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STEAM ENGINES
ELEVATORS

SCRANTON:
INTERNATIONAL TEXTBOOK COMPANY

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PREFACE

The International Library of Technology is the outgrowth of a large and increasing demand that has arisen for the Reference Libraries of the International Correspondence Schools on the part of those who are not students of the Schools. As the volumes composing this Library are all printed from the same plates used in printing the Reference Libraries above mentioned, a few words are necessary regarding the scope and purpose of the instruction imparted to the students of—and the class of students taught by—these Schools, in order to afford a clear understanding of their salient and unique features.

The only requirement for admission to any of the courses offered by the International Correspondence Schools, is that the applicant shall be able to read the English language and to write it sufficiently well to make his written answers to the questions asked him intelligible. Each course is complete in itself, and no textbooks are required other than those prepared by the Schools for the particular course selected. The students themselves are from every class, trade, and profession and from every country; they are, almost without exception, busily engaged in some vocation, and can spare but little time for study, and that usually outside of their regular working hours. The information desired is such as can be immediately applied in practice, so that the student may be enabled to exchange his present vocation for a more congenial one, or to rise to a higher level in the one he now pursues. Furthermore, he wishes to obtain a good working knowledge of the subjects treated in the shortest time and in the most direct manner possible.

In meeting these requirements, we have produced a set of books that in many respects, and particularly in the general plan followed, are absolutely unique. In the majority of subjects treated the knowledge of mathematics required is limited to the simplest principles of arithmetic and mensuration, and in no case is any greater knowledge of mathematics needed than the simplest elementary principles of algebra, geometry, and trigonometry, with a thorough, practical acquaintance with the use of the logarithmic table. To effect this result, derivations of rules and formulas are omitted, but thorough and complete instructions are given regarding how, when, and under what circumstances any particular rule, formula, or process should be applied; and whenever possible one or more examples, such as would be likely to arise in actual practice—together with their solutions—are given to illustrate and explain its application.

In preparing these textbooks, it has been our constant endeavor to view the matter from the student's standpoint, and to try and anticipate everything that would cause him trouble. The utmost pains have been taken to avoid and correct any and all ambiguous expressions—both those due to faulty rhetoric and those due to insufficiency of statement or explanation. As the best way to make a statement, explanation, or description clear is to give a picture or a diagram in connection with it, illustrations have been used almost without limit. The illustrations have in all cases been adapted to the requirements of the text, and projections and sections or outline, partially shaded, or full-shaded perspectives have been used, according to which will best produce the desired results. Half-tones have been used rather sparingly, except in those cases where the general effect is desired rather than the actual details.

It is obvious that books prepared along the lines mentioned must not only be clear and concise beyond anything heretofore attempted, but they must also possess unequaled value for reference purposes. They not only give the maximum of information in a minimum space, but this information is so ingeniously arranged and correlated, and the indexes are so full and complete, that it can at once be made available to the reader. The numerous examples and explanatory remarks, together with the absence of long demonstrations and abstruse mathematical calculations, are of great assistance in helping one to select the proper formula, method, or process and in teaching him how and when it should be used.

The numerous questions and examples, with their answers and solutions, which have been placed at the end of each volume, will prove of great assistance to all who consult the Library.

The first part of this volume treats on the construction, installation, care, and management of steam engines, both simple and compound. The various types of engines are discussed and explained, and the indicator and indicator diagrams are very fully treated. The treatment accorded to the subject of steam engines is amply sufficient to enable a fireman or engineer to pass any of the usual examinations preliminary to the granting of a license. Special attention is called to the latter portion of this volume, which contains a thorough treatment of the various types of elevators now in use. Full descriptions are given regarding their construction, care, and management, and this is believed to be the first attempt in print to give a connected account of this indispensable adjunct to the modern office building.

The method of numbering the pages, cuts, articles, etc. is such that each subject or part, when the subject is divided into two or more parts, is complete in itself; hence, in order to make the index intelligible, it was necessary to give each subject or part a number. This number is placed at the top of each page, on the headline, opposite the page number; and to distinguish it from the page number it is preceded by the printer's section mark (§). Consequently, a reference such as § 16, page 26, will be readily found by looking along the inside edges of the headlines until § 16 is found, and then through § 16 until page 26 is found.

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W

THE STEAM ENGINE.

INTRODUCTION.

CLASSIFICATION OF STEAM ENGINES.

1. The great number of the types of the steam engine may be classified as follows:

1. According to the kind of service, as *stationary*, *locomotive*, *marine*, *etc.*

2. According to number and arrangement of cylinders, as *simple*, *compound*, *triple expansion*, *quadruple expansion*, *duplex*, *etc.*

3. According to the type of valve used to control the distribution of steam, as *plain slide valve*, *automatic cut-off*, *Corliss*, *etc.*

4. According to the motion of the piston, as *reciprocating*, *rotary*.

Each of these types may be horizontal or vertical, condensing or non-condensing, and, except the rotary engine, single-acting or double-acting.

2. All the different types of reciprocating engines involve essentially the same principles, and therefore the description of a single type will be sufficient to give a general knowledge of these principles. For this purpose we shall choose the simple slide-valve engine, which is the engine in most common use.

THE RECIPROCATING STEAM ENGINE.

3. Steam may do work by acting on a piston working in a cylinder so as to lift weights or overcome the pressure of the atmosphere; in most cases, however, the work can best be done through the action of shafts and wheels having a continuous rotary motion; it is therefore essential that some

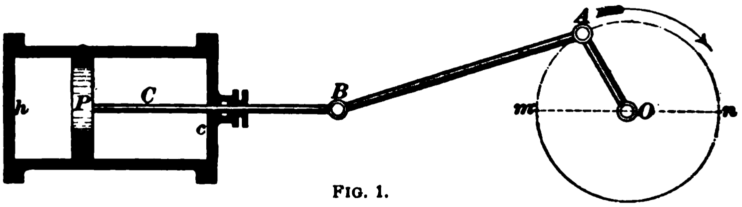


FIG. 1.

method of changing the to-and-fro, or reciprocating, motion of the piston into a continuous rotary motion should be devised. The form of mechanism used for this purpose in practically all types of reciprocating engines is shown in diagrammatic form in Fig. 1.

The steam from the boiler enters one end—say, in this case, the end h—of the **cylinder**, and pushes the piston to the other end. By means of another mechanism, called the **valve**, the steam is now admitted to the end c of the cylinder, while the end h is at the same time allowed to communicate with the atmosphere or with a condenser. The steam in h escapes, while that in c pushes back the piston to its original position, whence the same operation is repeated.

Attached to the piston, and forming a part of it, is the piston rod $C B$; to the end of $C B$ is fastened, by a joint, one end of the link $B A$. The other end of $B A$ is joined to the link $A O$; and the other end of $A O$ terminates in a shaft O, which is fixed in stationary bearings. It is evident that the end of $B A$ which is attached to $C B$ can move only in a straight line; and since the shaft O can only rotate in its bearings, the end of $A O$ which is attached to $B A$ can move only in a circle.

4. When the piston is at one extreme end of the cylinder, say at h, the joint A is at the point m, and all three links, $A\,O$, $B\,A$, and $C\,B$, lie in a straight line. As the piston moves to the right, the link $C\,B$ moves also to the right, while the joint A must move in a semicircle $m\,n$. When P arrives at the other end of the cylinder, the joint A is at n, and again $A\,O$, $B\,A$, and $C\,B$ are in a straight line. The piston now moves back to the end h of the cylinder, the joint A moving in the other semicircle from n to m.

The link $A\,O$ is called the **crank,** $B\,A$ the **connecting-rod,** and $C\,B$ the **piston rod.** Those parts that have a to-and-fro, or reciprocating, motion are called the **reciprocating parts.**

The end h of the cylinder is called the **head end,** and the end c the **crank end.** The distance passed over by the piston during half a revolution of the crank is called the **stroke,** and is plainly equal to the diameter of the circle described by the end of the crank—that is, the distance $m\,n$.

5. The engine may run in the direction shown by the arrow in the figure or it may run in the reverse direction. In the former case it is said to *run over* and in the latter case to *run under*.

The stroke from the head end to the crank end of the cylinder, that is, from left to right in the figure, is called the **forward stroke;** the one from crank end to head end, the **return stroke.**

The above simple mechanism perfectly fulfils the office of giving a continuous rotary motion in one direction. A pulley is keyed to the shaft O and the power is transferred by belting from the pulley to shafting or directly to the machinery to be run.

CONSTRUCTION OF A PLAIN SLIDE-VALVE ENGINE.

