinder, the steam is carried to the front end of the reheater and, leaving at the opposite end, is conducted to the low-pressure cylinder of the engine. From the low-pressure cylinder the steam is conducted to a feed-water heater and then to the condenser, neither of which are shown in the figure. In both superheater and reheater, the steam is made to flow against the direction of flow of the furnace gases so that the steam will enter the engine cylinder at a higher temperature.

The engine, as can be seen, is mounted directly on the boiler. The saddle *F*, bolted to the boiler shell, supports the engine bed, which is securely fastened to it. The engine frame is permitted to slide on the saddle *G* so as to allow for unequal expansion between the boiler shell and the engine frame. At the head end of the low-pressure cylinder the yoke *Y Y* supports the cylinder casting.

Fig. 38 shows the relative temperatures of the steam and the

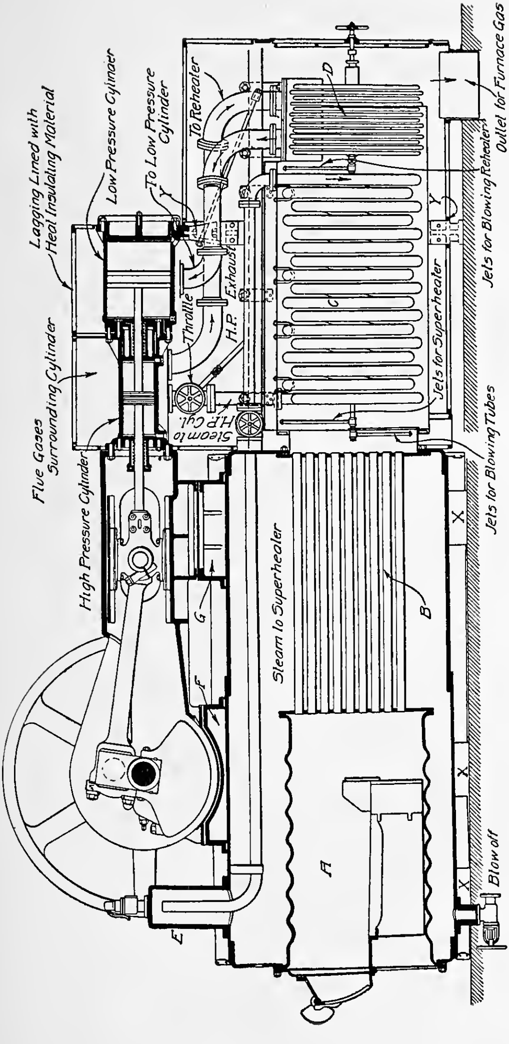


Fig. 37. Sectional View of First American Locomobile
Courtesy of Buckeye Engine Company, Salem, Ohio

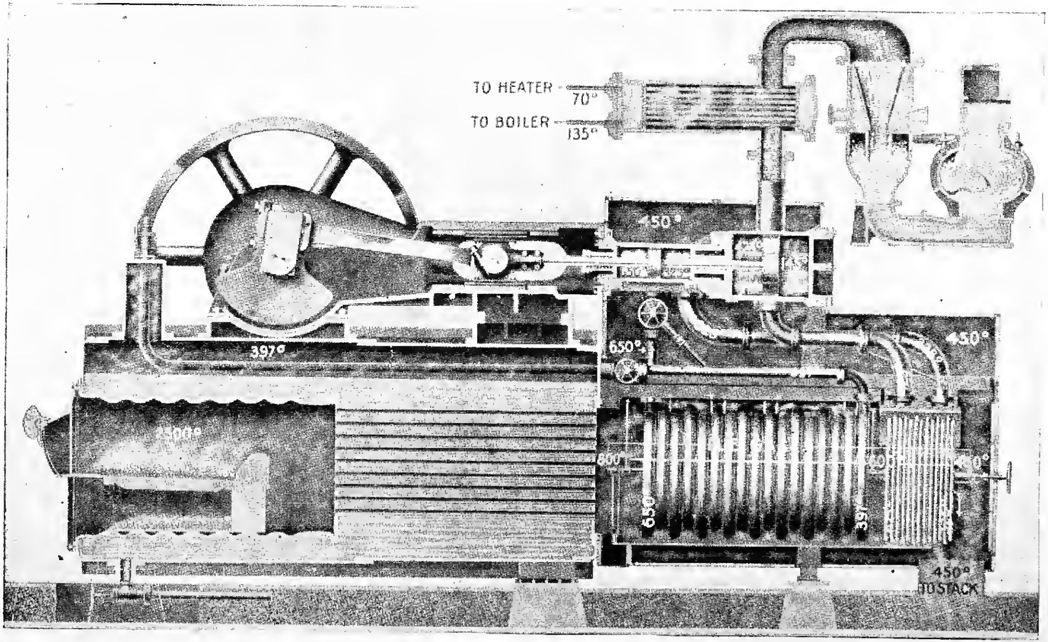


Fig. 38. Section of Buckeyemobile Showing Temperatures in Different Parts of the Power Units
Courtesy of Buckeye Engine Company, Salem, Ohio

gas at various points in their respective paths. The figure is self-explanatory, except that the feed-water heater, condenser, and vacuum pump shown in the upper right-hand corner are put there merely for convenience of illustration. In the actual locomobile they are situated alongside the boiler in some convenient manner. The temperatures here shown are approximate and would vary, of course, with different coals, combustion rates, steam pressures, etc. Fig. 38 also clearly shows the manner in which the waste gases circulate around the high- and low-pressure cylinders.

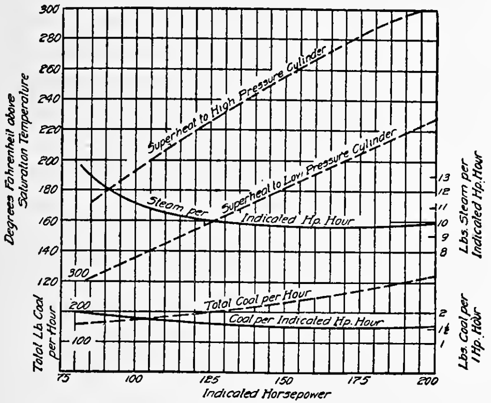


Fig 39 Graphical Results of Locomobile Economy Tests

The big saving aimed at in the locomobile is the reduction of radiation and condensation losses to a minimum. This would mean of course that a greater amount of work could be generated in the engine by the burning of the same amount of coal. In tests conducted in this country, the locomobile has generated an indicated horsepower on less than two pounds of coal, and has used approximately ten pounds of water. Fig. 39 shows graphically the results of tests, which are remarkable considering the size of the unit.

General Survey of Stationary Types. The treatment of the subject of mill or stationary engines, in so far as the scope of this work will permit, has been covered by the discussions given concerning the plain slide valve engine, the vertical engine of small units, the vertical engines for large installations, the compound and tandem engines which are being used more and more, and finally by a consideration of the most economical engine of all, the Corliss, whether operated simple or compound. In addition to the types mentioned there are a large number of other makes, which have distinguishing features and which give good service, but yet the principles enumerated in the types already discussed fulfill all the requirements likely to be made upon stationary plants. Hence a discussion of other makes is not thought necessary.

FARM OR TRACTION ENGINE

The advancement of scientific and progressive farming has made the farm engine of more interest and importance than ever before; in fact, the demands of the active farmers in recent years have taxed the builders of such equipment to the limit of their output. The steam engine is used for a large variety of purposes upon the modern large farm, and appears most commonly in the form of the so-called traction engine. It is used for plowing, digging ditches, building of roadways, logging purposes, running threshers, and numerous other purposes. Various types of stationary engines of small power are also to be found in use on the farm, the small gas engines now having been perfected to such a degree that they are rapidly replacing the steam engine.

General Description. The traction engine is really more than simply an engine; in fact, it is a self-contained power plant. It consists of a simple or compound engine, a boiler for supplying the steam required by the engine, and the transmission mechanism, together with all the auxiliaries necessary for a complete power plant. A good type of a general utility traction engine is shown in Fig. 40. It consists of a boiler of the locomotive type, carried by four wheels, the two front ones serving as a means for guiding, and the two rear being the ones which receive the power and known as the driving wheels. In order to prevent the slipping of the rear wheels when doing heavy hauling, they are made with heavy projecting lugs or cleats which are forced into the ground by the weight of the machine. The engine, which is mounted on the side of the boiler, as may be

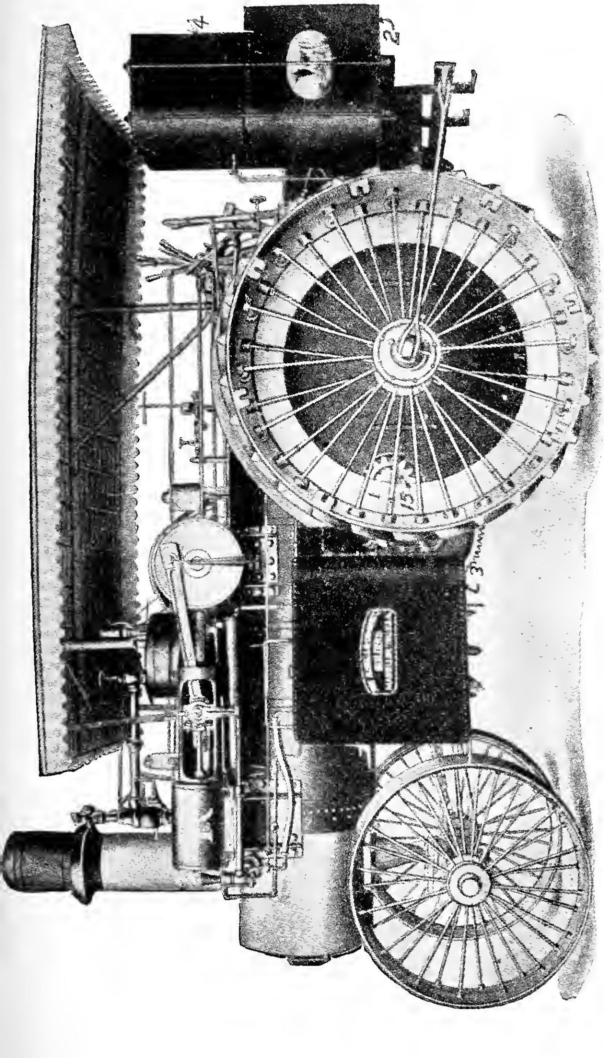


Fig. 40. Standard Type of Steam Traction Engine

seen in the illustration, is a plain slide valve engine of the side-crank type. The speed is regulated by an ordinary fly-ball, centrifugal governor. The construction of the various parts of the engine are similar to those previously described in this work. The same watchful care should be given to the lubrication, operation, and maintenance of this engine as to any other, when economy, durability, and reliability are desired. It should be noted that both cross-compound and tandem-compound engines are used as well as simple engines in this class of service, and that various types of valves find application.

In order to make clear the construction and operation of the traction engine, a view showing the rear wheels, platform, and side tanks removed is shown in Fig. 41. The means provided for guiding, reversing, and driving this engine is clearly illustrated. It is evident that the type of boiler used is similar to that of the locomotive boiler, having a narrow fire box. It has an extended front end *1* and stack *2* for carrying away the gases of combustion. The boiler is mounted upon the front wheels through the pivoted pedestal connection *3*. It is supported on the rear wheels, by having the rear axle extend beneath the fire box, or by having the supporting elements riveted to the side sheets as in Fig. 41.

Operation of Plant. *Reversing Mechanism.* The operation of the plant is about as follows: If the engineer desires to go forward the mechanism is placed in forward gear by means of the reversing lever *29*, the reversing being accomplished by means of a swinging eccentric, which can be thrown across the shaft at the discretion of the operator. (On some types of traction engines, a reversing link mechanism is used.)

Transmission. Having adjusted the reversing gear in accord with the desired direction, the throttle valve of the engine is opened by moving the lever *30*. The opening of the throttle valve starts the the engine shaft *12*, which carries the flywheel *11*. On the engine shaft behind the flywheel is keyed a small spur gear which is in mesh with the larger gear *13*, which in turn meshes with the gear *14*. As the engine shaft revolves, the small gear in the shaft revolves, which transmits its motion to *13* and on to the small gear *15*, which is keyed to the shaft driven by the wheel *14*. The gear wheel *15*, Figs. 40 and 41, is in mesh with an annular gear on the drive wheel *1*, Fig.

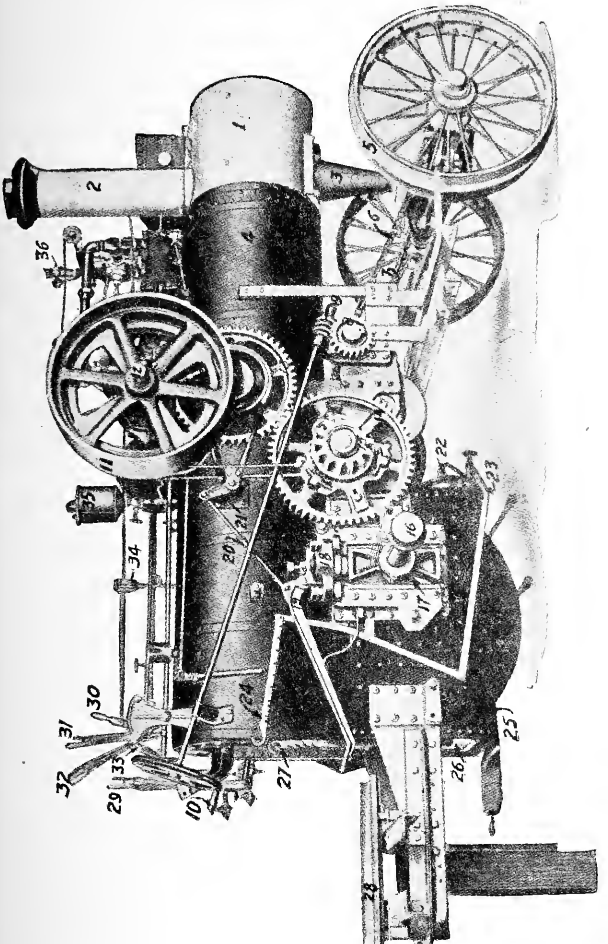


Fig. 41. Traction Engine with Rear Wheels, Platform and Side Tanks Removed to Show Steering and Driving Mechanism

40; hence, by reason of this connection, the large wheel is made to revolve. The shaft carrying gear wheel *15* extends beneath the boiler to the opposite side and drives a set of gear wheels which causes the other driving wheel to revolve with the one just considered.

Running Gear. The axle *16* of the wheel has a sliding head *17* attached to it. This head is free to move up and down in guides securely fastened to the fire box. This sliding head works against a spring, which is contained in the box *18*. This spring reduces the shocks to which the machine is subjected when on the road, hence, the engine is much easier to ride than it otherwise would be. In addition to the easy riding qualities, it also relieves the parts of the machine of stresses and strains due to sudden jolts, which would be detrimental to the durability of the machine as a whole.

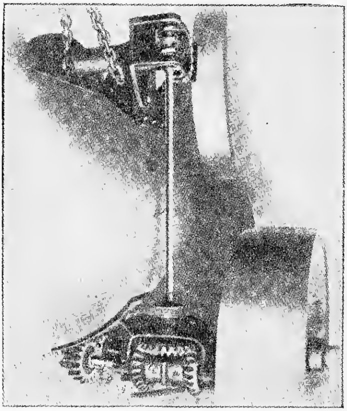


Fig. 42. Friction Gear Device for Steering Traction Engine

Steering Gear. The engine is guided by the hand wheel *10*. It will be noted that a chain is connected to the front axle on either side of the pivotal point. This chain wraps around a cam arrangement on the shaft which carries the small gear wheel *8*. The wheel *8* is in mesh with the worm *9*, which may be turned by the hand wheel *10*. If the driver when moving forward should wish to turn to the right, for instance, he would turn the hand wheel so the wheel *8* would be driven counter-clockwise, and in so doing the chain *6* would be shortened, the chain *7* lengthened, the wheel *5* would be cut in, and the machine would turn to the right. If it was desired to go in the opposite direction, the reverse operation would be carried out, that is, gear *8* would be revolved clockwise.

It is sometimes difficult to operate the steering gear by hand, especially in large traction engines and in places where a heavy load is being driven over rough ground; hence, some engines are provided with a friction gear device. This attachment, Fig. 42, is exceedingly simple, and when it is used the engine furnishes power to guide itself. It consists of a shaft *2* extending from the worm gear *1* to a bracket on the side of the boiler in front of the main shaft. On top of this vertical shaft is a horizontal miter gear *3* arranged to engage alternately with two vertical gears, one at the right *5* and the other at the left *4*. These vertical gears are on a shaft run by a chain of small gearing *6* from the engine shaft. They are thrown in or out, at the pleasure of the operator, by means of a shifting yoke which is worked by a straight rod extending back to the right-hand side of the engineer. A lever at the end of this rod is within easy reach all the time. By moving it forward or backward, the engine is guided to the right or left, as desired. If the lever remains at the center, the engine guides straight. An extension rod is placed on the rear end connecting with a hand lever at the left side of the platform, so that the engine may be guided equally well from either side. To operate this steering lever requires no appreciable exertion on the part of the engineer.

Friction Clutch. A friction clutch is provided in the flywheel, which permits the engine to be operated without driving the machine forward on the road. With the engine running at full speed, the clutch can be gradually thrown into action, and the machine will start forward on the road without any sudden shocks. The clutch is operated by the lever *31*, Fig. 41. By disconnecting the engine from the flywheel, a high speed can be obtained, so that by throwing the clutch in gear quickly the engine is often able to pull the machine out of difficult places. Oftentimes it is desired to operate the engine independently of the traction wheels for the purpose of running the thresher, saw mill, electric generator, or for other purposes, hence some form of clutch is necessary.

Brake. A friction brake is operated by a system of levers and rods as *19*, *20*, and *21*, Fig. 41. The operator can apply the brake by pushing downward upon the foot piece on the lever *19*. The amount of air admitted to the fire box is controlled by the two dampers *22* and *26*, which may be manipulated by the levers *24* and *27*.

Water Tanks. In Fig. 40, large tanks *2*, *3*, and *4*, are shown. These tanks are water reservoirs from which the supply pumps take water and deliver it to the boiler. Opposite the tank *2* is a bin for holding the fuel, which may be wood, coal, or straw, depending upon the location and character of work to be done. If the traction engine is used for threshing purposes, it would have a fire box arranged for burning straw; whereas if it was being used in a logging camp or a saw mill, the available fuel would be wood, hence the fire box would be constructed accordingly.

Boiler. Since it is necessary to have a high-grade, durable, and economical boiler in order to have an efficient and reliable machine, it is thought advisable to call especial attention to the type of boilers used in this connection and point out some of their good and bad features. It was mentioned in the description of the traction engine, Fig. 40, that a locomotive type of boiler with some modifications was used. Fig. 43 illustrates such a boiler. It is of the fire tube horizon-

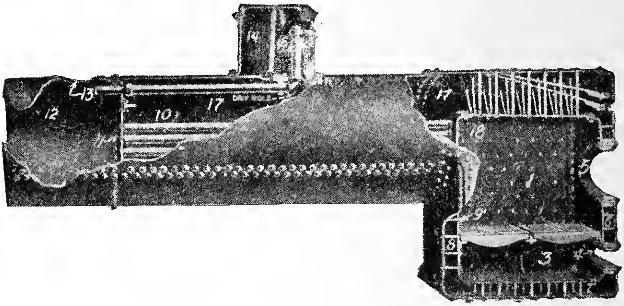


Fig. 43. Traction Boiler of the Locomotive Type

tal type. The fire box *1* is of a horizontal rectangular construction with open grate bars *2* and ash pit *3*, below. The fuel, either wood or coal, is fed through the fire door *5*, and the ash is removed through the door *4*. The products of combustion, such as smoke, hot gases, etc., pass through the tubes *10* into the front end *12*, from whence they are exhausted through the opening *13* and the smoke stack into the atmosphere. If straw is to be used as fuel, a brick arch is placed in the fire box which deflects the gases toward the fire door so that, after passing over the arch, they are drawn out through the tubes in the usual manner. It is also necessary to put in different grate bars where straw is used, as the bars must be closer together so the fuel will not drop through into the ash pan.

It will be noted that the fire box is surrounded by water legs *6*, *7*, and *8*, and the water and steam space *17*. Water is also circulating around the tubes and is several inches deep above the crown sheet *18*. As combustion takes place in the fire box and the hot gases pass through the tubes, the plates of the fire box and the tubes become heated. As a consequence, the water in contact with these hot surfaces becomes heated also, and steam is formed which rises to the top of the boiler, entering the steam dome *14* from whence it is taken by the pipe *15* through a throttle valve to the cylinder. By using a steam dome a better quality of steam is obtained, because it is so far above the water level that less water is carried over by the steam into the steam pipe *15*.

This type of boiler has many advantages as well as some disadvantages. It has a large amount of heating surface and it is well distributed. Due to the large amount of heating surface and the excellent draft arrangement, a high evaporation per square foot of heating surface is obtained. It is well adapted to various classes of service and operating conditions. Its disadvantages consist largely in the cost incurred in its maintenance especially in localities where bad water must be used. When this condition is imposed upon it, the flues give trouble by leaking around the joints where they enter the flue sheets *9* and *11*. This leakage may at times become troublesome and in the end costly if proper preventive measures are not taken regularly. Some criticism is also made of this boiler on account of the necessity of using stay bolts in the crown sheet and water legs. It must be admitted that stay bolts are also an item of considerable expense in bad water districts where high steam pressures are used. But by watchful care and manipulation this boiler will give splendid results and for some classes of service it has no equal.

The type of boiler, shown in end view in Fig. 44 and in longitudinal cross section in Fig. 45, is a modification of the well-known and efficient Scotch marine boiler. The boiler consists primarily of a cylindrical fire box *1* enclosed by a circular shell. About midway of the fire box is placed a bridge wall *7*, which deflects the hot gases upward against the shell of the fire box. Ordinary cast-iron grate bars are inserted as at *4*, with the ash pit below. It is to be noted there is a water space *6*, which extends the entire distance around the circular fire box. Above the fire box there are a number of return tubes *3*, which take the hot gases from the rear end *8* of the boiler to the smoke stack. The path of these gases is indicated by the arrows. To protect the rear sheet from the heat of the gases, a protection plate *9* is riveted or bolted to the plate. As steam is generated it rises, enters the steam dome *12*, passes into the steam pipe *13*, and on to the engine.

It should be noted that this boiler contains no stayed portions and that all the surfaces are circular in form and securely riveted. There being no stayed surfaces the circulation of the water is not interfered with —which is an important consideration—and the opportunity for scale and sediment to collect is greatly reduced, hence there is less likelihood of portions of the boiler becoming heated to the point of injuring the boiler or impairing its safety. Still another feature of interest in the boiler is that the gases are made to traverse the entire length of the boiler twice before being ejected at the stack. This being the case an opportunity is given for a greater portion of the heat contained in the gases to be absorbed by the water, thus securing a higher thermal efficiency than obtained from boilers of the locomotive type. Having no stayed surfaces and a small number of flues results in a small maintenance cost of this type of boiler.

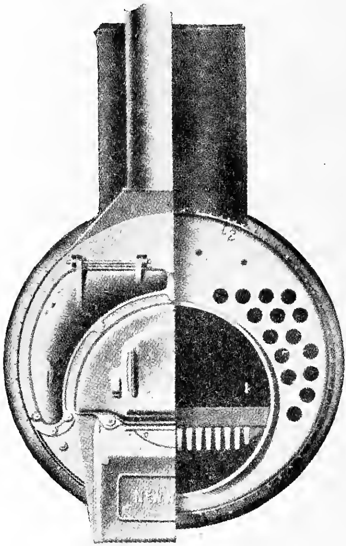


Fig. 44. End View of Modified Scotch Marine Boiler

Traction engines run in sizes from about $7\frac{1}{4}$ inches $\times$ 10 inches to 12 inches $\times$ 12 inches for single engines, and for compound engines the common sizes are $5\frac{3}{4}$ inches $\times$ $8\frac{1}{2}$ inches $\times$ 10 inches to $9\frac{1}{4}$ inches $\times$ 13 inches $\times$ 11 inches. The corresponding horsepower

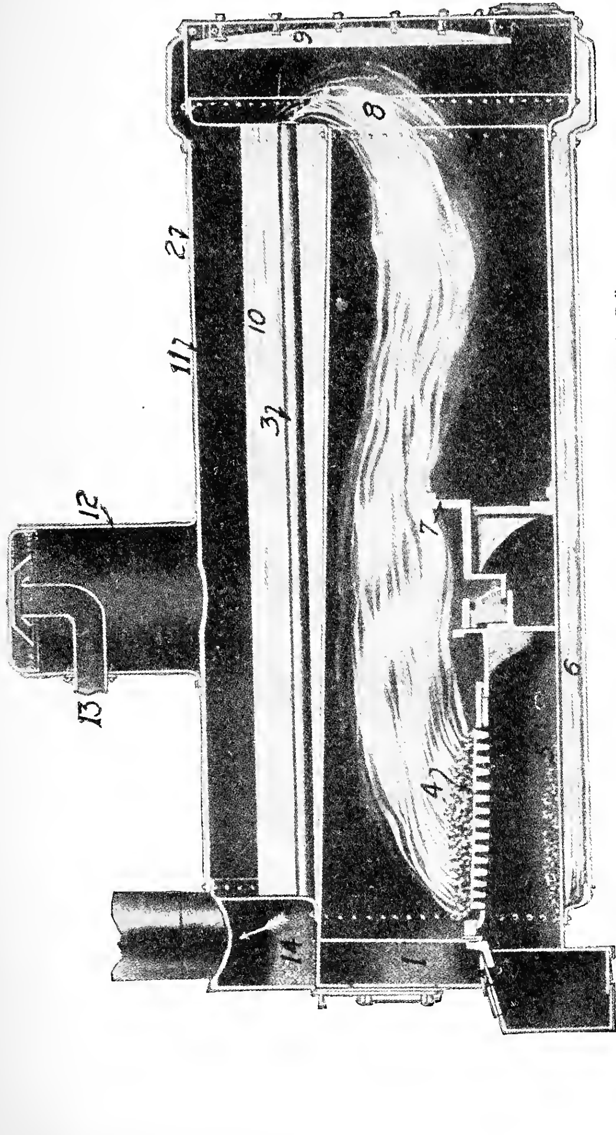


Fig. 45. Longitudinal Cross-Section of Modified Scotch Marine Boiler

developed will run between 15 and 100. The speed attained on the road in miles per hour is about $2\frac{1}{2}$ to 5.

Road Roller Type. The traction engine just considered as an agricultural engine may also be considered as a portable engine or a road locomotive. A portable engine is, therefore, one that can be easily moved about from place to place, or as in the case of the traction engine it may be mounted upon wheels and self-propelled.

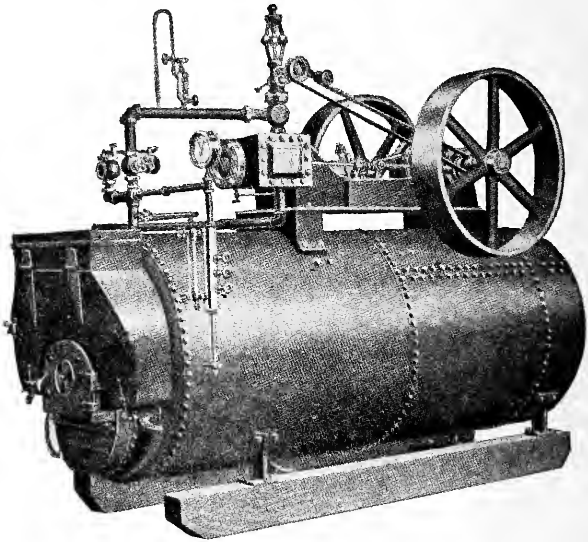


Fig. 46. Semi-Portable Engine and Boiler

Another illustration of a similar type is the ordinary road roller, or road locomotive as it is sometimes called. The principle of its construction and operation is similar to the traction engine, the chief difference between the road roller, or road locomotive, and the ordinary traction engine being that the two front wheels of the traction engine are replaced by a large smooth roller, or cylindrical weight, which revolves as the engine moves. The drive wheels of the road roller are also made large, heavy, and contain no cleats or lugs. These rollers are used in the making of macadamized or other forms of roads.

Semi-Portable Type. The semi-portable engine is usually connected with a small boiler and the two together may be moved from place to place as required. It is not mounted upon wheels but rather on large wood skids, and is moved by being placed either in a wagon or on rollers. It is largely used for hoisting purposes in connection with the construction of large buildings, bridges, etc.

Since the portable and semi-portable engines have no transmission mechanism, they are lighter and considerably cheaper to construct than traction engines.

A very neat, compact, and serviceable type of semi-portable engine is illustrated in Fig. 46. It is mounted upon skids so that it may be easily moved about. The engine is mounted on top of a Scotch marine boiler, similar to the boiler last described, and is of the plain slide valve, center-crank type, with a centrifugal governor. The boiler is equipped with a pressure gauge, water glass, and such other appliances as are usually found in a boiler room of moderate size. The boiler used is sometimes of the locomotive type and, oftentimes, both engine and boiler are of the vertical type. The smaller units are usually of the vertical type, the larger ones of the horizontal type. The semi-portable plant is built in sizes ranging from about 20 to 70 horsepower. If the semi-portable plant, Fig. 46, be mounted on wheels and drawn by horses or some other means, then it is usually classed as a portable engine as distinguished from a semi-portable or traction engine.

LOCOMOTIVE ENGINES

It is not within the province of this work to fully discuss the modern railway locomotive, but suffice it to say that no other power-developing unit has been so rapidly developed with such economical results. Considering the exacting demands made upon a locomotive, its performance is remarkable. The locomotive consists of two primary elements, namely, the boiler which generates the steam and the engines which convert the energy of this steam into useful work by giving motion to the transmission mechanism.

Boiler. Fig. 47 illustrates a modern locomotive boiler. It consists of a cylindrical barrel and an enlarged rear end which contains the fire box. The fire box is securely fastened in the boiler shell by stay bolts and radial stays. A few rows of sling stays are sometimes

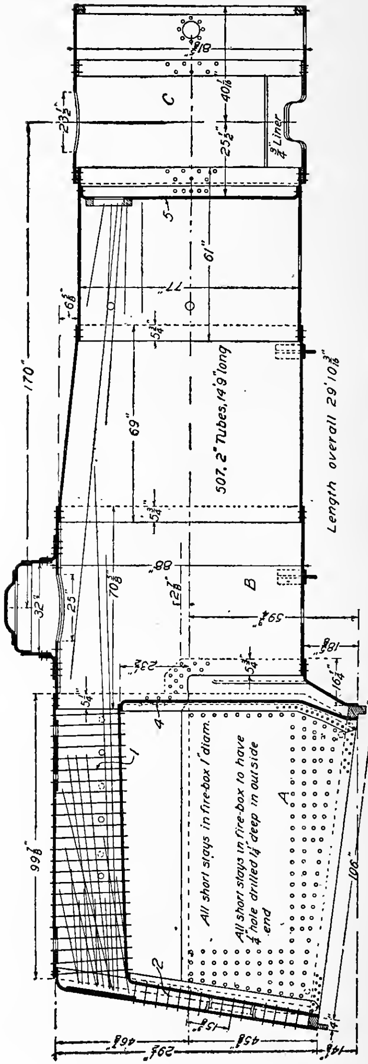


Fig. 47. Detailed Drawing of Modern Locomotive Boiler

used at the front end of the fire box to allow for expansion and contraction of the sheets. The boiler is divided into three distinct departments, as the fire box *A*, the water space *B*, and the smoke box *C*. The sheets *4* and *5*, which separate these departments, are known as the back and front flue sheets, respectively. The flue sheets are drilled with holes to receive the flues.

Flues. In the particular boiler illustrated about 400 2-inch flues are used. These flues extend from flue sheet to flue sheet and form a passage for the gases to travel from the fire box to the smoke box. Surrounding the flues in the space *B* and surrounding the fire box is water, which is vaporized into steam due to the combustion of fuel in the fire box. The total amount of heating surface will vary from 2,500 to over 4,000 square feet, according to the type and size of the locomotive. Of this total amount of heating surface only a very small per cent is furnished by the fire box, there being usually only about 200 square feet of heating surface contained in the fire box. It is evident, therefore, that the flues are a very important part of the locomotive boiler.

Grate Area. The amount of grate area varies from about 40 to 60 square feet. It must be obvious that in order for so small a grate area to supply sufficient heat to such a large amount of heating surface there must be a very high rate of coal consumption per square foot of grate area. A series of tests made at St. Louis during the Exposition in 1904 demonstrated that the amount of dry coal fired per square foot of grate area per hour varied from 20 to as high as 130 pounds. These results were obtained from several different types of locomotives operated under widely different speeds and loads, hence the above figures may be taken as approximating the maximum and minimum consumption under ordinary running conditions. Under these very widely different operating conditions it was found that the equivalent evaporation per pound of dry coal varied from $6\frac{1}{2}$ to 12 pounds, which compares very favorably with stationary boiler performance which gives an average evaporation of about 8 pounds of water per pound of coal.

Mechanical Efficiency. The mechanical efficiency of a locomotive is also very good. Through a long series of tests conducted on a well-equipped locomotive testing plant, a mechanical efficiency of 65 to 85 per cent was obtained. The same degree of efficiency has been obtained in various other tests and under more adverse conditions. The locomotive is also very efficient in the use of steam. The St. Louis tests showed that simple freight locomotives gave an average minimum water consumption per indicated horsepower per hour of 23.67 pounds. The water consumption per indicated horsepower per hour under maximum load was 23.83 pounds, whereas the maximum rate was 28.95 pounds. For compound freight locomotives the average steam consumption was: minimum load 20.26 pounds, maximum load 22.03 pounds, and maximum consumption 25.31 pounds. The average steam consumption for simple passenger locomotives was: minimum load 18.86 pounds, maximum load 21.39 pounds, and maximum consumption 24.41 pounds. When these figures are compared with those of the best stationary engines, some idea of the economy of the locomotive can be obtained. The steam consumption of an automatic, tandem-compound, condensing stationary engine with piston valves under full load is about 18 pounds per indicated horsepower per hour, whereas the compound non-condensing locomotive is about 21 pounds. A Corliss engine or a medium speed, four-valve simple engine will give a minimum steam consumption of about 22 pounds per indicated horsepower per hour under full load. A simple freight engine under full load will use about 23.5 pounds of steam per indicated horsepower per hour. The foregoing figures speak well in favor of the economy of a steam locomotive, which is operated under conditions unfavorable to the securing of good economy.

Engine Characteristics. The engines used on locomotives may be simple or compound; in fact, both are used extensively, although the simple type predominates. It is to be noted that the steam locomotive is equipped with two separate and distinct engines—one being attached to each side of the boiler, and both attached to the driving wheels through the medium of the frames, etc.

The mechanical construction of these engines is quite similar to that of the type already described in this work. Certain features are made necessary in order to properly tie together the engine, boiler, and transmission mechanism. Perhaps the most noticeable change in detail is in the construction of the cylinders and valve seats, otherwise there is little variation from the well-established principles of engine design. The valves, rods, crossheads, guides, etc., are made of the same high-grade material and constructed in the same first-class manner as is required for a good stationary engine. This being true, much of the discussion of the steam stationary engine and its parts already given is applicable to the engines of a locomotive. There are, however, many perplexing questions that arise with reference to the performance and operation of the locomotive as a whole that are never encountered in stationary practice, due to the unusual and sometimes trying conditions under which the locomotive must be operated. The solution of these problems demands a great amount of ingenuity and engineering ability.

To discuss the various types of locomotives and tell the many interesting and important points connected therewith would require entirely too much space, so the discussion must be confined to narrow limits.

Types of Locomotives. There are certain types of locomotives common in American practice which have special names. The eight-wheel or "American" passenger type of locomotive has four coupled driving wheels and a four-wheeled truck in front. The "ten-wheel" type has six coupled drivers and a leading four-wheel truck. This type is used for both freight and passenger service. The "Mogul" type is used altogether for freight purposes; it has six coupled drivers and a two-wheel or pony truck in front. The "Consolidation" type is used for heavy freight service. It has eight coupled drivers and a pony truck in front. There are also a great many special types for special purposes. In switch yards a type of engine is used which has four or six drivers with no truck. The Forney type has four coupled driving wheels under the engine and a four-wheel truck carrying the water tank and fuel. This type is used on elevated roads largely. "Decapod" engines are a type used for heavy freight service, having ten coupled driving wheels and a two-wheel truck in front. A tank engine is one which carries the feed water in tanks on the engine itself instead of in the tender, as in other engines.

A locomotive of modern design that is being largely used for fast freight service and for heavy passenger service is illustrated in Fig. 48. It is commonly known as the Atlantic type locomotive, having four leading truck wheels, four coupled drivers, and a two-wheel trailing truck. The leading truck wheels serve in a guiding

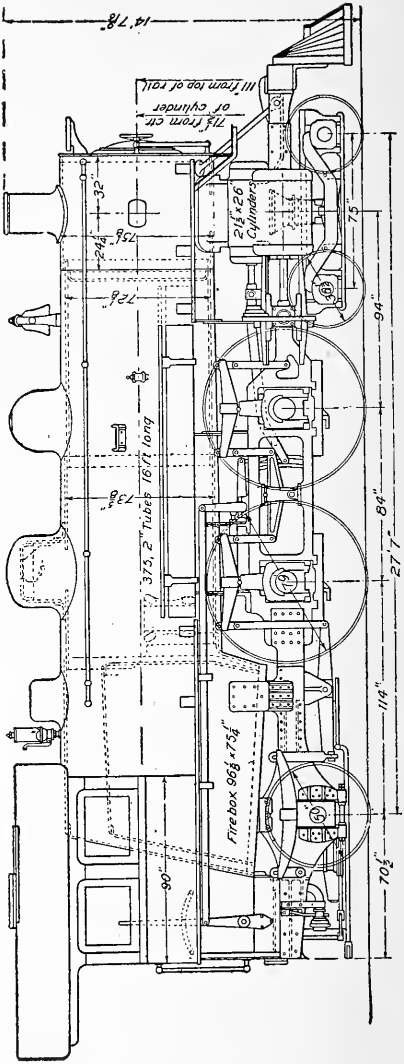


Fig. 48. Detailed Drawing of Modern Locomotive Engine of the "Atlantic" Type

capacity. This engine is a compound type with piston valves, is well designed, is neatly proportioned, and admirably fulfills every requirement.

Compound Type. In connection with the subject of compounding just mentioned it may be said that in recent years the compound locomotive has been found in increased numbers on American railroads. A type of compound that has given especial good service and which is being adopted by many roads for heavy hill climbing duty is the Mallett Articulated Compound. The adoption of the compound locomotive has been due to a general opinion among railroad officials that the findings of a committee of the American Master Mechanics Association were true, as demonstrated by practice. This Committee says of compounding:

(a) It has achieved a saving in the fuel burned, averaging 18 per cent at reasonable boiler pressures.
(b) It has lessened the amount of water to be handled.
(c) The tender can, therefore, be reduced in size and weight.
(d) It has increased the possibilities of speed beyond sixty miles per hour, without unduly straining the engine.
(e) It has increased the haulage power at full speed.
(f) In some classes of engines it has increased the starting power.
(g) It has lessened the valve friction per horsepower developed.

A number of other reasons are given in their report. Notwithstanding these facts, however, the compound locomotive has not come into very general use on railroads.

WATER PUMPS

The subject of pumping engines is a very broad one, and one which has received the thought and study of the most eminent engineers for many decades. From the earliest history of man there is gleaned the fact that human ingenuity and skill had been devoted in those early times to the perfection of some kind of power pump. It would be a difficult matter to mention an industry of any character or description but what a pump was needed somewhere in the enterprise. It was first used in a large way in the mining industries for pumping water out of the mines. Today it is found in all power houses, mines, and factories of various kinds. Both the large and small cities depend upon it for their water supply. The heating and ventilating systems of modern apartment houses and office buildings use the pump, and mention might be made of many other instances where the water pump is indispensable.