6. In Fig. 2, the construction of a **plain slide-valve engine** is shown, and in Fig. 3 is shown a section of a steam cylinder. Referring to these figures, H is the head end and C the crank end of the steam cylinder; B and B' are the steam ports; D is the steam chest; E is the exhaust

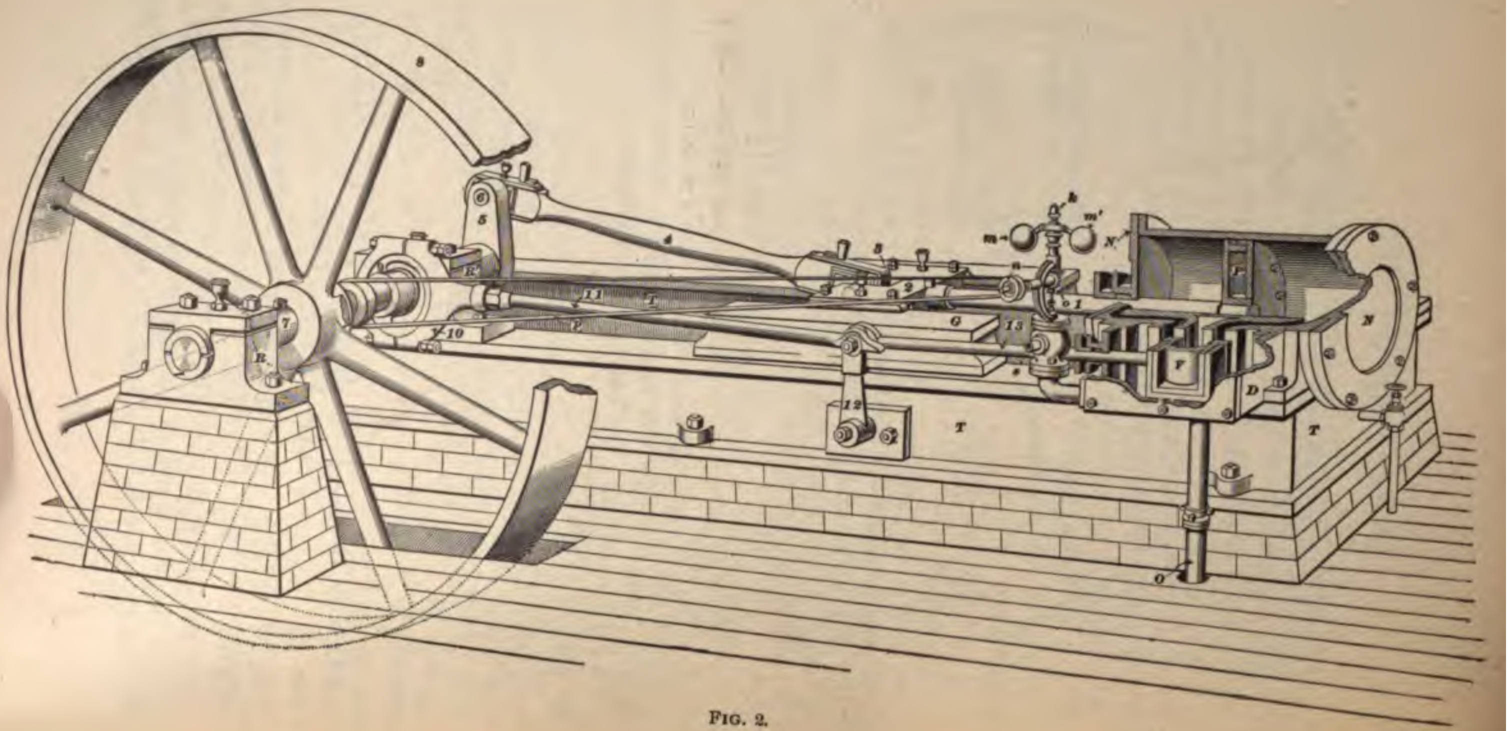


FIG. 2.

t; N and N' are the cylinder heads; S is the steam-supply pe; O is the exhaust pipe, and connects with the exhaust rt E; G is one of the two guide bars; R and R' are the aft bearings; and T is the bed, or frame, of the engine. he above are all **stationary parts** of the engine, or parts hat do not change their relative positions when the engine in motion. P is the piston; *1* is the piston rod; *2* is the rosshead; *3* is the crosshead pin, which is often called the wristpin; *4* is the connecting-rod; *5* is the crank; *6* is

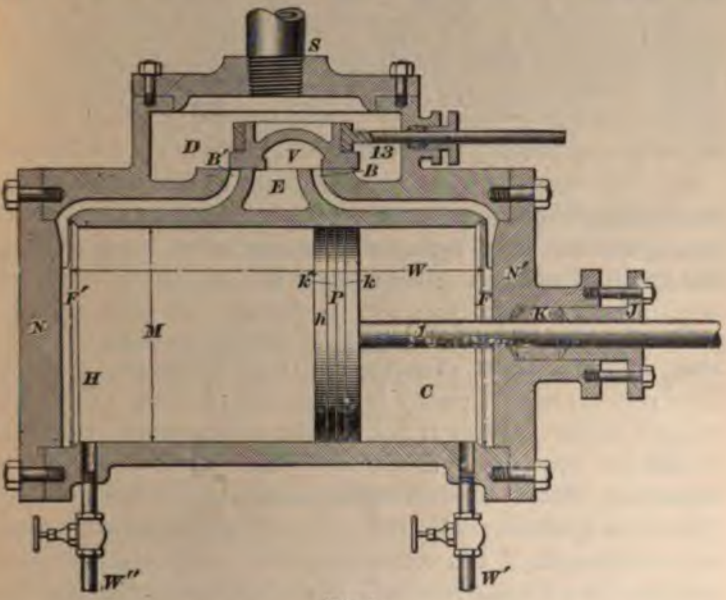


FIG. 3.

the crankpin; *7* is the crank-shaft; *8* is the flywheel; *9* is the eccentric; *10* is the eccentric strap; *11* is the eccentric rod; *12* is the rocker; *13* is the valve rod or stem; and V is the slide valve. These are all movable parts of the engine, or parts that change their relative positions when the engine is in motion. The eccentric, eccentric strap, eccentric rod, rocker, valve stem, and slide valve, which form the mechanism by means of which the steam is distributed, when considered as a whole, are termed the **valve gear** of the engine.

7. The working length of the cylinder is shown by the dimension line *W*. It is slightly less than the distance between the cylinder heads, since a small space must be left between the head and the piston, when the latter is at the end of its stroke. The stroke of the engine is the travel of the piston *P*; since the piston and crosshead are rigidly fastened to the same rod, the stroke must also be equal to the travel of the crosshead. It was shown in Fig. 1 that the stroke is also equal to the diameter of the circle described by the crankpin *6*, or, what is the same thing, it is equal to twice the length of the crank *5*, this length being measured from the center of the crankpin *6* to the center of the crankshaft *7*. The diameter or bore of the cylinder is represented by *M*.

8. The **size** of an engine is generally expressed by giving the diameter of the cylinder and the stroke in inches. Thus, an engine having a cylinder diameter of 16 inches and a stroke of 22 inches is called a 16″ × 22″ engine.

9. At the ends *F* and *F'* the cylinder is *counterbored*—that is, for a short distance the bore is greater than *M*. The piston projects partly into this counterbore at the end of each stroke. Were it not for the counterbore, the piston would not wear the cylinder walls their entire length, and shoulders would be formed at each end of the cylinder. When the wear of the joints in the connecting-rod is taken up, the length of the connecting-rod is slightly increased, and the piston is shoved back slightly towards the head end of the cylinder. In this case, the shoulder would cause an undesirable pounding of the piston.

Drain cocks *W''* and *W'''*, Fig. 3, are fitted to each end of the cylinder, through which any condensed steam may be discharged.

10. The piston fits loosely in the cylinder and has split rings *k* and *k'* inserted, which spring out so as to press against the walls of the cylinder and prevent leakage of steam between the wall of the cylinder and piston. Pistons are usually supplied with a follower plate *h*, which is bolted to the head end of the piston *P* in order to hold these split rings *k* and *k'* in place. The piston rod *1* is a round bar rigidly connected to both the piston *P* and the crosshead *2*.

11. A stuffingbox *K* in which packing is placed is fitted with a gland *J*, which, when bolted down, compresses the packing around the piston rod *1* and makes a steam-tight joint. This packing is usually made in the form of split rings, which are so placed that the split of the first ring is covered by the solid part of the next ring. When repacking, care should be taken not to cause unnecessary friction by too much pressure from the gland. The guide bars, as *G*, Fig. 2, constrain the crosshead *2* to move exactly in line with the axis of the cylinder, thus relieving the piston rod of all bending stresses.

The connecting-rod *4* forms the connecting link between the crosshead and crank *5*. The joint between crosshead *2* and connecting-rod *4* is made by the crosshead pin *3*, and that between the connecting-rod and crank by the crank-pin *6*. Connecting-rods are usually made from 4 to 6 times the length of the crank, or from 4 to 6 "cranks" in length.

THE ECCENTRIC.

12. Fig. 4 shows the eccentric that imparts motion to the slide valve *V* in Figs. 2 and 3. It consists of a circular disk of iron *a*, which is keyed or fastened by setscrews to the shaft and revolves with it. The center of this disk, which is called the **eccentric sheave,** is at *O*. It is evident that, as the shaft revolves, the center *O* of the sheave *a* will describe the dotted circle *b*, whose center is the center of the shaft. Consequently, the eccentric strap *c* and the eccentric rod *d*, to which it is fastened, will be moved horizontally, during a half revolution, a distance equal to the diameter *e* of the dotted circle. This distance *e* is commonly called the **throw** of the eccentric. The distance *O Q* between the center of the eccentric and center of the shaft is called the **radius** of the eccentric or the **eccentricity.** It is plain that the throw is *twice* the radius. Attention is here called to the fact that practice varies somewhat in the definition of

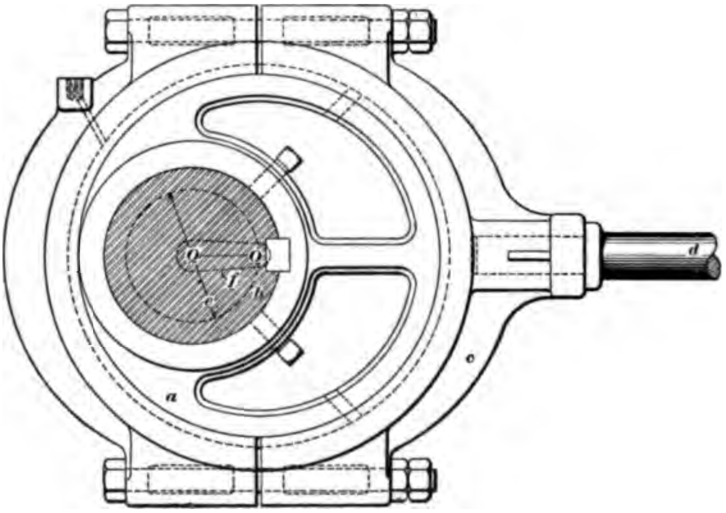


Fig. 4.

the term *throw*. Some engineers call the radius the throw, but by far the greater part define throw as here given.

13. The eccentric is equivalent to a crank whose length is equal to the radius of the eccentric. Thus, if the end of the eccentric rod *d* were attached at *O* to the crank *f* (shown in dotted lines), the crank would give the same motion to the rod that the eccentric does. In plain slide-valve engines, the eccentric is usually keyed to the shaft after being properly adjusted.

The connection between the eccentric rod *11*, Fig. 2, and the valve stem *13* is accomplished in a variety of ways. In Fig. 2, a rocker-arm *12* is used to support the joint between the eccentric rod *11* and the valve stem *13*. The latter must be supported in some manner to prevent its binding in its stuffingbox.

14. Motion of Eccentric and Valve.—As the motion of the valve is given by the eccentric, the valve is in mid-position in a horizontal engine when the radius of the eccentric is in a vertical position. When $Q\,O$, Fig. 4, lies horizontally on the right side of Q, the valve V (see Fig. 2) is in its position nearest the head end of the steam chest, and when $O\,Q$ lies horizontally on the left side of Q, the valve is at the end of its stroke towards the crank end of the steam chest.

THE D SLIDE VALVE AND STEAM DISTRIBUTION.

ACTION OF SLIDE VALVE.

15. Description of the Slide Valve.—Of the different kind of valves used to distribute the steam in the engine cylinder, the **D** slide valve is the most common. A section of such a valve is shown in Fig. 5 in its central position;

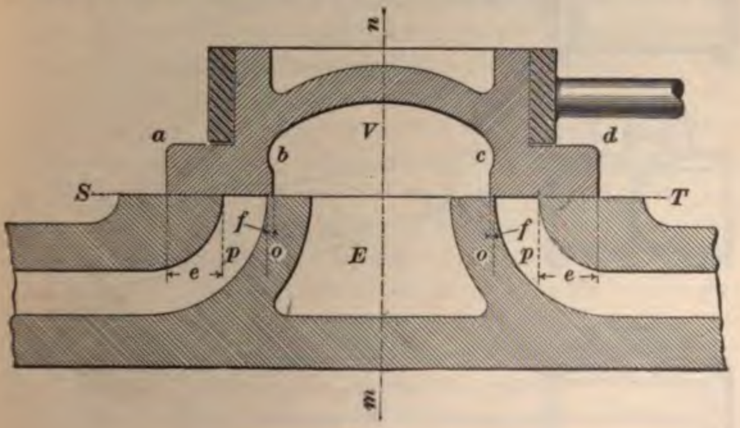


FIG. 5.

p, p are the **steam ports,** o, o the **bridges,** E the **exhaust port,** $S\,T$ the **valve seat.** The flanges of the valve, $a\,b$ and $c\,d$, are seen to be wider than the ports that they cover. Of this extra width, the parts e, e are called the

outside lap, and the parts *f, f* the **inside lap.** The valve is here shown in mid-position, i. e., the center line *n* of the valve coincides with the center line *m* of the exhaust port.

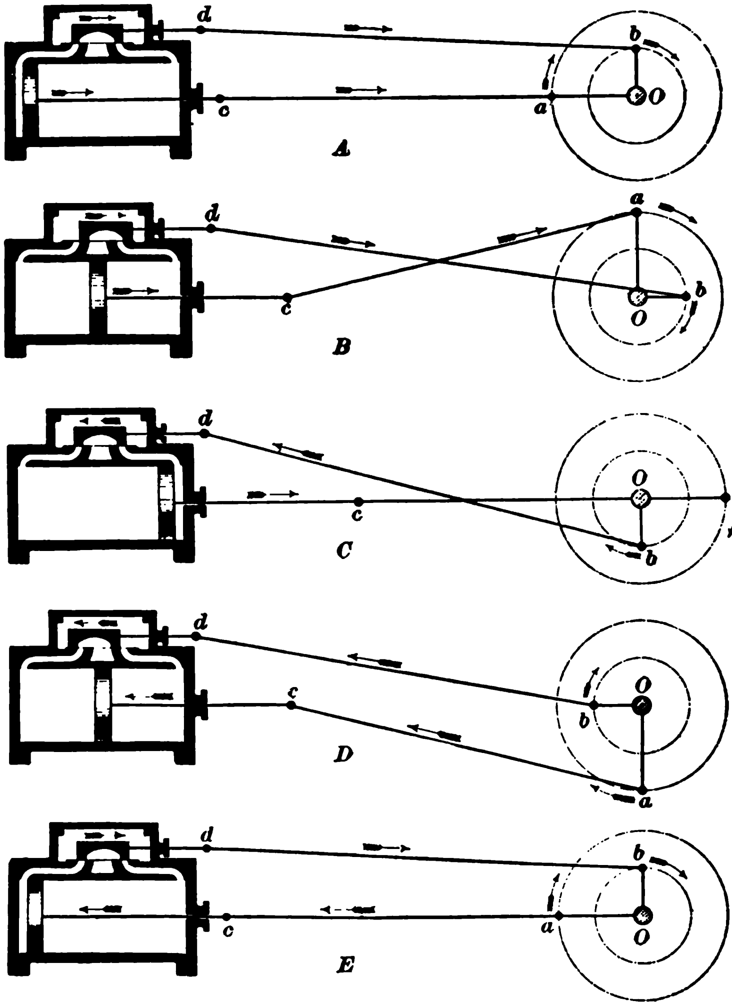


FIG. 6.

16. Action of Valve Without Lap or Lead.—Fig. 6. shows five diagrams that represent a **D** slide valve without lap or lead. *Oa* represents the crank; *Ob* the eccentric, which was shown to be equivalent to a crank; *ac* the connecting-rod; and *bd* the eccentric rod. It should be remembered that the relative sizes of some of the parts have been greatly exaggerated, particularly the radius of the eccentric circle and the amount of clearance. Diagram *A*, Fig. 6, represents the piston just on the point of beginning the forward stroke. The valve is moving in the direction of the arrow, and the outer edge is just about to admit steam to the left-hand port. As will be seen, the valve is in its central position, and, consequently, the line joining the center of the shaft and the center of the eccentric (this line will hereafter be called the **eccentric radius**) is vertical. All the parts are about to move in the direction of the arrows. Diagram *B* shows the positions of the parts when the crank has moved through 90° from its position in *A*. The piston is at the middle of its stroke, or very nearly there. It would be exactly at the middle of its stroke but for the fact that the connecting-rod makes an angle with the horizontal. It will be assumed here that this has no effect on the position of the piston. The valve has reached the extreme limit of its travel to the right and the eccentric radius *Ob* is horizontal. The left steam port is fully opened for the live steam and the right steam port is fully opened for the exhaust. Another crank movement of 90° places the different parts as shown in diagram *C*. The piston has reached the end of its forward stroke; the valve is in its central position moving towards the left, and having just closed the left steam port and the right exhaust port, is about to open the right port for the admission of live steam and the left port for the release of exhaust steam. The piston has now traveled one full stroke. Diagram *D* shows the piston in its central position on the return stroke. The crank is in the position *Oa*; the eccentric is horizontal, as represented by *Ob*, and the valve is at the farthest point of its travel to the left, the right port being fully open for live steam and the left port fully open for exhaust. In the diagram *E* the piston has reached the extreme point of the return stroke, the piston rod, connecting-rod, and crank being all in one straight line; this also occurs in diagrams A (which is the same as E) and C. The valve has been moving to the right and is now in its central position, just on the point of admitting steam to the left port.