There are two general classes of pumps, namely, crank or flywheel type and direct acting pumps.

Crank or Flywheel Type. The crank or flywheel type was the first form to be developed. These pumps vary greatly both in their design and in the details of their construction. They are of varying sizes, including some of the largest and most expensive in the world. As a general thing they are used in heavy hydraulic enterprises, for furnishing water supply for cities, and in various other enterprises where a large and constant supply of water is demanded. In this class of pumps or engines the application of the power in the steam cylinders in driving the pump plunger or piston varies greatly both in design and detail of construction. Long or short beams or bell cranks may be used and sometimes gearing may be employed, but in all cases the limit of the stroke of the steam piston and of the pump plunger is governed by the crank of a revolving shaft. In pumping engines it is not absolutely necessary to have a revolving shaft, the only requirement being that the piston in the pump cylinder shall be driven back and forth with a plain reciprocating motion which may be exactly like that of the steam piston. For this reason, in early pumping engines and also in modern engines, the reciprocating motion of the steam piston is applied directly, or through a beam, to produce the reciprocating motion of the pump piston or plunger without the use of any revolving part. Frequently, however, it is desirable to use a flywheel so that the steam may be used expansively, and in these cases, of course, a revolving shaft must be used.

Cameron Belt-Driven Pump. The power pump used as an illustration, Fig. 49, is a belt-driven one. The belt is placed on the pulley *1* and can be shifted to a loose pulley by the shifter *2*, when desired. The shaft *4*, which is driven by the belt pulley, extends across the frame and has attached to it a flywheel *5* and a small gear wheel, which meshes with the large gear wheel *3*. The gear wheel *3* is keyed to the crank shaft *6*, hence, when it is driven, the crank shaft is made to revolve, which in turn gives a back and forth movement to the piston as in the ordinary steam engine. The flywheel *5*, attached to the revolving shaft, may be of greater or less diameter and weight, depending on the condition under which the pump is to be operated.

In addition to assisting the crank to pass the dead center at each end of the stroke of the piston, it can be employed as a reservoir in which any excess energy may be stored at the beginning of each stroke and drawn out during the latter part of the stroke, where the force of the water column is greater than that of the steam. By this means it is possible to use shorter cut-offs in the cylinder than could otherwise be permitted; hence, a resulting saving in steam. Many means may be used to drive the power pump. While the illustration shows

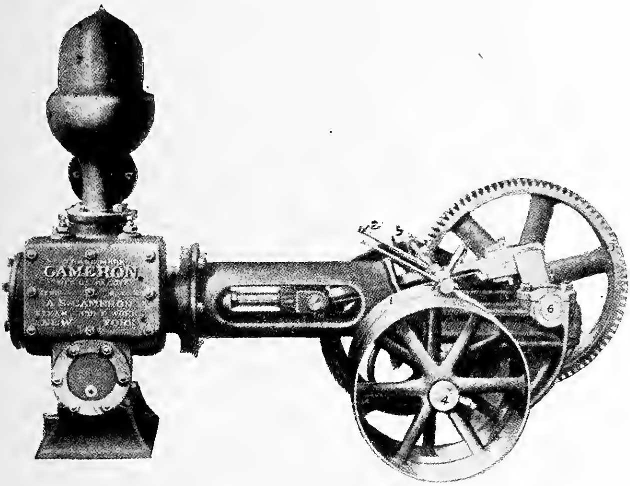


Fig. 49 Belt-Driven Power Pump

one belt driven, yet they are frequently electrically driven, and sometimes the revolving shaft is attached to the shaft of a gas or steam engine.

Deep-Well or Mine Pump. For deep-well or mine pumping, the cylinders are often set in a vertical position directly over the pump cylinder. The piston rod extends from the steam eylinder directly below to the pump plunger. Sometimes it is possible to use steam expansively in these pumps by reason of the weight of the reciprocating parts. When the weight is sufficient, the steam can be cut off before the end of the stroke and the momentum of the parts will be enough to just finish the stroke, consequently these pumps are sometimes compounded. They are used only in pumping from very deep wells.

Direct-Acting Type. A direct-acting steam pump is one in which there are no revolving parts, such as shafts, cranks, and fly-wheels, the power of the steam in the steam cylinder being transferred to the piston or plunger in the pump in a direct line through the use of a continuous rod or connection. In pumps of this construction there are no weights in the moving parts, other than those required to produce sufficient strength in such parts for the work they are required to perform and, as there is consequently no opportunity to store up power in one part of the stroke to be given out at another, it is impossible to cut off the steam in the steam cylinder during any

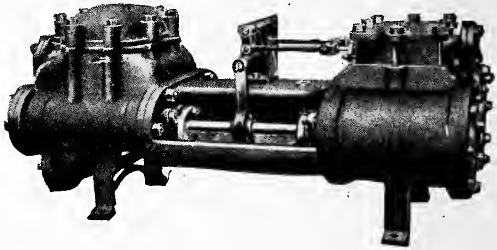


Fig. 50. Direct-Acting Duplex Pump with Rocker and Bell-Crank Lever

part of its stroke. The uniform and steady action of the direct-acting steam pump is dependent alone on the use of a steady uniform pressure of steam through the entire stroke of the piston, against a steady, uniform resistance of water pressure in the pump; the difference between the power exerted in the steam cylinders and the resistance in the pump governs the rate of speed at which the piston or plunger of the pump will move. The length of the stroke of the steam piston within the steam cylinders of this class of pumps is limited, and is controlled alone by the admission, compression, and release of the steam used in the cylinders.

Duplex Pump with Rocker and Bell-Crank Lever. The direct-acting steam pump, Fig. 50, is known as a duplex pump and consists simply of two direct-acting steam pumps placed side by side. The steam pistons are at one end and the water pistons at the other.

The steam pressure acts directly on the pistons; no flywheel is used; and since the reciprocating parts are comparatively light and there is no revolving mass to carry by the dead points, it is evident that in the ordinary form there can be no expansion of steam. The pump is inexpensive and gives a positive action. It uses a relatively large quantity of steam, but for small work the absolute amount is not very great.

On the piston rod of each pump is a bell-crank lever which operates the valve of the other pump. There must be a rocker on one side and a bell-crank lever on the other, because of the relative motion of the valves and pistons. The first piston, as it goes forward, must use a rocker, because it draws the second valve back. The second piston, as it goes back, must use a bell-crank lever because it must push the first valve back in the same direction as its own motion. The two pistons are made to work a half-stroke apart, thus one begins its stroke when the other is in the middle. In this way a steady flow of water is obtained, as both pumps discharge into the same delivery pipe. In large pumps of this kind, and even in some small ones, the motion described above merely admits steam to a small auxiliary piston on each steam cylinder, which then moves the main steam valve by steam pressure.

Duplex Pump with Tappet. Some pumps operate the steam valve by means of a tappet instead of a rocker and a bell-crank lever, Fig. 51. Its construction and operation is as follows:

A is the steam cylinder; *C*, the piston; *L*, the steam chest; *F*, the chest plunger, the right-hand end of which is shown in section; *G*, the slide valve; *H*, a lever, by means of which the steam-chest plunger *F* may be reversed by hand when expedient; *II* are reversing valves; *KK* are the reversing valve chamber bonnets; and *EE* are exhaust ports leading from the ends of the steam chest direct to the main exhaust and closed by the reversing valves *II*.

The piston *C* is driven by steam admitted under the slide valve *G*, which, as it is shifted backward and forward, alternately connects opposite ends of the cylinder *A* with the live steam pipe and exhaust. This slide valve *G* is shifted by the auxiliary plunger *F*, the latter having hollow ends which are filled with steam, and this, issuing through a hole in each end, fills the spaces between it and the heads of the steam chest in which it works. Pressure being equal at each end, this plunger *F*, under ordinary conditions, is balanced and motionless; but when the main piston *C* has traveled far enough to strike and open the reverse valve *I*, the steam exhausts through the port *E* from behind that end of the plunger *F*, which immediately shifts accordingly and carries with it the slide valve *G*, thus reversing the pump. No matter how fast the piston may be traveling, it

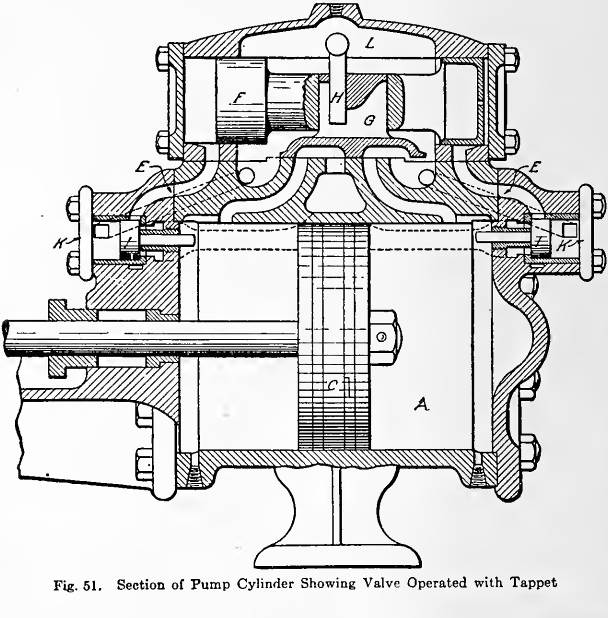


Fig. 51. Section of Pump Cylinder Showing Valve Operated with Tappet

must instantly reverse on touching the valve *I*. In its movement the plunger *F* acts as a slide valve to close the port *E* and is cushioned on the confined steam between the ports and steam-chest cover. The reverse valves *II* are closed as soon as the piston *C* leaves them by a constant pressure of steam behind them conveyed direct from the steam chest through the ports shown by dotted lines.

The motion of the piston *C*, Fig. 52, is transmitted through the rod *M* to the water piston in the cylinder *R*. As the piston moves back and forth, water enters through the intake valves O and leaves through the discharge valves immediately above, and finally leaves through the delivery pipe P. In order to create a more continuous flow of water, an air chamber Q is provided. Any sudden variation in the pressure in the line is taken up largely by the air chamber. It also serves to lessen the effect of water hammer.

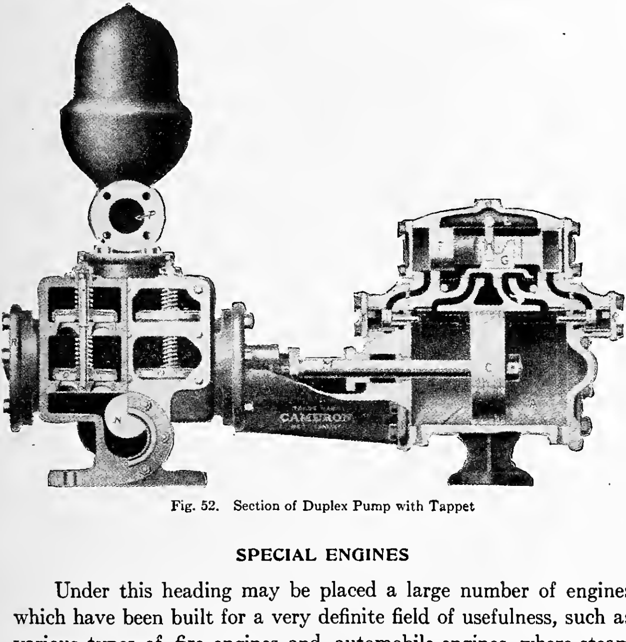


Fig. 52. Section of Duplex Pump with Tappet

SPECIAL ENGINES

Under this heading may be placed a large number of engines which have been built for a very definite field of usefulness, such as various types of fire engines and automobile engines, where steam is used as the motive force. Again a number of experimental engines have been built, commonly known as freak engines, having peculiar construction and design, which never got beyond the experimental stage. Rotary engines as well as rotary pumps have been used to some extent, but the rotary engines thus far developed have been so extravagant in steam consumption that their use has been discontinued. It is thus seen that under the head of special engines many of the engines already discussed, as well as an untold number of others of more or less merit, may be properly classed.

The special engines referred to above were not mentioned for the purpose of studying them, but rather to indicate that outside and distinct from the steam engines classified and considered, there are a large number of special types that should not be entirely ignored.

MARINE ENGINES

The subject of Marine Steam Engines is a broad and important one, and to treat it properly would require one or two volumes the size of this one. However, it seems desirable to discuss the subject in a very general way in connection with the still broader subject of The Steam Engine, and thus give the student a general idea of marine engine parts.

Definition of Terms. Before taking up the subject, it is thought advisable to present a brief statement of nautical terms used in

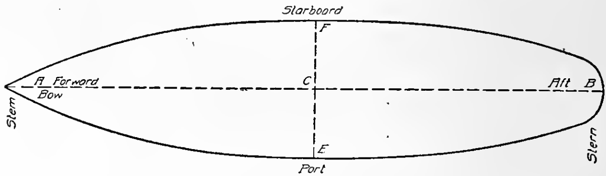


Fig 53. Plan of Vessel Showing Different Parts

describing a vessel. Fig. 53 shows a plan of a vessel. The front part A is called the *bow;* the extremity B is called the *stern.* An object placed near the bow is said to be *forward;* if near the center C, it is *amidship;* and if near the stern, it is *aft.* An article, if placed so that its major dimension is parallel to the line AB, is said to be placed *fore and aft.* Thus the crankshaft of a triple expansion engine of a vessel is located along the line AB and is sometimes spoken of as a fore-and-aft engine. An article located crosswise of the vessel, that is, at right angles to AB, is said to be placed *athwartship.* To one standing on the deck facing the bow, the *starboard* side is on his right and the *larboard,* or *port* side, on his left.

The width of a vessel FE is its *beam,* and the perpendicular distance from its lowest part to the surface of the water is called the *draft.* The length of a vessel is the horizontal distance between perpendiculars drawn at its extreme ends. The displacement of a vessel is equal to the weight of water it displaces and is usually expressed in long tons.

The *speed* of a vessel is usually expressed in *knots* per hour, but is sometimes given in miles per hour. A knot is equal to about $1\frac{1}{6}$ miles.

Methods of Propulsion. Speaking in a general way, the propulsion of a steam vessel is accomplished by causing a mass of water adjacent to the ship to move in a direction opposite to that of the ship. Motion is imparted to the water in one of the following three ways: (1) by paddle wheels; (2) by screw propellers; and (3) by jets of water or hydraulic propulsion.

The oldest of these three forms of propulsion, the paddle wheel, is still much used in lake and river steamers and ferry boats; but for ocean-going vessels and in many boats on inland waterways, the screw propeller has supplanted it. Jet or hydraulic propulsion has not proved to be practical and for this reason has never been used in commercial work.

TYPES OF ENGINES

Beam Type. The first steam vessels were fitted with paddle wheels, and as beam engines were the most common, this form of engine was used. Its construction, however, was somewhat modified for this service. This arrangement of beam engine and paddle wheel was used for many years and was applied to ocean vessels as well as to small river boats. It is still used, especially in this country, on river steamers and some coast steamers. The beam is supported by a large A-frame on the deck, and the engines are about on a level with the shaft.

Engines of this type take up rather more room than those now in common use, partly because of great size, and also because of the shaft and paddle wheels. Another disadvantage is that in heavy weather when one paddle wheel is thrown out of the water the other is deeply immersed and takes all the strain, so that there is a tendency to rack the boat. Then again if the boat is loaded heavily, the paddle blades are very deeply immersed; while if light, they barely touch the water. It is difficult to handle the engines satisfactorily under either condition.

Inclined Type. The introduction of the screw propeller overcame these difficulties very largely and at the same time required a high speed engine. At first, the increased speed was supplied by the use of spur-wheel gearing, but gradually higher speed engines were built and connected directly to the propeller shaft. It was, of course, difficult with small width at each side of the shaft to use horizontal engines, therefore various arrangements of inclined engines were used before the vertical engine was finally chosen by all as the standard form for marine work. It is only in recent years that the vertical engine has become general in naval work and in merchant steamers.

Vertical Type. In merchant ocean steamers the common form has three cylinders set in line, fore and aft, above the shaft, the cranks being set 120 degrees apart in order to give a more even turning moment. The three cylinders are worked triple expansion, the valves being usually of the piston type on the high and intermediate and double-ported slide type on the low. Sometimes piston valves are used on all the cylinders. Plain slide valves are not suitable for high-pressure work of any kind. While steam turbines are used to some extent in ocean-going vessels, the majority of ships in this service are equipped with high-speed, vertical, multicylinder engines direct connected to the propeller shaft.

Cylinder Arrangement. The different arrangement of marine engine cylinders commonly found in service is shown in Figs. 54 to 57.

Tandem- and Cross-Compound Types. In Fig. 54, *A* is the tandem-compound arrangement with its single crank; *B* is the cross-compound with cranks set 90° apart; and *C* is the three-cylinder compound with cranks set 120° apart. In arrangement *C*, the high-pressure cylinder is sometimes placed between the two low-pressure cylinders.

Triple-Expansion Type. The cylinder arrangement, Fig. 55, is found only on the larger vessels, and is spoken of as the triple-expansion type. In this type there are three cylinders to each engine, and they are called the high-, intermediate-, and low-pressure cylinders, each succeeding one being of larger volume than the one preceding. Fig. 55 illustrates two arrangements of the cylinders of triple-expansion engines. In arrangement *A* the cylinders follow

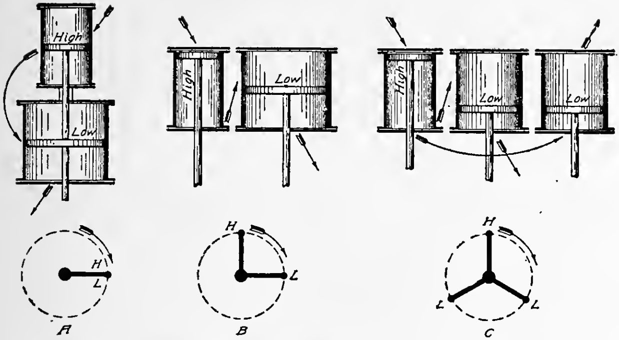


Fig. 54. Diagrams of Tandem and Cross-Compound Cylinder Arrangements

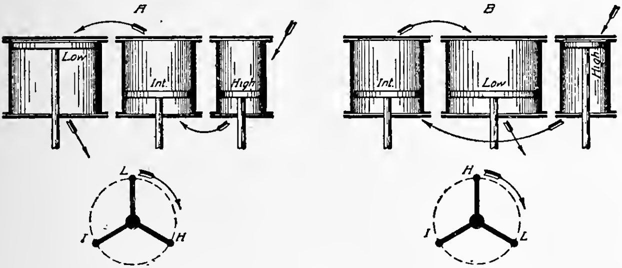


Fig. 55. Diagrams Showing Triple-Expansion Cylinder Arrangements

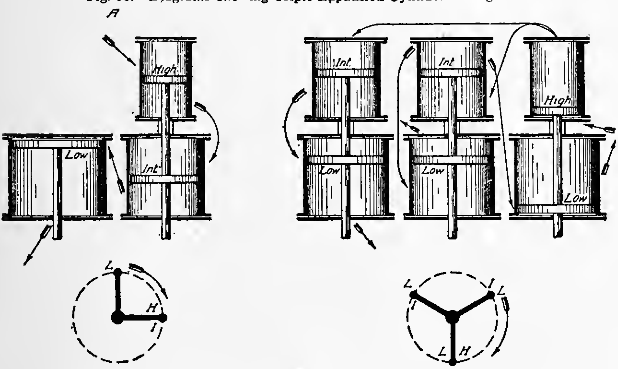


Fig. 56. Diagrams Showing Other Triple-Expansion Cylinder Arrangements

each other in natural sequence; this requires the least length of piping. Arrangement *B* is frequently used, but requires more piping than arrangement *A*. Another common arrangement is to put the high-pressure cylinder in the center of the group. In any of these systems the cranks would be set at 120°, giving a more nearly uniform turning movement to the shaft, since each cylinder will develop approximately one-third the total horsepower of the engine.

Still other arrangements of the cylinders of triple-expansion engines are found in Fig. 56. Arrangement *A* gives the effect of a tandem-compound between the high- and the intermediate-pressure cylinder and a cross-compound between these two and the low-pressure cylinder—an arrangement which results in cranks being

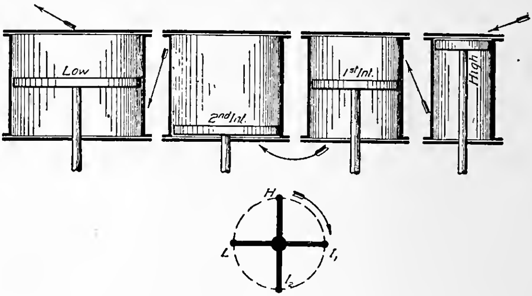


Fig. 57 Diagram Showing Quadruple-Expansion Cylinder Arrangement

set at 90° with the consequent uneven turning effect, but it is sometimes resorted to because of lack of space for all three cylinders in line. Arrangement *B* is a triple-expansion engine, having six cylinders. Here the volume of one intermediate-pressure cylinder is divided among two cylinders, and the volume of one low-pressure cylinder among three cylinders. This form is very expensive and is not often used. The arrangement requires less floor area than would be required for the same power in a three-cylinder engine.

Quadruple-Expansion Type The last cylinder arrangement to be considered is found on the quadruple-expansion engine In this type the steam goes from the high-pressure cylinder to the first intermediate, then to the second intermediate, and finally to the low-pressure cylinder The volume of each cylinder is larger than that of the preceding cylinder. There are many different arrangements of cylinders possible with quadruple-expansion engines. Fig. 57 shows the arrangement of cylinders in their natural sequence with the four cranks set 90° apart, which gives a slightly more even turning effort

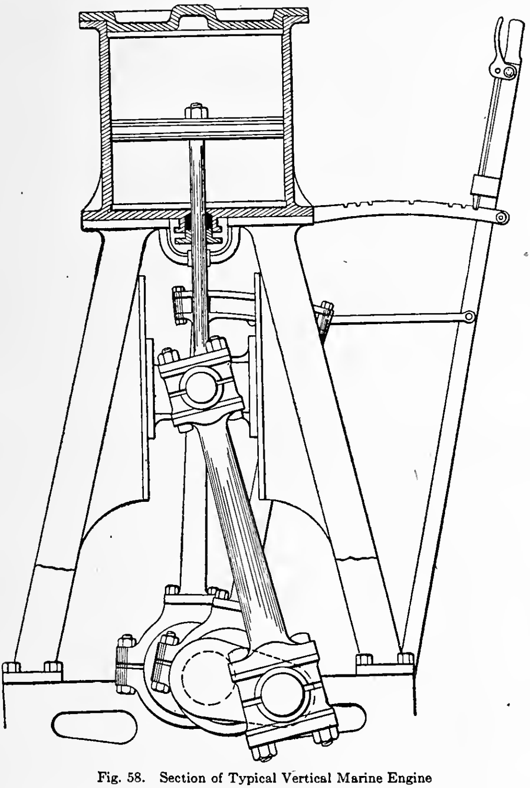


Fig. 58. Section of Typical Vertical Marine Engine

than is obtained with cranks at 120°, as in the triple-expansion engine.

The idea in the design of a quadruple-expansion engine is to produce an engine more economical in the use of steam than is obtained in any other type. With high-pressure steam, say 200 pounds and over, it gives a better economy in the use of steam than does the triple-expansion engine. However, the saving effected in the use of less steam is, to a very large extent, offset by an increase in first cost, operating cost, and general upkeep.

Comparison of Marine with Stationary Types. Fig. 22, page 29, and Fig. 58 show cross-sectional views of marine engines. In marine work many different designs of engines are used. These two views are intended to present merely the general features and characteristics of the marine engine. In comparison with stationary engines attention is called to the different form of frame used, lighter frames, different details of the connecting rod, and in the latter figure the separate crankshaft for each cylinder and the single crosshead guide. Also the cylinders are of complicated form and have double walls, and the pistons are of a cup shape. These points will be brought out more in detail in what follows.

ENGINE DETAILS

Cylinder. The general type of steam cylinder for a marine engine consists of three distinct parts, namelv, the shell, the liner, and the cover.

Shell. In Fig. 59, the shell is the outer casting forming the outside cylinder wall, the lower cylinder head, and the steam ports. As its complicated form makes the casting of the shell a difficult matter, an iron is used that runs freely in the mould. Sometimes the lower cylinder head is not cast integral with the shell, but is fitted to it separately like the cover.

Liner. The liner is the plain cylindrical casting or bushing, which forms the inner cylinder wall. Its use is made necessary because the metal in the shell is of such composition that it will not wear well if the piston is permitted to work directly on it. The material of the liner is usually hard, close-grained cast iron. In some cases forged steel is used. It is secured to the shell by bolts through a flanged end, or by stud bolts. But one end is fastened to the shell, the other end being left free to expand under the influence of the higher temperatures to which the liner is exposed.

Cover. The cover forms the upper end of the cylinder. Usually it is made of steel to combine lightness and strength. Sometimes the cover is cast hollow so as to form a steam jacket for the cylinder head, but more often it is made of a single wall of metal, reinforced by radial ribs on the outside.

Marine Details Resemble Stationary. Many of the details of marine engines are so nearly like those of stationary engines in essential features, and the minor points of difference are so varied that special mention of them will not be made here. For illustrations of different parts, reference may be made to the earlier sections of this book. For example, Fig. 9 shows a typical marine piston and connection to the piston rod; Fig. 14 a typical crosshead and crosshead pin; Fig. 16 a connecting rod; and Figs. 18 and 19 typical main bearings.

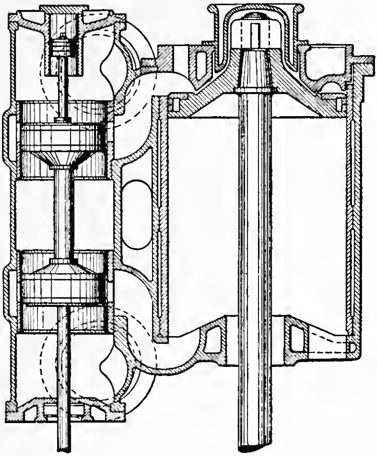


Fig. 59. Sectional View of Marine Engine Cylinder, Piston and Steam Ports

Crosshead Guides. A form of marine crosshead guide, differing from that ordinarily used in stationary work, is shown in Fig. 60. The crosshead used with this guide is known as the slipper type. It has but one bearing surface, and this runs in the space between *SS* and *A*. In the guide the plate *P* is bolted to the engine frame so that it receives all the crosshead pressure when the engine is running ahead. For backward motion of the engine the flanges *FF* are provided to receive the thrust.

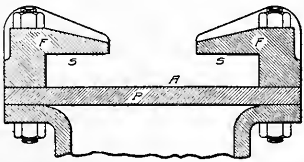


Fig. 60. Type of Marine Crosshead Guide

Cranks. In marine work side cranks are not used. The connecting rod is always connected between two crank arms. Furthermore, each cylinder of an engine has a separate crankshaft. These separate shafts are bolted together by flanges, as shown in Fig. 61. The dotted lines in this figure show how, in large shafts, the center is sometimes made hollow. This is done to make a saving in weight and to remove inperfect portions usually found in the center. The

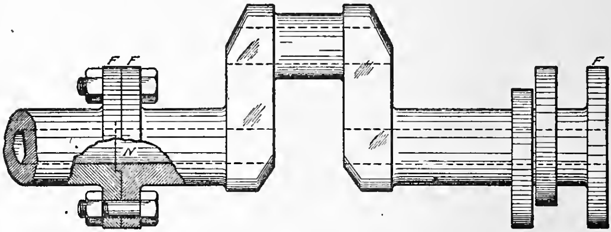


Fig. 61. Portion of Marine Engine Crank Showing Method of Bolting Sections Together

center of the shaft is the least effective of any part of it in resisting twisting forces, and the outside is the most effective. By using a little larger shaft, therefore, and removing considerable metal from around its center, a shaft of the same strength as a solid one is obtained, with a material saving in weight. The crankshafts are

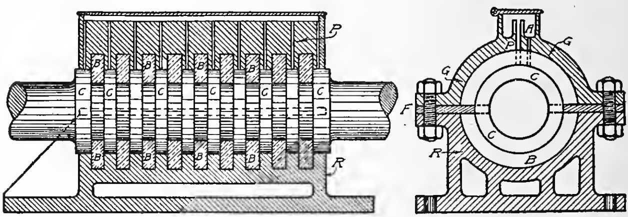


Fig. 62. Typical Marine Thrust Bearing

usually all made of the same size, so as to be interchangeable, and thus require fewer parts to be kept in stock.

Bearings. The bearing, Fig. 62, while not part of the engine, will nevertheless be discussed at this point. This is called the thrust bearing, and is used to relieve the engine of the thrust caused by the revolving propeller in screw-propelled vessels. The propeller shaft is turned with the collars *C* as a part of it. The cast-iron box *R* is secured to the frame of the vessel just aft of the main engines, and the cap *G* is bolted to it, as shown. The collars *C* press against rings *B* of gun metal or brass and transmit the propeller thrust to them, and thence to the vessel. Rings *B* are split and are prevented from turning by the tongue piece *F*. Holes *P* are for lubrication and holes *A*

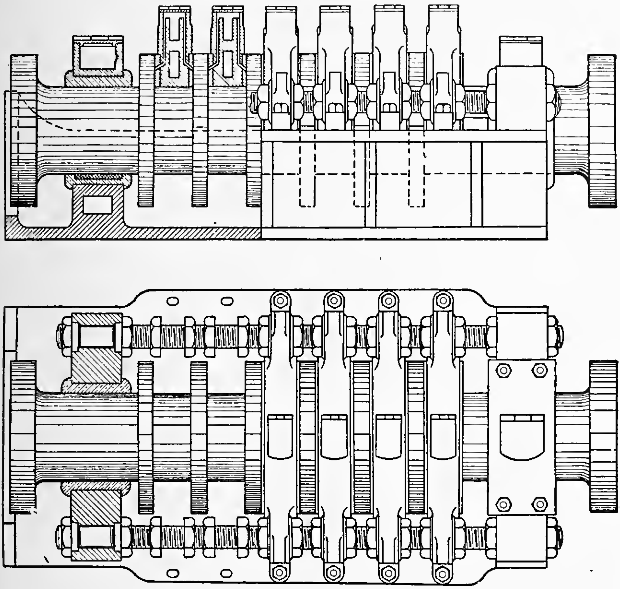


Fig. 63. Type of Thrust Bearing in Which Provision Is Made for Taking Up Wear

are provided for water cooling when needed. Water may also be circulated through the base.

Fig. 62 shows the principle of the thrust bearing, but it is not much used because no provision is made for taking up unequal wear between the brasses. Fig. 63 shows a type of bearing in which provision is made for this feature, the wear being taken up by means of the nuts fitted to the long screws at either side of the thrust bearing.

Thrust Bearing Calculations. The number of collars required in any given thrust bearing depends primarily on the total thrust that will come on them. There may be a large number of collars of small diameter or a small number of large diameter. The experience of the designer is usually the determining factor as to the number used.

Knowing the number of collars required, their diameter may be computed from the following formulas, in which n is number of collars; D is diameter of collars; d is diameter of shaft; P is total thrust; and p is safe allowable pressure per square inch of area, which is usually taken as 60 pounds per square inch.

First taking the formula expressing the total thrust, we have

$$P=p\left(\frac{\pi D^2}{4}-\frac{\pi d^2}{4}\right)n$$

and substituting for p the value of 60 pounds per sq. in., there results the formula

$$P=60\times\frac{\pi}{4}(D^2-d^2)\,n$$
$$=47\,(D^2-d^2)\,n$$

Transposing in the last formula and solving for the value of D, there results the equation

$$D=\sqrt{d^2+\frac{P}{47n}}$$

which gives the diameter of collars required for the conditions assumed.

AUXILIARY APPARATUS

The auxiliary apparatus aboard a ship is far more numerous than would be suspected by one not acquainted with it or even by one familiar with the apparatus in stationary power plants. The general features of some of the more important pieces of apparatus, only, will be described.

Reversing Mechanism. The reversing mechanism of large marine engines is so large and heavy and, at times, has to be moved so quickly that it cannot be done by hand. Consequently, in some instances a small steam power cylinder is attached to the reversing gears to move them. This apparatus is called the steam-starting gear and is under the control of the engineer.

The action of this gear, Fig. 64, is as follows: When the reversing lever, or handle, is moved from the mid-position A to B, the rod CE is moved to the left. This movement raises the rod H, which is connected to the lever fulcrumed at T. As the rod H raises, the rod

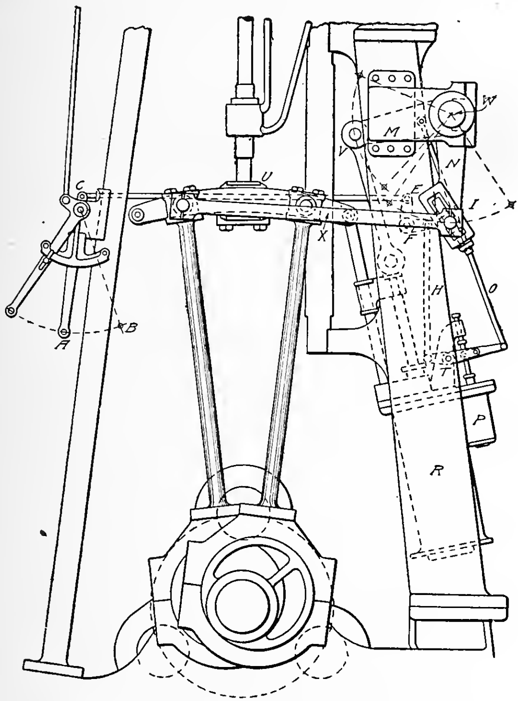


Fig. 64. Details or Steam Starting Gear

O moves downward, thus causing the arm M to move downward and the arm N to move to the right. This movement of the arm N and pin I causes a corresponding movement to the right of the reach rod and link, to which it is connected. Thus it is readily seen that the movement of the reversing lever A moves the link slightly and at the same time causes steam to be admitted to the power cylinder, which acts on the piston and aids in the movement of the links.

Condensers. *Surface Type.* In marine work the surface condenser is used almost exclusively, because with this type the cooling water (sea water) does not come in contact with the steam, and the latter can then be used over and over in the boilers. Jet condensers on ocean vessels would prevent the continued use of the condensed steam because of the deposit the salt of the water would leave on the boiler tubes and shell.

Keel Type. In small boats, such as steam launches, the surface condenser would occupy much valuable room and add considerable weight, so a substitute, called the keel condenser, is frequently used. This consists of several rows of copper tubes placed outside the hull along the keel of the boat. The engine exhaust enters at one end of these tubes, is condensed by the sea water in contact with the outside of the tubes, and is then drawn out of the condenser by the air pump and pumped back to the boiler. This form of surface condenser requires no circulating pump.

Pumps. *Centrifugal Type.* The pump most often used on shipboard to circulate condenser cooling water is of the centrifugal type, driven by an independent engine or motor. The absence of valves in this kind of pump is of advantage, as is also the fact that it can be run through a greater range of speed and, consequently, give greater volumes of water when occasion demands. Oftentimes the piping is so arranged that these pumps can draw from the engine room bilge and discharge without passing the sludge through the condenser.

Air and Vacuum Types. Of the many kinds of air or vacuum pumps used, the one shown in Fig. 65 has been chosen for description as being a good example and one easy to understand. The operation of the pump is as follows: The inlet E is piped to the outlet of the condenser. On the up-stroke of the piston P, a partial vacuum is formed below it, enabling the condensed steam and air in E to rush through the foot valves F and into the pump cylinder B below the piston. After reaching the upper limit of its stroke, P descends, producing a slight pressure on the air and water entrained in the cylinder, which closes the foot valves against the escape of the cylinder contents. As the piston continues, the bucket valves H in the piston are forced open, permitting the escape of air and water to the space above. On the next up-stroke this air and water are forced out of the air pump through the delivery valves A and the outlet N.

A small check valve, or pet-cock (not shown in the figure), is usually located in the cylinder wall B just below the delivery valves. When insufficient air comes through with the condensed steam to properly cushion the piston on the up-stroke, this valve is opened to provide the required air for cushioning. This air does not affect the degree of vacuum, because it is on the discharge side of the pump, where pressure on the piston is immaterial as regards vacuum in the condenser.

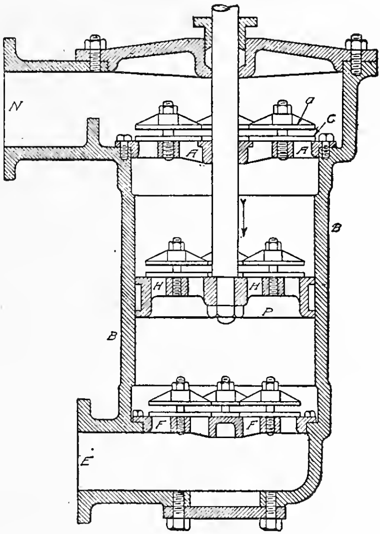


Fig. 65. Section of Air or Vacuum Pump

Vacuum is measured by gages similar to those used for measuring high pressures, but calibrated to read in inches of mercury instead of pounds per square inch. A column of mercury under atmospheric pressure will stand about 30 inches high. Consequently, since 30 inches of mercury is equal to about 15 pounds per square inch, one inch of mercury will be equal to 15 divided by 30, or nearly one-half pound per square inch, the exact value being 0.49 pounds per square inch. Suppose the vacuum gage of a condenser reads 26. This means there has been a reduction of pressure corresponding to 26 inches of mercury or 26×0.49, or 12.74 pounds per square inch. If the atmospheric pressure is 14.7 pounds per square inch, then there remains in the condenser $14.7 - 12.74$, or 1.96 pounds per square inch absolute pressure.

Besides the auxiliary apparatus already mentioned, there are many more on large and small vessels which cannot be discussed here, such as machines used for ventilation, forced draft, steering. weighing anchor, operating hoists and capstans, compressed air and refrigeration machines, and electric lighting.

PROPULSION

Process of Starting. When the engines are started and the screws or paddle wheels of a ship begin turning, there is no appreciable motion of the ship for a short time. During this short time the work done by the propellers is all used in overcoming the inertia of the vessel. As the inertia is overcome, the ship gradually begins to move and increase its speed. As the speed increases, the resistance offered to the motion of the ship through the water also increases. When all the power of the screws or paddle wheels is used in overcoming the resistance of the water to the passage of the ship through it, then the ship will be moving along at an approximately constant speed.

Resistance Factors for Ship in Motion. In smooth, quiet water the resistance offered to the ship's motion may be divided into three elements, namely: (1) frictional resistance of the hull; (2) eddy-making resistance; (3) wave-making resistance.

The most important of these is the frictional resistance or skin friction. The amount of this resistance depends on the area and the length of the immersed surface of the hull, the roughness of this surface (whether covered with barnacles, sea-weed, etc.), and the speed of the ship.

Eddy-making resistance, which is usually small, is caused by eddy currents following just astern of the ship and by the churn of the propellers.

Wave-making resistance is caused by the waves made at the ship bow.

Winds and waves also offer resistance to a ship, but the amount of resistance due to these causes is difficult to estimate.

Variations of Resistance with Speed of Vessel. It has been shown by experiment that for a given ship, the resistances vary almost directly as the square of the speed, and that the power required to overcome these resistances varies almost as the cube of the speed. That is, if at a speed of 10 knots an hour a ship encounters a certain resistance R and requires a certain power P, if the speed be increased to 20 knots, the resistance will be increased to R^2 and the power to P^3.

Indicated Thrust. Indicated thrust is a mathematical expression denoting the ratio of the total work in foot-pounds done by the main engines to the distance through which this force acts. Expressed as a formula, this ratio becomes

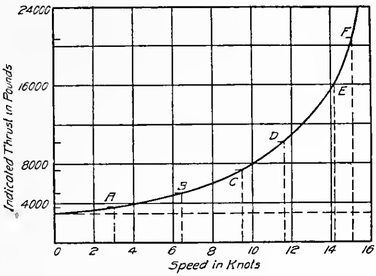


Fig. 66. Curve Showing Indicated Thrusts for Different Speeds

$$T = \frac{33{,}000 \times I.H.P.}{pN}$$

where T is indicated thrust in pounds; $I.H.P.$ is indicated horse-power of engines; p is pitch of screw in feet; and N is number of revolutions per minute. Since $I.H.P. = \frac{2\ PLAN}{33{,}000}$, this formula may be reduced to the form

$$T = \frac{2\ PLA}{p}$$

Where P is equivalent mean effective pressure in pounds per square inch; L is length of stroke in feet; and A is area of low-pressure piston in square inches.