These diagrams show that, with no lap or lead, the steam is admitted to the cylinder for the full stroke of the engine; consequently, there can be no cut-off, and therefore no expansion of steam.

The following conclusion is now evident: *With an ordinary* **D** *slide valve, operated by an eccentric, there can be no cut-off, and therefore no expansion of steam, unless the valve has outside lap.*

17. Lap and Angle of Advance.—The effect of lap on the relative movement of the valve and piston, and also on the movement of the eccentric and crank, is clearly shown in Figs. 7 to 14. In these figures, the valve has both outside and inside lap, but no lead. These diagrams have been distorted, as was done in Fig. 6, in order that the eccentric radius might be long enough to show up well. In Figs. 7 to 14 the eccentric radius is three times as long as it should be for the amount of valve movement shown by the figure. The diameter of the crank circle is also a little greater than the stroke of the piston, for the same reason.

18. Diagram of Pressures in Cylinder.—In order to show the distribution of steam by the valve, a diagram has been drawn above and below each cylinder, those above being marked M and those below N. These diagrams are supposed to be drawn in the following manner: Imagine it to be possible to connect two small pipes to the piston, one on each side. Suppose each pipe to have a steam-tight piston working in it, the lower side of the pistons being subjected to the steam pressure in the cylinder and the upper side to the atmospheric pressure. Suppose, further, that there is a coiled spring on top of the piston; that a piston rod passes through the center of the spring; and that a pencil is attached to the end of the piston rod. If a pressure of 10 pounds is required to compress the spring 1 inch, it is evident that for every 10 pounds pressure in the cylinder, the pencil will move upwards 1 inch, and if it touched a sheet of paper, would mark a line on that paper. It will now be presumed that an arrangement like that just described is attached to the steam-engine piston, and that the pencil touches a sheet of paper that is held stationary. Then, when the steam-engine piston moves ahead, the pencil will make straight lines at heights corresponding to the steam pressure on the under sides of the little pistons, except when the pressure of the steam in the cylinder varies, in which case the pencil will move up or down, according as the pressure increases or diminishes.

19. Having made these suppositions clear, let QX, Figs. 7 to 14, represent the line that the pencil would trace if there were a perfect vacuum in the cylinder; i. e., QX is the line of no pressure; also let AB represent the line the pencil would trace if the pressure in the cylinder was just equal to that of the atmosphere, and QY the line of no volume. Then the point Q represents no volume and no pressure. Finally, let $1D$ represent the volume of clearance.

20. Angle of Advance.—Consider Fig. 7. The piston is represented as just beginning the forward stroke and the valve as just commencing to open the left steam port, both moving in the same direction, as shown by the arrows. If the valve had no outside lap, the position of the eccentric center would be at e, but on account of the lap, the valve had to be moved ahead of its central position in order to bring its edge to the edge of the port. To accomplish this, the eccentric center has been moved from e to b, Ob being the position of the eccentric radius. The angle bOe that the eccentric radius makes with the position it would be in if there were no lap or lead is called the **angle of advance.**

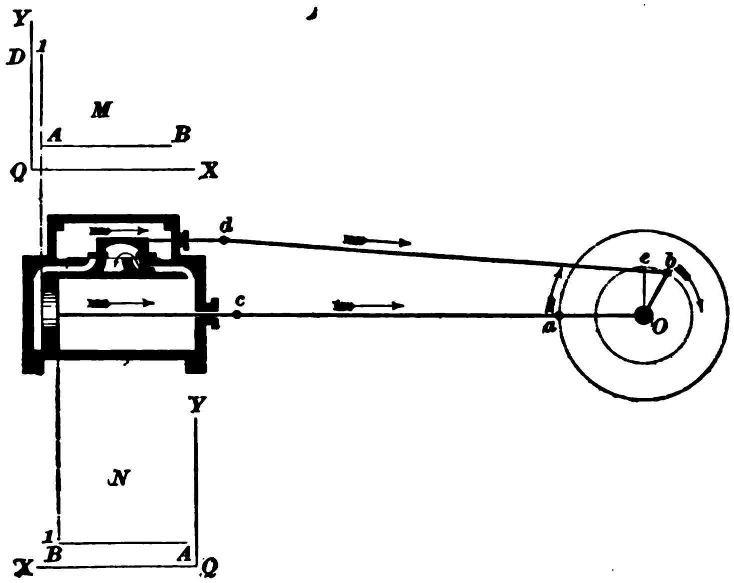


FIG. 7.

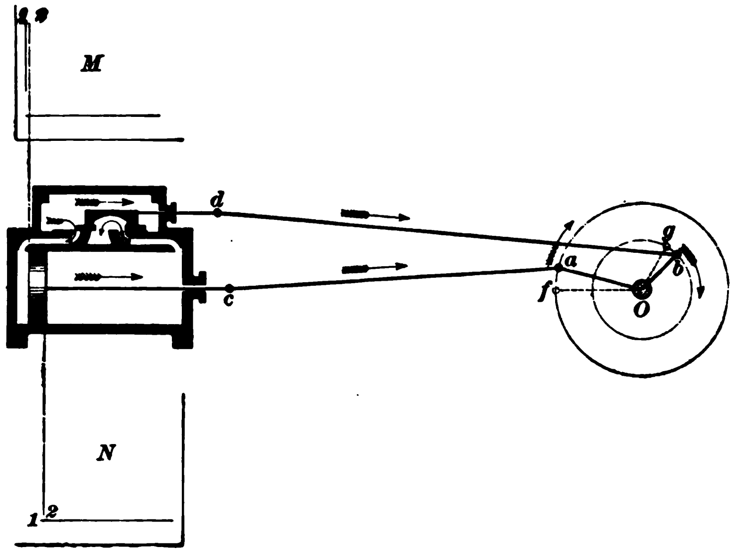


FIG. 8.

21. Back Pressure.—Assume that the piston and valve have moved a very small distance, just sufficient to admit steam to fill the clearance space on the left of the piston, so that the steam acts on the piston at full boiler pressure. If the length of the line *A 1* represents the boiler pressure (gauge), the pencil that registers the pressure on the left side of the piston will be at *1*. The steam on the right side of the piston is flowing (exhausting) into the atmosphere through the exhaust port, as shown by the arrow. As the size of the exhaust port is limited by practical considerations, the exhaust is not perfectly free, and there is a slight pressure on the exhaust side of the piston in addition to the atmospheric pressure. This is termed **back pressure.** Therefore, in the diagram *N*, let *1* be the position of the second pencil; then, *1 B* is the back pressure.

22. Exhaust Port Fully Open.—Fig. 8 shows the position of the piston and valve when the exhaust port is fully open. The crank has moved from the position *O f* (shown by dotted line) to *O a* and the eccentric center from *g* to *b*. Steam is entering the head end of the cylinder and exhausting at the crank end. The pencils have moved from *1* to *2* on both diagrams *M* and *N*.

23. Valve at the End of Its Stroke.—In Fig. 9, the piston has advanced far enough to enable the valve to reach the end of its stroke and open the port its full width. The crank and eccentric have moved to the positions *O a* and *O b*, the dotted lines showing their last position in Fig. 8. The eccentric radius is horizontal, and any further movement of the crank will cause the eccentric to travel in the lower half of its circle and make the valve move back. In the diagrams *M* and *N*, the pencil has traced the lines *2–3*.

24. Valve on Return Stroke, Steam Port Partly Closed.—Fig. 10 shows the piston still farther advanced on its stroke and the valve as having its inside edge in line with the outside edge of the exhaust port. The left end of

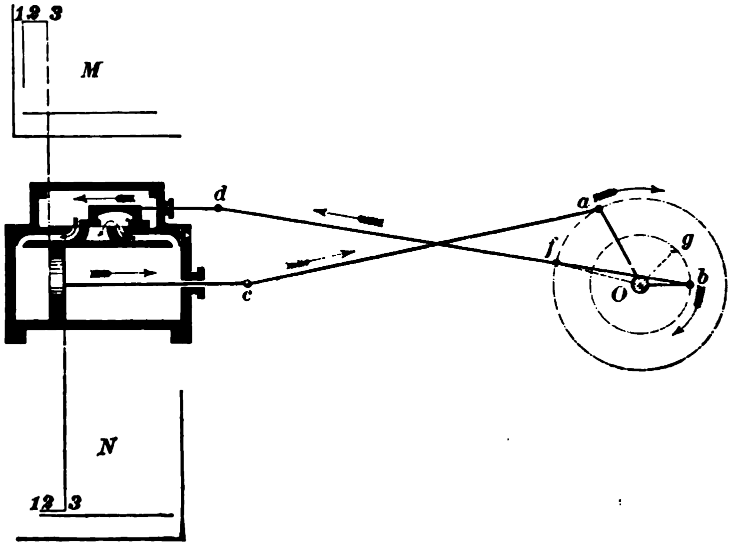


FIG. 9.