EXAMPLE. What is the indicated thrust of a 1200 I.H.P marine engine driving a propeller of 20-foot pitch at 90 r.p.m.?

SOLUTION.

$$T = \frac{33{,}000\ \text{I.H.P.}}{pN} = \frac{33{,}000 \times 1200}{20 \times 90} = 22{,}000 \text{ pounds}$$

The indicated thrust for any given ship may be taken from a curve, such as is shown in Fig. 66.

Economical Speed. The most economical speed of a ship is that speed at which it can travel a given distance with the least consumption of fuel. At speeds either above or below this particular speed, the fuel consumption will be increased. To determine the most economical speed, the amount of coal used at different speeds is determined by trial. These amounts are then plotted, as shown in Fig. 67. As it stands, this curve shows merely the coal consumed at different speeds, but by drawing a line from O tangent to the curve, the most economical speed is found at the point of tangency, or in this case at N, or about 8 knots per hour. If the coal used by the auxiliary machinery is to be considered, then OX, the amount of this coal, is laid off as shown, and the tangent drawn from the new

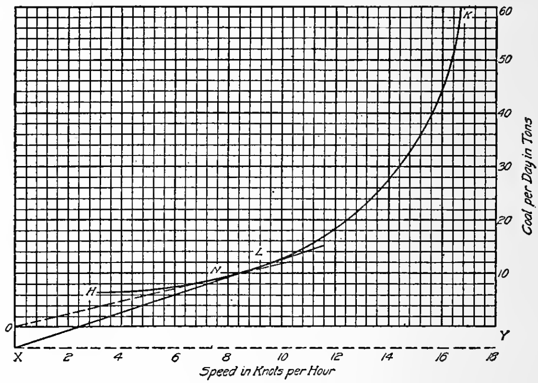


Fig. 67 Curve Plotted to Show Most Economical Speed of a Ship

origin at X. This new line, tangent at L, gives a higher speed for the most economical one than that given for the main engines only.

PROPELLERS

Although propellers are not, strictly speaking, a part of marine engines, yet the two are so closely related that a brief discussion at this point seems desirable. Screw propellers only will be considered, because they are used more extensively than any other device for propelling vessels of various kinds.

Details of Screw Propeller. A screw propeller is a set of blades, usually constructed of iron or bronze, which are made to revolve in the water at the stern of the ship, by being connected to an extension of the main engine shaft.

Small propellers are usually cast with the hub and blades in one piece, but large ones have a central boss to which the blades are bolted. Propellers are made of a variety of metals, including iron, steel, bronze, and gun metal.

Blades. The general appearance of a blade may be seen from Fig. 68. Propellers may have two, three, or four blades. In merchant vessels the latter is most common.

Pitch. The pitch of a screw propeller is the distance in the direction of the axis of the screw that would be traveled by a point on the blade during one revolution if there were no slip. It is similar to the pitch of an ordinary lathe feed screw, but of course is much larger.

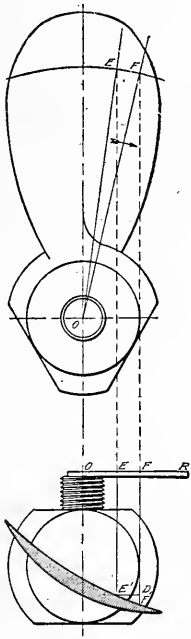


Fig. 68. Typical Shape of Propeller Blade

Diameter. The diameter of a screw propeller is simply the diameter of a circle described by the extreme ends of the blades. The ratio of the diameter to the pitch of a propeller is ordinarily from 1 to 1.1 and up to 1 to 1.5. Thus for a 14-foot diameter propeller the pitch would likely be from $14 \times 1.1 = 15$ feet to $14 \times 1.5 = 21$ feet.

Propelling Action of Screw Propeller. When a screw propeller is revolving in a given direction (for go-ahead motion for instance), the blades press on the water as the threads of an ordinary screw do upon the threads in the nut. The pressing of the blades on the water causes the water to be driven backward. There is, however, a reaction caused by projecting this mass of water sternward which results in the ahead motion of the boat. The useful work done by the propeller is the work which forces the water directly sternward; of course, the movement of water in any other direction than sternward results in a waste power.

If the screw worked in an unyielding medium, it would advance a distance equal to its pitch at each revolution. Hence, the speed of the screw per minute is the product of the pitch and the number of revolutions per minute.

Example. Suppose a screw is of 18-foot pitch and makes 72 revolutions per minute. What is the speed of the screw in feet per minute and knots per hour?

Solution.

$$18 \times 72 = 1{,}296 \text{ feet per minute}$$
$$1{,}296 \times 60 = 77{,}760 \text{ feet per hour}$$
$$\frac{77{,}760}{6{,}080} = 12.78 \text{ knots per hour}$$

Slip. Water is a yielding medium and for this reason the pressure of the blades causes the water acted on to be driven back instead of remaining firm. Then the actual speed of the ship (when referred to the undisturbed water at a slight distance from the ship) is less than the speed of the screw. This difference is called slip. *Slip is the difference between the speed of the screw and the speed of the ship, relative to still water.* It is expressed in feet per minute and as a per cent of the speed of the screw.

Example. A ship is moving at the rate of 16 knots per hour. The screw has a pitch of 19 feet and makes 97 revolutions per minute. What is the slip?

Solution.

$$19 \times 97 = 1{,}843 \text{ feet per minute} = \text{speed of screw}$$
$$\frac{16 \times 6{,}080}{60} = 1{,}621 \text{ feet per minute} = \text{speed of ship}$$
$$\text{Slip} = 1{,}843 - 1{,}621 = 222 \text{ feet per minute}$$
$$= \frac{222}{1{,}843} = .1204 = 12.04 \text{ per cent}$$

This may be expressed algebraically as follows: Let S equal speed of screw; s equal speed of ship; and L equal slip in feet per minute. Then

$$L = S - s$$
$$\frac{S - s}{S} \times 100 = \text{slip expressed in per cent}$$

The slip thus found is not the actual slip, but the apparent slip. It is not the actual or real slip, because the screw does not act in still water, but in water that has been set in motion by the screw itself or by the hull.

While the hull moves through the water, it sets in motion the water in contact with it, the direction being the same as that of the ship. The water close to the ship has a greater forward velocity than that at a distance. Since this water has a velocity a little less than that of the ship, it soon falls behind the hull and is found at the stern. Thus the water in which the propeller acts has a forward velocity. Also the velocity is influenced by the waves and eddies, due to the lines of the vessel. On account of the many conditions that make the velocity of the wake variable, it is difficult to calculate it.

When the propeller is considered, it is evident that the condition of the water in which it works should be considered. Since the velocity is difficult to obtain, the real slip is not easily found.

When slip is referred to, it is generally the apparent slip that is intended and not the real slip. The apparent slip varies from 5 to 25 per cent—15 to 20 per cent being a fair average. The actual slip is usually from 5 to 15 per cent greater than the apparent.

MANAGEMENT OF MARINE ENGINES

It is of great importance that the chief engineer and all of the assistants should be familiar with the machinery of the ship. The steam and exhaust pipes, both main and auxiliary, and the location of the valves should be carefully traced; also the feed pipes to the boilers, and the piping to the condensers. It is important that each officer should know the function of every pump and the piping from the bilges. Unless the engineer on watch is well acquainted with all the machinery, he cannot act promptly in case of emergency, but will be compelled to send for the chief or find someone under him who can furnish detailed knowledge of the part in question. The promptness and confidence with which he can act at all times depend upon his knowledge of all the parts of the machinery.

Before Starting. Just what to do before starting depends largely upon the prevailing conditions and the arrangement of the machinery. In general, the following should be observed:

All gear used in port or for repairs should be stowed away and all covers replaced. Such valves as the inlet and the outlet valves of the circulating pump and all valves to bilge pipes should be tried and put in proper condition. The outboard delivery valves from all pumps should receive especial attention. The valves to jackets and the bulkhead and regulating valves should be opened and inspected. The valves in the main steam pipe should not be closed tightly or they will be set fast when steam enters.

The oil cups and lubricators should be examined and put in good working order and the necessary worsteds adjusted.

The various joints should be inspected and the glands packed.

Pressure and vacuum gages should be connected and the shut-off cocks tried.

The bright parts of the machinery that are likely to become splashed with water should be oiled.

Auxiliary engines should be tried by steam if possible; if not, by hand. Such auxiliaries as the steering engine, circulating engines, and the electric-lighting engines should receive careful attention. In all cases, the reversing engine should be tried before using the main engines and before entering port it should again be tried to make sure that it works properly.

The main engine should be oiled at all the rubbing and rotating parts.

An important item is the examination of the crank pits and all the working parts. If these parts are not examined, some obstruction may prevent the engine from starting. The main engines should be turned through at least one revolution, both ahead and astern, by hand.

In case forced draft is used with closed stokeholds, the draft gages should be cleaned and filled with water and the air-tight doors should be examined and rigged. The fans should be carefully oiled and adjusted.

To Start Engine. In starting an engine the engineer in charge must use the knowledge gained from experience, as no set rules will apply to all engines. For instance, a small single-cylinder engine is not started in the same manner as a large triple-expansion engine. In the following we will consider the types of machinery most used—the triple-expansion engine and surface condenser.

In general, to start an engine it is first necessary to warm the cylinders and form a vacuum in the condenser; the engine can then be started by admitting steam to the cylinders.

To Form Vacuum. It is usual to fit an independent circulating pump, so the Kingston or sea-valve should be opened and the discharge valve tested to see if it lifts readily. The circulating pump is then started so that the condenser will not become heated by the drains and exhaust steam. The auxiliary air-pumps should then be started to keep the main and auxiliary condensers free from water and to form a partial vacuum. If the air-pump for the main condenser is independent, it may be started so as to form a vacuum.

To Warm the Engines. To warm the engines, all cylinder, receiver, and steam chest drains are put in communication with the condenser. In order to ascertain whether or not the drains are working properly, a by-pass arrangement is often fitted. This arrangement connects the drains to the bilges. The jackets are usually trapped to the hot well or feed tanks, but can be drained directly to the bilges. If all the drains are in order, open slightly the throttle valve and all valves in the main steam pipe. This will admit a little steam to the high-pressure steam chest. Steam is also admitted to the jackets to assist in warming the cylinders.

Now open the by-pass valves a little to admit steam to the receivers. The steam in the receivers finds its way into the cylinders and helps in the warming up. To warm both ends of the cylinders move the valve gear back and forth slowly from full gear ahead to full gear astern. The throttle may now be opened a little wider, enough to set the engine in motion. By means of the reversing gear, the cranks can be made to move back and forth without making a complete revolution.

Opening the Throttle. We will assume that the engine is thoroughly warm and (as the drains are open) free from water. Steam is in the jackets and the starting engine and starting valves ready. The centrifugal pump is at work circulating water through the condenser and either the auxiliary air-pump or an independent air-pump is at work.

To start the engines, run the links into full gear ahead or astern and open the throttle valve. In case the engines do not start, use the by-pass or auxiliary starting valves. The engines should be started slowly and the speed gradually increased by admitting more steam. After the engines have made 200 revolutions or more, the drain cocks may be closed.

Causes of Failure to Start. Marine engines may fail to start from many causes, but if proper precautions are observed before trying to start there should be no difficulty. Among the causes which are not apparent from the exterior are:

The throttle valve spindle may be broken.

The high-pressure valve (if a slide valve) may be off its seat and admit steam to both ends.

The engine may be gagged; that is, the throttle will supply steam to one side of the high-pressure cylinder and the by-pass valves admit steam to the opposite side of the intermediate or low. In this case the engine will not move, as the pressures are equalized. In using the by-pass valves, the valve or valves should be used which will produce a turning moment on the shaft. Let us suppose that both the high- and low-pressure valves cover the ports, and the intermediate slide valve is in such a position that steam can enter that cylinder. If now the throttle is opened, the engine will not start, because both ports are closed. If the by-pass valves to both receivers are opened, steam will be admitted to the proper side of the intermediate piston. Also the steam in the low-pressure receiver will find its way through the exhaust cavity of the low-pressure slide valve to the other side of the intermediate cylinder. The result will be that the engine will not start because the high and low are not available for starting and the pressures on the intermediate piston will balance. In this case steam should be admitted to the intermediate receiver only. If steam is admitted to the low-pressure receiver only, it tends to force the intermediate valve off its seat.

The opening of the wrong starting valves will frequently produce a similar situation.

If the engine has become gagged, it should be freed from steam. This may be done by closing the throttle and moving the link to the opposite extreme position. The engine can then be started in this direction and then be quickly reversed; or it may be started in the proper direction if the mistake is not repeated. In case the engine will not start, one of the following conditions may be the cause:

(a) The valve stem may have become broken inside the chest or the valve may have become loose on the stem.
(b) One of the eccentrics may be broken or slipped on the shaft.
(c) Bearings set up too tightly or too much compression on the packing in stuffing boxes often prevent starting.
(d) The propeller may be fouled by a rope or other obstruction.
(e) The turning gear may not be disconnected; that is, the worm may still be in gear with the worm wheel.

Adjustments After Starting. After the engine has been running for a short time, the following adjustments should be made:

The speed of the feed pumps to maintain the proper water level in the boilers.

The supply of circulating water to the condensing equipment.

The amount of circulating water around the main bearings should be reduced as low as possible to relieve the work of the bilge pumps.

The pressures in the steam jackets and the valves in the drains should be regulated.

Lubrication. The oil cups on bearings require special attention. The caps of lubricators should be kept in place on the oil cups to prevent dirt and water from entering. The lubricators should be examined frequently because the pipes and passages are likely to become clogged.

For cylinder lubricator as little oil as possible should be used, so as to keep the boilers free from grease. The lubricators used for this work are discussed and described in "Steam Engines", Part II.

Hot Bearings. There are many causes for hot bearings, the most common of which is dirt. To prevent the accumulation of dirt in the bearings, the engine room, oil cups, and pipes, should be kept clean.

Insufficient and improper lubrication will almost always cause heating. If the oil enters at the top, where the pressure is greatest, suitable oilways should be cut to allow the entrance of the oil. Another method is to lead the oil to a point of low pressure.

Other causes are improper adjustment or alignment and deficient surface. These defects lead to excessive pressure in some parts, which causes heating.

In many large, modern engines, the main bearings have the castings cored out so that water circulates through the bearing continuously, but does not come in contact with the rubbing surface. In the caps there are holes to allow the hand to feel of the bearings and to allow air to circulate. The temperature of the circulating water and the hand test indicate the condition of the bearing.

In case a bearing tends to become too warm, the amount of circulating water is increased. In extreme cases of heating, the bearing may be flooded with water, thus washing out all of the dirt and reducing the temperature. If this water douche is used, plenty of oil should be supplied and the bearing given careful attention.

It may be necessary to slack back the nuts on the caps for a short time, but they should be slacked but little or there will be pounding. Sometimes the power distribution may be temporarily altered, that is, the power given out by any one cylinder may be decreased, and the power given out by the others increased by running the link in or out and adjusting the expansion gear. It may even be necessary to reduce the speed for a time, but this is not done unless necessary, as it causes delay.

If the bearing is discovered to be hot, the water service should not be applied, as the sudden cooling may cause fracture. In this case the engine should be slowed down or stopped and the bearing cooled with oil, sulphur, or a mixture of soft soap, water, and oil.

Bearings that are lined with white metal should receive special attention, as the white metal soon becomes plastic and melts at about 400° F.

The water douche should be used only in extreme cases and with caution, because it may cause fracture and is likely to corrode and destroy the bearings. If water must be used, the parts should be cleaned and oiled as soon as the engines stop.

Hot Rods. Piston rods and valve rods are often kept lubricated by means of a large brush, called a swab. Frequently in starting, a man with a swab is stationed to keep the rods cool. If these rods become warm because of tight glands, they may be cooled by slacking back the gland and applying water and oil by means of a swab or syringe. If the rod is hot and water is applied, one side may be cooled and shortened; the result will be a bent rod. Instead of using water, the engines should be eased. If the rod cannot be felt, a few drops of oil or water syringed on the rod will show whether or not it is hot. If hot, the water will hiss or the oil will burn and cause smoke.

As with bearings, piston rods that are packed with metal packing should receive careful attention, as the packing may run and cut the rods. The principal causes for hot rods are glands too tight or not properly packed, piston rod not in line, and insufficient lubrication.

Knocks. Bearings should be adjusted while the engines are running. If a bearing is loose, it will knock at both ends of the stroke. Usually knocks can be located by the sound or by the feeling. Knocking in the cylinder may be due to a loose or broken piston ring, piston loose on the rod, or a nut or bolt loose. If knocking occurs, open the cylinder and jacket drains to be sure it is not due to an accumulation of water. If the noise continues at various speeds, it is probably due to looseness of the piston rings. If this is the case, the ring must be re-scraped and fitted.

Jackets. The pressures in the jackets should be maintained at the desired amount. The jacket drains are led either to the condenser or to the feed tank. If led to the feed tank, the temperature of the feed water is then raised. The jackets should be well drained, as water causes a crackling noise at each stroke. The remedy is to open the drains wide and, when clear of water, regulate the drain valves by increasing the opening.

Bilges. The bilge pumps should be at work constantly while the vessel is steaming, so that water will not accumulate in the bilges or crank pits. The crank pits should not be in communication with the bilges, or the oil from the crank pits will be spread over the bilges. If the stokehold bilges empty into the engine room bilges, the bilge water should be strained on account of the fine coal in the stokeholds. Strainers should be carefully attended to, as fine coal, waste, and articles carelessly left in the bilges are likely to choke them. It is considered good practice to pump from wells formed in the bilges and covered with strainers.

Linking Up. When starting, the links are placed in full gear. When running at the required speed, the engine is linked up so that the expansive working of the steam may be utilized. The best position of the links for a given speed is determined by experience. Trial will show at what position the engine will run smoothly, economically, and without too much noise. The throttle valve should be wide open, so that steam will enter the high-pressure chest at nearly boiler pressure. If the engine is running at reduced speed, it is a good plan to link up the high-pressure engine by the use of the block in the slot of the arm on the weight shaft. This will increase the total ratio of expansion, but will not reduce the port opening of the intermediate- and low-pressure cylinders. If there is any probability of a change in speed, the engineer in charge should see that the starting engine is warmed and drained from time to time and be sure that it is ready for use. Grunting of the slide valves is sometimes stopped by running the links into full gear for a short time, then adjusting them in a slightly different position.

Marking Off Nuts. In order to have a record of adjustments and to aid in adjusting bearings, the following marks are made. At each corner of the hexagonal nut near the face that bears on the washer, a number is stamped, as shown in Fig. 69. The washer is prevented from moving by some device. A part of the circumference of the washer is marked off in, say, 10 divisions about one-half inch apart. These divisions are then sub-divided and numbered. It is then easy to record the position of the nut by noting what number on the washer coincided with the corner of the nut. Thus 1 on $1\frac{1}{2}$ or 2 on $8\frac{1}{2}$.

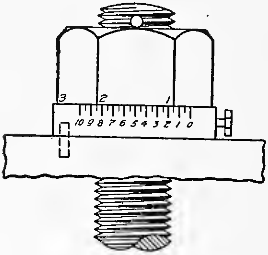


Fig. 69. Marking Nuts and Washers

Refitting Bearings. To find out whether or not a bearing needs refitting and to ascertain the amount of play, a lead is taken. The cap is first removed and a piece of lead wire is laid along the journal parallel to the axis. Some engineers place two pieces around the journals near the ends and others place them diagonally. The cap is then replaced and screwed down hard on the liners. The cap is again removed and the leads taken out and examined. They should be flattened uniformly. The thickness shows the clearance. If the marks on the nuts at which the leads were taken are noted, they may be compared with the marks and leads taken sometime afterward and the location and extent of wear known.

If the leads show that the bearing needs refitting, the caps are first removed and the journal, caps, and oilways cleaned. The journal is then carefully calipered and, if found oval, cut, or rough, should be filed all over until smooth and true. This process requires considerable care and skill for the new surface must be concentric with the axis. The filed surfaces are smoothed by an oil stone or emery. If emery is used, care must be taken to clean all surfaces.

After the journals are in proper condition, the brasses, if used, are fitted by filing and scraping. A little red lead smeared on the journal will assist in the fitting. The brasses should be eased away at the sides, as the metal at those points is of no assistance, but increases the friction.

If the bearings are lined with white metal, they must be relined when the white metal is worn through. To do this a mandrel of the same size as the journal is placed in position in the bearing and the molten metal poured in or the strips of white metal are hammered into the recesses. The metal stands clear of the brass about $\frac{1}{4}$ inch when finished.

Stopping the Vessel. *Before Entering Port.* When near port, the fires should be burning light, so that there will be no difficulty in keeping the steam pressure down. If the pressure rises when the engines are slowed down, there may be an unnecessary waste of fresh water on account of the blowing of the safety valve; the loss of fuel will also be considerable.

Before entering port all the ashes should be dumped overboard and all the water possible should be pumped out from the bilges. The reversing and capstan engines should be warmed ready for use. When the engines are slowed down, the water service should be shut off and the cil supply increased to prevent rusting of the bearings while in port. The pressures in the receivers and jackets should be watched, as they have a tendency to rise when the engines slow down.

Adjustments After Stopping. When the engines are done with, the valves in the main steam pipe and the jacket valves should be closed, but not too suddenly; the steam should then be allowed to escape from the pipe or used up by the reversing or other auxiliary engine. All drains and receiver relief valves should then be opened, and the steam should be shut off from the steering and reversing engines.

The hand-turning gear may be put in gear as soon as there is no steam left in the engine room main steam pipe. The engines should now be cleaned while warm by wiping down the rods and shafting with cotton waste and oiling the bright parts to prevent rusting.

In case the engines are stopped suddenly, notice should be immediately given in the fire room so that the draft may be checked and the evaporation reduced. If the water level is low, water should be pumped into the boilers. Every precaution should be taken to prevent an oversupply of steam, but if it is impossible to prevent the rise of pressure, the excess of steam may be used in the evaporators, distillers, etc., and in pumping out bilges and crank pits. The engines should be kept warm and well drained so as not to cause delay in starting. If the air-pump is worked by an independent engine, it should be kept working for a time, so that the condenser will not be flooded with water and injure the air-pump. If the air-pump is worked from the main engine, it will of course stop as soon as the engines stop; in this case put on a feed-pump to keep the condenser free from water. The circulating engines may be stopped soon after the engines stop.

As in case of entering harbor, watch receiver and jacket pressures, and stop the supply of water to bearings, etc. If there is any chance of starting again soon, keep the reversing engine warm and well drained.

Precautions for Long Stay in Port. If the stay in port is to be long, the main condensers and air-pumps should be well drained and several of the boilers may be cleaned and repaired if necessary. The fires should be allowed to burn themselves out gradually. If the stop is for a short time, the fires should be banked.

Emergencies. What to do in emergencies depends upon the arrangement of the machinery. The kind and number of engines and their arrangement and capacities of the condensers and auxiliary machinery often determine what course to pursue in case any part breaks or gets out of position.

Cylinder Head Broken. If a cylinder head breaks, it should be repaired if proper means are at hand. If it cannot be repaired, the steam port which admits steam to that end may be blocked up by driving in plugs of soft pine and the engine run single-acting. This is comparatively simple if the valve is a plain slide, but with a piston valve the many ports make it more difficult. If a cylinder head of a triple-expansion engine breaks, and one engine must run single-acting, the expansion gear should be arranged so that the work will be properly divided.

Fracture in the Crankshaft. What to do in this case depends upon many conditions. If the engine is of the multicylinder type, and the crankshaft is made in interchangeable lengths, fit the spare length in place of the disabled one. In case no spare length is carried and the crankshaft of the low-pressure engine is damaged slightly, change the low-pressure length to the high-pressure engine and place the high-pressure length in place of the low. The low-pressure length transmits the most power. If the damage is considerable, such as the breaking of the crankpin, the length cannot be used and the high-pressure engine must be disconnected. If the pumps are worked from the high-pressure crosshead, repair the broken shaft, place it in the high-pressure engine, and block up the steam ports to the high-pressure cylinder. The power is then developed in the intermediate- and low-pressure cylinders; the amount of power transmitted to the high-pressure crankshaft being just sufficient to work the pumps. Probably it will be necessary to run the engines slowly because of the weak shaft.

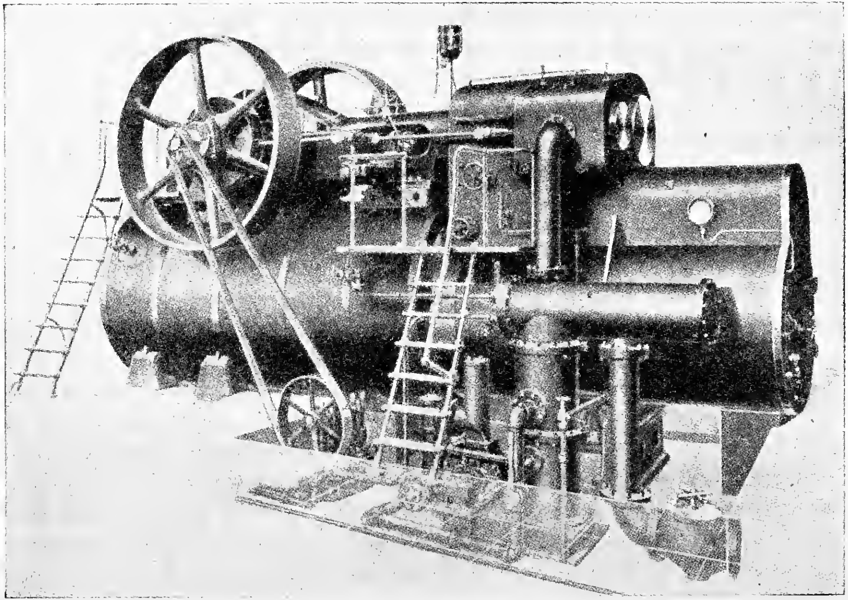
Piston Broken. If the piston, piston rod, or valve stem become broken and cannot be repaired, the damaged engine must be disconnected and the power furnished by the others.

Air-Pump Broken. In case the air-pump breaks and cannot be repaired, the exhaust may be carried to the deck and the engines run non-condensing. This is a great disadvantage if the amount of fresh water carried is slight and the ship is far from port. In case no separate exhaust is possible, the auxiliary air-pumps may be connected and the ship proceed. In most cases, however, the auxiliary air-pumps are not of sufficient capacity to remove all of the condensed exhaust steam and the air; therefore, no vacuum will be carried, but the condensation may be returned to the boilers.

Bent Piston-Rod. In the case of a small rod and a long, slight bend, the rod may be straightened by placing it in a lathe and applying a powerful lever. A large rod, or one with a quick bend, should be heated to a dull red in a wood fire. The rod is then placed in a large lathe and straightened by an hydraulic jack. In doing this work care must be taken that the rod is not heated too hot, does not scale, and that the points of contact are protected by copper plates.

Eccentric Broken. If the go-ahead eccentric or eccentric rod breaks and cannot be repaired, the go-astern eccentric can be shifted in its place. The engine will now run ahead, but cannot be reversed. The go-astern end of the links must be kept from dropping by some flexible support, such as a rope or chain.

Another method is to disconnect the connecting rod from the crankpin and crosshead of the disabled engine, and block up the steam ports so that the steam will flow to the other cylinders by the shortest passage. The piston should be secured on the bottom of the cylinder. The valve should be removed. After removing the broken valve gear, the engine is ready to start. This method may be used if the pumps are worked from the low-pressure crosshead and the low-pressure engine is intact. If, however, the high-pressure eccentric is broken and the pumps are worked from that crosshead, the same method may be pursued as described for a fractured crank-shaft. That is, the valve gear should be removed, the ports blocked, and the piston, the piston rod, crosshead, and connecting rod left in place. The moving parts of the high-pressure engine will then work the pumps by means of the power transmitted to the high-pressure crank. The engine must be run slowly, but can be reversed.



SIDE VIEW OF BUCKEYE TYPE OF LOCOMOBILE SHOWING ACCESSORIES BELOW THE BOILER
Courtesy of Buckeye Engine Company, Salem, Ohio

STEAM ENGINES

PART II

MECHANICAL AND THERMAL EFFICIENCY

The brief historical review and the study of the various types of engines have served to unfold the degree of perfection that has been attained in the design and details of construction of the modern steam engine. From a mechanical standpoint, the modern engine is highly efficient. A mechanical efficiency, that is, $\frac{\text{Brake horsepower}}{\text{Indicated horsepower}}$, of from 85 to 95 per cent is not infrequently obtained. An actual test of a 12-inch$\times$19$\frac{3}{4}$-inch$\times$15-inch tandem-compound Corliss engine operating non-condensing gave a mechanical efficiency of 94 per cent. That is to say, if the engine was developing 120 horsepower in the cylinders, that 112.8 horsepower would be delivered by the engine to the flywheel. In other words, the horsepower used in overcoming the friction of the various moving parts was only 7.2 or 6 per cent of the total horsepower developed.

Low Thermal Efficiency Inherent. From the standpoint of thermal efficiency, however, the modern engine is very inefficient, but it is much more efficient than the older types. Even the maximum thermal efficiency obtained is only about 15 per cent, and, under favorable conditions, this very low figure may be so reduced that the engine is operated at a great economic loss. It is now proposed to briefly point out some of the causes for the very low thermal efficiency obtained and to indicate some of the means that have been employed to increase the thermal output of the steam engine. In order to make this study it becomes necessary to again refer to steam and its properties. It is well known that steam contains a great deal of heat, and that this heat can be converted into useful work by allowing the steam to pass from the high temperature of the heat generator to the lower temperature of the refrigerator, during this change giving up heat. There are several forms of heat engines, all of which convert the heat contained in some substance into work. The theoretically perfect engine shall be considered first, and after that the modifications that go to make up the steam engine of today.

Ideal Engine. The theoretical engine, Fig. 70, is supposed to receive heat from the generator at constant temperature T_1 until communication is interrupted at B. The working substance expands to C without losing or gaining any heat from external sources until the temperature of the refrigerator is reached. The engine now rejects heat at the constant temperature T_2 of the refrigerator and then compresses the working substance without loss or gain in the quantity of heat until the temperature of the heat generator is reached. These are ideal conditions and, if fulfilled, the efficiency of the perfect engine will depend only on the difference between the temperature at which heat is received and rejected or, in other words, it depends only upon the difference in temperature between the generator and the refrigerator.

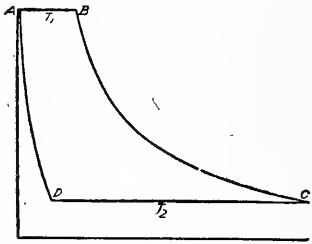


Fig. 70. Theoretical Indicator Diagram

If T_1 equals the absolute temperature of the heat received and T_2 equals the absolute temperature of the heat rejected, then the thermal efficiency E of the engine will be represented by the formula

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

Or, in other words, the efficiency equals the absolute temperature of the heat rejected, subtracted from the absolute temperature of the heat received, and the remainder divided by the absolute temperature of the heat received.

Example. Given an engine using steam at a 120 pounds absolute pressure, and exhausting at atmospheric pressure. What is the thermal efficiency?

Solution. The absolute temperature corresponding to 120 pounds pressure is 341.31+459.5, or 800.81°, and the absolute temperature of the exhaust is 212+459.5, or 671.5°. Then

$$E=\frac{800.81-671.5}{800.81}$$
$$=.161, \text{ or } 16.1 \text{ per cent}$$

Losses in Practical Engine. *In General.* In actual engines this efficiency can not be realized, because the difference between the heat received and the heat rejected is not all converted into useful work. Part of it is lost by radiation, conduction, condensation, leakage, and imperfect action of the valves. The cylinder walls of the theoretical engine are supposed to be made of a non-conducting material, while in the actual engine the walls are of metal, which admits of a ready interchange of heat between cylinder and steam. This action of the walls can not be overcome and is so important that a failure to consider its influence will lead to serious errors in computations, and no design can be made intelligently if based on the theory of the engine with non-conducting walls. In theoretical engines steam expands without the loss of any heat, while in the actual engine a large amount of heat is lost by radiation. There is also a considerable loss of pressure between the boiler and the engine, due to the resistance offered by the pipes and cylinder passages. In a slow-speed engine with large and direct ports and valves this trouble is reduced to a minimum. The imperfect action of the valve gears may also be lessened with due care, but the action of the cylinder walls still remains to be overcome.

Theoretical and Actual Card Analysis. In the theoretical card, admission is at constant boiler pressure, cut-off is sharp, expansion is complete—that is, expansion continues until the temperature falls to that of the condenser and the exhaust is at condenser pressure—and the piston always sweeps the full length of the cylinder.

In the actual engine there is a considerable loss of pressure between boiler and engine, and the wire drawing of the ports and valves tends to cause a sloping steam line. Condensation at the beginning of the stroke causes the real expansion line to fall below the theoretical, while re-evaporation causes it to rise above the theoretical toward the end of expansion. In the actual engine, release takes place before the end of the stroke, expansion is not complete, that is, the pressure at release is above that of the condenser, and the resistance of exhaust ports causes the back pressure to be above the actual condenser pressure. Moreover, the piston does not sweep the full length of the cylinder, and the clearance space must be filled with steam, which does very little work. The theoretical and actual cards are shown in Fig. 71.

Mechanical Losses. It has been shown that the efficiency of the theoretical engine is purely a thermal consideration; the efficiency of the actual engine, however, is largely a mechanical matter. The unit of work is the horsepower, which corresponds to the development of 33,000 foot pounds per minute. As 778 foot pounds are equivalent to one British Thermal Unit, 33,000 foot pounds per minute, or one horsepower, is equivalent to 33,000÷778 or 42.42 British Thermal Units. Now if a certain engine uses 84.84 British

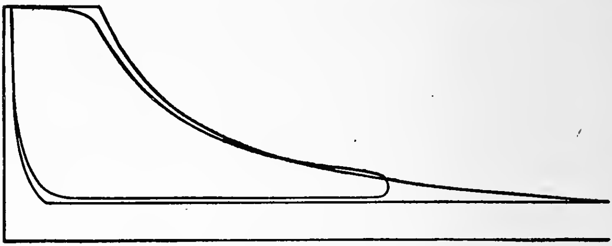


Fig. 71. Superposed Ideal and Actual Indicator Diagrams

Thermal Units per horsepower per minute, it is evident that its efficiency will only be one-half, or 50 per cent, because 42.42 is one-half of 84.84. Hence, it may be said that the efficiency of the actual engine is equal to $\dfrac{42.42}{\text{British Thermal Units per horsepower per minute}}$

This efficiency is always much less than that of the perfect engine.

ANALYSIS OF LOSSES

The effect of some of the losses in the steam engine and the methods for decreasing them will now be considered.

Radiation. In the first place, the metal walls of the cylinder, being good conductors of heat, become heated by the steam within and transmit this heat by conduction and radiation to the air or external bodies. With the cylinder well lagged, much less heat is lost by radiation. If the lagging were perfect and the temperature of the cylinder remained the same as the temperature of the steam throughout the stroke, there would be no loss by radiation, but heat would still be lost by conduction to the different parts of the engine.

Cooling by Expansion. During expansion, the temperature and pressure of the steam decrease as the volume increases, and the temperature at exhaust is much less than the temperature at admission. In the perfect engine, the working substance after exhaust is compressed to the temperature at admission, but in the actual engine much of this steam is lost and the compression of a part of it is incomplete, so that its temperature is less than the temperature at admission.

Steam Condensation and Re-Evaporation. Consider an engine operating with admission at 100 pounds absolute and exhaust at 18 pounds absolute. From steam tables the temperature at admission is found to be 327.86°, and at exhaust 222.4°. The metal walls of the cylinder, being good conductors and radiators of heat, are cooled by the low temperature of exhaust, so that the entering steam in passing through ports and into a cylinder is subjected to a temperature of more than 100° cooler than the steam. This means that heat must flow from the steam to the metal until both are of the same temperature. This causes the steam to give up part of its latent heat, and as saturated steam can not lose any of its heat without condensation, the cylinder walls become covered with a film of moisture, usually spoken of as initial condensation. This condensation in simple unjacketed engines, working under fair conditions, may easily be 20 per cent or more of the entering steam. The moisture in the cylinder has, of course, the same temperature as the steam; it has simply lost its heat of vaporization.

Although metal is a good conductor of heat, it can not give up or absorb heat instantly; consequently during expansion, the temperature of the steam falls more rapidly than that of the cylinder. This allows heat to flow from the cylinder walls to the moisture on them. As fast as the steam expands so that the pressure in the cylinder becomes less, this condensation will begin to evaporate. As the pressure falls it requires less and less heat to form steam and, therefore, more and more of this moisture will be evaporated. At release the pressure drops suddenly, more heat at once flows from the cylinder walls, and re-evaporation continues throughout the exhaust. Probably all of the water remaining in the cylinder at release is now re-evaporated, blows out into the air of the condenser, and is lost as far as useful work is concerned.

The steam that is first condensed in the cylinder does no work; its heat is used to warm up the cylinder, and later, when it is re-evaporated, it works only during a part of the expansion and at a reduced efficiency, because it is re-evaporated at a pressure and, consequently, at a temperature very much lower than that of admission. If the cut-off is short, perhaps 20 per cent of the steam condensed may be re-evaporated during expansion; if the cut-off is long, 10 per cent may be re-evaporated, the rest remaining in the cylinder at release, still in the form of moisture. Thus some of the entering steam passes through the cylinder as moisture until after cut-off, and still more passes entirely through without doing any work.

Suppose an engine is using 30 pounds of steam per horsepower per hour and admission is at 100 pounds absolute. The latent heat of vaporization at this pressure is 884 British Thermal Units per pound. If the condensation amounts to $33\frac{1}{3}$ per cent, then 10 pounds are condensed and there is lost 10 times 884, or 8,840 British Thermal Units per hour, or 147.3 per minute; and since 42.42 British Thermal Units represent 1 horsepower, there is lost by condensation 147.3 divided by 42.42, or $3\frac{1}{2}$ horsepower (nearly). If the cut-off is shortened, the condensation increases and may amount to 50 per cent. Of course, very much less steam is used at a short cut-off than with a long cut-off, and doubtless in many cases 50 per cent of the steam at short cut-off is not as great an absolute quantity as 30 per cent at a long cut-off.

Exhaust Waste. In addition to the actual loss from condensation in the cylinder, there is still another loss due to re-evaporation. Suppose, as before, that 10 pounds of steam are condensed in the cylinder, and that 20 per cent of this is re-evaporated during expansion. This will leave 8 pounds to be re-evaporated during exhaust. Suppose the exhaust is at 3 pounds above atmospheric pressure, or 18 pounds absolute (about). Then the heat of vaporization is 963.1 British Thermal Units per pound of steam, and it will require 8 times 963.1, or 7704.8 British Thermal Units, to evaporate the 8 pounds. All of this heat is taken from the cylinder, leaving the engine much cooler than it would be were it not for this re-evaporation. This gives some idea of the great amount of heat passing away at exhaust, which is known as the *exhaust waste.*

Clearance. In all cylinders it is necessary to have a little space between the cylinder cover and the piston when at the end of the stroke. In vertical engines the space is greater at the bottom than at the top. The volume of this space, together with the volume of the steam ports, is called the clearance. It varies from 1 to about 15 per cent, depending upon the type and speed of the engine—the higher the speed, the greater the clearance. This clearance space must be filled with steam before the piston receives full pressure; and the volume of the clearance offers additional surface for condensation.

Friction. Another important loss is that due to friction. It is well known that it takes considerable power to move an unloaded engine; if fitted with a plain, unbalanced slide valve, the power necessary to move the valve alone is considerable. The piston is made steam-tight by packing rings, and leakage around the piston rod is prevented by stuffing boxes. All these devices cause friction as well as wear at the joints. The amount of power wasted in friction varies greatly, depending upon the kind of valves, general workmanship, state of repair, and lubrication.

OPERATION ECONOMIES

The foregoing discussion has served to indicate that the larger part of the heat loss occurring in the steam engine is due to initial condensation, exhaust waste, and clearance, although the effect of the latter has been greatly reduced by improvement in design. Regarding the methods devised for reducing the amount of initial condensation, the high-speed engine has in a measure decreased this difficulty because of the very high piston speed employed. Since the piston speeds are high, the length of time the steam remains in the cylinder has been greatly lessened; hence the transference of heat is considerably reduced. The piston speed is limited, however, by the performance of the valve gear, it being well known that the most efficient valve gears are those employed on the low-speed engines. Increased piston speed also calls for more clearance space, hence the possible gain in economy from high piston speed is limited by the performance of the valve gear and the clearance required for the higher speeds.

The application of the idea of multiple expansion, or compounding, has materially reduced the losses both by lessening the amount of condensation and also by utilizing the re-evaporated steam and the steam that leaks by the piston, which in some cases may be considerable, and this important improvement will be discussed first. In addition, other means have been employed for the purpose of increasing the economic performance of the steam engine, as for instance, *jacketing*, *superheating*, and the *use of condensers*.

MULTIPLE EXPANSION

Two engines may be used together on the same shaft, partly expanding the steam in one of the cylinders and then passing it over to the other to finish the expansion. One advantage from this arrangement is that the parts can be made lighter. The high-pressure cylinder can be of much less diameter than would be possible if the entire expansion were to take place in one cylinder. This, of course, makes the pressure exerted on the piston rod much less, and the piston rod and connecting rod can thus be made much lighter. The low-pressure cylinder must be larger than it otherwise would be, but its parts need not be much heavier, because the pressure per square inch is always low.

This arrangement gives not only the advantage of lighter parts, but a decided increase of economy over the single-cylinder type. If attention is given to the matter, a loss of economy would be expected, because the steam is exposed to a much larger surface through which to lose heat, but the gain comes from another source and is sufficient to entirely counterbalance the effect of a larger cylinder surface.

Less Condensation. When very high pressure steam and a large ratio of expansion is used, the difference between the temperature of the entering and of the exhaust steam is great. For instance, suppose steam at 160 pounds (gauge) pressure enters the cylinders and the exhaust pressure is 2 pounds (gauge), the difference in temperature as taken from steam tables is 370.7°−218.2°, or 152.5°. This difference becomes nearly 230 degrees if the steam is condensed to about three pounds absolute pressure. The cylinder and ports of the engine are cooled to the low temperature of the exhaust steam and, as we have seen, a considerable quantity of the entering steam is condensed to give up heat enough to raise the temperature of the cylinder to that of the entering steam. As the ratio of expansion increases, the difference in temperature increases, and consequently the amount of steam thus condensed also increases. To keep this initial condensation as small as possible, the range of temperature must be limited, that is, it must not have as great a difference between admission and exhaust. To do this the expansion of the steam must be divided between two or more cylinders.

It will be remembered that the great trouble Watt found with Newcomen's engine was its great amount of condensation, and he stated as the law which all engines should try to approach, that *the cylinder should be kept as hot as the steam which enters it.* This is to avoid condensation when steam first enters. If, instead of expanding the steam in one cylinder, it be expanded partly in one and then finished in another, it will have passed out of the first cylinder before its temperature has dropped a great deal, and consequently the cylinder walls will be hotter than they would have been if the expansion had taken place entirely in one cylinder. This would then reduce the amount of steam condensed. The importance of this may not be evident at first, but it makes a great difference in the economy of the engine. If there is less condensation, there will be less moisture to re-evaporate, and consequently less exhaust waste, hence there will be a saving in two ways.

Methods of Compounding. In a compound engine the steam is first admitted to the smaller, or high-pressure, cylinder and then exhausted into the larger, or low-pressure, cylinder.

Suppose steam at 160 pounds (gauge) pressure is admitted to a cylinder, and the ratio of expansion is such that the steam is exhausted at about 60 pounds (gauge) pressure; then the difference of temperature is $370.7° - 307.4°$, or $63.3°$

If now the steam when exhausted from the first cylinder enters a second and is allowed to complete its expansion, so that the exhaust pressure is about two pounds (gauge) pressure, the difference of temperature in the cylinder will be $307.4° - 218.2°$, or $89.2°$.

Then for the simple engine, if the exhaust pressure is two pounds (gauge), the difference of temperature is 152.5 degrees, while in the compound engine this difference is divided into two parts, 63.3 degrees and 89.2 degrees. The cylinder condensation for both cylinders of the compound engine will be much less than if the total expansion took place in a single cylinder. The cylinders should be so proportioned that the same quantity of work may be done in each.

If there are two stages of expansion, the engine is called simply *compound;* three stages, *triple;* and four, *quadruple.*

Exhaust Waste Utilized. Besides reducing the excessive condensation, there is still another gain in using multiple expansion. It has been shown how much heat is lost by the exhaust waste, which in the simple engine blows into the air or into the condenser and is entirely lost. In the multiple-expansion engine the exhaust and re-evaporation from one cylinder passes into the next and does work there; furthermore, any leakage from the high-pressure cylinder is also allowed to do work in the low-pressure cylinder.

JACKETING

The most primitive method of effecting steam economy is by jacketing, which principle Watt early recognized and adopted. This method reduces the loss due to cylinder condensation by supplying heat to the steam while it is in the cylinder, that is, by surrounding the cylinder with an iron casting and allowing live steam to circulate in the annular space thus formed. The cylinder covers are also made hollow to permit a circulation of live steam. A cylinder having the annular space *A*, Fig. 72, filled with steam is said to be jacketed. A lining *L* is often used in jacketed cylinders.

Function of Jacket. The function of the jacket is to supply heat to the cylinder walls to make up for that abstracted during expansion and exhaust, so that at admission the cylinder will be as hot as possible. The result is, that the difference in temperature between the cylinder walls and the entering steam is considerably less than in engines where no jacket is used. Condensation is therefore reduced and, since heat flows from the jacket to the cylinder during expansion, a much larger amount of this condensation is re-evaporated before release and it thus has an opportunity to do some work in the cylinder. This leaves a comparatively small amount of exhaust waste and the heat thus abstracted is made up from the steam in the jacket. Since a large amount of heat is given up by the jacket steam, a good deal of it must be condensed. Thus the question is asked: "What is the advantage of this method over that of allowing the entering steam to supply the heat by its own condensation?" This question is answered briefly as follows:

The loss of heat by condensing the steam would be less if the inside of the cylinder could be kept dry. It has been indicated how the moisture that collects by condensation is re-evaporated during expansion and exhaust because the pressure falls and the cylinder walls are hotter than the steam. This re-evaporation takes place at the expense of the heat in the cylinder walls and they are thus cooled. It has already been shown that a great many British Thermal Units

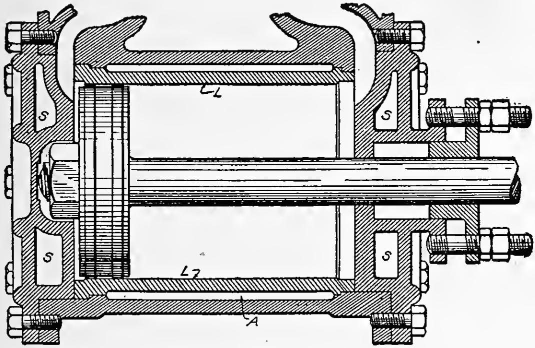


Fig. 72. Section of Steam Engine Cylinder, Showing Method of Jacketing

are thus taken from the cylinder and thrown out at exhaust at every stroke. Now if the inside of the cylinder can be kept dry so that there will be little or no re-evaporation at exhaust, it will cause a considerable saving. The steam that condenses in the jacket does not re-evaporate in it; but is returned to the boiler as feed water, so that the only heat lost is the latent heat given up during condensation. If the cylinder is heated from within, both the latent heat given up by condensation and the latent heat required for re-evaporation are lost.

In a triple-expansion engine there is one distinct advantage in allowing condensation in the cylinder, for this moisture acts as a lubricant, and as the heat of re-evaporation passes into the next cylinder and there does work, there is very little loss.

Saving Due to Jacketing. It is evident that a large part of the heat of the steam jacket flows to the cylinder during exhaust and is thus entirely lost in the simple engine. In the triple engine, however, this heat passes into the intermediate and low-pressure cylinders; consequently we might expect a greater gain from using a jacket on a triple engine than on a large, simple engine. The main advantage of the jacket has been previously pointed out, and as in all cases the gain is small, there is to be found a considerable diversity of opinion as to its real advantages. On some engines there is undoubtedly little if any gain, the largest gain being in the smaller engines of, say, 200 horsepower and under. On very small engines, such as a 5-inch×10-inch engine when developing only one and one-half horsepower under light load, the gain is as much as 30 per cent. On a 10-horsepower engine the gain might be as much as 25 per cent, while on engines of about 200 horsepower the gain would probably be 5 to 10 per cent for simple condensing and compound condensing, and from 10 to 15 per cent for triple expansion. The saving on large engines of, say, 1,000 horsepower is very small, the reason being that large engines offer less cylinder surface per unit of volume than small ones, and hence have proportionately less cylinder condensation. The very small engines, in which the gain would be the greatest, are seldom jacketed, because they are built for inexpensive machines and the first cost is of more consequence than the economy of operation. Owing to the cost of construction and the care necessary to keep jackets operative, the use of the jacket has gradually diminished. Furthermore, the introduction of the high-speed and compound engines, as well as the use of superheated steam, has reduced the advantage of jacketing to relative insignificance.

SUPERHEATING

General Practice. The use of superheated steam is rather a modern practice, although for many years previous to its adoption engineers had appreciated its value in producing steam engine economy. The reason for its delayed adoption in a practical way was due to the mechanical difficulties met with in superheating the steam and also to the increased cost of maintenance produced by its use. In recent years both of the objectionable features above mentioned have been, in a large measure, overcome, so that today superheat is being used in a large number of power plants, and also in steam locomotives.

Before describing a superheater, it may perhaps be well to clearly define what is meant by superheated steam. Water, when confined in a vessel and heated sufficiently, turns into steam, which, if some water still remains, is spoken of as saturated steam. Saturated steam when further heated becomes superheated steam, if it is separated from the water. To bring about this separation, a superheater is necessary. Superheaters vary considerably in details of construction according to the service for which they are designed,

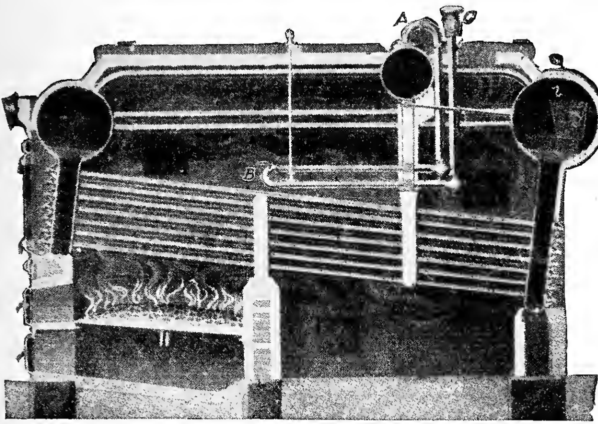


Fig. 73. Section of Water-Tube Boiler Showing Application of Foster Superheater

there being, for instance, quite a difference between the superheater designed for a stationary plant and one designed for a locomotive.

Foster Superheater. A Foster superheater as applied to a water-tube boiler is illustrated in Fig. 73. The superheating element is shown at *B*, which is connected to the steam space of the boiler by the pipe *A*. The saturated steam from the boiler passes through the pipe *A*, through the superheater, and then is conveyed to the engine through the valve *C*. In this installation the superheater is placed in the passage provided for the transmission gases to the chimney, hence it is heated by what would otherwise be lost heat. The manner of installing superheaters varies a great deal. Some are entirely separated from the boiler, being self-contained and supplied with a grate for separate firing.

The Foster superheater, Figs. 73 and 74, is made up of a number of elements placed parallel to each other, each of which consists of two straight steel tubes, one inside of the other. The elements are joined at one end to manifolds or connecting headers, and at the other end to return headers for which return bends are often substituted. On the outside of the tubes *B*, Fig. 74, are fitted a series of cast-iron annular flanges *D*, placed close to each other and carefully fitted to the tube so as to be practically integral with it, at the same time exposing an external surface of cast iron, which metal is best adapted to resist the action of the heated gases. The rings are carefully bored to gauge, and shrunk on the tubes. Once being in position, the rings and tubes act virtually as a unit. As the coefficient of expansion of steel is a trifle greater than that of cast iron, the rings grip the tubes even tighter when in service. This form of construction is flexible and durable. It provides a section of great strength and entire freedom from internal strains. The mass of metal in the tubes and covering acts as a reservoir for heat, which is imparted to the steam evenly, tending to secure a constant temperature of steam, even though the temperature of the hot gases does fluctuate. The seamless drawn tube secures great initial strength, which is reinforced by the rings shrunk on the outside. Inside of the elements there are placed other tubes *C* of wrought iron, which are centrally supported by means of knobs or buttons regularly spaced throughout their length. These inner tubes are closed at the ends. A thin annular passage *E* from the steam is thus formed between the inner and the outer tubes. The steam clinging closely to the heating surface is quickly heated in the most efficient manner.

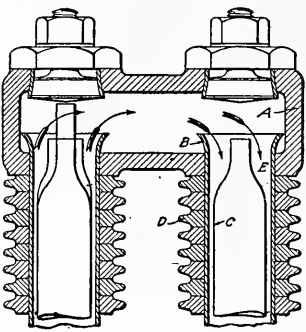


Fig. 74. Section of Foster Superheater Tubes

The superheater must be as free as possible from the liability of burning out in case of a chance of overheating of the exposed surfaces. The circulation must be properly distributed throughout the superheater at full load as well as at partial loads. The various parts must be accessible for inspection, both externally and internally, and must be readily renewable or easily repaired. There must be provision for free expansion and contraction of the various parts. The supporting arrangement must be carefully worked out. Cast iron has given excellent results in producing durable superheaters

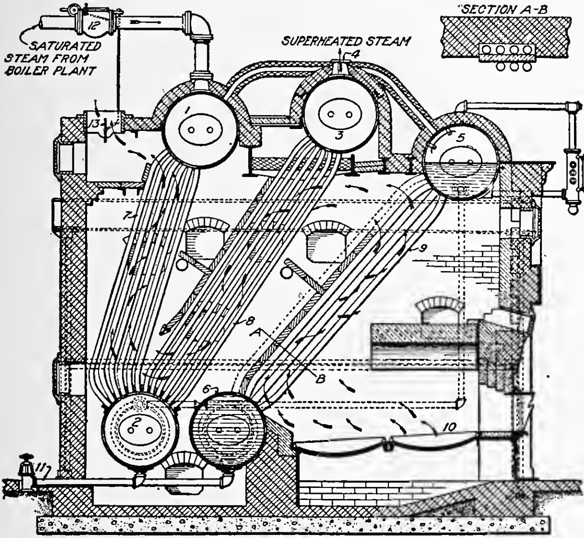


Fig. 75. Section of a Separately-Fired Type of Superheater

and has been extensively used because of its ability to withstand high temperatures. For high steam pressures, however, cast iron is not considered safe and has given way to the use of seamless steel tubes, which are homogeneous and strong, but lack the heat-resisting qualities of cast iron. It is evident that a combination of these two metals will preserve the good qualities of both.

Separately-Fired Superheater. A type of superheater differing radically from the one previously described is illustrated in Fig. 75. It is a separately-fired superheater and its construction is very similar to the Stirling water-tube boiler. The saturated steam from the main boiler plant enters the rear superheater drum *1*, passes through the rear bank of tubes *7* into the lower drum *2*, thence to the upper drum *3*, from which it passes into the pipe line through the opening *4*. The furnace is similar to that used in the standard design of Stirling boiler. To protect the superheater tubes from high temperatures of the furnace, a sufficient amount of boiler heating surface, as drums *5* and *6* and bank of tubes *9*, is located in front of the superheater proper in order to reduce the temperature of the gases to about 1,500 degrees by the time they reach the superheater. The builders state that when the gas temperature reaches 1,500 degrees in the standard boiler, 19 per cent of the boiler heating surface has been swept over by the gases, 50 per cent of the steam produced by the boiler has been generated, and the boiler heating surface per horsepower is 3.8 square feet. Consequently in the independently fired superheater shown in Fig. 75, 50 per cent of the heat absorbed is used to generate the steam, which is added to steam furnished by the main boiler plant and hence increases the capacity of the plant in proportion. The remaining 50 per cent of the heat, a portion of which passes out the stack, is absorbed by the superheater and superheats both the steam from the main boiler plant and that from the front bank of water tubes. The superheater, because of the front generator set, will produce about 12 per cent of the amount of steam furnished by the main boiler plant. As a further precaution against any possible overheating of the superheater tubes near the furnace, a flap valve *12* is placed in the pipe conveying saturated steam to the superheater, as shown in Fig. 75. The spindle of this valve is connected by links to the superheater damper *13*, so that the damper to the opening is regulated according to the quantity of steam flowing into the superheater. If the steam flow stops, the valve *12* drops to its seat and the damper *13* is closed. Independently fired superheaters are furnished in any desired capacity, suitable for any degree of superheat up to about 300° F. The upper water drum *5* and the lower superheater drum *2* are connected by piping, hence, if desired, the superheater sections may be flooded, converting the whole into a saturated steam boiler.

Purposes of Superheaters. These two types of superheaters illustrated and described will suffice, and we may now direct our attention to a study of the purposes of the superheater and to some consideration of the economy secured by its use. The purposes of superheating steam, as practiced in the past and as recognized at present, are, according to Thurston, the following:

(1) Raising the temperature which constitutes the upper limit in the operation of the heat-engine in such a manner as to increase the thermodynamic efficiency of the working fluid.

(2) To so surcharge the steam with heat that it may surrender as much as may be required to prevent initial condensation at entrance into cylinder and still perform the work of expansion without condensation or serious cooling of the surrounding walls of the cylinder.

(3) To make the weight of the steam entering the condenser and its final heat charge a minimum, with a view to the reduction of the volume of the condensing water and of the magnitude and cost of the air pump and condenser system to a minimum.

(4) To reduce the back pressure and thus to increase the power developed from a given charge of steam and efficiency of the engine.

(5) To increase the efficiency of the boilers both by the reduction of the quantity of the steam demanded from the original heating surface and by increasing the area of the heating surface employed to absorb the heat of the furnace and flue gases, and also by evading the waste consequent upon the production of wet steam.

If the steam entering a cylinder is only superheated enough to give dry saturated steam at cut-off, the range of temperature $\frac{T_1-T_2}{T_1}$ of the Carnot cycle is interchanged and there is, therefore, no increase of economy from item 1. The other four sources of economy depend upon one fundamental fact—the poor conductivity of dry steam. To the property of non-conductivity of heat of superheated steam is due its great advantage. On entering a cool cylinder it slowly gives up its heat, and if the degree of superheat is sufficient there will be little or no initial condensation. The degree to which steam should be superheated is still a debated point, some engineers contending that only a very moderate degree of superheat of about 100 degrees is sufficient, whereas others maintain that no real economy is obtained with less than 200 degrees or over. When a high degree of superheat was first used, difficulties were encountered such as the disintegration of the valves, valve seats, packing rings, and other parts subjected to the action of the superheated steam. Lubrication was also interfered with, since many of the oils used were not suited for such high temperatures. All of these difficulties no doubt account for the one time widespread objection to high degrees of superheat, but in recent years they have in a large measure been overcome. The author is familiar with the performance of a simple slide valve locomotive which has been in operation for several years under degrees of superheat ranging from 80 degrees to 214 degrees, during which time no trouble has been experienced with the valves or with the lubrication. Many European locomotives have been satisfactorily operated with high degrees of superheat, which insures the passage of steam through the cylinder with but little or no condensation.

Economic Advantages. The economy obtained by the use of superheat has been clearly demonstrated by a large number of practical tests both upon stationary engines and upon locomotives. It is to be noted also, that about the same per cent of economy has been obtained on the various types of engines tested, the stationary tests corroborating the results obtained upon the locomotive and *vice versa.* The various tests indicate a saving of from 12 to 15 per cent of the amount of steam used by the engine per indicated horsepower per hour, and a saving of coal from 20 to 25 per cent. Another very significant thing that has been determined is that the output of power has been increased from 20 to 30 per cent, depending upon the conditions. These three items of saving have hastened the installment of a large number of superheaters, so that at the present time thousands of locomotives in Europe are equipped with superheaters, and in the United States and Canada over 1,500 locomotives are so equipped. It seems that the railroads have been quicker to take up the idea of installing superheaters than have other industries, so that not nearly so many superheaters are found in stationary service.

It is to be noted that the greatest gains from the use of superheaters are to be expected in the more uneconomical plants. That is, the per cent of saving by the use of superheated steam in a simple engine would be greater than for a compound engine, and for a compound engine as compared with a triple-expansion engine. Several prominent engineers have advised the reduction of steam pressures with a relative increase in diameter of cylinders and the use of superheated steam. The combination of a simple engine with low steam pressure and superheated steam will give an increased output of power at a small cost, a result desired by all operators.

CONDENSERS

When low-pressure steam is cooled, it gives up its latent heat, that is, it changes from a vapor to a liquid, and, as a liquid occupies much less space than an equal weight of its vapor, the changing of the steam to water greatly reduces the pressure. Therefore, by cooling the steam in an engine cylinder in front of the piston, the back pressure, or resistance, is reduced, which, in turn, reduces the pressure necessary to push the piston through the stroke and, therefore, lessens the steam required to do the work. This cooling is accomplished by some form of condenser.

Theory of Condenser Action. *Back Pressure.* In the ordinary non-condensing engine, steam can not be exhausted below a pressure of 14.7 pounds absolute, because the atmosphere exerts that amount of pressure at the opening of the exhaust pipe. In fact, this 14.7 pounds is the theoretical limit only, and in practice the exhaust is always a little above this because of resistance in the exhaust ports and exhaust pipe; so that 17 or even 18 pounds absolute back pressure is more nearly the conditions of actual service.

During the forward stroke, steam expands from the pressure at admission to a much lower pressure at release; then the valve opens for the return stroke giving full steam pressure on one side of the piston and the pressure of exhaust on the other side, the latter acting against the piston and against the force of the incoming steam. If all of this back pressure could be removed so that there would be a vacuum on the exhaust side of the piston, the power of the engine would be increased by just so many pounds of mean effective pressure, and in addition to this the steam could expand to a very much lower pressure and therefore work with greater economy.

Effect of Condensation. One pound of steam at 17 pounds absolute pressure occupies 23.38 cubic feet of space in the cylinder of the engine, but one pound of water in the condenser occupies only about 0.016 cubic feet, which makes the steam occupy nearly 1,462 times as much space as the water into which it condenses. If then, the exhaust steam could be condensed instantly, the back pressure would be reduced almost to zero and the engine would exhaust into a vacuum.

Unfortunately the mere condensation of the steam will not give a perfect vacuum because of the air, always present in the water, which comes over from the boiler. Moreover, the condensed water is hot, and the vapor rising from it in the condensing chamber, together with the air and some leakage would spoil the vacuum were

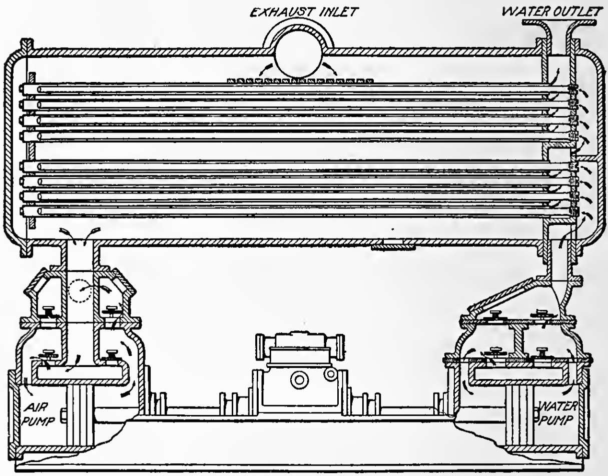


Fig. 76. Section of Steam Condenser of the Surface Type

it not for the air pump, which removes the air and condensed steam. Even with the best air pump it would be impossible to maintain a perfect vacuum, but a vacuum of 26 inches, which corresponds to about 2 pounds absolute pressure, can readily be maintained in good practice.

It is well known that a certain amount of heat is required to change one pound of water at a given temperature into steam at the same temperature; this is called the latent heat of vaporization. If the steam condenses, it must give up this latent heat. The easiest ways of doing this are either to let the steam come in contact with pipes through which cold water is circulated, as in a surface condenser, or mingle with a spray of water, as in a jet condenser. These two types will now be discussed.

Types of Condensers. Condensers may be divided into two general classes as follows:

(1) Surface condensers in which the cooling water is separated from the steam, usually by metallic surfaces in the form of tubes, the cooling water circulating on one side of this surface and the steam coming in contact with the metal on the other side.

(2) Jet condensers, including barometric condensers, siphon

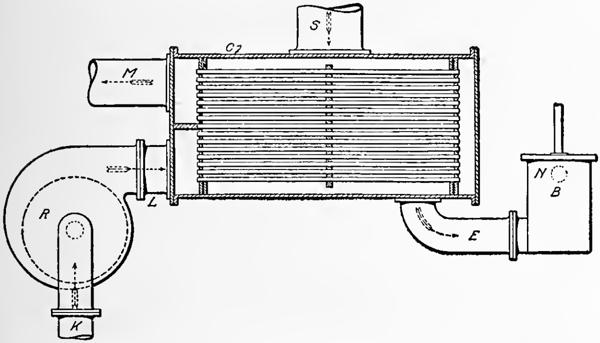


Fig. 77. Diagram Showing Relation of Surface Condenser to the Pumps Necessary for Proper Operation

condensers, ejector condensers, etc., in which the cooling water mingles with the steam to be condensed.

Surface Type. The condenser shown in section in Fig. 76 is one form of the surface type, in which the air pump and the circulating pump are both direct acting and both operated by the same steam cylinder. The cool condensing water is drawn from the supply into the circulating or water pump and is forced up through the valves and water inlet to the condenser. It flows, as indicated by the arrows, through the inner tubes of the lower section, then back through the space between the inner and the outer tubes. The water then passes upward and through the upper section, as it did in the lower, then it passes out of the condenser through the water outlet, taking with it the heat it has received from the steam.

The exhaust steam from the engine enters at the exhaust inlet and comes in contact with the perforated plate, which causes it to spread. The steam expanding in the condenser comes in contact with the tubes, through which cool water is circulating, and condenses. The air pump draws the air and condensed steam out of the condenser and thus maintains a partial vacuum. This causes the exhaust steam in the engine cylinder to be drawn into the condenser, at the bottom of which it collects as it condenses and is drawn into the air pump cylinder and discharged while heated to the hot well of the boiler. The use of this hot water as feed water effects a considerable saving, but the great advantage of the condenser is the reduction of the back pressure.

Hot water can not be used by an ordinary pump as easily as cold water because of the pressure of the vapor which arises from the hot water. In the condenser shown, the water and air pumps are run by the piston in the steam cylinder. Sometimes these pumps are connected to the main engine and receive motion from the shaft or crosshead.

The general arrangement of the surface condenser with the necessary pumps is shown in Fig. 77. The cooling water enters through the pipe K and flows to the circulating pump R, which forces the water into the condenser through the pipe L. In case the water enters the condenser under pressure from city mains, no circulating pump is necessary. After flowing through the tubes it leaves the condenser by means of the exit M and flows away. Exhaust steam enters at S and is condensed by coming in contact with the cold tubes; the water (condensed steam) then falls to the bottom of the condenser and flows to the air pump B by the pipe E. The air pump removes the air, vapor, and condensed steam from the condenser and forces it through the pipe N into the hot well, from which it goes to the boilers or to the feed tank.

Circulating Pump. The circulating pump, when separate from the condenser, is usually of the centrifugal type. This pump consists of a fan or wheel which is made up of a central web (or hub) and arms (or vanes). The vanes are curved and as the water is drawn in at the central part, the vanes throw it off at the circumference. A suitable casing directs the flow. This type of pump is advantageous because there are no valves to get out of order and, as the lift is little, if any, the pump will discharge a large volume of water in a nearly constant stream. The circulating pump is usually so placed that the water flows to it under a slight head. The pump is driven by an independent engine so that the circulating water may cool the condenser even if the main engine is not working.

Jet Type. Fig. 78 illustrates the longitudinal section of an independent jet condenser and pump. The cold water used to condense the steam enters at A, passes down the spray pipe B, and is broken into a fine spray by means of the spray cone C. This action insures a rapid and thorough mixing of the steam and water and consequently a rapid condensation. The exhaust steam enters at D with a comparatively high velocity, which is imparted to the water. The whole mixture of water, steam, and vapor passes at high velocity through the conical chamber E to the pump cylinder F, where it is forced into the pipe G. The spray cone is adjusted by means of the stem which passes through the stuffing box at the top of the condenser. The valves are shown at H and K. The steam end of the pump is at L.

In Fig. 79 a jet condenser is shown connected to a stationary engine. The exhaust pipe leads from the engine to the condenser,

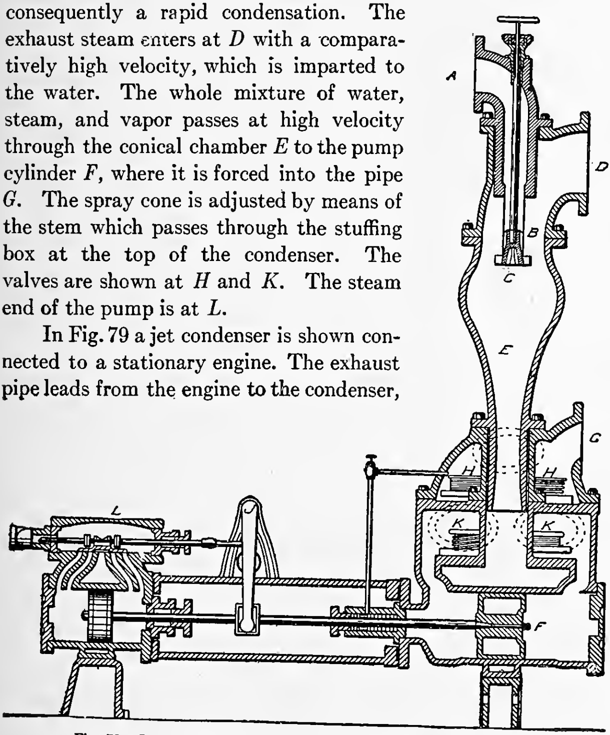


Fig. 78. Longitudinal Section of Independent Jet Condenser and Pump

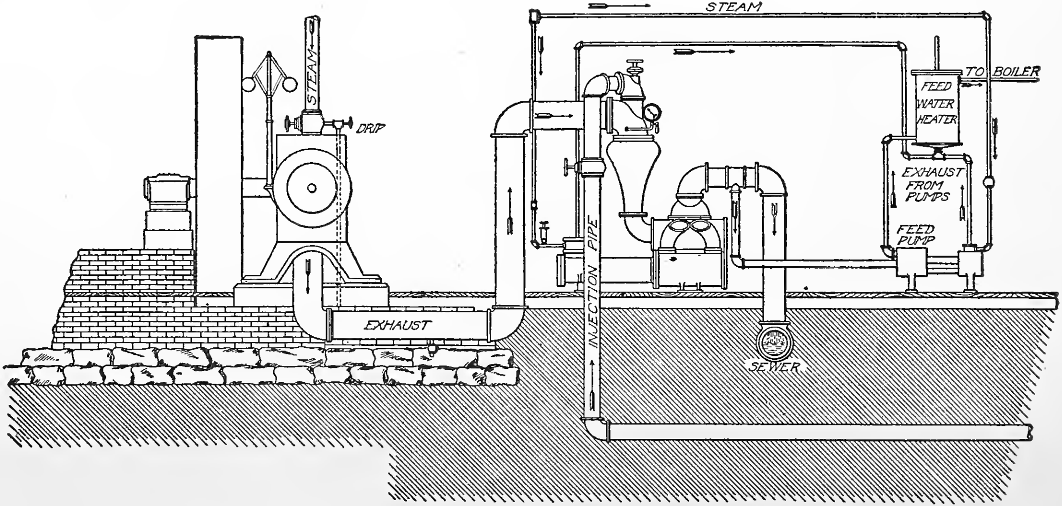


Fig. 79. Diagram Showing Stationary Engine with Connection to Jet Condenser, and Other Necessary Appliances

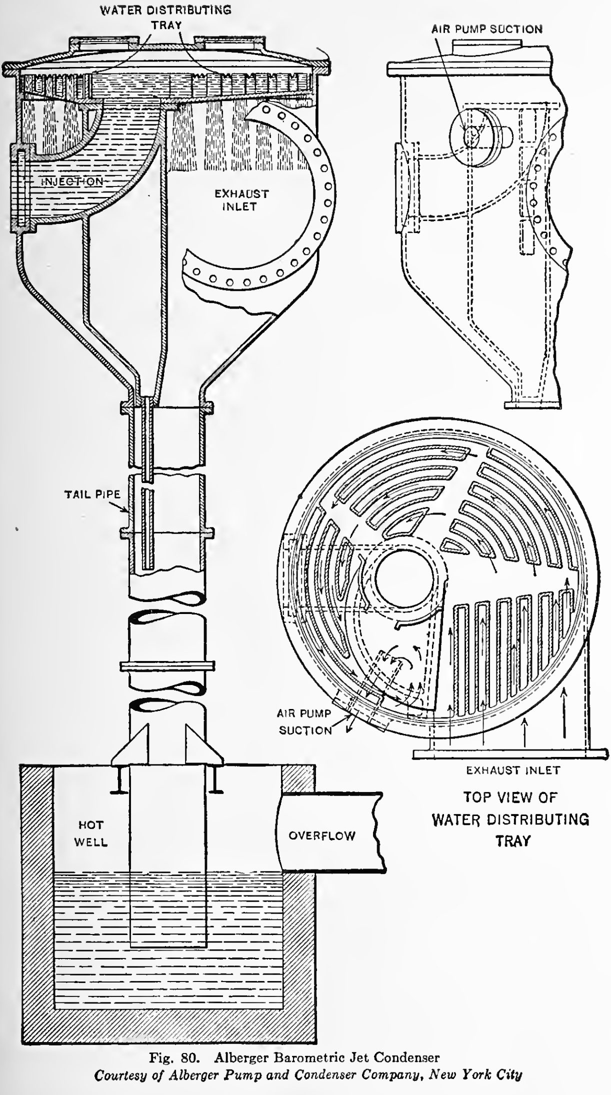


Fig. 80. Alberger Barometric Jet Condenser
Courtesy of Alberger Pump and Condenser Company, New York City

the arrows indicating the direction of the flow. Cold water enters the condenser through the pipe shown. Part of the mixture of exhaust steam and condensed water goes to the feed-water heater, which is kept nearly full; the rest passes to the sewer. The heater is placed a little above the feed pump, in order that the water may enter the pump under a slight head. This is necessary because the pump can not raise water which has been warmed by exhaust steam as readily as cold water.

Barometric Condenser. A type of condenser much used with reciprocating engines, and to a limited extent with steam turbines, is the barometric condenser, shown in Fig. 80. This condenser is one of the jet type. Steam enters at the point marked "exhaust inlet" in the left-hand figure and completely fills the exhaust steam chamber, while the condensing water enters through the injection pipe. The water rises into a distributing tray where it is broken up into many finely divided streams as seen in the left-hand figure. This spray condenses the steam in the exhaust chamber and passes down the tail pipe, carrying the condensed steam with it to the hot well. Air entering with the exhaust steam is cooled and collected in the air collector inside of the condensing chamber. A vacuum is maintained in the upper part of the condenser so that any air which has been collected during the process of condensing the steam is carried away through the pipe marked "air pump suction". A small amount of the cooler injection water is allowed to mix with this air so as to cool it before it passes on to the air pumps.

Westinghouse Leblanc Condenser. With the ordinary type of reciprocating engine, a vacuum of 26 to 27 inches is usually all that is desired. With modern steam turbines, however, a vacuum of 28 to 29 inches is common practice, and in many plants even these figures are exceeded. These figures, however, cannot be attained unless a very efficient air pump is used. The Leblanc condenser is considered one of the most efficient types of the many forms of jet condensers.

Fig. 81 shows a cross section of the Leblanc jet condenser, as manufactured by the Westinghouse Machine Company. This type is especially used in the larger steam turbine installations. In this condenser steam enters through the large opening E at the top, and the cooling water through B. This water is carried all around the circumference of the top of the condenser by the annular chamber C and is drawn inside the cone P through helical spray nozzles D, by the vacuum in the condenser. Inside of the cone P the water

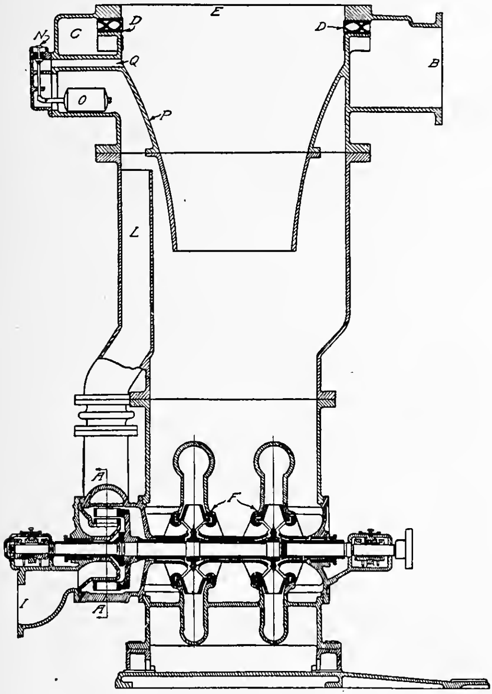


Fig. 81. Cross Section of Leblanc Jet Condenser
Courtesy of Westinghouse Machine Company, East Pittsburgh, Pennsylvania

is intimately mixed with the steam, condenses it, and falls to the bottom of the condenser. From here the water pumps F discharge the water from the condenser. The air released in the condenser by the water and condensed steam rises underneath the cone P, and is drawn off through the pipe L by the air pump A A. The inlet

I is the separate water supply for this air pump, shown more in detail by Fig. 82, which is a cross section on A A.

In Fig. 82, the water entering through the center of the pump is discharged through the orifice J into a tapering pipe, the water being emitted in a succession of layers, as indicated at G. These layers of water are sometimes spoken of as being water pistons. The air coming through pipe L is caught between these layers of water and carried to the atmosphere through the long diffuser pipe K.

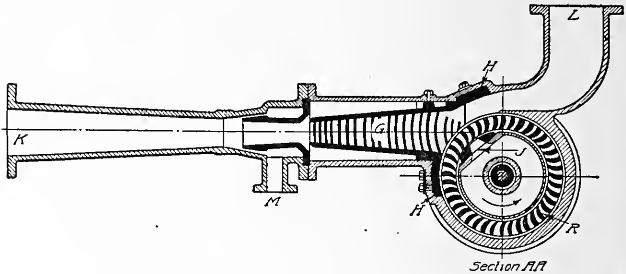


Fig. 82. Section of Leblanc Condenser Taken through AA, Fig. 81
Courtesy of Westinghouse Machine Company, East Pittsburgh, Pennsylvania

Since the cooling water enters by virtue of the vacuum, an accidental stopping of the pumps might cause serious trouble, due to the water rising above the top of the condenser. To take care of such emergencies, a very simple form of vacuum breaker is provided. In case the water rises in the condenser to an undesirable height, the float O, Fig. 81, opens the valve N and admits air to enter through passage Q directly into the condensing zone. This immediately stops the inflow of water by breaking the vacuum, and prevents damage to the turbine.