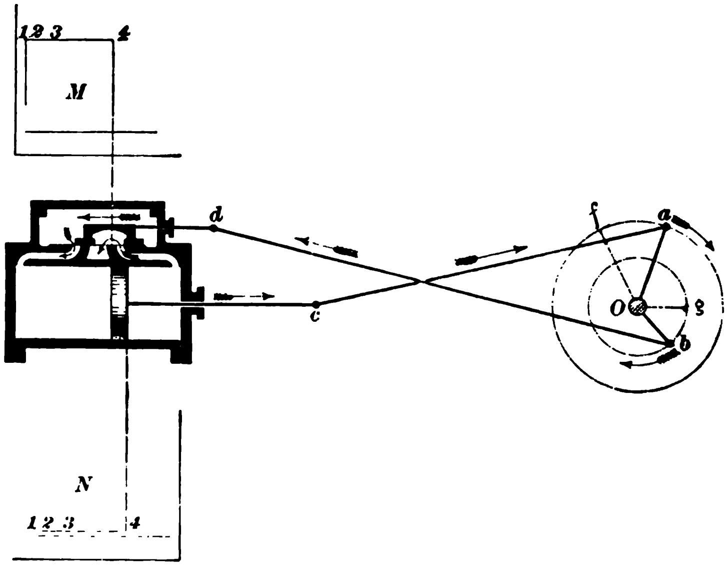


FIG. 10.

the valve has partially closed the steam port. The amount of advancement of the crank and eccentric from their last positions is shown by their distances from the dotted lines. The pencils have traced the lines *3–4* on the diagrams *M* and *N* during this movement of the piston from the last position.

25. Point of Cut-Off.—Fig. 11 marks one of the most important points of the stroke. Here the valve has closed

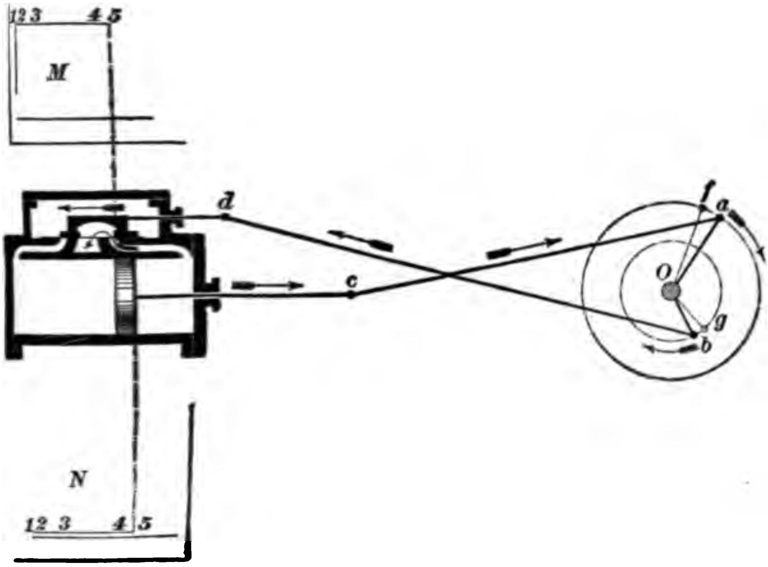


FIG. 11.

the steam port, i. e., cut off the steam, and from here to the end of the stroke the steam in the cylinder expands. This is called the **point of cut-off.** The exhaust port is now partially closed. The crank and eccentric have moved through the angles indicated. During this movement, the pencils have traced the lines *4–5*.

26. Point of Compression.—Fig. 12 shows another very important valve position. Here the inside edge of the valve closes the exhaust port, and from now to the end of the stroke, the steam in front of the piston is compressed. This point in the stroke is called the **point of compression.**

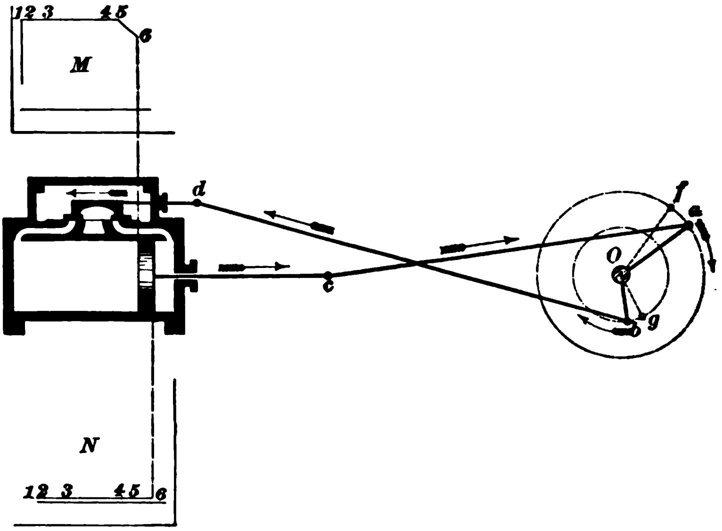


FIG. 12.

In the diagrams *M* and *N*, the lines *5–6* are traced by the pencils. The line *5–6* on the diagram *M* is an expansion line, the pressure falling as the piston moves ahead.

27. Point of Release.—In Fig. 13, the piston has advanced far enough to cause the left inside edge of the valve to be in line with the inside edge of the left port. The slightest movement of the valve to the left will open the left port so that the steam in the left end of the cylinder will pass into the exhaust port. This point of the stroke is called the **point of release.** The work done by expansion theoretically ends here, although, on account of the limitation in the size of the ports, there will still be a slight amount of work done by expansion, owing to the inability of the steam to escape instantly. During this last movement of the piston, the pencils trace the lines *6–7* on the diagrams *M* and *N*. On the diagram *M* the line *6–7* is a continuation of the expansion line *5–6*, while in the dia-

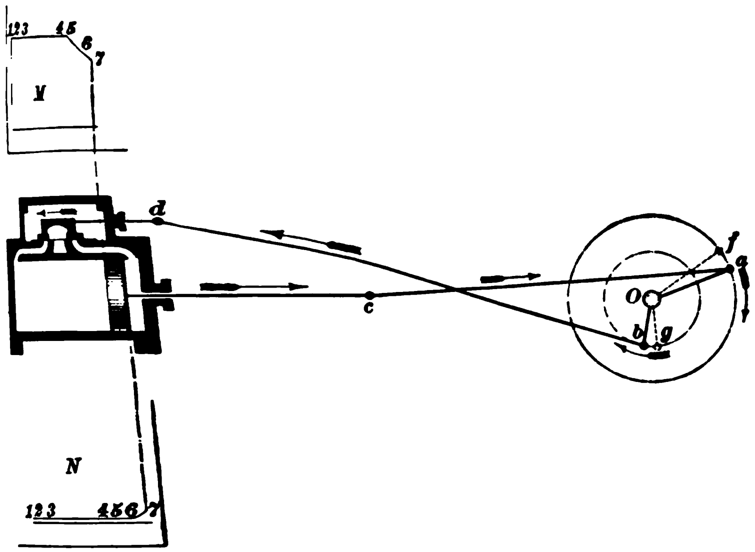


FIG. 13.

gram *N* it shows part of the compression line, the pressure rapidly increasing as the piston nears the end of the stroke.

28. Piston at End of Stroke.—In Fig. 14, the piston has reached the end of its forward stroke and is about to begin the return stroke. The right outside edge of the valve is in line with the outside edge of the right port. The steam is exhausting from the head end of the cylinder, as shown by the arrows. The crank and eccentric are both diametrically opposite their positions in Fig. 7. In the diagrams *M* and *N*, the pencils have traced the lines *7–8*. *M* shows that the pressure has fallen very rapidly from *7* to *8*, while in *N* it has risen from *7* to *8*. The very slightest movement of the piston to the left will admit steam to the crank end of the cylinder and cause the pencil to rise to the point *1'*.

During the return stroke the above-described actions of the steam will be repeated, the pencils tracing the dotted lines on the diagrams *M* and *N* in Fig. 14 and the exhaust going through the left port and the steam through the right

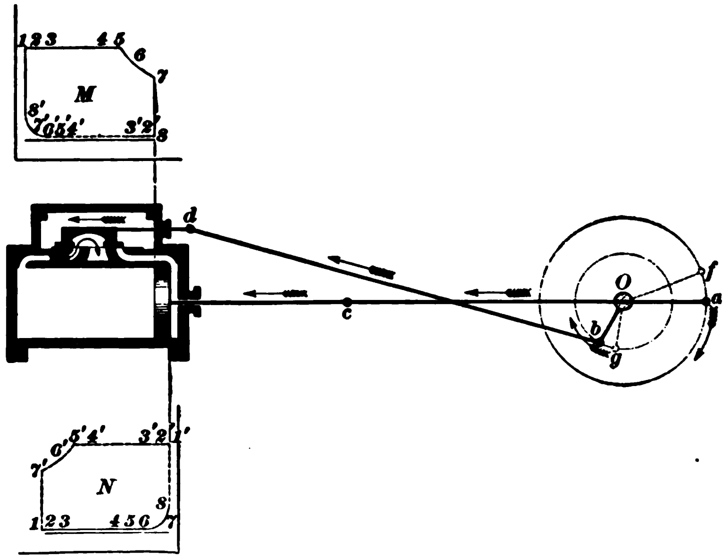


FIG. 14.

port. As the process is so nearly like the preceding, the diagrams have not been drawn, but the student should follow the valve through the different positions and note the effects on the diagrams. To assist him in this, the corresponding points have been numbered as in the foregoing figures.

29. Effects of Lap.—A study of Figs. 7 to 14 should show the effects caused by varying the lap. Thus, in Fig. 11, it is evident that if the outside lap had been less, the valve would not close the left port when its center was in the position shown; consequently, the piston must move farther ahead before the valve can move back far enough to close the port. This, of course, makes the cut-off take place later in the stroke and shortens the expansion. It is likewise evident that if the valve had more lap, this extra lap would extend beyond the port when the center of the valve was in the position shown. Therefore, the valve would cut off earlier in the stroke and the expansion would be lengthened. Hence, *increasing the outside lap means an earlier cut-off and an increasing expansion, while decreasing the outside lap means a later cut-off and a diminished expansion.*

30. Considering the inside lap, it is evident from Fig. 12 that if the inside lap had been less, the exhaust port would not have closed so soon, and consequently the compression would have been later; had the inside lap been greater, the compression would have begun earlier. Fig. 13 shows that with a diminished inside lap, the release would begin earlier, while with an increased inside lap, the release would have taken place later in the stroke. *Increasing the inside lap causes the compression to begin earlier in the stroke and causes the release to take place later. On the other hand, diminishing the inside lap causes the compression to begin later and the release to take place earlier in the stroke.*

31. Lead.—In Fig. 7 the piston is just commencing the forward stroke and the valve is just about to uncover the

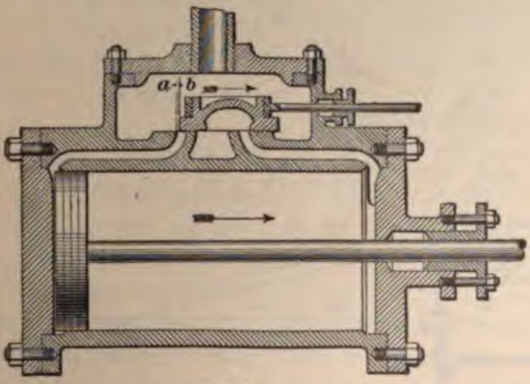


FIG. 15

left steam port. Most engineers, however, prefer to have the port a little open when the piston is at the end of the stroke. That is, the valve, instead of being just at the edge of the port, as shown in Fig. 7, is moved $\frac{1}{16}$ inch or $\frac{1}{8}$ inch to the right, so that the clearance space is filled with fresh steam before the piston begins its stroke. A valve with lead is shown in Fig. 15. Here the piston is at the end of the stroke and the port is open a distance *a b*. This distance *a b* is the **lead.**

Since, when a valve has lead, it is moved farther to the right than in the position shown in Fig. 7, it is evident that the center *b* of the eccentric must also be moved a little farther to the right, Fig. 7. That is, to give a valve lead, the angle of advance must be increased.

32. Position of the Eccentric.—When the plain slide valve has neither lap nor lead, as in the skeleton diagrams, Fig. 6, it was shown that the eccentric must make an angle of 90° with the crank. Further, when the engine "runs over," as in Fig. 6, the eccentric is *ahead* of the crank. That is, following the direction of the arrows, the eccentric *b* reaches any point on its circle a quarter of a revolution before the crank *a* does. Referring now to Figs. 7 to 14, it is seen that when the valve has lap (or lap and lead), the angle *a O b* between the crank and eccentric is greater than 90°. Following the direction of the arrows, it is seen, however, that the eccentric *b* reaches, say, the lowest point on the circle earlier than the crank *a* reaches the lowest point on its circle. That is, the eccentric is *ahead* of the crank, as in the above case.

Take now the case of an engine that "runs under," as shown in Fig. 16. The crank is in position *a* and is about

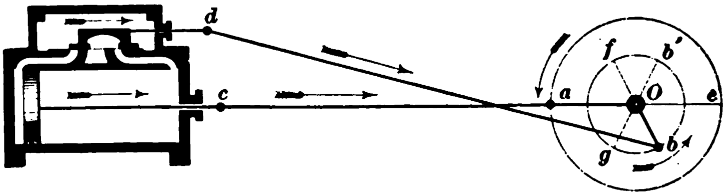


FIG. 16.

to move downwards. Now, the eccentric cannot be in the position *O b'*, for then it would move the valve to the left.

It cannot be opposite, in the position $O g$, for in that case, the valve would not be far enough to the right. It must be in the position $O b$. An inspection of the diagram shows that, following the direction of the arrows, the eccentric is set *ahead* of the crank, and the angle between the crank and eccentric is $a O b = 90° +$ the angle of advance.

Hence, for the ordinary slide valve, the following general direction applies: *When the valve rod and eccentric rod move in the same direction, the eccentric is set* **ahead** *of the crank, and the angle between the crank and eccentric is 90° + the angle of advance.* This law is true whether the engine runs "over" or "under."

33. Rocker-Arms.—It frequently happens that the eccentric cannot be so located on the shaft (in the direction of its length) that the eccentric rod and valve stem will be in the same straight line. It can never be done when the valve is on top of the cylinder without inclining the valve seat, now very seldom done, and with the valve on the side of the cylinder, other considerations, such as the location of the flywheel, may interfere. In such cases as this, a lever or rocker-arm may be used.

An example is shown in Fig. 2. It is perfectly evident that when the eccentric rod *11* moves to the left, the valve rod *13* will also move to the left, being compelled to do so by reason of the rocker-arm *12*. It is also plain that the amount of horizontal movement of the valve rod will be the same as it would be if the eccentric were attached directly to the valve rod, thus getting rid of the rocker-arm. The reason for using the rocker-arm in this case is that the eccentric-rod axis and valve-stem axis are not in the same straight line, the eccentric then being thrown too far to the left by the main bearing R'. The valve seat could, in this case, have been placed farther from the center of the cylinder, so as to bring the axes of the two rods in line. This, however, would have made the steam and exhaust ports that much longer. Since it is considered an advantage to have ports as short as possible, a rocker-arm was used.

34. Again, it is sometimes desirable to make the throw of the eccentric less than the valve travel. This may be accomplished by the use of a rocker-arm, as shown in Fig. 17. This rocker is pivoted at g and rotates about that point as a center. The valve rod is joined to the rocker at the end e and the eccentric rod is joined at d, a point between e and g.

Then, the eccentric throw must be smaller than the valve travel by the ratio $g d : g e \left(= \frac{g d}{g e}\right)$. For example, suppose the valve travel to be 4 inches, the distance $g d = 12$ inches, and $g e = 15$ inches. Then, the throw of the eccentric $= 4 \text{ inches} \times \frac{g d}{g e} = 4 \times \frac{12}{15} = 3.2$ inches.

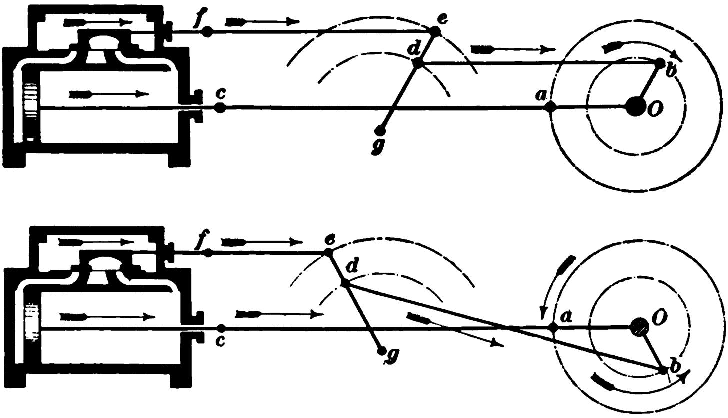


FIG. 17.

When the rocker is arranged as in Fig. 17, whether the engine runs over, as in the upper figure, or under, as in the lower figure, the valve rod and eccentric rod move in the same direction. Consequently, by the direction previously given, the eccentric is set 90° + angle of advance *ahead* of the crank.

35. It is often convenient to pivot the rocker near the center, as shown in Fig. 18. Here the points e and d, where the valve and eccentric rods, respectively, join the rocker, lie on opposite sides of the pivot g. As before, we have the

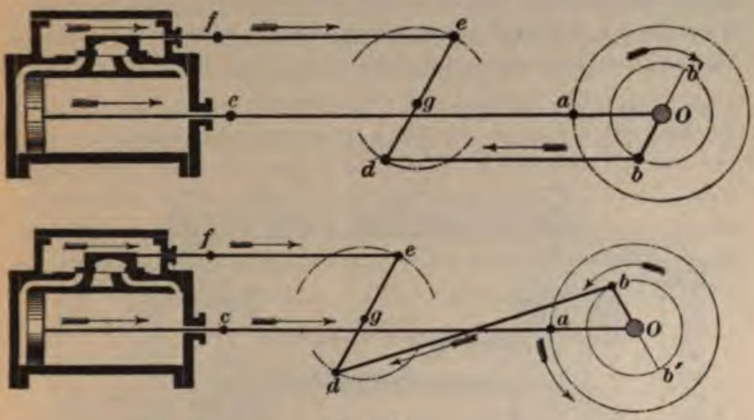


FIG. 18.

proportion—throw of eccentric : valve travel $= gd : ge$, or throw of eccentric $=$ valve travel $\times \frac{gd}{ge}$.