Relative Merits of Jet and Surface Condensers. In the jet condenser the steam, as soon as condensed, becomes mixed with the cooling water, and if the latter should be unsuitable for boiler-feed because of scale-forming impurities, acids, salts, etc., the pure distilled water represented by the condensed steam is wasted, and if it were necessary to purchase other water for boiler-feeding, this might represent a considerable waste of money. On the other hand, if the cooling water is suitable for boiler-feeding or if a fresh supply of good water is easily obtainable, the jet condenser, because of its simplicity and low cost, is unexcelled. Surface condensers are recommended where the cooling water is unfitted for boiler-feed and where no suitable and cheap supply of pure boiler-feed water is available. Condensed steam from a surface condenser makes the best boiler-feed water, being in fact pure distilled water entirely free from scale-forming matter and containing a considerable amount of heat, as compared with cold feed water. If the exhaust from reciprocating engines is to be condensed and used as boiler-feed water, a suitable oil separator should be interposed in the exhaust pipe between the engine and the condenser. Another advantage of the surface condenser as compared with the jet condenser is that there is no danger, in case of failure of vacuum pumps, of the circulating water backing up into the engine cylinder and wrecking the engine.

Effect of Condenser on Efficiency. It has already been stated that there is a gain in thermal efficiency by running an engine condensing, but it will be more clearly seen by considering a few figures. The thermal efficiency may be expressed by the previously mentioned formula

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

This efficiency may be increased if T_1 can be made larger—which would happen if the boiler pressure were increased—or if T_2 can be made smaller, which would result from reducing the back pressure by condensing. If the boiler pressure is raised, both the numerator and denominator of the fraction will increase, and the value of the fraction will be but slightly greater. If, however, the back pressure is reduced, the numerator T_1-T_2 will be larger, while the denominator T_1 will remain the same. It is apparent that this will cause a much greater increase in efficiency than raising the boiler pressure a like amount.

Suppose an engine is supplied with steam at 85.3 pounds (gauge) pressure and it exhausts at 3.3 pounds (gauge) pressure. The absolute temperature corresponding to 85.3+14.7, or 100 pounds pressure, is 327.86+459.5, or 787.36 degrees, and the absolute temperature corresponding to 3.3+14.7, or 18 pounds pressure, is 222.40+459.5, or

TABLE I

Increase in Efficiency by Use of Condenser for Various Engines

Type of Engine	Feed Water per Indicated Horsepower				Per Cent Gained by Condenser
	Non-Condensing		Condensing		
	Probable Limits Pounds	Assumed for Comparison Pounds	Probable Limits Pounds	Assumed for Comparison Pounds	
Simple High Speed	35 to 26	33	25 to 19	22	33
Simple Low Speed	32 to 24	29	24 to 18	20	31
Compound High Speed	30 to 22	26	24 to 16	20	23
Compound Low Speed	— —	24	20 to $12\frac{3}{4}$	18	25
Triple Exp. High Speed	27 to 21	24	23 to 14	17	29
Triple Exp. Low Speed	— —	—	18 to 12	—	—

681.9 degrees. Then the thermal efficiency determined from the formula becomes

$$E=\frac{T_1-T_2}{T_1}=\frac{787.36-681.9}{787.36}$$

$$=.134, \text{ or } 13.4 \text{ per cent}$$

If the boiler pressure were raised to 140 pounds absolute, the efficiency would be

$$E=\frac{812.59-681.9}{812.59}$$

$$=.161, \text{ or } 16.1 \text{ per cent}$$

If instead of increasing the boiler pressure a condenser is used and the exhaust pressure reduced to 4 pounds (absolute), the efficiency becomes

$$E=\frac{787.36-612.5}{787.36}=.222, \text{ or } 22.2 \text{ per cent}$$

Thus it is seen that if the exhaust pressure is lowered 14 pounds absolute there will be a greater increase in efficiency than if the boiler pressure is raised 40 pounds.

The per cent of efficiency that is obtained by the use of a condenser is shown in Table I.

Cost of Cooling Water Determines Condenser Economy. While the above figures are very encouraging, yet conditions may arise where the per cent of gain may be materially lessened or entirely lost, due to the cost of water. Condensing engines require from 20 to 30 pounds of cooling water to condense each pound of steam used, depending on the necessary temperature. Thus it can be seen that the quantity of cooling water is relatively very large, and if it is purchased from a water company, quite an item is added to the yearly expense account for the one item of water. If, however, some means could be provided whereby the circulating water as it issues from the condenser could be cooled and then used over again in the condenser, the non-condensing engine could be run condensing, thus taking advantage of all the benefits due to the use of reduced back pressure and heating of the feed water. This has been attempted by conducting the heated discharge water to a pond, where it is allowed to cool to a lower temperature before being used again. Another plan is to place in the yard or on the roof of the building large shallow pans, in which the water is cooled by being exposed to the atmosphere. These methods are unsatisfactory on account of the considerable area necessary and the slow action. In addition, they are uncertain, because they are dependent upon atmospheric conditions.

Cooling Tower and Water Table. A more efficient and at the same time more expensive process is to use a cooling tower or a water table. Fig. 83 illustrates the general arrangement of a cooling tower located upon the roof of a building. The discharge from the condenser is led, as shown by the arrows, to the top of the cooling tower, where it is cooled before being returned to the condenser. This cooling is effected by distributing the water, by a system of piping, to the upper edge of a series of mats or slats, over the surface of which the water flows in a thin film to a reservoir which is situated in the bottom of the cooling tower. The mats partially interrupt the flow and, by breaking up the water in small streams, cause new portions to be exposed to the cooling effect of the air currents. The water from the reservoir then flows downward through the suction pipe and is pumped by the circulating pump through the condenser. After passing through the condenser and absorbing heat from the exhaust steam, it rises through the discharge pipe and commences the circuit over again.

The tower may have several arrangements and be made of various materials. A satisfactory form is constructed of steel plates within the tower, or a large number of mats of steel wire cloth galvanized after weaving. The tower may be supported upon a proper foundation or upon legs, instead of being situated on the top of a building, as the one shown in the illustration.

To assist in the cooling of the water, the air is often made to circulate rapidly by means of a fan, which forces the air into the

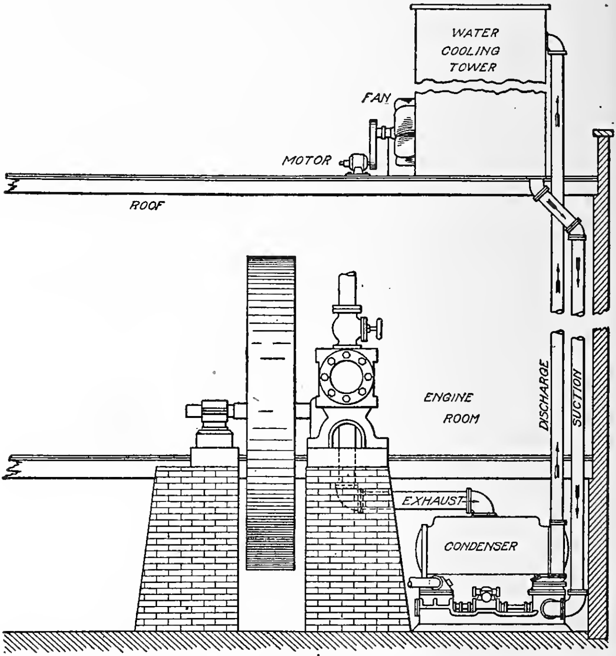


Fig. 83. Diagram of Stationary Engine with Connections to Water Cooling Tower on Roof of Building

lower part of the tower and upward through the mats. This fan may be driven by an electric motor, by a line of shafting, or by a small independent engine.

In case the fan is not used, the mats are arranged so that they are exposed to the atmosphere. This of course necessitates the removal of the steel casing. Usually the fanless tower must be placed at the top of a high building or in some position where the currents of air can readily circulate through the mats.

With an efficient type of cooling tower, the water may be reduced from 30 to 50 degrees, thus allowing a vacuum of from 22 to 26 inches. This will, of course, greatly increase the economy of the plant and allow the heated feed water to be returned to the boiler.

The water table is usually made of wooden slats placed in the ground near the plant. After trickling over the slats and becoming cooled by the air, it collects in the bottom of the reservoir and is then pumped into the condenser.

Amount of Cooling Water Per Pound of Steam. Besides condensing the steam, the injection water cools it still further, so that more than merely the latent heat is removed from it. If exhaust steam enters the condenser at a temperature t_1, it contains a certain amount of heat, known as *total heat at temperature* t_1. If it is condensed and cooled to a temperature t_2, at which it leaves the condenser, it then contains a certain amount of heat, known as *total heat at temperature* t_2.

If A represents the total heat at t_1 and B represents the heat of the liquid at t_2, then the heat given up by one pound of condensed steam is equal to $(A-B)$ British Thermal Units, provided the exhaust that enters the condenser is dry saturated steam. If C is the temperature of the injection or cooling water and D is the temperature of the discharge water, then every pound of cooling water absorbs approximately one British Thermal Unit for every degree rise in the temperature, or we may say that the heat absorbed is equal to $(D-C)$ British Thermal Units per pound of cooling water. Then it will take as many pounds of water W to absorb $(A-B)$ heat units as $(D-C)$ is contained in $(A-B)$. This may be expressed thus

$$W=\frac{(A-B)}{(D-C)}$$

Therefore, W represents the number of pounds of water required per pound of steam condensed.

EXAMPLE 1. Suppose steam is expanded in an engine to 4 pounds absolute pressure. If the initial temperature of the cooling water is 45 degrees, and the condenser is of the surface type, discharging water at 120 degrees, and the temperature of the condensed steam is 130 degrees, how many pounds of cooling water are required per pound of steam?

SOLUTION. By consulting the steam tables, we find the total heat of steam at 4 pounds pressure to be 1126 5 British Thermal Units. The heat of the liquid in the condensed steam at 130 degrees is 98.0 British Thermal Units. Then

$$W=\frac{1126.5-98.0}{120-45}$$
$$=13.71 \text{ pounds}$$

EXAMPLE 2. Suppose steam at 6 pounds absolute pressure exhausts into a jet condenser. The temperature of the injection water is 50 degrees and the discharge is 120 degrees. How many pounds of water are necessary to condense 8 pounds of steam?

SOLUTION. In the jet condenser the temperature of the condensed steam and the discharge water is the same. We find from the steam tables that the total heat of steam at 6 pounds absolute is 1133.6 British Thermal Units, and the heat of the liquid in the condensed steam at 120 degrees is 88.0 British Thermal Units. Then as before.

$$W=\frac{1133.6-88.0}{120-50}$$
$$=14.94$$

Therefore, 8 pounds of water will require 14.94×8, or 119.52 pounds.

The above calculation can not be relied upon to any great extent for we seldom know the true condition in the condenser, and it would be of little value to us if we did know, as the exact condition will change considerably. In practice it is customary to allow for about twice as much water as the above calculation would require. These figures give us a fair idea of the necessary sizes of the pipes and passages leading to the condenser, and give a basis for estimating the dimensions of the air pump.

Cooling Surface in Surface Condensers. The amount of surface required to condense the steam in surface condensers depends upon the conductivity of the metal, the condition of the tubes and their thickness, and the difference in temperature between the two sides. The tubes of a condenser are much thinner than boiler tubes, hence we might expect them to be more efficient in condensing the steam than the boiler tubes are in evaporating water. It has been found in actual practice, that a surface condenser receiving cooling water at 60 degrees and discharging it at 120 degrees will condense from 10 to 20 pounds of steam per square foot of the tube surface per hour. An average of 13 pounds per square foot of surface per hour is considered a fair one. With exhaust pressure from 6 to 30 pounds absolute, it has been found that an allowance of 1.5 to 3.0 square feet of cooling surface per indicated horsepower is sufficient, when the initial temperature of cooling water is 60 degrees and the final temperature is 120 degrees.

It is evident that the amount of surface will depend upon the quantity of steam used per hour by the engine, the pressure and temperature of the exhaust, and the temperature of the cooling water and discharge. There must also be an allowance for inefficient work after the condenser has become fouled with service. All these conditions make the problem so uncertain that calculations by means of formulas are likely to be untrustworthy, and it is best at all times to make estimates from the figures given for similar conditions in actual service.

Feed Water Heaters. In many places where water is expensive and the condensing engines can not be run economically, a very considerable saving can be effected either by allowing the exhaust steam to condense into a feed water heater, thus saving the heat that would otherwise be wasted, or by using the exhaust steam for heating purposes. Of course in such cases the steam consumption of the engine is high, but if proper allowance is made for the heat used for other purposes, the actual fuel consumption rightfully charged to the engine is not excessive. If the feed water is heated by waste gases, then the gain belongs to the boiler and not to the engine.

ANALYSIS OF ENGINE MECHANISMS

CRANK EFFORT

In the steam engine the steam exerts a pressure on the crank pin through the piston rod and connecting rod. When the crank is at the dead center, the entire pressure is on the bearing of the crank shaft, and there is no tendency to turn the crank. As the crank pin moves from the dead center, the tendency increases until it reaches a maximum and then decreases until, at the other dead center, it is zero again. If the connecting rod were of infinite length and steam were admitted throughout the whole stroke, the maximum tendency, or the maximum turning moment as it is called, would occur with the crank at right angles to the line connecting the dead points.

Variable Thrust. In the actual engine the thrust along the rod is constantly varying even though the pressure on the piston remains the same. This is due to the angularity of the connecting rod. The turning moment is always equal to the thrust along the connecting rod multiplied by the perpendicular distance from the connecting rod to the center of the shaft. If the steam pressure on the piston remains constant, the maximum turning moment occurs when the connecting rod is at right angles to the crank, for in this position the perpendicular distance from the rod to the center of the shaft is a maximum and equal to the length of the crank; and, as the rod makes its greatest angle with the line connecting the dead center at this point, the thrust along it will also be a maximum. If the cut-off is very early, one-quarter stroke for instance, the maximum thrust along the rod will occur earlier than at the point previously mentioned, but the leverage of the force will be less, so that really there will be little change in the point of maximum turning moment no matter where the cut-off may occur.

Diagrams. To represent this turning moment, diagrams of crank effort may be drawn, with rectangular co-ordinates, having the crank angles represented as abscissas and the turning moments corresponding to these angles as ordinates.

FLYWHEEL

Besides the thrust of the connecting rod there must be taken into account friction and the inertia of the reciprocating parts. At first this may be thought of small consequence but with a fairly heavy piston and connecting rod it is obvious that at high speed the momentum would be great. In the case of a vertical engine, on the up stroke the steam must lift this heavy mass and impart a very considerable velocity to it, while on the down stroke the acceleration of the mass is added to the steam pressure. This makes the effective force on the up stroke less than that due to the actual steam pressure, and greater on the down stroke.

Function. In the case of a horizontal engine it is evident that while the piston can push the crank around during part of the stroke, and pull it along during another part, yet at the end of the stroke the pressure on the piston, no matter how great, can exert no turning moment on the shaft. Therefore, if some means is not provided for making the shaft turn past these points without the assistance of the piston, it may stop. This means is provided in the flywheel which is merely a heavy wheel placed on the main shaft. On account of the momentum of the flywheel it can not be stopped quickly and therefore carries the shaft around until the piston can again either push or pull.

Size of Wheel. If a long period be considered, the mean effort and the mean resistance must be equal; but during this period there are temporary changes of effort, the excesses causing increase of speed. To moderate these fluctuations several methods are employed.

The turning moment on the shaft of a single cylinder engine varies, *first*, because of the change in steam pressure, and *second*, on account of the angularity of the connecting rod. Before the piston reaches mid-stroke the turning moment is a maximum, as shown by

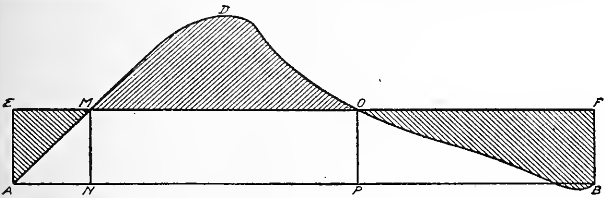


Fig. 84. Graphical Representation of Turning Moment of Crank Shaft of a Single-Cylinder Engine for One Stroke

the curve, Fig. 84. Near the ends of the stroke the turning moment diminishes and finally becomes zero. This, of course, tends to cause a corresponding change in the speed of rotation of the shaft. In order to have this speed as nearly constant as possible and to give a greater uniformity of driving power, the engine may be run at high speed. By this means the inertia of the revolving parts, such as the connecting rod and crank, causes less variation. When the work to be done is steady and always in the same direction, a heavy flywheel may be used. The heavier the flywheel, the steadier will be the motion. It is desirable, of course, in all engines to have steady motion, but in some cases it is more important than in others. For instance, in electric lighting plants it is necessary that the machinery shall move with almost perfect steadiness. It is undesirable to use larger wheels than are absolutely necessary, because of the cost of the metal, the weight on the bearings, and the danger from bursting.

Methods of Reducing Size. If the turning moment which is exerted on the shaft from the piston could be made more regular and if dead points could be avoided, it would be possible to get a steadier motion with a much smaller flywheel.

If the engine must be stopped and reversed frequently, two or more cylinders are used, being connected to the same shaft. The cranks are placed at such angles that when one is exerting its minimum rotative effort, the other is exerting its maximum, or when one is at a dead center, the other is exerting its greatest effort. These cylinders may be identically the same in dimension as is the case with most hoisting engines and with many locomotives; or the engine may be compound or triple expansion. This arrangement is also used on engines for mines, collieries, and for hoisting of any sort where ease of stopping, starting, and reversing are prerequisites. Simple expansion engines with their cranks at right angles are usually spoken of as being coupled.

The governor adjusts the power of the engine to any large variation of the resistance. The flywheel has a duty to perform which is similar to that of the governor. It is designed to adjust the effort of the engine to sudden changes of the load which may occur during a single stroke. It also equalizes the variation in rotative effort on the crank pin. The flywheel absorbs energy while the turning moment is in excess of the resistance, and restores it while the crank is at or near the dead points. During these periods the resistance is in excess of the power.

Action of Flywheel. The action of the flywheel may be represented as in Figs. 84 and 85. It will be noticed that in Fig. 84, the curve of the crank effort runs below the axis toward the end of the stroke. This is because the compression is greater than the pressure near the end of expansion, and produces a resultant pressure on the piston. In Fig. 85 the effect of compression has been neglected. Let us suppose that the resistance, or load, is uniform. In Fig. 84, the line $A\ B$ is the length of the semi-circumference of the crank pin. or the circumferential distance the crank pin moves during one stroke. The curve $A\ M\ D\ O\ B$ is the curve of turning moment for one stroke. $M\ N$ is the mean ordinate and, therefore, $A\ E\ F\ B$ represents the constant resistance. The effort and resistance must be equal if the speed is uniform; hence the area $A\ E\ F\ B$ equals

A M D O B. Then area *A E M* plus area *O F B* equals area *M D O*. At *A* the rotative effort is zero because the crank pin is at the dead point and from *A* to *N*, the turning moment is less than the resistance. At *N* the resistance and the effort are equal. From *N* to *P* the effort is in excess of the resistance. At *P* the effort and the resistance are again equal. From *P* to *B* the resistance is greater than the effort. In other words, from *A* to *N* the work done by the steam is less than the resistance. This shows that the work represented by the area *A E M* must have been done by the moving parts of the engine. From *N* to *P* the work done by the steam is greater than the resistance, and the excess of energy is absorbed by or stored in the moving parts. From *P* to the end of the stroke the work represented by the area *O F B* is done on the crank pin by the moving parts.

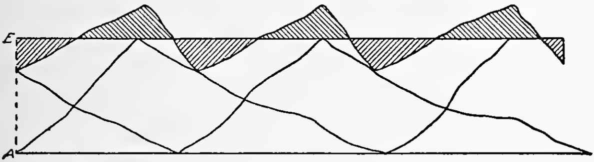


Fig. 85. Simultaneous Crank Effort Curves of Two Engines Acting at Right Angles to Each Other

It is known that energy is proportional to the square of the velocity from the formula

$$E = \frac{WV^2}{2g}$$

in which E is energy in foot pounds, W is weight in pounds, V is velocity in feet per second, and g is acceleration of gravity in feet per second^2. Hence as W and g remain the same, the velocity must be reduced when the moving parts are giving out energy and increased when receiving energy. Thus it is seen that the action of the crank pin is to move slowly, then more rapidly. The weight of the revolving parts of an engine is not sufficient to absorb sufficient surplus energy, hence a heavy flywheel is used.

In case there are two engines at right angles, two crank effort curves may be drawn, as shown in Fig. 85. The mean ordinate *A E* is equal to the mean or constant resistance. There are two minimum and two maximum velocities in one stroke. The diagram shows that the variation is much less than for a single cylinder, hence a lighter wheel may be used.

Calculations of Mass. The weight of the flywheel depends upon the character of the work done. For pumping engines and ordinary machine work the effort need not be as constant as for electric lighting. In determining the proper weight of a flywheel the diameter of the wheel must be known. If the wheel is too large, the high linear velocity of the rim will cause too great a centrifugal force and the wheel will not be safe. In practice, about 6,000 feet per minute is taken as the maximum linear velocity of cast-iron wheels. When made of wood and carefully put together the velocity may be taken as 7,000 to 7,500 feet per minute.

The linear velocity of a wheel is expressed in feet per minute by the formula $V=2\pi\ R\ N$, or $\pi\ D\ N$, in which V is velocity in feet per second, R is radius of wheel in feet, D is diameter of wheel in feet, and N is revolutions per minute.

Then if a wheel runs at 100 revolutions per minute, the allowable diameter would be obtained from the equation

$$6000=3.1416\times D\times 100$$

Therefore

$$D=\frac{6000}{3.1416\times 100}$$
$$=19.1 \text{ feet}$$

If a wheel is 12 feet in diameter the allowable speed is found to be

$$N=\frac{V}{\pi D}$$
$$=\frac{6000}{3.1416\times 12}$$
$$=159 \text{ revolutions per minute}$$

It is usual to make the diameter less than the calculated diameter.

Having determined the diameter, the weight may be calculated by several methods. There are many formulas to obtain this result given by various authorities, one formula being

$$W=\frac{C\times d^2\times b}{D^2\times N^2}$$

in which W is weight of rim in pounds; d is diameter of cylinder in inches; b is length of stroke in inches; D is diameter of flywheel in feet; N is number of revolutions per minute; and C is a constant having a value which varies for different types of engines and for different conditions as follows:

Slide valve engines, ordinary work	$C=$ 350,000
Corliss engines, ordinary work	$C=$ 700,000
Slide-valve engines, electric lighting	$C=$ 700,000
Automatic high speed engines	$C=$ 1,000,000
Corliss engines, electric lighting	$C=$ 1,000,000

EXAMPLE 1. Find the weight of a flywheel rim for an automatic high speed engine used for electric lighting. The cylinder is 24 inches in diameter; the stroke is 2 feet. It runs at 300 revolutions per minute, and the flywheel is to be 6 feet in diameter.

SOLUTION.

$$W=\frac{1000000\times(24)^2\times 24}{36\times 90000}$$
$$=4266 \text{ pounds}$$

EXAMPLE 2. A plain slide valve engine for electric lighting is 20 inches $\times$ 24 inches. It runs at 150 revolutions per minute. The flywheel is to be 8 feet in diameter. What is the weight of its rim?

SOLUTION.

$$W=\frac{700000\times 400\times 24}{64\times 22500}$$
$$=4666 \text{ pounds}$$

The weight of a flywheel is considered as being in the rim. The weight of the hub and arms is simply extra weight. Then, if the weight of the rim and its diameter be known, the width of the face and thickness of the rim can be found. Assume the given diameter to be the mean of the diameter of the inside and outside of the rim. Let b equal width of face in inches; t equal thickness of rim in inches; d equal diameter of flywheel in inches; and .2607 equal weight of 1 cubic inch of cast iron. Then

$$W=.2607\times b\times t\times \pi d$$
$$=b\times t\times .819\, d$$

EXAMPLE 3. Suppose the rim of a flywheel weighs 6,000 pounds, is 9 feet in diameter, and the width of the face is 24 inches. What is the thickness of the rim?

SOLUTION.

$$t=\frac{W}{.819\, db}$$
$$=\frac{6000}{.819\times 108\times 24}$$
$$=2.83 \text{ inches}$$

In this case the rim would probably be made $2\frac{13}{16}$ inches thick. The total weight, including hub and arms, would probably be about 8,000 pounds.

GOVERNOR

The load on an engine is never constant, although there are cases where it is nearly uniform. While the engine is running at constant speed, the resistance at the flywheel rim is equal to the work done by the steam, disregarding friction. If the load on the engine is wholly or partially removed and the supply of steam continues undiminished, the force exerted by the steam will be in excess of the resistance. Work is equal to force multiplied by distance; hence, with constant effort, if the resistance is diminished, the distance must be increased. In other words, the speed of the engine will be increased, and the engine will "race." Also, if the load increases and the steam supply remains constant, the engine will "slow down."

It is evident, then, that if the speed is to be kept constant some means must be provided so that the steam supply shall at all times be exactly proportional to the load. This is accomplished by means of a governor.

Methods of Action. Steam-engine governors act in one of two ways (1) they may regulate the pressure of steam admitted to the steam chest, or (2) they may adjust the speed by altering the amount of steam admitted. Those which act in the first way are called *throttling governors,* because they throttle the steam in the main steam pipe. Those of the latter class are called *automatic cut-off governors,* since they automatically regulate the point of cut-off.

Theoretically, the method of governing by throttling the steam causes a loss in efficiency, but the throttling superheats the steam, thus reducing cylinder condensation. By the second method the loss in efficiency is very slight, unless the ratio of expansion is already great, in which case shortening the cut-off causes an increasing cylinder condensation.

Control by Centrifugal Force. In most governors of the throttling type and those applied to Corliss engines, centrifugal force counteracted by some other force is employed. A pair of heavy masses (usually iron balls or weights) are made to revolve about a spindle, which is driven by the engine. When the speed increases, the centrifugal force increases and the balls tend to fly outward, that is, they revolve in a larger circle. The controlling force, which is usually gravity or springs, is no longer able to keep the balls in their former path. When, therefore, the increase is sufficiently great, the balls in moving outward act on the regulator, which may throttle the steam or cause cut-off to occur earlier.

With the throttling governor, a balanced throttle valve is placed in the main steam pipe leading to the valve chest. If the engine runs faster than the desired speed, the balls are forced to revolve at a higher speed. The increase in centrifugal force will cause them to revolve in a larger circle and in a higher plane. By means of levers and gears, the spindle may be forced downward, thus partially closing the valve. The engine, therefore, takes the steam at a low pressure, and consequently the speed falls slightly.

Similarly, if the load is increased, the engine slows down, causing the balls to drop and open the valve more widely; steam at higher pressure is then admitted and the speed is increased to the regular number of revolutions.

With the Corliss or other four-valve engines, the governor acts differently. Instead of throttling the steam in the steam pipe, the governor is connected to the releasing gear by rods. An increase of speed causes the releasing gear to unhook the disengaging link earlier in the stroke. This causes earlier cut-off, which of course decreases the power and speed, since the amount of steam admitted is less. If for any reason the load increases, the governor causes the valves to be held open longer. The cut-off, therefore, occurs later in the stroke.

Pendulum Governor. One of the most common forms of governor is similar to that invented by James Watt. It is called from its appearance the pendulum governor and is illustrated in principle in Fig. 86. To consider the theory of the pendulum governor, the masses of the balls are assumed to be concentrated at their centers and the rods are made of some material having no weight.

When the governor is revolving about its axis at a constant speed, the balls revolve in a circle having a radius r. The distance from this plane to the intersection of the rods, or the rods produced, is called the height and is equal to h.

If the balls revolve faster, the centrifugal force increases, r becomes greater, and h diminishes. The mathematical expression for centrifugal force is

$$F=\frac{Wv^2}{gr}$$

in which F is force in pounds; W is weight of one ball in pounds: v is velocity in feet per second; g is acceleration due to gravity; and r is radius in feet. From the above equation it is seen that force varies inversely as the radius.

While the pendulum is revolving, centrifugal force acts horizontally outward and tends to make the balls fly from the center; and the action of gravity tends to make the balls drop downward. In order that the balls shall revolve at a certain height, the moments of these two forces about the point of suspension must be equal, or

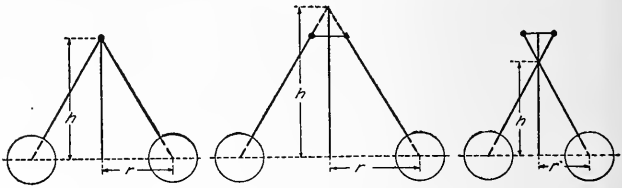


Fig. 86. Diagrams Showing Action of Pendulum Governor

the weight of the balls multiplied by their distance from the center must equal the centrifugal force multiplied by the height, or

$$W \times r = F \times h$$

from which

$$\frac{h}{r} = \frac{W}{F}$$

Substituting value of F just given, we have

$$\frac{h}{r} = \frac{W}{\frac{Wv^2}{gr}}$$

$$= \frac{gr}{v^2}$$

Therefore,

$$h = \frac{gr^2}{v^2}$$

Now since v, the linear velocity of a point revolving in the circumference of a circle, is expressed as $2 \pi r N'$ feet per second, where N'

is revolutions per second, this value may be substituted in the above formula, giving

$$h = \frac{g\,r_2}{4\,\pi^2\,r^2\,(N')^2}$$

$$= \frac{g}{4\,\pi^2\,(N')^2}$$

and since the values of g and π are known, the formula may be written

$$h = \frac{32.16}{4 \times 3.1416^2 \times (N')^2}$$

$$= \frac{.8146}{(N')^2} \text{ feet.}$$

$$= \frac{9.775}{(N')^2} \text{ inches}$$

If it is desired to use N, the r.p.m., instead of N', the r.p.s., the former may be substituted in the formula by multiplying the fraction by $\overline{60}^2$, or 3600, giving

$$h = \frac{2932.56}{N^2} \text{ feet}$$

$$= \frac{35190.7}{N^2} \text{ inches}$$

From the above formula it is evident that the height is independent of the weight of the balls or the length of the rod, depending entirely upon the number of revolutions. The height varies inversely as the square of the number of revolutions.

The ordinary pendulum governor is not isochronous, that is, it does not revolve at a uniform speed in all positions, the speed changing as the angle between the arms and spindle changes.

Fly-Ball Governor. The early form consisted of two heavy balls suspended by links from a pin connection in a vertical spindle, as shown in Figs. 87 and 88. The spindle is caused to revolve by belting or gearing from the main shaft, so that as the speed increases, centrifugal force causes the balls to revolve in a circle of larger and larger diameter. The change of position of these balls can be made to affect the controlling valves so that the admission or throttling will vary with their position. With this governor it is evident that for a given speed of the engine there is but one possible position for the governor, consequently one definite amount of throttling or one point of cut-off, as the case may be. If the load varies, the speed of the engine will change. This causes the position of the governor balls to be changed slightly, thus altering the pressure. But in order that the pressure or cut-off shall remain changed, the governor balls must stay in their new position. That is to say, the speed of the engine

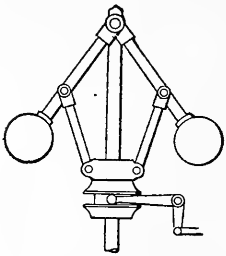


Fig. 87. Simple Type of Fly-Ball Governor

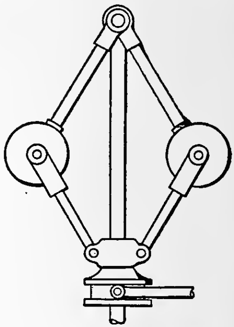


Fig. 88. Later Type of Fly-Ball Governor

must be slightly changed. Thus with the old ball governors there was a slightly different speed for each load. This condition has been greatly improved by various modifications until now such governors give excellent regulation.

While the engine is running with a light load, the valve controlled by the governor will be open just enough to admit steam at a pressure that will keep the engine running at a given speed. Now if the engine is heavily loaded, the throttle valve must be wide open. The change of opening is obtained by a variation in the height of the governor, which is caused by a change of speed. Thus it is seen that the governor can control the speed only within certain limits which are not far apart. The difference in the extreme heights of the governor must be sufficient to open the throttle its entire range. In

TABLE II

Heights of Governor for Different Speeds of Engine

Number of Revolutions Per Minute	Height in Inches	Variation of Height in Inches 4 Per Cent
250	.563	0225
200	.879	.035
175	1.149	.046
150	1.564	.062
125	2.252	090
100	3.519	.140
75	6.256	250
50	14 076	563

most well-designed engines, equipped with a throttling governor, the speed will not vary more than 4 per cent, that is, 2 per cent above or below the mean speed.

From the formula $h=\frac{35190.7}{N^2}$, the heights corresponding to given speeds can be computed as shown in the second column of Table II. The third column is the variation in height for a speed variation of 4 per cent or 2 per cent either above or below the mean.

Disadvantage of Ordinary Fly-Ball Type. From Table II it will be seen that for a considerable variation of speed there is but slight variation in the height of the governor, this being too small to control the cut-off or throttling mechanism throughout the entire range. Also for high speeds the height of the governor is so small that it would be difficult to construct it.

Other disadvantages of the fly-ball governor are as follows: It is apparent that the valves must be controlled by the weight of the governor balls. In large engines this requires very heavy balls in order to quickly overcome the resistance of the valves. But these large balls have considerable inertia and will therefore be reluctant to change their speed with that of the engine. The increased weight will also increase the friction in the governor joints and the cramping action existing when the balls are driven by the spindle will increase this friction much further. All these things tend to delay the action of the governor, so that in all large engines the old-fashioned governor became sluggish. The balls had to turn slowly because they were so heavy; this was especially troublesome in high-speed engines.

Porter Improved Type. To remedy these defects the weighted or Porter governor, Fig. 89. was designed. It has a greater height for a given speed, and the variation in height for a given variation of speed is greater and, consequently, more sensitive. By increasing this variation in height, the sensitiveness is increased. Thus, if a governor running at 50 revolutions has a variation in height of .57 inch, it is not as sensitive as one having a variation of 1 inch for the same speed.

In the weighted governor, the weight is formed so that the center of gravity is in the axis. It is placed on the spindle and is free to revolve. The weight adds to the weight of the balls, and thus increases the moment of the weight. It does not, however, add to the centrifugal force, and hence the moment of this force is unchanged. It may then be said that the weight adds effect to the weight of the governor balls but not to the centrifugal force, and as a consequence the height of the governor for a given speed is increased. If W equals the weight of the ball as before, and W' equals one-half the added weight, the equated moments are

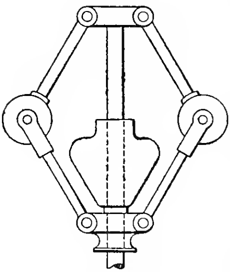


Fig. 89: Porter Improved Type of Fly-Ball Governor

$$(W+W')\,r = Fh$$

Substituting for F its value obtained from the formula, p. 147, we have

$$(W+W')\,r = \left(\frac{Wv^2}{gr}\right) h$$

$$h = (W+W')r\left(\frac{gr}{Wv^2}\right)$$

$$= \frac{(W+W')r^2g}{W\times 4\pi^2r^2(N')^2}$$

$$= \frac{(W+W')}{W}\times\left(\frac{g}{(4\pi^2(N')^2}\right)$$

Since it is known that

$$\frac{g}{4\pi^2(N')^2}=\frac{.8146}{(N')^2}$$

$$h=\left(\frac{W+W'}{W}\right)\times\frac{.8146}{(N')^2}$$

Hence the height of a weighted governor is equal to the height of a simple pendulum governor multiplied by $\left(\frac{W+W'}{W}\right)$, or $\left(1+\frac{W'}{W}\right)$.

For instance, if the height of a simple pendulum is 10 inches and

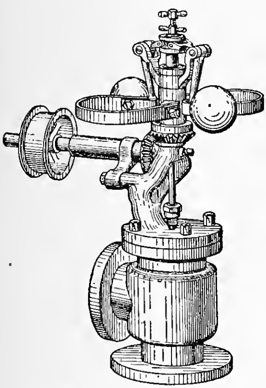


Fig. 90. Waters Governor with Safety Stop

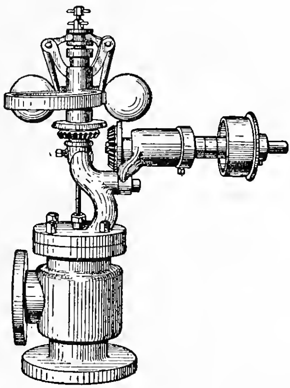


Fig 91. Waters Spring Type of Fly-Ball Governor

the weight of the balls equal to the added weight, the height of the weighted governor will be

$$h=\left(1+\frac{1}{1}\right)\times 10$$

$$=2\times 10$$
$$=20$$

Thus it is evident that if a weight equal to the combined weight of the balls is added, the height of the governor will be doubled. If the belt driving the governor slips off or breaks, the balls will drop, with the result that the engine will "run away." To diminish this danger many governors are provided with some kind of safety stop which closes the valve when the governor loses its normal action. Usually a trip is provided which the governor does not touch in its normal positions, but which will be released if the balls drop down below a certain point.

Spring Type. In many cases a spring is used in place of the weight. This type of governor is frequently used on throttling engines, and it consists of a pendulum governor with springs added to counteract the centrifugal force of the balls. Thus the height and sensitiveness are increased. Fig. 90 shows the exterior view of a Waters governor and Fig. 91 shows the same governor having the safety stop. In this governor the weights are always in the same plane, the variation in height being due to the action of the bell-crank levers connecting the balls and spindle. When the balls move outward, the spindle moves downward and tends to close the valve. The governor balls are caused to revolve by means of a belt and bevel gears. The valve and seat are shown in section in Fig. 92. The valve is a hollow cylinder with three ports through which steam enters. The seat is made in four parts, that is, there are four edges that the steam passes as it enters the valve. The valve, being cylindrical and having steam on both sides, is balanced, and because of the many openings only a small travel is necessary.

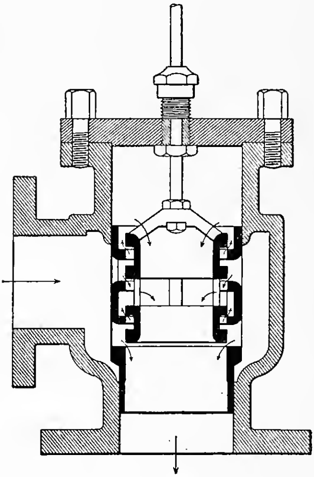


Fig. 92. Section of Valve and Valve Seat of Waters Governor

Shaft Governor. Usually some form of pendulum governor is used for throttling engines. For governing an engine by varying the point of cut-off, shaft governors are generally used, although the Corliss and some other engines use pendulum governors for this purpose. Cut-off governors, which are called shaft governors because they are placed on the main shaft; are made in many forms, but their essential features are the same. Two pivoted masses or weights are arranged symmetrically on opposite sides of the shaft and their tendency to fly outward when the speed increases is resisted by springs. When in action the outward motion of the weights causes the admission valve to close earlier, and the inward motion causes it to close later. This change is effected by altering the position of

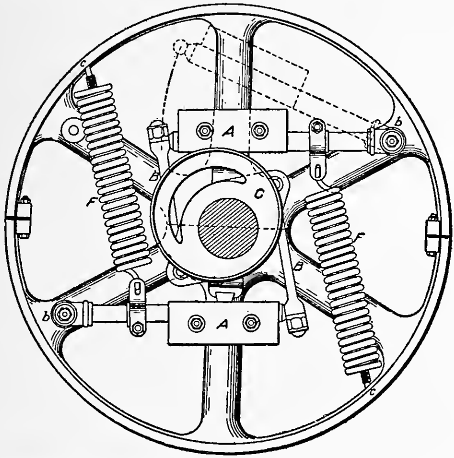


Fig. 93. Diagram Showing Action of Buckeye Shaft Governor

the eccentric, either by changing the eccentricity or the angular advance.

Shaft governors are made in a great variety of ways, no two being exactly alike. If the principles of a few types are understood, it is easy to understand others.