It is easily seen, however, that when the rocker is pivoted near the center, as in Fig. 18, the valve rod and eccentric rod move in opposite directions. Consequently, to give the valve the proper motion, the eccentric rod must at all times move in a direction exactly opposite the direction of a rod attached as shown in Fig. 17. This can only be accomplished by placing the eccentric exactly opposite the position shown in Fig. 17. That is, instead of placing the eccentric 90° + angle of advance ahead of the crank, it must be placed 90° − angle of advance behind the crank. We have, therefore, the general direction: *In the case of the plain slide valve, if a rocker is so pivoted as to make the valve rod and eccentric rod move in opposite directions, the eccentric must be placed* **behind** *the crank, and the angle between the two is 90° − angle of advance.*

36. Direct and Indirect Valves.—A slide valve is said to be **direct** when it opens the left port upon moving to the right and closes it by moving to the left. A valve is said to be **indirect** when it opens the left steam port by moving to the left and closes it by moving to the right.

The plain slide valve already described is a direct valve. It opens the left port by moving to the right, admits steam past the outside edge, and exhausts it past the inside edge.

37. The **piston valve** shown in Fig. 19 is an example of an indirect valve. It consists of a hollow cylinder sliding

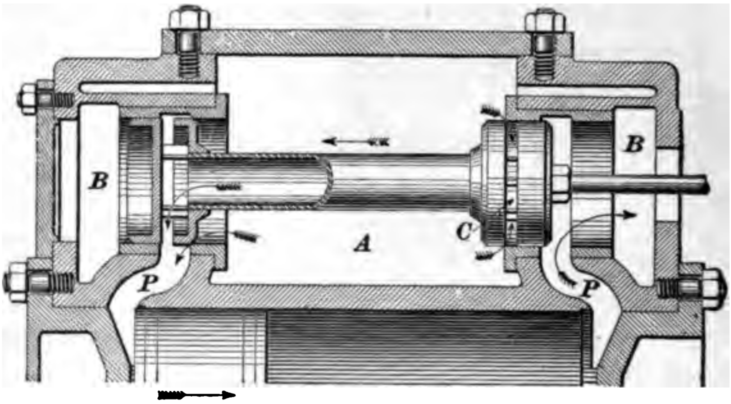


FIG. 19.

in a cylindrical valve seat. The ports *P*, *P* extend clear around the valve. The steam is admitted into the central chamber *A* and the exhaust steam escapes out of the two ends *B*, *B*. As shown in the figure, the piston is just about to start to the right and the valve is moving to the left, thereby uncovering the left steam port and allowing the steam to enter past its inside edge. The valve is, therefore, indirect. To give a larger admission, steam also passes into the center of the valve through the channel *C* and thence into the left port. The exhaust steam meanwhile escapes directly through the right steam port into the chamber *B*.

Attention is called to the fact that a piston valve is not necessarily an *indirect* valve; piston valves are often made as *direct* valves. In the latter case their action is exactly the same as that of the ordinary plain slide valve.

38. Eccentric Positions With Indirect Valves.—It is plain that the direction of motion of an indirect valve is precisely opposite that of a direct valve. Hence, as before explained, the eccentric must be set exactly opposite the position it would have were a direct valve used. We have, then, the following direction for the position of the eccentric: *When an indirect valve is used, set the eccentric* **behind** *the crank and make the angle between them equal 90° – the angle of advance. If a rocker is used that makes the valve rod and the eccentric rod move in opposite directions, set the eccentric* **ahead** *of the crank and make the angle between them equal to 90° + the angle of advance.* This rules applies whether the engine runs "under" or "over."

39. Table of Eccentric Positions.—The position of the eccentric relative to the crank for both the direct and indirect valves, direct and reversing rocker-arms, is given in the table. A rocker of the character shown in Fig. 17 will be called a *direct* rocker. One that changes the direction of the motion, as in Fig. 18, will be called a *reversing* rocker.

ECCENTRIC POSITIONS.

	Kind of Valve.	Kind of Rocker-Arm.	Angle Between Crank and Eccentric.	Position of Eccentric Relative to Crank.
I	Direct	Direct	90° + angle of advance	Ahead of crank
II	Direct	Reversing	90° – angle of advance	Behind crank
III	Indirect	Direct	90° – angle of advance	Behind crank
IV	Indirect	Reversing	90° + angle of advance	Ahead of crank

The above table may be applied equally well whether the engine runs over or runs under. It is simply necessary to remember that to set the eccentric ahead of the crank is to set it so that it reaches a given point in its revolution before the crank reaches the same point in its revolution. For example, in Fig. 16, suppose the engine to run under, as shown by the arrow. Then, the eccentric $O\ b$ is set ahead of the crank $O\ a$, because it will reach the line $O\ c$ before the crank will. If the eccentric were in the position $O f$, it would be behind the crank, because the crank would reach $O\ c$ first. If, now, the engine should be supposed to "run over," then, if the eccentric were in position $O\ b$ or $O\ g$, it would be behind the crank; if in position $O\ f$ or $O\ b'$, it would be ahead of the crank.

DISTURBANCE OF CUT-OFF BY THE CONNECTING-ROD.

40. In Fig. 20, let $a\ b$ represent the path of the center of the wristpin and $c\ d$ the circle described by the center of

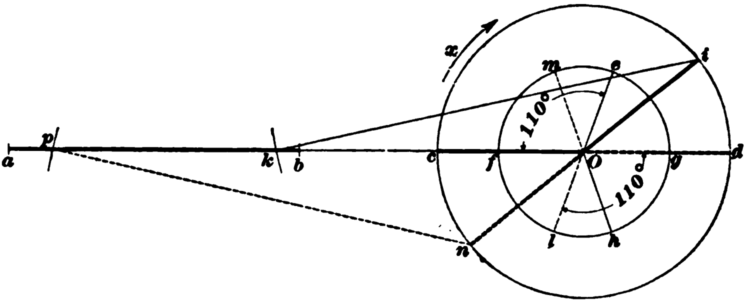


FIG. 20.

the crankpin. Let the diameter of the circle $f g$ equal the throw of the eccentric. (This is shown greatly exaggerated.) Assuming the crank to be in the position $O\ c$, that is, on the interior dead center, the length of the line $a\ c$ will represent the length of the connecting-rod. We shall assume that the angle of advance is 20°; further, that the slide valve is set so as not to have any lead. $O\ c$, then, is the position of eccentric when crankpin is at c. Now, let the crankpin move in the direction of the arrow x; that is, let the piston commence its forward stroke. Since the valve has no lead, the slightest movement of the crankpin in the direction of the arrow will cause the valve to open the left steam port. When the eccentric has reached the position $O\ g$, the valve has moved to its farthest position to the right, and any further movement of the crank will cause the valve to begin to close the steam port. To close the steam port fully, the valve will have to move the same distance to the left that it moved to the right to uncover the port. From this it follows that the eccentric must move through the same angle to close the port that it moved through to open the port. Laying off the angle $g\,O\,h = g\,O\,c$, $O\,h$ will represent the position of the eccentric at the time cut-off takes place. Laying off the angle $h\,O\,i = c\,O\,e$, we find the corresponding crank position. From the point i (the center of the crank-pin) as a center, with a radius equal to the length of the connecting-rod (the length of the line $a\,c$), describe an arc intersecting the line $a\,b$ at k; the point k will be the position of the center of the wristpin at the time of cut-off on the forward stroke. When the crank passes the exterior dead center, the right steam port will be opened; and at the moment that the crank occupies the position $O\,d$, the eccentric will be at $O\,l$; that is, $90° + 20° = 110°$ ahead of the crank. From what has previously been explained, it will be clear that the cut-off takes place on the return stroke when the eccentric reaches the position $O\,m$. The corresponding crank position will be $O\,n$. From n as a center, with a radius equal to the length of the connecting-rod, describe an arc intersecting $a\,b$ at p, which will give the position of the center of the wristpin at the time of the cut-off on the return stroke.

41. It will be seen at a glance that the cut-off has taken place considerably later on the forward stroke than on the return stroke, since $k\,b$ is less than $a\,p$. From this we see that a valve having an equal lap and set so as to have an equal lead cannot cut off equally on the forward and return stroke. If the valve is set so that the cut-off will be equal, the lead will be unequal.

This is due to the use of a connecting-rod. As a general rule, it may be stated that the longer the connecting-rod, the less will be the difference in the points of cut-off; and the shorter the connecting-rod, the greater the difference.

The effects of the connecting-rod on the steam distribution of a simple slide valve may be summarized as follows: *It will cause the valve to cut off and release the steam, as well as close the exhaust port, later on the forward stroke of the piston than on the return stroke.*

FORMS OF SLIDE VALVES.