Buckeye Type. The valve of the Buckeye engine is hollow and of the slide valve type. The cut-off valve is inside. The change of cut-off is due to the alteration of the angular advance, the arrangement of the parts which effect this alteration being shown in Fig. 93. A wheel which contains and supports the various parts of the governor is keyed to the shaft. Two arms, having weights *A A* at the ends, are pivoted to the arms of the wheel *b b*. The ends having the weights are connected to the collar on the loose eccentric *C* by means of rods *B B*.

When the weights move to the position indicated by the dotted lines, the eccentric is turned on the shaft about a quarter of a revolution in the direction in which the engine runs, that is, the eccentric is advanced, or the angular advance is increased; this makes cut-off occur earlier, as shown by the table presented in "Valve Gears." If the engine had a single plain slide valve, the variation of the angular

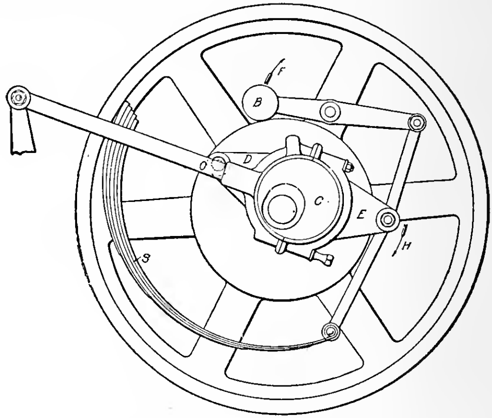


Fig. 94. Diagram Showing Action of Straight-Line Type of Shaft Governor

advance would produce too great a variation of lead; but as this engine has a separate valve for cut-off, admission is not altered by the cut-off valve.

The springs *F F* balance the centrifugal force of the weights; the weights *AA* are varied to suit the speed; and the tension on the springs is altered by means of the screws *c c*. Auxiliary springs are added in order to obtain the exactness of regulation necessary for electric lighting. These springs tend to throw the arms outward, but act only during the inner half of this movement.

Straight-Line Type. Fig. 94 shows the governor of the Straight-Line engine. It has but one ball *B*, which is linked to the spring *S* and to the plate DE, on which is the eccentric C. When the ball flies outward in the direction indicated by the arrow F, the eccentric is shifted about the pivot O, the links moving in the direction of the arrow H. The ball is heavy and at a considerable distance from the center, hence it has a great centrifugal force and the spring must be stiff. The governor of the Buckeye engine alters the cut-off by changing the angular advance, while the Straight-line engine governor changes the travel of the valve. The latter type of valve is very common.

Inertia Form. The well-known Rites inertia governor, Fig. 95, is a form of shaft governor largely used for certain types of engines. This governor regulates the speed of the engine by shifting the eccentric, thus changing the valve travel and increasing or decreasing the angular advance, depending on the speed conditions. It differs in its operation from the centrifugal shaft governor previously considered, in that it makes use of the inertia of two large weights instead of centrifugal force. To understand this action, it first becomes necessary to know something about its construction.

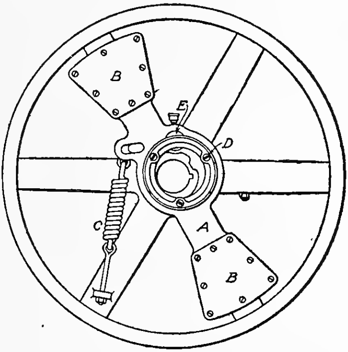


Fig. 95. Diagram of Rites Inertia Type of Shaft Governor

The governor consists essentially of a heavy arm A pivoted at E to the flywheel. This arm carries two heavy weights at B. The eccentric D is fastened to the arm by three countersunk screws, as shown, and moves with reference to the engine shaft whenever the weights B cause the arm A to move about its pivot point. Fastened to the flywheel arm and the governor arm A is the spring C, which brings the arm A back to its normal position when the engine is not operating. This spring also has certain other functions to perform in the operation of the governor.

The action of the governor is such that the valve experiences very much the same movement as in the centrifugal governor. As the engine speeds up, the tendency of the heavy arm *A* is to lag behind the flywheel. This lagging action controls the position of the eccentric so that the valve travel is reduced, thus limiting the amount of steam that enters the cylinders. If, after the engine is operating at a uniform rate of speed, an increase of load suddenly occurs, the motion of the engine shaft and flywheel will be slightly retarded and the engine will commence to "slow down." On account of the energy stored up in the governor arm and the weights *BB*, they will not be so quickly affected, hence the governor will be moving slightly faster than the shaft. As a result the eccentric position with reference to the shaft will be changed, and the valve travel increased, thus permitting more steam to enter the cylinder, increasing the power commensurate with the added load. If for any reason the engine takes a sudden spurt in speed, the tendency of the governor is to fall backward, so to speak; and if the engine is suddenly slowed down for any cause, the tendency of the governor is to plunge forward; hence the valve travel is shortened or lengthened according to which action takes place. This type of governor gives very close regulation when properly constructed.

ERECTION AND OPERATION OF STEAM ENGINES

The limited scope of this work will not permit of an exhaustive study of these two important details—the erection and operation of steam engines; only the general principles governing each will be pointed out.

ERECTION

Foundations. When about to erect an engine the first requisite is the foundation, the character of which will, of course, depend upon the type and the size of the engine. It should be built according to plans submitted by the engine builders, no changes of material consequence being made without the approval of the builders. It should be neither connected with nor in close proximity to any supporting column or columns of the building, as vibrations of the engine will be transmitted to the building which might prove to be disastrous. The foundation should be built upon a solid bottom, but if this is not obtainable at the depth required by the foundation plans, the base of the foundation should be extended in all directions in order that the bearing surface may be increased. In the case of the horizontal engines the nearer the center of gravity of the foundation is placed to the center line of the engine, the more effective will be the foundation. In such cases, therefore, it is preferable to have an extended bearing surface rather than one of considerable depth. The foundation bolts and washers should be carefully located in accordance with the furnished plans. A space of one inch or more should be left around each of the foundation bolts. This may be obtained by using pieces of short iron pipe or old boiler tubes around the bolts, care being taken that they do not extend above the foundation, so as to prevent the proper tightening of the bolts after the engine is placed in position. After the engine is properly set, the space left around the foundation bolts should be filled with the best cement mortar, so as to insure their permanency. The foundation should be a solid one and built of brick, stone, or concrete.

Brick. When brick is used, a hollow square effect may be constructed and the open space filled with a mixture of concrete, consisting of one part cement and three parts sand and gravel.

Concrete. When making a concrete foundation, suitable forms must first be constructed to receive the concrete. Crushed stone or clean gravel or both may be used, care being taken to wash the gravel free of all clay. A good mixture for ordinary foundations is one having the proportions: $1:2\frac{1}{2}:5$. That is, 1 barrel, or 4 bags, cement, $2\frac{1}{2}$ barrels, or 9.5 cubic feet, of sand, and 5 barrels, or 19 cubic feet, of gravel or stone. If the foundation is to be waterproof, careful consideration must be given to the proportioning of the mixture. If the foundation covers considerable area and is not very deep, the mixture should be richer in cement; if, however, the foundation is very deep, a poorer mixture may be used at the bottom and a richer one near the top.

The cement, gravel or stone, and sand should first be thoroughly mixed in the dry state and the water added while the mixing process continues until the mass is well mixed and thoroughly wet. After the mixing is complete, the concrete should be laid in layers from 6 to 9 inches deep and well rammed until solid. The ramming of the concrete is an absolute necessity in order that a solid foundation may be secured.

When the foundation has been completed in accordance with the furnished plans, sufficient time must elapse before any machinery is placed thereon in order to insure a proper setting of the cement. When the concrete has set sufficiently, it should be inspected to see that no omissions or errors have been made, after which the engine may be unpacked and prepared for setting. If the foundation is a large one, an inspector should be on hand at all times to follow the work and see that no errors are made.

Setting the Engine. Upon the accuracy and thoroughness of the setting of the engine, in a large measure depends its successful operation as to smoothness and efficiency of running. In this process there are a great many things to be considered. First, the base and sub-base must be carefully cleaned and set in position. Next, the crank shaft, cylinders, piston, crosshead valves, and other details must be carefully placed in position and alignment made according to the plans of the builders. As all of these details require skill, an inexperienced person should not attempt the setting up of an engine. It is always preferable, when possible, to obtain an experienced man from the engine builders.

Installation of Attachments. In addition to the erection and setting of the engine proper there are various attachments and auxiliaries that require care and skill in their proper installation. The steam and exhaust piping as well as the cylinder drainage should be carefully attended to. The piping should be of ample size, all bends should be easy, and gate valves should be used whenever possible. The piping should have a gradual fall from the boiler to the engine, at or near which should be placed a separator.

Separator. The separator should be of approved design, and care must be taken to carefully provide for drainage in order to insure the removal of the water, otherwise the separator might form a reservoir for water and thus endanger the engine more with its use than without. In addition to being a safeguard against water hammer, when properly attached, the separator also improves the steam economy of the engine, since it removes the most of the entrained moisture which is carried from the boiler through the steam pipes.

Exhaust Pipes. The exhaust pipes should be of ample area to take care of all exhaust steam, and safeguards should be used to insure no backing up of the condensed exhaust into the cylinders. To this end, sharp bends should be avoided and gate valves should be used if valves are necessary, as by their use the area of the pipe is less reduced than by other forms of valves. Check valves should be avoided whenever possible.

Cylinder Drains. The cylinder drains should be of sufficient area to care for all condensed steam in the cylinders and so attached to the cylinders and the exhaust pipe or receiver that no pockets will be formed for the accumulation of water. In the case of compound engines the cylinder drains of the high and the low pressure cylinders should not be connected together, but separately connected to the exhaust or other main drain. In condensing engines the cylinder drains should always go into the exhaust drain if it is low enough to admit of proper drainage.

OPERATION

Let us now turn our attention to the operation and management of an engine. It should be borne in mind that many suggestions as to the proper alignment and adjustment of bearings, the adjustment of valves, and the consideration given lubrication will be applicable both to the first setting up of the engine and also to the daily operation afterwards.

Competent Engineer a Requisite. The operation of an engine should be committed to a careful, skillful, and reliable man. This is especially true in the case of modern well-equipped plants which represent quite an outlay of capital. In many of the smaller plants, however, not much attention is given to the matter and we find, as a result, men holding positions as operators who know very little about their business. Under such conditions the plants are seldom operated efficiently.

As a suggestion of some of the duties of a man in charge of a modern plant, which also suggest the amount of judgment and experience required, the following general instructions are presented.

Care of Bearing Caps. The caps on the main bearings should always have sufficient liners underneath to enable the nuts on the bearing studs to draw the cap down tightly upon them and not pinch the shaft, which should be free to revolve in its bearings without unnecessary play.

The caps should be removed occasionally as conditions demand in order to clean out the oil grooves which are chipped in the babbitt metal, as the passages may become clogged with dirt or other foreign matter.

Adjustment of Connecting Rod Box. In adjusting the connecting rod box at the crank pin end, the same general rules should be observed regarding the liners under the cap—the large nuts drawn solidly upon it, the small nuts firmly jammed and the cotter pins placed in position. The adjustment of the box should then be tested with a lever about 12 inches in length, the adjustment being so made that with a lever of this length the operator can easily move the end of the connecting rod sufficiently to take up the side play between the flanges on the crank pin and the end of the box. The adjustment should never be made so close that this side movement can not be observed.

The adjustment of the connecting rod box at the crosshead pin should be made by placing the crank on the center nearest the cylinder; then with a wrench provided for that purpose, slack off both wedge screws at the upper and lower sides of the connecting rod, and draw the wedge up until it is solid against the box; then slack off one screw about a sixth of a turn, and draw up the other so as to firmly lock the wedge.

Lining Up Crosshead. The crosshead should be lined up between the guides, while disconnected from the connecting rod. When in this condition the crosshead should be so lined that it can be easily pulled from one end of the guides to the other with a short lever.

The crosshead should never be run very close, and should always be free enough to allow long and continuous runs without heating the guides to the degree that they would be uncomfortably warm to the touch.

When making any adjustments of the crosshead, the operator should assure himself that the lock nut which prevents the piston rod turning in the boss of the crosshead is securely placed.

Adjusting Eccentric Strap. The eccentric strap adjustment is made by liners placed between the halves of the strap and double nutted bolts. When adjustment is necessary, the other end of the eccentric rod should be disconnected and, after drawing up the strap bolts, it should be tested by giving the strap a half revolution about the eccentric. If it is found that the friction between the strap and the eccentric is sufficient to support the weight of the rod, the bolts should be loosened and liners replaced until the strap moves freely without lost motion. The double nuts should then be locked and the cotter pins replaced in the ends of the bolts.

Governor. The governor should be adjusted to meet the different conditions of speed and steam pressure and the degree of regulation required. As governors differ so much in design and detail of construction, it is not possible to give any general rule for their adjustment. The operator, if desired, can usually obtain instructions from the engine builder for the particular type of governor in question.

Valve Setting. As a discussion of the setting of the valves and their adjustment for wear will be found given in "Valve Gears," no consideration of the subject will be presented here.

Lubrication. The lubrication of a steam engine, and especially of high speed engines, is a very important consideration with both the designer and the operator, for it is upon proper lubrication that they must largely depend for a constant and satisfactory operation. The designer must, therefore, provide ample and efficient facilities for lubricating the bearings, cylinders, and valves, whereas the operator must use discretion in selecting his lubricants and the amount to use after selection has been made.

Choice of Oils. It might be said that only the best oils should be used. Cheap oils are usually considered expensive at any cost and should be avoided as they promote excessive wearing of the parts—causing noisy operation—and may cause serious cutting of the cylinders. There are two general classes of liquid lubricants now in the market, namely, mineral and animal oils. There is also a compounded lubricant which is made up of about 5 to 15 per cent of animal matter and the balance of mineral oil. This compound makes a very efficient lubricant for some classes of service, as it withstands the action of the condensation and adheres to the surface of the cylinders, thus giving better results than larger quantities of mineral oil.

In plants where open heaters are used and where the exhaust steam is condensed and used for boiler feed water, the compounded oils can not be used, on account of the danger of the animal matter getting into the boilers and causing considerable trouble. In such cases mineral oil must be used, although it may require considerable more mineral than compounded oil to accomplish the lubrication.

Solid Lubricants. Several solid lubricants are used, such as graphite, metalline, soapstone, and fiber graphite.

Graphite when mixed with certain oils is well adapted for heavy pressures. It is especially good for heavy pressures and low velocities. Under conditions which require a large amount of cylinder oil, a small amount of crystal or flake graphite may be used with good results. Care must be exercised, however, if the exhaust steam is used for feed water, as the graphite may get into the boilers and cause inconvenience and perhaps serious trouble.

Metalline is a solid compound containing graphite. It is made in the form of solid cylinders, which are fitted to the holes drilled into the surface of the bearing. When a bearing is thus fitted, no other lubricant is necessary.

Soapstone in the form of powder and mixed with oil or fat is sometimes used as a lubricant. Soap mixed with graphite or soap-stone is often used where wood is in contact with wood or iron.

A preparation called *fiber graphite* is used for self-lubricating bearings. It is made of finely divided graphite mixed with fibers of wood. It is pressed in molds and afterwards fitted to bearings.

For great pressure at slow speed, graphite, lard, tallow, and other solid lubricants are suitable. If the pressure is great and the speed high, castor, sperm, and heavy mineral oils are used.

For low pressure and high speed, olive, sperm, rape, and refined petroleum give very satisfactory results.

In ordinary machinery, heavy mineral and vegetable oils and lard oil are good. The relative value of various lubricants depends upon the prevailing conditions. Oil that is suitable for one place might not flow freely enough for another.

The quality of oil is of great importance. In many branches of industry it is imperative that the machinery run as perfectly as possible. On this account and because of the high cost of machinery, only first class oil should be used. The cylinder oil especially should be high grade, because the valves, piston, and piston rods are the most delicate parts of the engine.

Qualities of a Good Lubricant. From the foregoing brief discussion of lubricants it will be evident that they must possess certain qualities which may be enumerated as follows:

The lubricant must be sufficiently fluid, so that it will not in itself make the bearing run hard.

It must not be too fluid or it will be squeezed out from between the bearing surfaces. If this happens, the bearing will immediately heat and begin to cut. The heating will tighten the bearing and increase the pressure and the cutting.

It must not gum or dry when exposed to the air.

It must not be easily decomposed by the heat generated. If it should be decomposed, it might form substances which would be injurious to the bearings.

It must not take fire easily.

It must contain no acid and should form no acid in decomposing, as acids corrode the bearings.

Both mineral and animal oils are used as lubricants. Formerly animal oils were used entirely, but they were likely to decompose at high temperatures and form acids. It is important in using high pressure steam to have "high test oils," that is, oils which will not decompose or volatilize at the temperature of the steam. It was the difficulty of getting such oils which made great trouble when superheated steam was first used. Mineral oils will stand high temperatures very readily, and even if they do decompose, they form no acids.

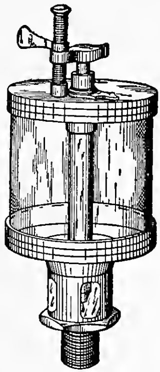


Fig. 96. Standard Type of Simple Oil Cup

Common Oilers. Engines are lubricated by means of oil cups and wipers placed on the bearings wherever required. They are made in many forms. Formerly, the oil cup was made with a tube extending through the oil. A piece of lamp wick or worsted leads from the oil in the cup to the tube. Capillary attraction causes the oil to flow continuously and drip down the tube. When not in use, the lamp wick should be withdrawn. This type of oil cup is now seldom used.

The oil cup shown in Fig. 96 is simple and economical. The opening of the valve is regulated by an adjustable stop. The oil may be seen as it flows drop by drop. The cylindrical portion is made of glass, so that the operator can see how much oil there is in the cup without opening it.

A form of wiper crank pin oiler is shown in Fig. 97. The oil cup is attached to a bracket. The oil drops from the cup into the sheet of wicking or wire cloth and is removed at each revolution of the crank pin by means of the cup which is attached to the end of the connecting rod. This form of oiler works very satisfactorily at slow speeds.

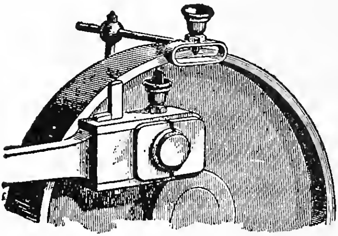


Fig. 97. Form of Wiper Crank Pin Oiler

Centrifugal Oilers. Fig. 98 shows a centrifugal oiling device which operates very satisfactorily at all speeds. The oil flows from the oil cup through the tube to the small hole in the crank pin by centrifugal force. It reaches the bearing surface by means of another small hole.

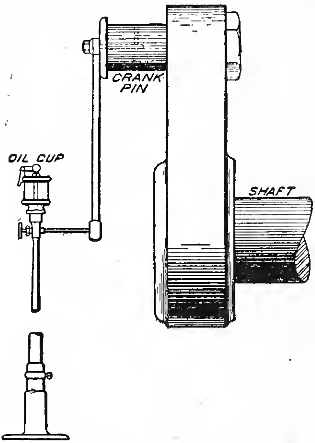


Fig. 98. Centrifugal Oiler

Cylinder Lubrication. In oiling the valve chest and the cylinder, the lubricant must be introduced against the pressure of the steam. This may be done in several ways, in each of which it is introduced into the steam before it reaches the valve chest and is carried by the steam to the surfaces to be lubricated.

By Oil Pumps. The oil may be forced into the steam pipe by a small hand pump or, in large engines, by an attachment from the engine itself. The supply of oil is, of course, intermittent if the pump is driven by hand, but continuous and economical if driven by the engine.

By Sight-Feed Lubricators. The most common device for feeding oil to the cylinder is that which introduces the oil drop by drop into the steam when it is in the steam pipe or steam chest. The oil becomes vaporized and lubricates all the internal surfaces of the engine.

Fig. 99 shows the section of a sight-feed lubricator, which must be placed on the steam supply pipe in a vertical position above the throttle. The reservoir *O* is filled with oil. The pipe *B*, which connects with the steam pipe, is partly filled with condensed steam which flows down the small curved pipe *E* to the bottom of the chamber *O*. A small portion of the oil is thus displaced and flows from the top of the reservoir *O* down the tube *F* by the regulating valve *D*, and up through the glass tube *S*, which is filled with water. It enters the main steam pipe through the connection *A*. The gauge glass *G* indicates the height of water in the chamber *O*. To fill the lubricator, close the regulating valve *D* and the valve in pipe *B*; the oil chamber can thus be drained through the cock *C*, and filled. If the glass *S* becomes clogged, it may be cleaned by closing valve *D* and opening the small valve *H*. This will allow the steam to blow through the glass. After cleaning close valve *H* and allow glass *S* to become filled with water before opening the feed valve. The amount of oil fed to the cylinder can be regulated by opening the valve *D* the proper amount. The exact quantity of oil necessary for the engine is not easily determined. For ordinary sizes it is from one to four drops per minute, depending on the conditions.

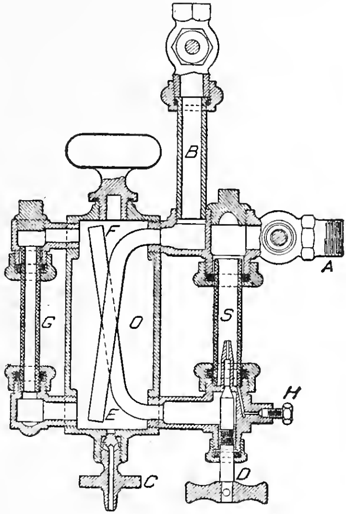


Fig. 99. Section of Sight-Feed Lubricator

Instructions for Proper Lubrication. In slow speed engines it is not a difficult matter to attend to the oiling; all the parts are moving slowly and can be readily examined and oiled. Many high speed engines run so fast that it is impossible to examine the various parts, and special means must be provided for lubrication. It is especially important in high speed engines that there be no heating.

In order to avoid the danger of neglecting to oil a bearing of a high speed engine, it is customary to have all the bearings oiled from one central source. All the oil is supplied to one reservoir, from which pipes lead to all bearings. If this is not done, large oil cups are used so that oiling need not be done so frequently.

In some high speed engines the moving parts are enclosed and the crank runs in a bath of oil. This secures certain oiling and is very effective. All the bearings may be inside this crank case, so that all are oiled in this way. It is thus impossible for a careless operator to overlook one point and so endanger the whole engine.

Large steam engines and turbines are now usually furnished by the manufacturers equipped with some form of oiling system. The large turbines are generally equipped with a continuous oiling system, but in these systems no provision is made for the scientific removal of water and foreign matter from the oil.

Complete Lubrication Systems. We may classify steam turbine oiling and filtering systems as follows:

(1) *Continuous Circulating Systems.* Those in which all the oil used on the bearings is continuously passed through a filter, Fig. 100. Such systems are successful only on small machines.

(2) *Partial Filtration Systems.* Those in which part of the dirtiest oil is continually removed, passed through a filter and returned.

(3) *Batch Filtration Systems.* Those in which all the oil contained in the system is removed and purified in a separate filter, the system being supplied with a fresh batch of clean oil to permit it to operate while the dirty oil is being purified.

Operation of Typical System. The method of operation of the lubricating system of the modern steam turbine is illustrated in Fig. 101 Oil from the different bearings flows by gravity into oil reservoir *B*. A small rotary pump *A*, usually driven from the governor shaft, pumps the oil through the cooler *C*, thence through pipes *D* to the various bearings. The relief valve *L* by-passes any excess oil back into the storage reservoir *B*. In some systems the oil is pumped into an overhead reservoir from which it is allowed to flow by gravity to the bearings. In the illustration shown, the turbine has four main bearings, *E*, *F*, *G*, and *H*. These bearings are made hollow and are water cooled. The oil is fed into the

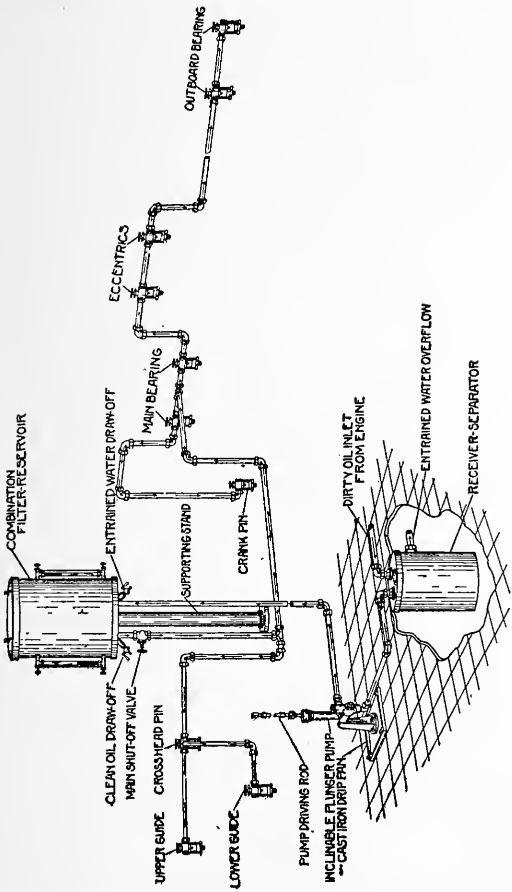


Fig. 100. Diagrammatic Drawing of Continuous Oiling and Filtering System for Simple Engine Having Ten Points of Lubrication

bearings at the top and flows out at each end, where it drops down into chambers in the turbine casing and is collected by the pipe J and returned to the reservoir B. A screen K is provided in the reservoir which removes any large particles of foreign matter.

In such a system there are several places where water finds its way into the oil, the principal place being at the packing gland M at the high-pressure end of the turbine casing. Different manufacturers use different forms of packings at this point, but in practically every case some steam escapes which, on coming in contact with the water-cooled bearing at E, condenses and mixes with the oil. Again, when turbines are operated with a back pressure, there will be a leakage of condensed steam into the oil at the packing gland N, at the exhaust end of the casing. Also the oil cooler C and the water-cooled bearings may develop small leaks.

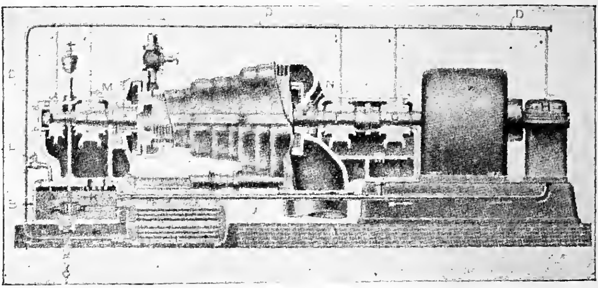


Fig. 101 Section of Modern Steam Turbine, Showing Scheme of Operation of Self-Contained Oil Circulating System

As a means of getting rid of the water which gets into the system, the cock P is provided at the bottom of the reservoir B, from which water collecting at this point can be drawn off. The oil usually passes through this tank so rapidly that there is not sufficient time for complete separation of the entrained water

What has been said concerning the lubrication of large steam turbines applies equally as well to the lubrication of large steam engines.

In Figs. 102 and 103 is shown the plan of operation of a power plant oil filter with the various parts named.

Starting the Engine. Before starting an engine, the oil cups should be started feeding, grease cups screwed down, and the governor and other parts of the valve gear oiled. The cylinder lubricator should be started before the engine so that the oil passages will contain oil. The cylinder drain cocks should be open so that any condensed steam in the cylinder will be removed without injury to the cylinder. These precautions having been observed,

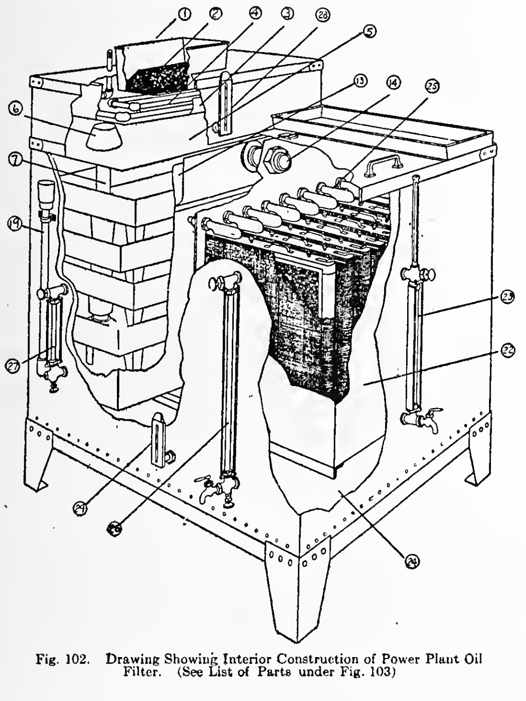


Fig. 102. Drawing Showing Interior Construction of Power Plant Oil Filter. (See List of Parts under Fig. 103)

the throttle may be opened slowly and the engine started and gradually brought to the required speed.

After starting the engine, notice should be taken of the governor and all the lubricating apparatus to see that each is properly performing its function.

When the engine is to operate condensing, the condenser should be started first, if it is in such a position that the water in the exhaust can drain into it. If the condenser is above the engine and no means are provided for removing the water, the engine should be started non-condensing. When a jet condenser is used, the quantity of injection water should be increased as the load is increased; the amount being determined by the conditions of the vacuum and temperature of the discharge water, which

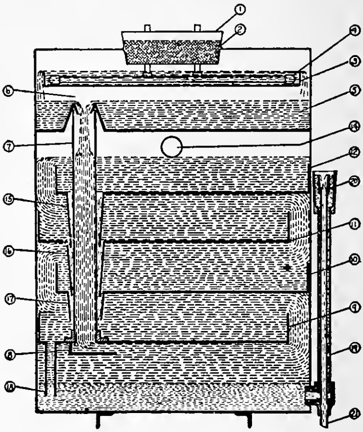


Fig. 103. Drawing Showing Section Through Precipitation Compartment and Automatic Water Ejector of Power Plant Oil Filter. (See List of Parts)

NAMES OF PARTS OF FIGS. 102 AND 103

1—Strainer box for receiving dirty oil; *2*—Removable strainer; *3*—Heating tray; *4*—Steam heating coil; *5*—Dirty oil receiving compartment; *6*—Funnel; *7*—Conductor pipe; *8*—Baffle plate; *9*, *10*, *11*, *12*—Trays; 13—Oil skimmer maintaining constant oil level in precipitation compartment; *14*— Pipe for conducting oil from precipitation compartment to filtering compartment; *15*, *16*, *17*, *18*—Funnels for conducting water direct to bottom of precipitation chamber; *19*—Automatic visible adjustable water overflow pipe; *20*—Overflow funnel; *21*—Water discharge pipe; *22*—Filtering compartment containing non-collapsible cloth-covered filtering units; *23*—Oil gauge for filtering chamber; *24*—Clean oil compartment; *25*—Filtering unit nozzle conducting filtered oil into the clean oil compartment; *26*—Oil gauge for clean oil compartment; *27*—Water gauge for precipitation compartment; *28*—Thermometer showing temperature of oil entering the precipitation compartment; *29*—Thermometer showing temperature of clean oil compartment.

should be from 100° to 110° F. If the water is colder than this, it would denote that more injected water is being used than is required.

The foregoing suggestions and indicated precautions are only a few of the more important things that will arise in the course of the erection, setting, and operation of an engine. The one performing these various duties must at all times exercise good judgment and act according to what his past experiences and that of others have taught under similar circumstances.

ENGINE SPECIFICATIONS

Selecting an Engine. The engineer who has the responsibility of selecting an engine for a given class of service has no small task to perform, if he carefully analyzes all the factors entering into the problem. If the installation contemplated is to be an extensive or expensive one, expert advice should be solicited. Since this is not always to be had, a few suggestions will be given as to how best to proceed when one has to specify an engine for a given service. Consider for the time being that an expert consulting engineer is not available and a rather inexperienced person, or non-technical man, who knows little about the theoretical questions that should be given consideration, has to select the engine. In this case the most satisfactory procedure to follow would be to go to some reliable engine builder and ask him to build or specify an engine that would perform the service required. Having only one builder intrusted, the item of expense would not be chief in his consideration since there would be no competition, therefore the builder would build or specify the best engine possible for the service. If the funds available are limited or must be closely conserved, the intended purchaser may state the limits of cost and then require the builder to come within those limits. It would also be wise on the part of the purchaser to require a guarantee as to the performance of the engine and its maintenance cost for a given period of one year or more.

Drawing Up Specifications. If the purchaser is a competent engineer or he has in his employ such a person, a complete set of specifications may be drawn up and submitted to several engine builders for competitive bids. The specifications submitted should cover in detail the service for which the engine is to be used, the speed at which it is to operate, the type of valves and valve gear desired, the per cent of variation permissible in its governing, and many other items as to the design and detail of construction. Most specifications also specify within what limits the engine must operate, as to the amount of steam used per indicated horsepower per hour, and the range of mechanical efficiency that must be attained. A provision should be made in the contract as to the conditions under which the acceptance test will be made and by whom.

The form of specification usually submitted by the builders and which in general will be like those written by an engineer when requesting bids, is submitted herewith. This may be taken as a typical specification, the items being changed to meet different conditions of service as the particular case demands.

SPECIFICATIONS OF A VERTICAL CROSS-COMPOUND, SIDE-CRANK, ENGINE, ARRANGED FOR 1000-K.W. DIRECT CONNECTED GENERATOR, 60 CYCLE ALTERNATOR

Size, Power, and Dimensions

Diameter of high pressure cylinder, 27 inches.
Diameter of low pressure cylinder, 54 inches.
Stroke, 42 inches.
Revolutions per minute, 120.
Initial steam pressure, 125 pounds, 26 inches vacuum, condensing.
Rated load in indicated horsepower, 1,520; cut-off, 26/100.
At $\frac{1}{2}$ cut-off, indicated horsepower, 2,100; maximum cut-off, 7/10.
Estimated total weight of engine, 346,000 pounds.
Weight of wheel, 92,000 pounds. Diameter, 16 feet. Face, —— inches.
Diameter of bearings, 19 inches. Length, 35 inches.
Diameter of shaft between bearings, 22 inches.
Diameter of crank pin, 9 inches. Length, 8 inches.
Diameter of crosshead pin, 8 inches. Length, 8 inches.
Bearing surface of crosshead, 17 inches by 20 inches.
Diameter of piston rod, 5 inches.
Diameter of throttle valve, 12 inches.
Diameter of exhaust opening, 22 inches.

Workmanship and Materials

The workmanship, finish, fitting, and materials will be first-class in every particular. All forgings will be of open-hearth steel or hammered iron, as hereafter specified. All castings subject to wear, such as cylinders, guides, pistons, etc., will be poured from mixtures containing charcoal iron, graded according to the size of casting in order to secure the proper hardness and closeness of grain.

The engine will be made to gauge and interchangeable. This feature will be thoroughly carried out.

Flat surfaces will be scraped to surface plates, and surface and cylindrical grinding will be used where advantageous.

Guarantee

We guarantee the workmanship and materials in the engine to be first-class and in fulfillment of our guarantee we will give a duplicate to take the place of any part that may prove defective in material, workmanship, or design within one year after the engine is started.

We guarantee the engine to regulate from no load to full rated load within 2 per cent variation of speed.

We guarantee the engine to run in a smooth and proper manner without undue heating or vibration.

CYLINDERS

The cylinders and steam chests will be neatly covered with sheet iron lagging, enclosing a thick layer of the best quality of asbestos or magnesia fiber. The cylinder and steam chest covers will also be provided at each end with thin iron castings or covers. The cylinders will be provided at each end with a patent combination relief-valve and drip-cock of large diameter, adjustable to open automatically at any pressure desired. Being operated by hand as drip-cocks, these will not stick or become inoperative from disuse, but will relieve dangerous pressure from water or other causes.

JACKETS AND RECEIVER

The high pressure cylinder will be steam-jacketed and there will be a receiver of large capacity between cylinders.

The receiver will be filled with seamless brass heating coils containing steam at boiler pressure. The high pressure jacket and these coils should be piped in series, so that steam will pass through in the order named, and since the steam in the low pressure coils is hotter than the receiver steam, the latter will be considerably superheated upon entering the low pressure cylinder, and enough of the former will be condensed in the coils to cause brisk circulation in the high pressure cylinder jacket which is necessary to its efficiency. It is the aim of this arrangement to keep the steam dry throughout its course through the engine without the loss of any portion of heat of the jacket to the exhaust steam. The water condensed in jackets and in the coils should be returned to the boiler.

VALVES

Both cylinders will be four-ported and provided with valves of the flat gridiron type of our standard form.

The valves slide crosswise of the cylinder upon gridiron seats, which are separate and removable from the cylinder itself. Since the valves are of the gridiron type, a very small stroke is necessary to give full opening, and they move with an intermittent motion, standing still when closed, and only require power to operate when open and relieved of steam pressure. The clearance is reduced to about one-half of that necessary with valves of the Corliss type.

These valves possess the following advantages:

They give rapid opening of port with the least amount of wear and power required to operate.

The clearance space is reduced to a minimum.

They will not stick when the engine is started, and are easy to keep lubricated.

They wipe over and wear evenly, are unbalanced, and hence *will be tight* when old as well as when new.

VALVE GEAR

The main valves will be driven by a fixed eccentric controlling the admission of the steam and the opening and the closing of the exhaust. The cutting off of the steam will be effected by the cut-off valves which are controlled by the governor.

The valve gear is positive, composed of simple levers and links, and the cut-off can take place at any point between zero and the maximum cut-off. The cut-off, except at light loads, occurs when the main and cut-off valves are moving in opposite directions, and the cut-off is as sharp as with a releasing type of valve gear notwithstanding the short stroke used.

The cut-off is varied simultaneously upon all the cylinders in such a manner that the work done in each is approximately equal, as is also the drop in temperature of steam in each. This adds to smooth running and gives best distribution of steam for economy at all cut-offs under variable loads.

The valve gear will be constructed in the most substantial and durable manner, and in such a way as to equalize the cut-off at both ends of the cylinders for all cut-offs. Rock-shafts, pins, and links will be made of open-hearth steel. Connecting links will be fitted with bronze ends having quick taper key adjustment. The eccentric straps will be lined with babbitt hammered in and bored out. The rock-shaft bearing will be babbitted and adjustable.

Governors

The governor will be situated on the main shaft of the engine. A change in position of the centrifugal weights revolves the eccentric controlling the position and motion of the cut-off valves around the shaft and varies the point of cut-off.

All the bearing pins in the governor will be made of tool steel hardened and ground, turning in bearings bushed with phosphor bronze. The centrifugal force of each governor weight is resisted by a plate spring through a pin having hardened steel points resting in phosphor bronze cups, one at the end of the spring and the other at the center of gravity of the governor weight. The centrifugal force of the governor weights is thus opposed in a direct and frictionless manner without causing pressure or friction on the pins upon which the governor weights swing. This governor will regulate the speed of the engine with a closeness and certainty impossible with a fly-ball governor, and its action is unaffected by wide and sudden fluctuations of load. The governor will control both cut-off eccentrics.

Pistons, Piston Rods, and Stuffing Boxes

The pistons will be cored out and provided with internal ribbing, making them very light and strong. They will be secured to the piston rod by being forced upon a taper, with shoulder beyond, and by a nut, with a simple but efficient locking device. The pistons will be provided with cast-iron packing rings.

The piston rods will be of open-hearth steel running through deep stuffing boxes and babbitted glands. The rods will not touch the heads—which will be bored large—bronze rings fitting the rods in the bottom of each stuffing box and preventing escape of packing to the interior of the cylinder.

Low pressure piston will be of steel.

Framing

This will consist, for each cylinder, of a deep and massive base containing the main bearings. On the back of each base will stand a very heavy rectangular column, as shown in the blue print, securely bolted to a heavy frame head. In front the frame heads will be connected to the bases by forged steel columns bolted by flanges forged solid with the columns. The rear column will support the cylinders when the forged columns in front are removed, facilitating the placing of shaft and other parts.

GUIDES, CROSSHEADS, AND CROSSHEAD PINS

The guides will be separate from the frame and adjustable for wear with an oil dish at the bottom which, together with a thin brass fringe upon the bottom of the crosshead, forms an efficient self-oiling device.

The crossheads will be of open-hearth steel fitted with babbitted cast-iron shoes.

The crosshead pins will be of open-hearth steel flattened on two sides to prevent wearing oval.

CONNECTING RODS AND BOXES

The connecting rod will be of forged steel, provided with gib and key ends. The straps will be provided with pinching bolts which will prevent spreading. Both crank and crosshead pin boxes will be lined with babbitt hammered in and bored out.

The body of the connecting rod will be made of larger section than the piston rod, being designed properly for the added strain due to its length and angular motion.

SHAFT, CRANK PIN, AND DISK

The shaft will be piled and faggoted hammered iron forging.

The crank disk will be made with counterbalance, of a mixture containing charcoal iron. The crank pin will be made of forged steel. The shaft and crank pin will be forced into the disk by hydraulic pressure and the disk will also be keyed securely to the shaft.

MAIN BEARINGS AND REMOVABLE SHELLS

The main bearings will be fitted with cylindrical shells, lined with babbitt, hammered in and bored out. These shells can easily be taken out by removing the cap and simply jacking up the shaft sufficiently to take the weight off the bearings, when they can be revolved around the shaft and taken out without disturbing any other parts of the engine. The shells are made hollow for water circulation. This is not intended to be used ordinarily, but in case dirt or other unusual conditions should cause the bearing to heat, it often enables the engine to complete its run without stopping.

The main bearings will be provided with a self-oiling device which will keep them flooded with oil.

OIL FEED SYSTEM

The feed will be positive and adjustable and the system will be closed, so that there will be little waste and deterioration of oil. Rings at the ends of the bearings will throw off escaping oil into close-fitting shields with suitable drain pipes leading to a large settling reservoir beneath. A small pump driven from the valve gear will deliver the oil to a feed tank at each bearing. This tank will be provided with an adjustable feed outlet pipe leading to the bearings, and with a gauge-glass and by-pass overflow, and can be filled by hand and used as an ordinary oil cup if it is desired to cut off the automatic supply while the engine is running.

FLYWHEEL

The wheel will be cast in halves and will be bolted together at the hub with reamed bolts carefully fitted in holes drilled from the solid, and the parts will be planed where they join. Steel arrow head links will be used at the rim. The wheel will be carefully designed throughout in order to have a large factor of safety, and both edges and face of rim will be turned true.

PLATFORMS

Platforms convenient for handling and operating the engine will be provided as shown in print. These can be arranged to suit the location of the engine and will be made stiff to avoid vibration. The hand railings will be of seamless brass tubing, fitted into brass caps or iron posts. The platform plates will be diamond figured, planed where they join together and neatly fitted. Stairs will be made of channel iron, with cast-iron diamond threads.

FIXTURES

The following fixtures will be provided: throttle valve; indicator motion; complete outfit of sight-feed cylinder lubricators; glass body oil pumps; grease cups for valve gear; centrifugal crank pin oilers; reservoirs with sight-feed outlets; oil pipes and wipers for oiling the main parts of the engine conveniently and continuously; relief valves for each end of the cylinders; drip-cocks; wrenches, foundation bolts; and foundation plans.

Contract. After the engine has been selected and the builders determined, a written contract should be entered into in order to make it a legal document. A contract, according to Blackstone, is an agreement upon sufficient consideration to do or not to do a particular thing. In the case of the purchasing of an engine, the builder agrees to build, erect, and put into operation an engine in accordance with the specifications and drawings submitted, which items become a legal portion of the contract. The purchaser may also require that the engine be ready for operation in a given time and that it must also come up to certain requirements in its performance, as previously mentioned. In consideration of the foregoing, the purchaser agrees to pay the builders a specified sum of money, either in one payment or more as determined by them. The wording and statement of the contract should be carefully prepared, in order to avoid any possible misinterpretation of any of its provisions.

COST OF ENGINES AND OF THEIR OPERATION

The question of the cost of an engine and of its erection and operation is indeed a very vital one. This cost can not be classified in a brief way, since there are so many contributing factors that differ widely in different localities. For example, no well-defined indication of the cost of operation can be given, and the cost of labor and material are fluctuating items of expense; therefore, the cost of the engine can not be stated definitely, since in a brief interval of time it may be considerably more or less. Many articles appear

TABLE III

Price of Single Cylinder Corliss Engines, Set and Erected

Size of Cylinder, Inches	Horse-power	Cost of Engine	Cost of Foundation	Cost of Erecting	Cost of Piping	Total Cost
16×36	125	$1950	$325	$210	$180	$2665
18×36	155	2150	375	240	200	2965
18×48	200	2600	425	260	220	3505
20×48	230	2850	525	275	250	3900
22×42	250	3000	550	300	310	4160
24×48	320	4000	700	375	390	5465
28×48	425	5150	900	500	800	7650
30×48	490	5800	1200	600	1070	8670

from time to time in the leading engineering papers which give valuable information upon such matters and usually this information is correct since it is given currently with the ascertained cost of various items. It is, therefore, suggested that if the latest and perhaps most authentic information is desired upon these items of expense that such articles as appear in the papers mentioned should be consulted.

Engine Costs. As an indication of what such expense will be Tables III and IV, as devised by Dean C. H. Benjamin, are given.

TABLE IV

Cost of High Speed, Single Cylinder Engines

Horsepower	Size of Cylinders Inches	Steam Pressure Pounds Per Square Inch	R. P. M.	Cost Del. F. O. B.	Cost of Sub-Base	Cost of Engine Foundation	Cost of Superintendence—Labor	Cost of Handling	Total Cost (Engine set up on Foundation)
50	9×10	100	300	$695	$45	$65	$70	$10	$885
75	10×12	100	300	890	50	75	70	15	1100
100	12×12	100	290	1085	50	80	70	15	1300
125	13×14	100	275	1260	70	95	70	17	1512
150	15×14	100	245	1595	80	110	75	20	1880
	14×16								
200	18×16	100	225	2010	90	140	85	25	2350
250	19×18	100	200	2800	250	200	100	35	3385

Relative Cost of Operation Items. The cost of the operation of a steam plant is properly made up of several items, viz, rent or

TABLE V

Cost of Installation and Operation for One Year

Kind of Engine	Total Cost Engines and Boilers	Annual Cost of Both Engine and Boilers, Depreciation and Interest	Total Tons Coal Per Year of 3000 Hours	Lubricants	Labor, Engine and Boilers	Cost of Power, Coal at $2 Per Ton, 1 Year
Simple Slide Valve Non-condensing	$29.75	$4.03	6750	$1.02	$5.00	$23.55
Compound Slide Valve Non-condensing	31.50	4.38	5660	1.25	4.50	21.45
Compound Slide Valve Condensing	29.80	4.26	4050	1.25	3.80	17.41
Simple Corliss Non-condensing	32.25	3.84	6075	1.00	4.70	21.00
Compound Corliss Condensing	30.87	3.76	3375	1.25	3.50	16.25
Triple Corliss Condensing	34.25	4.28	3110	1.50	4.00	16.00

interest on real estate; interest on investment; maintenance, etc., of equipment; fuel; water; supplies; and attendance.

The relative value of these various items for a large central station lighting plant was given in the *Engineering Magazine*, May, 1905. Taking the total cost of maintaining the station as 100 per cent, the following were the average costs of the various items: Fuel 52.5%; wages 26.4%; water 2.2%; oil and waste 1.8%; rent 4.35%; station repairs 2.2%; steam repairs 5.45%; electric repairs 5.1%.

Annual Operation Expenses. Professor Carpenter in the *Economist* summarizes the cost of installation and the operation of an entire plant for one year of 3,000 hours as given in Table V. A coal consumption of 4.5 pounds per boiler horsepower per hour is assumed and the cost given per engine horsepower is for a 1,000 horsepower engine.

An illustration involving the items given in Table V will serve to make it clearer. The case of a simple slide valve non-condensing plant will be considered.

Cost of engines and boilers at $29.75 per horsepower = $29,750.
Annual cost of depreciation and interest at $4.03 per h.p. = $4,030.
Annual cost of coal at $2.00 per ton = 6750 × 2 = $13,500.
Annual cost of lubricants at $1.02 per horsepower = $1.02 × 100 = $1,020.
Annual cost of labor at $5.00 per horsepower = $5.00 × 1000 = $5,000.

Annual cost for the last four items = $23,550 or $23.55 per h.p.

The foregoing tables will serve to give some idea of the cost of engines, also of the cost of operation of a steam plant, but it must be remembered that the figures given will not be exact for all localities or for all times, due to the changing influences previously mentioned.

ENGINE TESTS

Importance of Tests. It was mentioned in connection with the discussion of specifications and contracts that often a guarantee is given by the builder as to the economical performance of a steam engine, hence it is required that the engine be tested in order to ascertain whether or not it meets the provisions of the guarantee. While this is one reason that may be assigned for testing an engine, yet there are several others of importance. The user from time to time may want to ascertain the condition of the engine as a whole and also the condition of particular features such as the valves, etc. For purely theoretical reasons an engine is often tested in order that an analytical study may be made of its performance under various conditions and in comparison with other engines of different classes. Many such tests have resulted in obtaining data, the facts of which have demonstrated to both the builder and the user possible economies. Because of the information thus obtained, the builder has been enabled to design a better engine, and the user to operate his engine more advantageously. The remarks given will suffice to indicate that the ultimate object of an engine test is the determination of the *economy* with which the engine produces a given amount of *power*. In steam engines the economy, as usually ascertained, relates to the weight of steam consumed, to the quantity of coal required to make the steam, or to the number of heat units supplied. The elementary quantities concerned are accordingly two in number, viz, the amount of steam, fuel, or heat (as the case may be) consumed, and the amount of power developed. How to determine these quantities is the problem.

A. S. M. E. Code. The American Society of Mechanical Engineers (A.S.M.E.) deemed the testing of engines according to some definite and standard method of such importance that a committee was appointed to devise a standard code. This, after much labor and diligent study, was presented to the Society and adopted. The full report appears in Volume 24 (1904), page 713, of the Transactions. Since the report of the first committee, various committees have studied the Code and offered revisions which were ultimately adopted in 1915. In so far as the conditions will permit, this code should be followed. The 1915 Code for Conducting Tests of Reciprocating Steam Engines is too lengthy to give in its entirety but is summarized as follows:

Method of Conducting Steam Engine Tests
Code of 1915

INSTRUCTIONS REGARDING TESTS IN GENERAL

(1) *Object.* Ascertain the specific object of the test, and keep this in view not only in the work of preparation but also during the progress of the test, and do not let it be obscured by devoting too close attention to matters of minor importance. Whatever the object of the test may be, accuracy and reliability must underlie the work from beginning to end.

If questions of fulfillment of contract are involved, there should be a clear understanding between all the parties, preferably in writing, as to the operating conditions which should obtain during the trial, the methods of testing to be followed, corrections to be made in case the conditions actually existing during the test differ from those specified, and as to all other matters about which dispute may arise, unless these are already expressed in the contract itself.

Among the many objects of performance tests, the following may be noted: Determination of capacity and efficiency and how these compare with standard guaranteed results; comparison of different conditions or methods of operation; determination of the cause of either inferior or superior results; comparison of different kinds of fuel; and determination of the effect of changes of design or proportion upon capacity or efficiency, etc.

(2) *Preparations.* Measure the dimensions of the principal parts of the apparatus to be tested, so far as they bear on the objects in view, or determine these from correct working drawings. Notice the general features of the apparatus, both exterior and interior, and make sketches, if needed, to show unusual points of design.

The dimensions of engine cylinders should be taken when they are cold, and, if extreme accuracy is required, as in scientific investigations, corrections should be applied to conform to the mean working temperature. If the cylinders are much worn, the average diameter should be found. The clearance of the cylinders may be determined approximately from working drawings of the engine. For accurate work, when practicable, the clearance should be determined by the water measurement method.

Make a thorough examination of the physical condition of all parts of the plant or apparatus which concern the object in view, and record the conditions found, together with any points in the matter of operation which bear thereon.

Ascertain the interior condition of all steam, air, gas, or water cylinders and the condition of their pistons, and of water plungers and impellers, together with the valves and valve seats belonging thereto. Examine for air leaks in exhaust piping, condenser, packings, etc., by using the vacuum-gage or candle-flame test or by filling the piping, etc., with warm water under a slight head. Examine steam, air, gas, or water piping, traps, drip valves, blow-off cocks, safety valves, relief valves, heaters, etc., and make sure that they do not leak.

If the object of the test is to determine the highest efficiency or capacity obtainable, any physical defects or defects of operation tending to make the result unfavorable should first be remedied, all fouled parts being cleaned and the whole put in first-class condition. If, on the other hand, the object is to ascertain the performance under existing conditions, no such preparation is either required or desired.

In steam tests make sure that there is no leakage through blow-offs, drips, etc., or through any steam or water connections of the plant or apparatus undergoing test which would in any way affect the results. All such connections should be blanked off, or satisfactory assurance should be obtained that no leakage is going on either out or in. This is a most important matter, and no assurance should be considered satisfactory unless it is susceptible of absolute demonstration.

(3) *Apparatus and Instruments.* Select the apparatus and instruments specified later, locate and install the same, and complete the preparations for the work in view. The arrangement and location of the testing appliances in every case must be left to the judgment and ingenuity of the engineer in charge, the details being largely dependent upon locality and surroundings. One guiding rule, however, should always be kept in view, viz, *see that the apparatus and instruments are substantially reliable and arranged in such a way as to obtain correct data.* The following is a list of apparatus and instruments needed, together with a description of their leading features, methods of application and use, and, where needed, methods of calibration.

Weighing Scales. For determining the weight of coal, oil, water, etc., ordinary platform scales serve every purpose. Too much dependence, however, should not be placed upon their reliability without first calibrating them by the use of standard weights and carefully examining the knife-edges, bearing plates, and ring suspension to see that they are all in good working order.

Water Weighing and Measuring Apparatus. In tests of complete steam power plants, where it is required to measure the feed water without unnecessary change in the working conditions, a water meter may be employed. Meter measurement may also be required in many other cases such as locomotive and marine service. The accuracy of meters should be determined by calibration in place under the conditions of use.

If a large quantity of water is to be measured, an automatic water weigher, a rotary, disk, or Venturi meter, a weir, or some form of orifice measurement may be employed. In any case the measuring apparatus should be calibrated under the conditions of use, unless its design is such that standard formulas and constants may be applied for determining the discharge. If recording mechanism is employed in connection with orifice or weir measuring apparatus, make sure its record is reliable.

In measuring jacket water or any supply under pressure which has a temperature exceeding 212° F., the water should first be cooled, which may be done by discharging it into a tank of cold water previously weighed or by passing it through a coil of pipe submerged in running and colder water, thereby preventing loss by evaporation, which occurs when such hot water is discharged into the open air. If such water is untrapped, the drain pipe should be provided with a gage glass and the outlet choked, so as to keep the water in sight in the glass.

Venturi meters, Pitot tubes, pitometers, and orifices may be used for measuring water discharged by pumps through pipes under pressure.

Steam-Measuring Apparatus. Various forms of steam meters may be employed for measuring steam, provided that such meters are properly calibrated under conditions of use and that the pulsations of pressure, if any, are not serious. For measuring the steam used by the auxiliaries of a steam plant, either individually or collectively, the orifice form of steam meter may be used, consisting of an orifice in a plate inserted between the two halves of a pair of flanges in the pipe through which the steam passes or placed in a by-pass through which the steam is diverted, with gage pipe on either side for determining the fall in pressure. The quantity of steam represented by the various differences of pressure which occur may be found by arranging the apparatus so as to draw steam through the orifice and discharge it into a tank of water resting on platform scales, by which its actual weight in a given time is determined.

A plate $\frac{1}{8}$ inch thick containing an orifice 1 inch in diameter, with square edges, will discharge the approximate quantities of dry steam per hour given in Table VI with various pressure drops, the pressure below the orifice being 100 pounds by gage.

The water-glass method affords an approximate means for determining the steam consumption of auxiliaries and for measuring the leakages of steam and water from the boiler and its connections. This method consists of shutting off all secondary feed valves (which must be known to be tight) and the main feed valve, thereby stopping absolutely the entrance or exit of water at the feed pipes to the boiler; then maintaining the steam pressure (by means of a very slow fire) at a fixed point, which is approximately that of the working pressure, and observing the rate at which the water falls in the gage glasses. It is well in this test, as

TABLE VI

Discharge through Orifice 1 Inch in Diameter at 100 Pounds Pressure

Pressure Drop (lb. per sq. in.)	Dry Steam (lb. per hr.)
$\frac{1}{2}$	430
1	615
2	930
3	1200
4	1400
5	1560
10	2180
15	2640
20	3050

in other work of this character, to make observations every ten minutes and to continue them until the differences between successive readings attain a constant rate. In many cases the conditions will have become constant at the expiration of fifteen minutes from the time of shutting the valves, and thereafter the fall of the water due to leakage of steam and water becomes approximately constant. It is usually sufficient, after this time, to continue the test for two hours, thereby obtaining a number of half-hour periods. When this test is finished, the quantity of leakage is ascertained by calculating the volume of water which has disappeared, using the area of the water level and the depth shown on the glass, making due allowance for the weight of one cubic foot of water at the observed pressure. The water columns should not be blown down during the time a water-glass test is going on nor for a period of at least one hour before it begins.

If there is opportunity for condensation to occur and collect in the steam pipe during the leakage test, the quantity should be determined as closely as desirable and properly allowed for.

Pressure Gages. For determining pressure, the gages belonging to the plant may be used, provided they are compared with a standardized gage of the spring or mercury type and verified, due allowance being made for the head of water, if any, standing in the connecting pipe. Such comparisons should be made with both gages at their respective normal temperatures. In the use of spring gages for steam, the gages should be protected by proper siphons of water seals and no leakage should be allowed at the gage cock. The gages should also be located so that they will not be unduly heated.

For measuring low pressure or vacuums, a U-tube gage may be employed or a spring gage may be used, provided it is referred to a standard and corrected for water in the connecting pipe. In cases where extreme vacuums are to be measured, as in turbine practice, the absolute-pressure gage is useful, provided the exhaustion is complete and no air is admitted afterward.

For determining steam pressure on the two sides of an orifice, two gages should be used which are carefully graduated to single pounds or, better, one gage should be used and should be piped up so as to connect at will to either side of the orifice. A differential gage may also be employed, indicating at once the pressure drop. If the pressure drop is small, a glass U-tube containing mercury may be used. If the drop is less than one-half pound, water columns may be substituted.

For determining the water pressure in the force main of a pumping engine, the gage should be one which is sensitive to changes amounting to one-half per cent of the pressure indicated. If such a gage is not a part of the equipment of the plant, a special test gage should be attached.

For calibrating gages indicating pressures above the atmosphere, the dead-weight testing apparatus, which is manufactured by many of the prominent gage makers, may be employed as a standard of comparison. It consists of a vertical plunger nicely fitted to a cylinder containing oil or glycerin, through the medium of which the pressure is transmitted to the gage. The plunger is surmounted by a circular stand on which weights may be placed and by means of which any desired pressure can be secured. The total weight, in pounds, on the plunger (including weight of plunger) divided by the average area of the plunger and of the bushing which receives it, in square inches, gives the pressure in pounds per square inch.

Another standard of comparison is the mercury column. If this instrument is used, assurance must be had that it is properly graduated with reference to the ever-varying zero point; that the mercury is pure; and that the proper correction is made for any difference between the actual temperature and the temperature at which the instrument was graduated.

For pressures below the atmosphere, an air pump or some other means of producing a vacuum is required and reference must be made to a mercury gage. Such a gage may be a U-tube having a length of 30 inches and with both arms properly filled with pure mercury.

Thermometers. Thermometers should be of the kind having graduations marked on the glass stem. Those used for temperatures above 500° F. should have nitrogen in the top of the bore. They should also have a small safety bulb at the top. Thermometers constructed in this way can be used satisfactorily up to 1000° F.

Thermometers which are used for important data should be calibrated before and after a test by reference to standard thermometers. They may be calibrated, if desired, by direct comparison with standard thermometers certified by the U.S. Bureau of Standards.

A thermometer well consists of a hollow plug threaded at the upper end and screwed into a threaded hole in the top of a horizontal pipe, the lower part extending vertically into the interior of the pipe as far as the center, if practicable. The inside diameter of the thermometer tube and the well should be filled with mercury or high-grade mineral oil for temperatures below 500 degrees and with soft solder for higher temperatures. For superheated steam the immersed portion should be fluted so as to increase the area of the absorbing surface.

When the stem is not immersed, the correction to be added to its reading is $0.000088\, n(T-t)$, in which n is the number of degrees on the scale not immersed, T the indicated temperature, and t the mean temperature of the air surrounding the stem, as shown by a thermometer suspended at the mean point.

For accurate work thermometers should be standardized for the immersion at which they are intended to be used, and such immersion should be recorded.

For ordinary work thermometers may be used without correction if they are of the type that are graduated at a given immersion, the degree of immersion being marked on the stem and the temperature of the exposed stem being approximately that at which it was graduated.

Barometers. For important or extremely accurate steam tests and gas engine tests, the pressure of the atmosphere should be taken either by a mercurial or aneroid barometer and the reading from this instrument, reduced to pounds pressure per square inch, should be employed in determining the absolute steam pressure. In many cases it is sufficient to refer to the daily records of the nearest station of the Government Weather Bureau. These records, which refer to sea level, should be corrected for altitude. Aneroid barometers may be readily calibrated by comparing them with a mercury barometer, making proper temperature corrections.

Steam Calorimeters. The most satisfactory instruments for determining the amount of moisture in steam are calorimeters that operate on the throttling principle or that combine the throttling and separating principles, the orifice used being of such size as to throttle to atmospheric pressure and the instrument being provided with two thermometers, one showing the temperature above the orifice and the other that below it. If no commercial make of calorimeter is available on a test, an instrument of the throttling type can be made of pipe fittings Instruments working on the separating principle alone may also be employed; also certain forms of electric calorimeters.

In using a steam calorimeter great care must be exercised in attaching the instrument to a properly located sampling tube or pipe. The sampling tube, pipe, or nozzle should be made of ½-inch iron pipe and inserted in the steam main at a point where the entrained moisture is likely to be most thoroughly mixed. The inner end of the pipe, which should extend nearly across to the opposite side of the main, should be closed and the interior portion perforated with not less than twenty ⅛-inch holes equally distributed from end to end and preferably drilled in irregular or spiral rows, with the first hole not less than ½ inch from the wall of the pipe.

If it is necessary to attach the sampling nozzle at a point near the end of a long horizontal run, a drip pipe should be provided a short distance in front of the nozzle, preferably at a pocket formed by some fitting, and the water running along the bottom of the main drawn off, weighed, and added to the moisture shown by the calorimeter; or, better, a steam separator should be installed at the point noted.

Indicators. To determine the amount of power developed in the cylinder of a reciprocating engine or that expended in a pump or compressor cylinder, the instrument required is the steam engine indicator. One or more of these instruments are attached to the cylinder or cylinders and operated from the crosshead or main shaft by the use of proper driving rig. As to the selection of the make of instrument, it should be one which is in all respects of first-class construction and is adapted to the purpose for which it is to be used.

Outside spring indicators are preferred for superheated steam and in other cases where the temperature of the gas or vapor is very high or low.

Planimeters. To determine the area of indicator diagrams, from which to ascertain the mean effective pressure, it is convenient to use some form of planimeter. The simplest, and probably the most desirable, instrument is the Amsler polar planimeter, in which the area is registered in square inches.

It is desirable to calibrate a planimeter from time to time by running it over a figure having a known area, such as a right-angle triangle of, say, 4 inches in length and 2 inches in height, observing whether it checks with the computed area.

Tachometers and Other Speed-Measuring Apparatus. For determining the speed of revolution of an engine shaft, especially where the speed exceeds 300 r.p.m., a convenient instrument is a tachometer which continually indicates on a dial the number of turns per minute. This instrument can be arranged to have a permanent location and to be operated continuously when the engine is running, or it can be a portable instrument which is held in the hand and applied for the time being to the end of the shaft. These instruments are of four general classes, viz, fly-ball, liquid, electromagnetic, and vibration.

These instruments should be calibrated by comparison with the record obtained by counting with the watch and a speed recorder or indicator the number of turns per minute.

The determination of variation of speed during a single revolution, or the effect due to sudden changes of the load, is desirable, especially in engines driving electric generators used for lighting purposes. There is no recognized standard method of making such determinations, and if they are desired the method may be devised to suit the requirements.

One method for determining the instantaneous variation of speed which accompanies a change of load is described as follows: A screen containing a narrow slot is placed on the end of a bar and vibrated by means of an electric current. A corresponding slot in a stationary screen is placed parallel to, and nearly touching, the vibrating screen, the two screens being placed a short distance from the flywheel of the engine in such a position that the observer can look through the two slots in the direction of the spokes of the wheel. The vibrations are adjusted so as to conform to the frequency with which the spokes of the wheel pass the slots. When this is done the observer viewing the wheel through the slots sees what appears to be a stationary flywheel. When a change in velocity of the flywheel occurs, the wheel appears to revolve either backward or forward according to the direction of the change. By careful observations of the amount of this motion, the angular change of velocity during any given time is ascertained.

Friction Brakes or Absorption Dynamometers. The power delivered by an engine may be determined by the application of a Prony brake to the rim of the flywheel. The friction device may consist of a simple band or rope, a number of ropes, or a series of blocks encircling the wheel. Weighing scales either of the platform or spring type are required for measuring the torque. For long runs the wheel is made with interior flanges for holding water to keep the rim cool.

The most satisfactory brake for absorbing and measuring power is some form of water friction brake. The advantage of a water brake is that it can be employed equally well for large or small amounts of power, and it is necessarily kept cool by the water upon which it depends for its operation. With this brake the determination of the quantity of water used and the number of degrees its temperature is raised (when corrected for radiation) furnishes a means of computing the amount of heat converted into work and thereby obtaining an additional measurement of power developed.

Another satisfactory form of brake is the electric dynamometer, in which the work is transformed into electric energy and the torque is measured in the same manner as in a Prony brake.

Transmission Dynamometers. Transmission dynamometers furnish means for determining the amount of power delivered by an engine under working conditions. In the case of a mill engine it is the power transmitted from the main shaft of the engine to the shaft of the mill. If this power is carried through a belt, the dynamometer measures the net amount of force transmitted. In a marine engine driving a screw propeller through a long shaft, the dynamometer shows the torsional strain on the shaft at a point as near as practicable to the engine. In a locomotive the dynamometer measures the amount of pull on the drawbar through which the power is transmitted to the first car of the train.

Steam Tables. Quantities depending upon the properties of saturated and superheated steam, which are used throughout the Code, such as B.t.u. per pound of steam, and temperatures corresponding to various pressures, etc., are based on Marks and Davis' tables (edition of 1909). The report of a test should state the authorship of the tables on which the calculations are based.

(4) *Miscellaneous Instructions.* The person in charge of a test should have the aid of a sufficient number of assistants, so that he may be free to give special attention to any part of the work whenever and wherever it may be required. He should make sure that the instruments and testing apparatus continually give reliable indications and that the readings are correctly recorded. He should also keep in view, at all points, the operation of the plant or part of the plant under test and see that the operating conditions determined on are maintained and that nothing occurs, either by accident or design, to vitiate the data. This last precaution is especially needed in guarantee tests.

Before a test is undertaken, it is important that the engine shall have been in operation a sufficient length of time to attain working temperatures and proper operating conditions throughout, so that the results of the test may express the true working performance. An exception should be noted where the object of the test is to obtain the working performance, in which case all the conditions should conform to those of regular service.

In preparation for a test to demonstrate maximum efficiency, it is desirable to run preliminary tests for the purpose of determining the most advantageous conditions.

(5) *Operating Conditions.* In all tests in which the object is to determine the performance under conditions of maximum efficiency or in which it is desired to ascertain the effect of predetermined conditions of operation, all such conditions which have an appreciable effect upon the efficiency should be maintained as nearly uniform during the trial as the limitations of practical work will permit.

On the other hand, if the object of the test is to determine the performance under working conditions, no attempt at uniformity is either desired or required unless this uniformity corresponds to the regular practice; the usual working conditions should, therefore, prevail throughout the trial.

(6) *Records.* A log of the data should be entered in notebooks or on blank sheets suitably prepared in advance. This should be done in such manner that the test may be divided into hour periods or, if necessary, periods of less duration and the leading data obtained for any one or more periods as desired, thereby showing the degree of uniformity obtained.

The readings of the various instruments and apparatus concerned in the test other than those showing quantities of consumption (such as fuel, water, etc.), should be taken at intervals not exceeding one-half hour and entered in the log. Whenever the indications fluctuate, the intervals should be reduced according to the extent of the fluctations. When it is essential that a number of instruments be read simultaneously, there should be an observer stationed at each one and the readings should be taken on a signal from a timekeeper.

Make a memorandum of every event connected with the progress of the test, however unnecessary it may appear at the time. A record should be made of the exact time of every such occurrence and the time of taking every weight and observation. For the purpose of identification the signature of the observer and the date should be applied to each log sheet or record.

In the simple matter of weighing water by the tank-full, which is required in many tests, a series of marks, or tallies, should never be trusted. The time each tank is emptied should be recorded. Such work should not be delegated to unreliable assistants and, whenever practicable, one or more men should be assigned solely to that work.

(7) *Plotting Data and Results.* If it is desired to show the uniformity of the data at a glance, the whole log of the trial should be plotted on a chart, preferably while the test is in progress, using horizontal distances to represent times of observation and vertical distances on suitable scales to represent various data as recorded.

RULES FOR CONDUCTING TESTS OF RECIPROCATING STEAM ENGINES

(1) *Introduction.* This code for steam engine tests applies to tests for determining the performance of the engine alone (including reheaters and jackets, if any), apart from that of steam-driven auxiliaries which are necessary to its operation. For tests of engines and auxiliaries combined and for tests of multiple-expansion engines from which steam is withdrawn for heating feed water

or otherwise, a separate and distinct code was prepared by the Power Test Committee of the A.S.M.E., called "The Code for Complete Steam Power Plants".

(2) *Object and Preparations.* Determine the object of the test, take dimensions, and note the physical condition not only of the engine but of all parts of the plant that are concerned in the determinations, examine for leakages, install testing appliances, etc., as pointed out in the general instructions.

(3) *Apparatus and Instruments.* The apparatus and instruments required for a performance test of a reciprocating steam engine are:

(a) Tanks and platform scales for weighing water (or water meters calibrated in place)
(b) Graduated scales attached to the water glasses of the boilers if the feed water is measured
(c) Pressure gages, vacuum gages, and thermometers
(d) Steam calorimeter
(e) Barometer
(f) Steam engine indicators
(g) Planimeter
(h) Tachometer, revolution-counter, or other speed-measuring device
(i) Friction brake or dynamometer if available

(4) *Methods Adopted.* Feed-Water Test. The determination of the heat and steam consumption of an engine by feed-water test requires the measurement of the various supplies of water to the boiler; that of water wasted by separators and drips on the main steam line; that of steam used for other purposes than the main engine cylinders; and that of water and steam which escape by leakage of the boiler and piping, all of these losses being deducted from the total feed water measured.

Air Pump Discharge Test. When a surface condenser is provided and the steam consumption is determined from the water discharged by the air pump, no such measurement of drips and leakage is required, but assurance must be had that all the steam passing into the cylinders finds its way into the condenser. If the condenser leaks, the defect should be remedied or suitable correction should be made. The water of condensation from jackets and reheaters, if not included in the air pump discharge, should be added thereto.

Steam Meter Test. When no other method is available the steam consumption may be determined by the use of a steam meter, bearing in mind the caution that it should be calibrated under the exact conditions of use.

Steam Used by Auxiliaries. The steam consumed by steam-driven auxiliaries which are required for the operation of the engine should not be included in the total steam from which the heat consumption is calculated, but the quantity of steam thus used should be determined and reported.

(5) *Operating Conditions.* Determine what the operating conditions should be to conform to the object in view and see that they prevail throughout the trial as previously explained.

Duration of Test. A test for steam or heat consumption with a substantially constant load should be continued for such time as may be necessary to obtain a number of successive hourly records during which the results are reasonably uniform. For a test involving the measurement of feed water for this purpose, five hours duration is sufficient. Where a surface condenser is used and the measurement is that of the water discharged by the air pump, the duration may be somewhat shorter. In this case, successive half-hour records may be compared and the time correspondingly reduced.

When the load varies widely at different times of the day, the duration should be such as to cover the entire period of variation.

Starting and Stopping. The engine and appurtenances having been set to work and thoroughly heated, as previously explained, note the water levels in the boilers and feed reservoir, take the time, and consider this the starting time. Then begin the measurements and observations and carry them forward until the end of the period determined on. When this time arrives, the water levels and steam pressure should be brought as near as practicable to the same points as at the start. This being done, again note the time and consider it the stopping time of the test. If there are differences in the water levels, proper corrections are to be applied.

Where a surface condenser is used, the collection of water discharged by the air pump begins at the starting time, and the water is thereafter measured or weighed until the end of the test.

Records. The general data should be recorded as previously pointed out. Half-hour readings of the instruments are sufficient, except where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 15 or 20 minutes and oftener if the nature of the test makes it necessary. Mark on each card the cylinder and the end on which it was taken, also the time of day. Record on one card of each set the readings of the steam and vacuum gages. These records should be subsequently entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are worked up.

(6) *Calculation of Results.* The following directions are given for computing the most important results:

Dry Steam. The quantity of dry steam consumed is determined by deducting the moisture, if any, found by the calorimeter test from the total amount of feed water (the latter being corrected for leakages and other losses) or from the amount of air pump discharge, as the case may be. If the steam is superheated, no correction is to be made for the superheat.

Heat Consumption. The number of heat units consumed by the engine is found by multiplying the weight of feed water consumed, corrected for moisture in the steam, if any, and for plant leakages and other exterior losses, by the total heat of 1 pound of steam (saturated or superheated) at the pressure in the steam pipe near the throttle, less the heat in 1 pound of water at the temperature corresponding to the pressure in the exhaust pipe near the engine.

Indicated Horsepower. In a single double-acting cylinder the indicated horsepower is found by using the formula

$$\frac{PLAN}{33000}$$

in which P is the average mean effective pressure in pounds per square inch measured from the indicator diagram; L is the length of the stroke in feet; A is the area in square inches of the piston less one-half the area of the rod if it passes through both cylinder heads; and N is the number of single strokes per minute.

Where extreme accuracy is required, the power developed by each side of the piston may be determined and the result added together.

Brake Horsepower. The brake horsepower is found by multiplying together the net pressure or weight in pounds on the brake arm (the gross weight minus the weight when the brake is entirely free from the pulley), the circumference of the circle whose radius is the horizontal distance in feet between the center of the shaft and the bearing point at the end of the brake arm, and the number of revolutions of the brake shaft per minute, and dividing the product by 33,000.

Thermal Efficiency. The thermal efficiency, that is, the proportion of the total heat consumption which is converted into work is found by dividing 2546.5 (B.t.u. equivalent of one horsepower hour) by the number of heat units actually consumed per horsepower hour.

The efficiency of the Rankine cycle is found by dividing the heat utilized per pound of steam in an ideal engine working on the Rankine cycle between the pressure and temperature in the steam pipe near the trottle and the pressure and temperatue in the exhaust pipe near the engine, by the difference between the total heat of 1 pound of steam at the throttle pressure and temperature (saturated or superheated, as the case may be), and the heat of 1 pound of water at the temperature of the steam in the exhaust pipe near the engine.

The Rankine cycle ratio (or the efficiency ratio referred to the Rankine cycle) is found by dividing the efficiency of the actual engine (referred to the i.h.p. or br.h.p., as the case may be), by the efficiency of the Rankine cycle.

Steam Accounted for by Indicator Diagrams at Points Near Cut-Off and Release. The steam accounted for, expressed in pounds per i.h.p. per hour, may readily be found by using the formula

$$\frac{13750}{\text{m.e.p.}}\left[(C+E)W_c-(H+E)W_h\right]$$

in which m.e.p. is mean effective pressure; C is proportion of direct stroke completed at points on expansion line near cut-off or release; E is proportion of clearance; H is proportion of return stroke incompleted at point on compression line just after exhaust closure; W_c is weight of 1 cubic foot of steam at pressure shown at cut-off or release point; and W_h is weight of 1 cubic foot of steam at pressure shown at compression point.

The points near cut-off, release, and compression referred to above are illustrated in Fig. 104.

In multiple-expansion engines the mean effective pressure to be used in the above formula is the aggregate m.e.p. referred to the cylinder under consideration. In a compound engine the aggregate m.e.p. for the high pressure cylinder is the

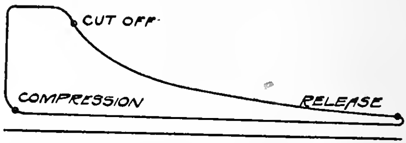


Fig. 104. Indicator Diagram, Showing Points Where "Steam Accounted for by Indicator" is Computed

sum of the actual m.e.p. of the h.p. cylinder and that of the l.p. cylinder multiplied by the cylinder ratio. Likewise the aggregate m.e.p. for the l.p. cylinder is the sum of the actual m.e.p. of the l.p. cylinder and the m.e.p. of the h.p. cylinder divided by the cylinder ratio.

Cut-Off and Ratio of Expansion. To find the percentage of cut-off, or what may best be termed the *commercial cut-off*, the following rule should be observed:

Through the point of maximum pressure during admission draw a line parallel to the atmospheric line. Through a point on the expansion line where the cut-off is complete draw a hyperbolic curve. The intersection of these two lines is the point of commercial cut-off, and the proportion of cut-off is found by dividing the length measured on the diagram up to this point by the total length.

To find the ratio of expansion, divide the volume corresponding to the piston displacement, including clearance, by the volume of the steam at the commercial cut-off, including clearance.

In a multiple-expansion engine, the ratio of expansion is found by dividing the volume of the l.p. cylinder, including clearance, by the volume of the h.p. cylinder at the commercial cut-off, including clearance.

Data and Results. The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for or omitting those not required, as may conform to the object in view. Unless otherwise indicated, the items should be the averages of the data.

DATA AND RESULTS OF STEAM ENGINE TEST CODE OF 1915

(1) Test of .... engine located at ....
To determine ....
Test conducted by ....

Dimensions, Etc.

(2) Type of engine (simple or multiple expansion) ....

(3) Class of service (mill, marine, electric, etc.) ....

(4) Auxiliaries (steam or electric driven) ....
- (a) Type and make of condenser equipment ....
- (b) Rated capacity of condenser equipment ....
- (c) Type of oil pump, jacket pump, and reheater pump (direct or independently driven) ....

(5) Rated power of engine ....
- (a) Name of builders ....
- (b) Kind of valves ....
- (c) Type of governor ....

		1st Cyl	2d Cyl.	3d Cyl.
(6)	Diameter of cylinders .... in.	....	....	....
(7)	Stroke of pistons .... ft.	....	....	....
	(a) Diameter of piston rod, each end .... in.	....	....	....
(8)	Clearance (average) in per cent of piston displacement	....	....	....
(9)	H.p. constant 1 lb. 1 rev .... h.p.	....	....	....
	(a) Cylinder ratio (based on net piston displacement) .... 1 to	....	....	....
	(b) Area of interior steam surface .... sq. ft.	....	....	....
	(c) Area of jacketed surfaces .... sq. ft.	....	....	....
(10)	Capacity of generator or other apparatus consuming power of engine .... h.p.	....	....	....

Date and Duration

(11) Date ....

(12) Duration .... hr.

Average Pressures and Temperatures

(13) Pressure in steam pipe near throttle, by gage..........lb. per sq. in.

(14) Barometric pressure..........in. of mercury.

(a) Pressure at boiler, by gage..........lb. per sq. in.

(15) Pressure in 1st receiver, by gage..........lb. per sq. in.

(16) Pressure in 2d receiver, by gage..........lb. per sq. in.

(17) Pressure in exhaust pipe near engine, by gage..........lb. per sq. in.

(18) Vaccuum in condenser..........in. of mercury.

(a) Corresponding absolute pressure..........lb. per sq. in.

(19) Pressure in jackets and reheaters..........lb. per sq. in.

(20) Temperature of steam near throttle..........deg.

(a) Temperature of saturated steam at throttle pressure..........deg.

(b) Temperature of steam leaving 1st receiver, if superheated..........deg.

(c) Temperature of steam leaving 2d receiver, if superheated..........deg.

(21) Temperature of steam in exhaust pipe near engine..........deg.

(a) Temperature of injection, or circulating, water entering condenser..........deg.

(b) Temperature of injection water leaving condenser..........deg.

(c) Temperature of air in engine room..........deg.

Quality of Steam

(22) Percentage of moisture in steam near throttle or number of degrees of superheating..........per cent or deg.

Total Quantities

(23) Total water fed to boilers..........lb.

(24) Total condensed steam from surface condenser (corrected for condenser leakage)..........lb.

(25) Total dry steam consumed (Item 23 or 24 less moisture in steam) ....lb.

Hourly Quantities

(26) Total water fed to boilers or drawn from surface condenser per hour..lb.

(27) Total dry steam consumed for all purposes per hour (Item 25 ÷ Item 12)..........lb.

(28) Steam consumed per hour for all purposes foreign to the main engine..lb.

(29) Dry steam consumed by engine per hour (Item 27 − Item 28)..........lb.

(a) Circulating water supplied to condenser per hour..........lb.

Hourly Heat Data

(30) Heat units consumed by engine per hour [Item 29 × (total heat of steam per pound at pressure of Item 13 minus heat in 1 lb. of water at temperature of Item 21)] ....B.t.u.

(a) Heat converted into work per hour..B.t.u.

(b) Heat rejected to condenser per hour [Item 20a × (Item 21b − 21a)] (approximate)..B.t.u.

(c) Heat rejected in form of uncondensed steam withdrawn from cylinders..B.t.u.

(d) Heat lost by radiation..B.t.u.

Indicator Diagrams

	1st Cyl.	2d Cyl	3d Cyl.
(31) Commercial cut-off in per cent of stroke ....per cent	....	....	....
(32) Initial pressure above atmosphere........................lb. per sq. in.	....	....	....
(33) Back pressure at lowest point above or below atmosphere........................lbs. per sq. in.	....	....	....
(a) Mean back pressure above atmosphere or zero ......lb. per sq. in.	....	....	....
(34) Mean effective pressure.............lb. per sq. in.	....	....	....
(a) Equivalent m.e.p. referred to 1st cylinder................lb. per sq. in.	....	....	....
(b) Equivalent m.e.p. referred to 2d cylinder................lb. per sq. in	....	....	....
(c) Equivalent m.e.p. referred to 3d cylinder................lb. per sq. in.	....	....	....
(35) Aggregate m.e.p. referred to each cylinder ....lb. per sq. in.	....	....	....
(36) Steam accounted for per i.h.p. hr. at point on expansion line shortly after cut-off......lb.	....	....	....
(37) Steam accounted for per i.h.p. hr. at point on expansion line just before release........lb.	....	....	....
(a) Pressure at selected point near cut-off........................lb. per sq. in.	....	....	....
(b) Pressure at selected point near release................lb. per sq. in.	....	....	....
(c) Pressure at point on compression curve shortly after exhaust closure................lb. per sq. in.	....	....	....
(d) Proportion of direct stroke completed at selected point near cut-off	....	....	....

	1st Cyl.	2d Cyl.	3d Cyl.
(e) Proportion of direct stroke completed at selected point near release	....	....	....
(f) Proportion of return stroke uncompleted at selected point on compression line	....	....	....
(g) Ratio of expansion	....	....	....
(h) M.e.p. of hypothetical diagram ....lb. per sq. in.	....	....	....
(i) Diagram factor	....	....	....

Speed

(38) Revolutions per minute..r.p.m.

(39) Piston speed per minute..ft.

(a) Variation of speed between no load and full load..........per cent

(b) Momentary fluctuations of speed on suddenly changing from full load to half-load.........................per cent

Power

(40) Indicated h.p. developed, whole engine..i.h.p.

(a) I.h.p. developed by 1st cylinder..i.h.p.

(b) I.h.p. developed by 2d cylinder..i.h.p.

(c) I.h.p. developed by 3d cylinder..i.h.p.

(41) Brake h.p..br.h.p.

(42) Friction of engine (Item 40 – Item 41)...h.p.

(a) Friction expressed in percentage of i.h.p. (Item 42 ÷ Item 40 × 100)..per cent

(b) Indicated h.p. with no load at normal speed....................i.h.p.

Economy Results

(43) Dry steam consumed by engine per i.h.p. per hr......................................lb.

(44) Dry steam consumed by engine per br.h.p. hr..lb.

(45) Percentage of steam consumed by engine accounted for by indicator at point near cut-off...per cent

(46) Percentage of steam consumed near release.....................................per cent

(47) Heat units consumed by engine per i.h.p. hr. (Item 30 ÷ Item 40)..B.t.u.

(48) Heat units consumed by engine per br.h.p. hr. (Item 30 ÷ Item 41)..B.t.u.

Efficiency Results

(49) Thermal efficiency of engine referred to i.h.p. (2546.5 ÷ Item 47) × 100...per cent

(50) Thermal efficiency of engine referred to br.h.p. (2546.5 ÷ Item 48) × 100...per cent

(51) Efficiency of Rankine cycle between temperatures of Items 20 and 21........per cent
(52) Rankine cycle ratio referred to i.h.p. (Item 49 ÷ Item 51)........per cent
(53) Rankine cycle ratio referred to br.h.p. (Item 50 ÷ Item 51)........per cent

Work Done per Heat Unit

(54) Net work per B.t.u. consumed by engine (1980000 ÷ Item 48)........ft. lb.

Sample Diagrams

(55) Sample diagrams from each cylinder........
(a) Steam pipe diagrams........

PRINCIPAL DATA AND RESULTS OF RECIPROCATING ENGINE TESTS

(1) Dimensions of cylinders........
(2) Date........
(3) Duration........hr.
(4) Pressure in steam pipe near throttle, by gage........lb. per sq. in.
(5) Pressure in receivers........lb. per sq. in.
(6) Vacuum in condenser........in. of mercury
(7) Percentage of moisture in steam near throttle or number of degrees of superheating........per cent or deg.
(8) Net steam consumed per hour........lb.
(9) Mean effective pressure in each cylinder........lb. per sq. in.
(10) Revolutions per minute........r.p.m.
(11) Indicated horse power developed........i.h.p.
(12) Steam consumed per i.h.p. hr........lb.
(13) Steam accounted for at cut-off each cylinder........lb.
(14) Heat consumed per i.h.p. hr........B.t.u.

Special Tests. For an engine driving an electric generator the form of test should be enlarged to include the electrical data, embracing the average voltage, number of amperes each phase, number of watts, number of watthours, average power factor, etc.; and the economy results based on the electric output embracing the heat units and steam consumed per electric h.p.hr. and per kw.hr., together with the efficiency of the generator.

Likewise in a marine engine having a shaft dynamometer, the form of test should include the data obtained from this instrument, in which case the brake h.p. becomes the shaft h.p.

Actual Engine Test. To illustrate the application of many of the items given as obtained from the Code, a full engine test will be taken and reported upon. This report will serve to give the order and manner in which data should be tabulated and also the method in which the report should be worked up.

DETERMINATION OF EFFICIENCY OF A BUCKEYE ENGINE UNDER DIFFERENT LOADS

Purpose

The purpose of this series of tests on the Buckeye engine located in the Engineering Laboratory of Purdue University was to determine the best efficiency under six different loads, ranging from zero to $1\frac{1}{4}$ load, by $\frac{1}{4}$ load steps, the engine running non-condensing and using 160 pounds of steam pressure, absolute.

Plan

The zero load was determined with the friction brake *I*, Fig. 105, removed and the engine running free. The full load was determined by the brake load

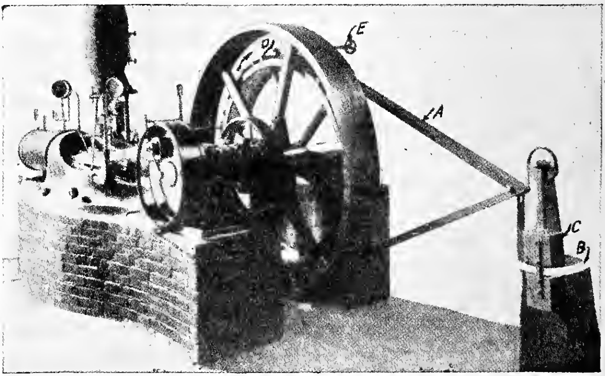


Fig. 105 Buckeye Engine Fitted with Prony Brake and Indicators

which the engine carried with 25 per cent cut-off, this being the builders' rating for this type of engine. The $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, 1, and $1\frac{1}{4}$ loads were taken as 25%, 50%, 75%, 100%, and 125%, respectively, of the full load.

Steam pressure was maintained constant at the pressure indicated for the test. Each test was of one hour duration, the engine having been run under conditions of the test a length of time sufficient to permit the conditions to become constant.

Method of Conducting Test

Constant steam pressure was obtained by throttling the 5-inch steam line leading to the engine by means of the pipe line valve. This throttling action was not sufficient to cause the steam to become superheated.

The revolutions per minute were obtained by means of a revolution counter.

Indicator diagrams were taken every five minutes, 13 sets of diagrams being obtained for each hour's run.

Barometer readings were taken every 15 minutes.

The amount of water was determined by condensing the exhaust steam at atmospheric pressure.

Preliminary Work

Before commencing the work the engine was placed in as good condition as was possible. The governor was adjusted in order to reduce friction; play was taken up in the valve gear and the valves were carefully set to give equal cut-off on both ends at full load; all stuffing boxes were repacked; the brake wheel was turned up and brake recalibrated.

The pressure in the engine supply line was obtained by tapping a $\frac{1}{4}$-inch pipe into the main, about 3 feet from the valve. This $\frac{1}{4}$-inch pipe was connected to a large steam gauge which faced the operator of the throttling valve, thus enabling him to watch the gauge all the time and maintain a constant pressure.

Observed Data

In each test the following observations were taken:

Steam pressure, constant throughout
Brake load
Revolutions per minute
Weight of condensed steam
Barometer
Indicator diagrams

Results

Having the above data it becomes possible to calculate the following:

(1) Per cent of cut-off, head end and crank end.
(2) Mean effective pressure (m.e.p.), head end and crank end.
(3) Indicated horsepower, head end and crank end and total.
(4) Brake horsepower (b.h.p.).
(5) Friction horsepower (f.h.p.).
(6) Mechanical efficiency.
(7) Pounds steam, per indicated horsepower per hour and per brake horsepower per hour.
(8) British Thermal Units per hour, per indicated horsepower and brake horsepower per hour.
(9) Thermal efficiency.

Constants and Formulas. The constants of the engine and formulas employed in obtaining the calculated items in the summary of results, are as follows:

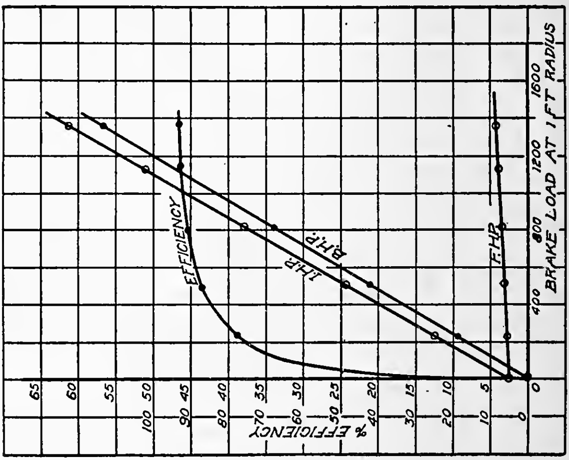


Fig. 106. Horsepower and Efficiency Curves for Buckeye Engine

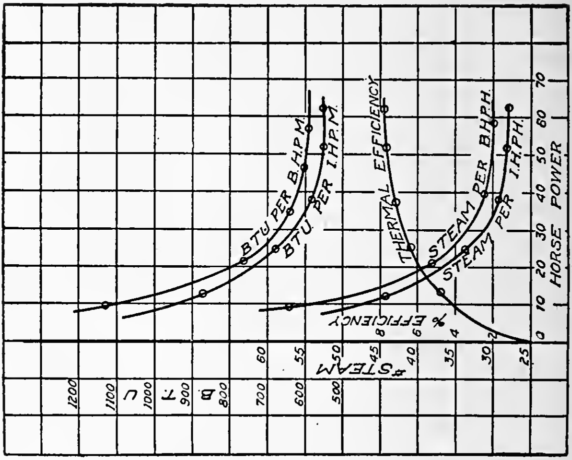


Fig. 107. Steam Consumption B.T.U. and Thermal Efficiency Curves for Buckeye Engine

Diameter of cylinder, 7.75 inches.
Piston rod diameter, 1.437 inches.
H.E. area, 47.173 square inches; c.e. area, 45.55 square inches.
Radius of brake arm, 38.25 inches, equals 3.185 feet.
Clearance, head end 6.15%; crank end, 6.765%
Normal speed, 220 revolutions per minute
Heat value of 1 horsepower, 42.42 British Thermal Units.
Heat value of 1 pound of steam, above 32°F for 160 pounds absolute 1,192.8 British Thermal Units.
Gauge pressure 15 pounds (approximate) less than absolute pressure

The horsepower constants are as follows:

H.E.–i.h.p. Constant = 001787 (See "Steam Engine Indicators.")
C.E.–i.h.p. Constant = .001726
B.H.P. Constant = .00060695.

Item (3). At observed revolutions per minute (r.p.m.):

H.E.–i.h.p. = .001787 × h.e. m.e.p. × r.p.m
C.E.–i.h.p. = 001726 × c.e. m.e.p × r.p.m
Total i.h.p. = h.e. i.h.p. + c.e i.h.p

It often happens that the engine is not operated at the desired speed just at the instant of taking the reading, hence a correction must be made if the indicated horsepower is to be expressed and recorded for the normal speed. Therefore i.h.p. = total i.h.p. × 220 ÷ observed r.p.m

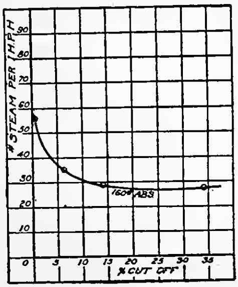


Fig. 108. Steam Consumption for Different Cut-Offs

Item (4) At observed r.p.m. determined as follows:

B.H.P = .00060695 × pounds brake load × r.p.m.
B.H.P = .0001904 pounds brake load at 1 foot radius × r.p.m.
At 220 r.p.m., corrected b.h.p. = b.h.p × 220 ÷ observed r.p.m.

Item (5) At 220 r.p.m., the f.h.p. = total i.h.p – b.h.p.
Item (6) At 220 r.p.m., the mechanical efficiency = b.h.p. ÷ total i.h.p.

The pounds of steam per hour at 220 r.p.m. = pounds of steam per hour at observed r.p.m. × 220 ÷ observed r.p.m The B T U supplied per hour = corrected pounds of steam per hour × total British Thermal Units in 1 pound steam, at given absolute pressure above 32°F

Item (7). The pounds steam per i.h.p. per hour = corrected pounds steam per hour divided by corrected i.h.p.
The pounds of steam per b.h.p. per hour = corrected pounds steam per hour divided by corrected b.h.p.
Item (8). The British Thermal Units per i.h.p per hour = total British Thermal Units supplied divided by 160 × corrected i.h.p.
The British Thermal Units per b.h.p., per hour = total British Thermal Units supplied divided by corrected b.h.p. × 160.
Item (9). The Thermal Efficiency = 42.42 British Thermal Units divided by British Thermal Units per i.h.p. per hour

TABLE VII

Indicator Diagram Data for Buckeye Engine Test

Steam Pressure 160 # Abs.							
Cyl. End	Card No.	No Load		¼ Load		½ Load	
		% C.O.	M.E.P.	% C.O.	M.E.P.	% C.O.	M.E.P.
Head End	1		.000	1.58	14.73	5.00	28.41
	2		.785	1.32	13.98	5.52	28.17
	3		.773	1.05	14.14	4.98	28.37
	4		.259	1.05	14.73	4.71	26.70
	5		1.040	1.05	13.65	5.00	26.82
	6		.000	1.05	14.45	4.46	26.78
	7		.000	.79	14.13	4.47	26.30
	8		.521	.79	13.41	4.71	26.18
	9		.675	1.05	12.90	4.71	27.22
	10		1.295	1.05	12.63	4.97	27.22
	11		.675	1.05	12.90	4.71	27.22
	12		.529	1.06	14.80	4.73	26.30
	13		.779	1.06	12.44	4.97	27.22
	Av.		.5638	1.073	13.757	4.841	27.14
Crank End	1		4.325	.77	18.58	7.41	36.59
	2		4.325	.76	18.50	7.66	36.80
	3		3.805	.77	18.62	7.65	36.46
	4		3.560	.77	18.58	7.65	36.20
	5		4.055	.77	19.13	7.65	36.20
	6		3.567	.77	18.92	7.66	36.30
	7		4.340	.77	18.43	7.65	36.20
	8		3.805	.77	18.37	7.41	36.30
	9		3.805	.76	17.70	7.64	37.63
	10		3.785	.76	17.95	7.41	37.81
	11		3.805	.76	17.95	7.65	36.75
	12		3.560	.76	17.75	7.40	36.20
	13		4.555	.76	17.95	7.69	37.18
	Av.		3.9445	.7653	18.338	7.578	36.655

TABLE VIII

Indicator Diagram Data for Buckeye Engine Test

STEAM PRESSURE 160 # ABS.							
CYL. END	CARD NO.	¾ LOAD		FULL LOAD		1¼ LOAD	
		%C.O.	M.E.P.	%C.O.	M.E.P.	%C.O.	M.E.P.
HEAD END	1	13.85	46.70	24.90	63.25	33.92	77.60
	2	13.85	44.95	24.05	63.15	35.22	77.60
	3	13.69	44.75	24.85	62.85	35.32	77.80
	4	13.88	44.75	25.15	63.35	34.90	77 00
	5	14.40	44.50	24.52	64.65	35.41	76.60
	6	14.21	45.50	25.15	64.15	35.26	77 10
	7	13.85	44.65	24.21	64.20	34.27	77.55
	8	13.57	45.15	25.05	63.76	34.87	77.05
	9	13.65	44.90	25.15	64.90	36.21	78.95
	10	14.15	46.20	24 35	63.35	34.05	77.50
	11	14.70	45.40	24.42	63.25	34.07	77.95
	12	13.92	44.40	25.15	64.15	34.80	78.05
	13	14.11	44.15	25.20	64.50	34.72	77.95
	AV.	13.98	45.073	24.78	63.80	34.84	77.537
CRANK END	1	14.05	53.15	24.80	67.85	31.87	82.05
	2	14.50	52.80	24.75	67 30	34.25	82.30
	3	14.05	52.15	25.10	68.70	33.59	81.80
	4	14.32	53.45	24.05	67 50	32.59	80.80
	5	14.54	51.80	24.67	67 50	33 32	81 31
	6	14.11	53.10	24.80	68.35	32.83	81.30
	7	14.22	52.55	24.28	68.40	32 83	80.80
	8	14.54	52.25	24.61	68.00	32.83	81.00
	9	14.28	51.80	24.80	68.80	34.25	81.80
	10	14.50	51.85	24.38	69.06	32.90	80.25
	11	14.54	52.25	24.60	68.65	33.07	80.75
	12	14.54	53.30	24.58	68.55	33.52	80.00
	13	14.76	52.15	24.90	68.50	32.63	80.20
	AV.	14.38	52.527	24.59	68.30	33.11	81.10

TABLE IX

PERFORMANCE OF
UNDER DIFFERENT CUT-OFF'S.
SUMMARY

OBSERVED						AV. M.E.P.		I. H.P.			
LOAD	#BRAKE LOAD AT 1 FOOT RADIUS	R.P.M.	BAROMETER INS. OF MERCURY	#STEAM CONDENSED PER HOUR	PER CENT CUT-OFF	HEAD END	CRANK END	HEAD END	CRANK END	TOTAL AT OBSERVED R.P.M.	TOTAL CORRECTED TO 220 R.P.M.
0	0	287.1	29.82	434.5		.564	3.945	.289	1.953	2.242	1.72
1/4	227.5	225.5	29.83	550.0	.97	13.76	18.39	5.45	7.13	12.66	12.35
1/2	531.0	222.9	29.81	849.0	6.21	27.14	36.66	10.82	14.10	24.92	24.58
3/4	815.0	221.9	29.80	1096.0	14.2	45.07	52.53	17.86	20.10	37.96	37.62
1	1126.0	217.4	29.78	1401.0	24.6	63 80	67.74	24.80	25.62	50.42	51.00
1 1/4	1363.0	209.7	29.79	1614.5	34.0	77.54	81.10	29.00	29.30	58.30	61.20

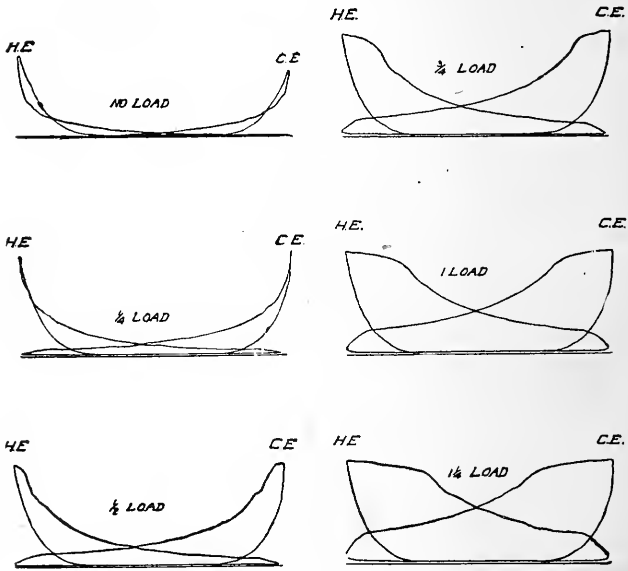


Fig. 109. Indicator Diagrams Taken During Test of Buckeye Engine

TABLE IX—Continued

THE BUCKEYE ENGINE STEAM PRESSURES 160# ABS. OF RESULTS.											
CALCULATED											
B.H.P.		F.H.P. AT 220 R.P.M.	MECHANICAL EFFICIENCY	#STEAM PER HR. CORRECTED TO 220 R.P.M.	TOTAL B.T.U. SUPPLIED PER HOUR	#STEAM PER I.H.P.H.	#STEAM PER B.H.P.H.	B.T.U. PER I.H.P.M.	B.T.U. PER B.H.P.M.	THERMAL EFFICIENCY
AT OBSERVED R.P.M.	CORRECTED TO 220 R.P.M.									
0	0	2.35	0	332.5	396600	141.4		2810		0
9.75	9.54	2.71	77.2	547.0	652500	44.25	57.4	879	1140	4.82
21.81	21.51	3.07	87.5	837.0	998350	34.1	38.95	678	774	6.29
34.41	34.18	3.44	90.8	1085.0	1294300	28.85	31.80	573	632	7.25
46.55	47.17	3.88	92.5	1418.0	1691500	27.8	30.5	552.5	606	7.68
54.45	57.07	4.13	93.3	1695.0	2022000	27.75	29.7	551	590	7.70

Plotted Results. On curve sheet shown as Fig. 106, are plotted to pounds brake load at 1′ radius, the i.h.p., b.h.p., f.h.p., and mechanical efficiency.

On curve sheet shown as Fig. 107, are plotted to horsepower the pounds steam per i.h.p. per hour, the pounds steam per b.h.p. per hour, the British Thermal Unit per i.h.p. per minute, the British Thermal Unit per b.h.p. per minute, and the thermal efficiency.

On curve sheet shown as Fig. 108, is plotted a curve which shows the steam consumption for the different per cents of cut-off.

Conclusions and Comparisons

An examination of the curves shows a marked increase in economy of the ½ load over the ¼ load; a smaller increase in economy of the ¾ load over the ½ load; and a still smaller increase in economy of the full load over the ¾ load; but the full load and 1¼ load have the same steam consumption per i.h.p. per hour indicating that the engine is operating most economically throughout this range.

The tests indicate a very good range of economical operation from ¾ load to 1¼ load, and although the steam consumption is higher than the best recorded results for other engines of greater horsepower, yet the results obtained are very good considering the relatively small size of the engine.

Appendix

The engine under test was a 7¾″×15″ type "B" Buckeye engine which had been rebuilt from the old type of flat valve to a piston valve engine. The following information was supplied by the Buckeye Engine Company:

Lap 11/16″, Lead 3/32″, Compression 2½″, Exhaust laps, ⅛ and 1/32″, Clearance 5.6%, Cut-off 25%. Weight of reciprocating parts 150 pounds.

The arrangement of the brake apparatus may be seen in Fig. 105, in which A is brake lever, B is calibrated brake load arc, C is brake pendulum, and D is brake wheel.

Cooling water for the brake enters through a hose not shown in the illustration. The direction of rotation of the brake wheel is indicated by the arrow near D. By means of the hand wheel E, the brake load is applied and regulated. The brake was carefully calibrated before beginning the test.

Computation of Constants

$$\text{H.E. piston displacement} = \frac{47.173 \times 15}{144 \times 12} = .41 \text{ cu. ft.}$$

$$\text{C.E. piston displacement} = \frac{45.55 \times 15}{144 \times 12} = .396 \text{ cu. ft.}$$

$$\text{H.E. clearance} = \frac{.0252}{.41} = 6.15\%$$

$$\text{C.E. clearance} = \frac{.0268}{.396} = 6.765\%$$

$$\text{H.E.} - \text{i.h.p.} \quad \text{Constant} = \frac{15 \times 47.173}{12 \times 33000} = .001787$$

$$\text{C.E.} - \text{i.h.p.} \quad \text{Constant} = \frac{15 \times 45.55}{12 \times 33000} = .001726$$

$$\text{B.H.P.} \quad \text{Constant} = .00060695$$

$$\text{Thermal Efficiency Constant} = \frac{33000}{778} = 42\ 42$$

Tables VII and VIII contain information from the indicator diagrams, and Table IX is a general summary of the observed and calculated results of the tests. Fig. 109 shows sample indicator diagrams taken during the test.

STEAM ENGINE TROUBLES AND REMEDIES

Manufacturers supplying reciprocating steam engines for commercial uses usually make it a part of their business to see that the machines are properly installed with suitable foundations, steam and exhaust piping, and accessories. They usually furnish a competent man to be present when the engine is first placed into commission to see if it operates satisfactorily. So at the beginning we usually find stationary reciprocating steam engines operating quietly and in a manner acceptable to all concerned. As time goes on, however, and a number of different operators have had charge of the plant, and natural wear of the various parts occur, operating trouble will be experienced. Then again, accidents are liable to occur from time to time which will require immediate attention from those in charge in order that the machines may be kept running. These conditions make it imperative that the operator in charge be prepared, as far as possible, to make the necessary repairs when, the emergency arises.

During the life of an engine which is put to continuous duty in a power plant or a manufacturing plant, any one of a large number of troubles or accidents may occur. Only a limited number of cases can be discussed.

Broken Cylinder Casting, Cylinder Head, or Piston. *Cause of Water in Cylinder.* An accident due to water in the cylinder usually occurs at a time when the machine is being started after having been shut down for a considerable period, in other words, when the piping and parts have had sufficient time to cool down. It is assumed that a steam separator is located in the steam line at a point just ahead of the throttle, if the engine is located any great distance from the boiler, and that cylinder drain cocks are provided if the engine is of the slide or piston valve type. The accident usually happens because of the carelessness of the operator. Sufficient time must be allowed to get the cylinder and parts warmed and all entrained water worked through the cylinder, before bringing the engine up to full speed. It sometimes happens that if the water in the boiler is being carried at too high a level and is dirty, water will be carried over into the cylinder in sufficient quantities to cause an accident. Some types of engines have pressure relief valves attached to the ends of the cylinder which will open when any undue pressure occurs within the cylinder.

Breakage of Cylinder Parts. Such accidents may result in a breakage of the cylinder head, cylinder casting, piston, or all three. Cases are on record where the machine was a complete wreck, rupturing, in addition to the parts mentioned, other parts, such as the crank pin, crank disc, connecting rod, piston rod, etc. Usually if but one part is broken, an additional one may be secured from the makers and be placed into position by the operator. If, however, the damage is very serious it is better to let the builder handle the matter of repairs as seems best.

Knocking or Pounding. *Difficulty of Locating Noise.* The operator in charge of a smoothly running engine takes great pride in showing it to visitors, but when a click or pound is heard he naturally feels that these defects reflect upon his skill, and, whether alone or in the presence of visitors, it is a constant source of annoyance. He will put in many hours of overtime for which he makes no charge in an effort to locate the cause of these noises, which to outsiders are of no consequence but to his ear become almost unbearable at times. When the engine is working under a light load the engine runs so smoothly that he fancies the trouble has been reached, but later when it is operating under normal load the exasperating pounding begins, and, to aggravate the case, the bearings that formerly had given no trouble begin to heat from being too closely adjusted.

There are few people outside of the engineering profession who really know how perplexing a knocking or pounding in an engine may become. In the first place it is frequently very difficult to locate. Go to the cylinder and it seems to be there. Stand at the crank and the noise is there. In such cases the sensitive ear of the earnest and sincere operating engineer can hear nothing else, and its continuance affects both his mind and body in a way that it is difficult to explain. Many such cases require much thought and patience to eventually locate the trouble.

Causes of Pounding and Knocking. Improper Alignment. Improper alignment is perhaps one of the most common causes of pounding, and this is a defect which every operator should be able to correct by adjusting the working parts to a line. The alignment may have been correct when the machine was first installed, but continued use has brought about changes which make realignment necessary.

Uneven Wear of Bearings. There are many other causes of pounding to which attention might be called, as the obscurity of some of these might cause them to be overlooked. For instance, wrist pins, crosshead pins, and crank pins naturally wear unevenly and eventually will not be circular in section. In such cases, if the bearings are properly adjusted at the dead center points they probably will be quite loose at the quarter points. If properly adjusted at the quarter points they will be too tight at the dead centers and will give trouble by heating. The only remedy in these instances is either to replace the pins or have them trued up in an engine lathe and the bearing readjusted to the new size.

Loose Flywheel or Belt Pulley. Perhaps the most difficult pound or knock to locate is that caused by a loose flywheel or belt pulley. When such conditions are found, the remedy to apply is at once evident.

Worn Shaft Governor Parts. Engines fitted with shaft governors, which usually have several joints and movable parts, frequently become noisy due to these parts becoming worn. To remedy this trouble, it is usually necessary to rebore the holes in the various parts and make new pins to fit. It may be necessary in certain cases to bush some of the holes. It is not uncommon to hear pounding from such engines caused by the governor fluctuating, the parts not remaining relatively at rest for the load being carried. Such pounding is usually traceable to one of two causes—either the governor ports themselves are set up too snug or the packing gland on the valve rod is turned up too tight.

Causes of Noises in Cylinder and Steam Chest. Occasionally one hears an engine operating with clicking noises emanating from the cylinder and steam chest. These noises may be caused by a number of different things. If the piston packing ring overtravels the steam port too great a distance, the steam pressure may compress the ring momentarily, which will assume its normal shape when the piston moves ahead or back, as the case may be, thus causing a "spring slam" at each end of the stroke. Improper valve setting may be the cause of such a knock or click as above mentioned. Should the compression pressure exceed the initial steam pressure, the valve will be lifted from its seat. The simplest way of correcting this trouble would be to take indicator cards and reset the valve or valves to give the proper compression. If the engine has been in service a number of years the piston packing rings may become sufficiently worn to cause a rattle or clicking noise. In such cases new rings should be fitted to the piston. In engines where the piston is made up of several parts, wear on these parts would eventually result in producing noises.

If the valve setting has become deranged through carelessness or otherwise to such and extent that the power of the engine is unequally divided between the two ends of the cylinder, the effect may be to produce a pound. This can readily be remedied by the aid of a steam engine indicator in readjusting the steam distribution. Adjustments should be made until the power is equally divided between the two ends. This principle applies to compound engines as well as simple engines.

Improper Adjustment of Main Crank-Shaft Bearing. The main crank-shaft bearing is quite often the cause of pounds or knocks. This part is made up in so many different forms that any specific directions seem impracticable. Suffice to say, that the pound is due to improper adjustment and that if the parts are not worn too greatly the correct adjustment should not be a difficult matter.

Broken Flywheel. *Effect of Sudden Changes in Speed.* Technical literature is profuse with records of disasters caused by the failure of flywheels in use on engines when in service. In many instances the destruction is so complete that no evidence remains as to the cause of the failure. Perhaps the most common cause of flywheel failure is due to some fault in the operating condition of the governor. On this account it is extremely important that the operating engineer give careful attention to the working condition of all parts of his engine and especially the governing system. It is true that in the casting of the flywheel internal stresses may be occassioned due to unequal expansion and contraction which might in time cause failure, but the wheel is always designed with a large factor of safety to take care of such possible conditions. So it seems reasonable to assume that usually flywheel failures are caused by too sudden changes in speed. This sudden change in speed may be caused, as previously mentioned, by a defect in the governing mechanism, a slug of water coming into the cylinder through the steam pipe, or by a failure of some part of the engine mechanism, which would tend to lock the movement of the parts, such as a broken piston or a piston rod becoming loosened sufficiently to permit the piston to strike one end of the cylinder.

Effect of Worn Valve Packing Rings. Cases are on record where engines of the piston valve type have been in service for a number of years and the valve packing rings have become worn with the result that the governor would not control the speed at light loads. This was due to the fact that the worn condition of the valve permitted enough steam to leak into the cylinder to cause the speed of the machine to greatly increase. In all such engines, it is desirable to keep the valve packing rings in perfect repair not only as a safeguard against accidents caused by overspeeding but also as an aid in maintaining the steam economy of the plant.

Maintaining Steam Economy. One of the important points to which an operating engineer should give constant attention is the steam economy of the engines placed under his charge. Information on this point can be secured from time to time by economy tests. Maintaining maximum steam economy of the plant is not only of inestimable value to the engineer himself but is dollars and cents to his employer The steam consumption of an engine should improve up to a certain time as the piston and valve or valves become worn to a more perfect bearing. Later, however, as the cylinder, piston and rings, valve or valves, etc., become worn, they cease to remain steam tight and, as a result, permit steam to leak through without doing any real work. For this reason the engineer should keep himself informed as to the condition of his engines If the cylinder is worn "out-of-round", it should be turned over to a reputable machine shop or returned to the maker for reboring. If the cylinder is in good condition but a piston ring is badly worn or broken, thus permitting steam to leak through, the broken ring should at once be replaced by a new one It is preferable to secure the new ring from the builder if possible, since the builder has the proper machines and tools for turning out perfect rings The ordinary shop, even with its best mechanics, if they have had little or no experience in such work, is very liable to turn out an imperfect job. It is very likely that an imperfectly made ring replacing a broken or worn one would show little or no improvement in steam economy

Enlarged Vacuum Pump Valves. In plants using surface condensers, we frequently find the direct-acting type of wet vacuum or air pumps installed. These pumps handle the condensate from the condensers as well as exhausting the air from the system to a greater or less extent. With such equipment it is possible to maintain a vacuum of about 26 inches. This type of pump is usually fitted with composition rubber valves which eventually become enlarged and distorted owing to the action of the oil in the steam used for cylinder and valve lubrication. As a result of this condition, it will be found that eventually the valves do not function properly and that the vacuum will be materially decreased, thus decreasing the power capacity of the plant and materially increasing the steam consumption. These conditions make it essential that the air end of the pump should receive frequent inspections.

Piston Rod and Valve Rod Packing Troubles. The piston and valve rods are made steam tight by one of two methods, namely, by the use of a stuffing box, where some form of packing is placed around the rod and securely held in place by means of a gland, or by the use of some form of metallic packing. Either form, if properly constructed and adjusted, will perform its function very satisfactorily. The metallic packing frequently will permit steam to blow through when the engine is started after a shutdown, but this steam blow usually ceases after the parts become thoroughly heated. When a steam blow occurs where a stuffing box is used, the operator will usually apply a wrench and tighten the gland until the blow stops. In such cases the wrench should be used very cautiously. Many times the rod will become overheated due to the gland being turned up too tight. This usually occurs when the packing has been in use a considerable length of time and has become very hard. When this condition exists the packing should be replaced. The safest policy to follow after repacking a stuffing box is to tighten the gland only as much as is necessary to prevent a steam blow. It will need watching for a time and occasional tightening.

Superheating and Lubrication. The use of superheated steam in power plant operation has in every instance demonstrated economic advantages. One of the things which operated to discourage the use of superheated steam at the start was the trouble experienced in lubrication. In almost every instance the trouble arose because of the fact that the same oil was being used as when saturated steam was being generated. It has been shown many times in practice that a grade of oil sufficiently high for saturated steam service may not prove satisfactory when used in installations where superheated steam is used. This fact is especially emphasized if a high superheat of, say, 200 degrees or more is employed. In such cases a high-grade oil is recommended. All reputable oil refineries now refine lubricating oils which they guarantee for such classes of service.

Lining an Engine. In the erection of a new engine, if it be of considerable size, the manufacturers usually send one of their experienced erecting engineers to superintend the work in order to insure satisfactory operation. In established plants, where the engines have become worn and where settlement has taken place in the foundations, the realignment of one or more of the machines may become necessary. Where such instances arise the operating engineer is usually given the responsibility of superintending the work. The following suggestions may be of some service to the inexperienced. The presentation must of necessity be abbreviated.

Building Foundation. To begin with, if a foundation is to be built, it should be made in accordance with the design of the engine builders. Its size will be governed largely by the type of engine and by the character of the soil in the particular locality. A substantial wood templet should first be prepared which would show the correct location of the foundation bolt holes in the engine base. This templet should be used in the construction of the foundation to properly locate the different positions of the foundation bolts, which should be embedded in the foundation to such a depth as would insure their security. For a portion of the distance down from the top of the foundation, the bolts should be surrounded by iron pipe of suitable dimensions to permit a small amount of lateral movement. This will permit correction for any slight errors of measurement in locating the bolt holes. The top of the foundation should be made level. When ready to receive the engine bed, it should be carefully placed in position and made level by the use of at least two spirit levels, holding the bed in the proper position by means of small wedges. Cement grout should then be poured under the bed and permitted to firmly set, when the small wedges can be removed and the holes filled.

Locating Center Line. It is assumed that the engine has already been dismantled, that is, the valve gear, connecting rod, cylinder head, piston and rod, and crosshead have all been removed. In securing the proper alignment the first step is to take a slotted stick or piece of metal and secure it across the head end of the cylinder by means of the cylinder head stud bolts. Draw a fine linen line or fine wire over this stick and through the center of the cylinder out between the guides and attach it to an upright stick at the crank end of the bedplate, nailed to the floor or clamped to the bedplate. This line should then, by careful calipering, be made to pass directly through the center of the counterbore at each end of the cylinder. The measurements are made from the counterbore because this portion of the cylinder has been subjected to the least amount of wear. It is not advisable to make measurements to locate the center line in the stuffing box at the crank end of the cylinder. Sufficient time should be taken and great care exercised that this line is accurately located as much depends upon the accuracy of this first work. After the line is finally located it is well to examine the guides and determine their parallelism.

Placing Engine Shaft in Position. The next step is to lower the main engine shaft into the bearings, the outboard bearing being first loosely located. Now turn the crank first one way then the other, shifting the outer end one way or the other, until that part of the crank pin that is to be in line with the center line of the connecting rod shall be exactly over the line; also, so that the turned surfaces, if turned, of the crank shall be parallel to the line. Next make a half-turn of the shaft as nearly as possible and see that the pin maintains the same relative position to the line. Finally, carefully place the shaft in a horizontal position by the aid of spirit levels and the plumb line. It is well in checking this last work to run a second line at right angles to the center line from which to make measurements. When the engine shaft is finally located as directed, fasten the outboard bearing block securely and attach the bearing caps in place.

Checking Work. Before removing the carefully located lines, it is well to go over the work from the very beginning to check the adjustments. When this has been completed the lines may be removed and the work said to be correct and the process of assembling the parts begun.

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