42. Double-Ported Valves.—The plain **D** slide valve, shown in Fig. 5, is largely used on small engines running

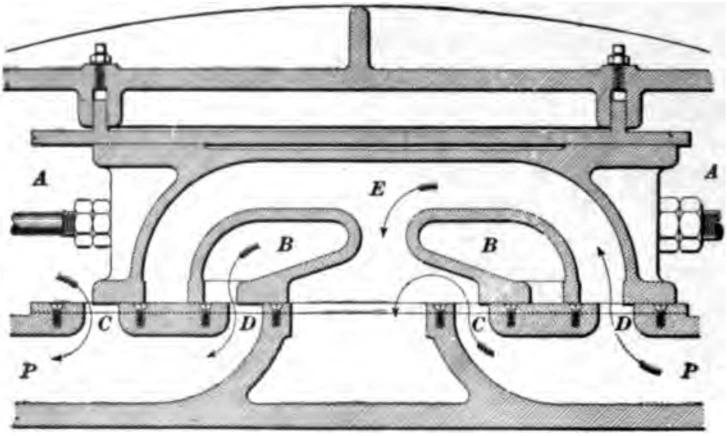


FIG. 21.

at moderately slow speeds. When, however, an engine has a high piston speed, the plain **D** valve does not open the port fast enough to allow the steam to follow up the moving piston and keep up full pressure in the cylinder. To overcome this difficulty, various forms of double-ported valves, one of which is shown in Fig. 21, are used. In Fig. 21, each port P has two openings C and D. The valve is made with two passages B, B extending through it; these passages connect with the steam chest A. In the position shown in the figure, the valve is opening the left steam port. The steam enters the passage C past the edge of the valve and enters the passage D through the opening in the chamber B. In the meantime, the exhaust is escaping from the right port into the chamber E beneath the valve. It is clear that, with the same travel, the double-ported valve gives double the opening to steam that the plain valve does. Otherwise, the two valves are alike in all respects.

43. The **Allen** or **Trick valve,** shown in Fig. 22, accomplishes the same object. The passage A is cast in the

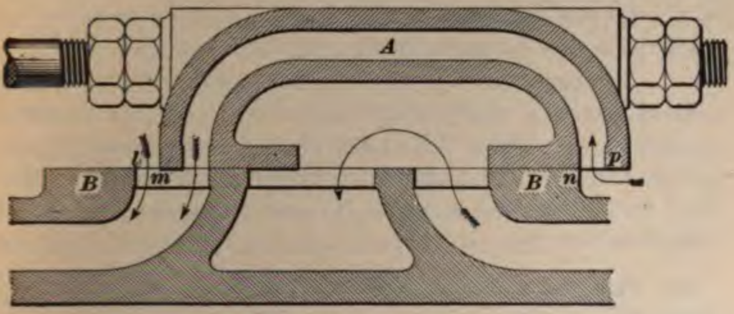


FIG. 22.

valve and extends clear through it. The shoulders B, B of the valve seat are so constructed that when the edge m of the valve is just even with the edge l of the port, the outer edge p of the passage A is just even with the edge n of the shoulder B at the other end of the valve seat. Now, when the valve moves a little to the right, into the position shown in the figure, steam enters the port directly between the edges l and m, as in the case of the ordinary valve. At the same time, the edge p of the passage has moved past the edge n of the valve seat; steam thus enters the passage A and finds there a direct path to the left steam port.

The piston valve shown in Fig. 19 is another example of a valve having a passage through it, by means of which the effective port opening for a given valve travel is doubled.

SETTING THE SLIDE VALVE.

44. Dead Centers.—Referring to Fig. 1, it is plain that when the piston P is at the end of its stroke at the end h of the cylinder, the crankpin A must lie at the point m in the crankpin circle. In this position the crank OA and connecting-rod AB lie in the same straight line. Likewise, when the piston is at the other end of the stroke, the pin A lies at the point n, and again the crank and connecting-rods are in the same straight line.

When the crank occupies either of these positions, the engine is said to be on its **dead center.** All the pressure of the steam on the piston is transmitted directly to the shaft O, because the reciprocating parts are in a straight line. Consequently, there is no tendency to turn the crank, and the engine cannot be started until turned into a different position. When the crank occupies the position Om, it is said to be on its **inner,** or **head-end,** dead center, and when it occupies the position On, on its **outer,** or **crank-end,** dead center.

45. To Place the Engine on Its Dead Center.—It is sometimes necessary to place the engine exactly on its center

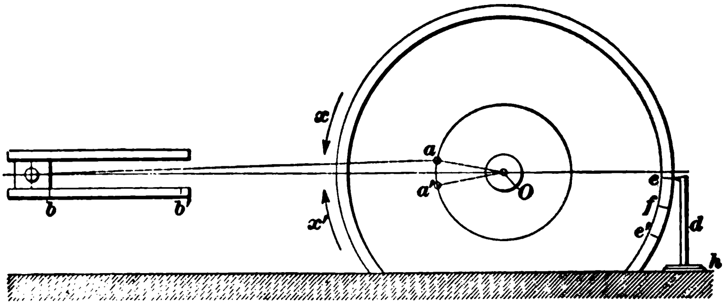


FIG. 23.

in order to set the valve. A common method of doing this is shown in Fig. 23.

When the crosshead is very near the end of its travel, make a mark b on one of the guides opposite the outer edge of the crosshead. Now turn the engine in the direction of the arrow x until the outer edge of the crosshead comes even with the mark b. While the engine is in this position, take a tram d, the length of which is about equal to the distance from the floor to the center of crank-shaft, place one end upon the floor (or, better, upon a solid block or part of the engine bed), and with the pointed end make a mark e upon the edge of the flywheel. The engine will probably not be exactly on the center; the crankpin will be, say, at a point slightly above the center. Now turn the engine in the direction of the arrow x' until the edge of the crosshead again comes even with the mark. The flywheel will have made nearly a complete revolution, and the crankpin will be at a', the same distance below the center that a was above it. Since the flywheel has made a little less than a full revolution, the mark e on the rim will not now be opposite the marking points of the tram, but the latter will make a new mark e' on the rim. Now, make a mark f half way between the marks e and e', and turn the wheel until the mark f comes opposite the point of the tram. The engine is then exactly on its dead center.

By taking another mark b' at the other end of the guide, the flywheel may be marked for the outer dead center. To insure accuracy, it is well to have both ends of the tram pointed. The lower point then fits into a prick-punch mark, made somewhere in the bed or foundation, and another punch mark on the rim determines the dead-center position.

46. Directions for Setting Slide Valve. — Put the engine on its dead center, place the valve on the seat and connect it with the eccentric rod. Shift the eccentric on the shaft until the valve has the desired lead. Turn the engine in the direction it is to run until it is on the other dead center. If the lead is the same as at the other end, the valve is correctly set; if it is not the same, the valve rod must be lengthened or shortened until the lead is the same at both centers. If, now, the lead is less than desired, shift the eccentric forwards a little on the shaft; if the lead is a little too great, shift the eccentric backwards.

After the valves are set and the engine is started, a pair of indicator diagrams should be taken. The diagrams will show any slight errors in the setting and corrections may be made accordingly.

CLEARANCE: REAL AND APPARENT CUT-OFF.

CLEARANCE.

47. Piston Clearance.—When the crank is on a dead center, the piston is always a short distance from the cylinder head; this allowance is made so that a slight change in the length of the connecting or piston rods will not cause the piston to strike the heads at the end of its stroke. It is also important to have a small space between the piston and head in which any small quantity of water in the cylinder may collect when the piston is at the end of its stroke; the incompressible nature of water would have the effect of breaking some part of the engine if there were no space in which it could collect. The short distance between the piston and the head when the piston is at the end of its stroke is called the **piston clearance.**

48. **To measure the piston clearance** at either end of the cylinder, first put the engine on its dead center for that end and make a mark on the guides corresponding to some convenient point of the crosshead. Next, disconnect the connecting-rod and push or pull the piston until it strikes the head. The distance of the chosen point on the crosshead from the mark made on the guides is the piston clearance for that end of the stroke.

49. The **clearance volume** or, simply, the **clearance,** is the volume of the space between the piston and cylinder head, when the piston is at the end of its stroke, plus the volume of the port leading to this space. Thus, in Fig. 15, the piston is at the end of its return stroke, and the clearance is the volume of the space between the piston and the left cylinder head plus the volume of the left steam port. In other words, the clearance may be defined as the volume of steam between the valve and the piston when the latter is at the end of the stroke.

50. Measuring the Clearance Volume.—The clearance volume of an engine may be found by putting the engine on a dead center and pouring in water until the space between the piston and the cylinder head and the volume of the steam port leading into it are filled. The volume of the water poured in is the clearance. Since water is likely to leak past the piston, some engineers advocate the use of a heavy oil for measuring the clearance volume.

51. Method of Expressing Clearance Volume.—The clearance volume may be expressed in cubic feet or cubic inches, but it is more convenient to express it as a percentage of the volume swept through by the piston. For example, suppose the clearance volume of a 12" × 18" engine is found to be 128 cubic inches. The volume swept through by the piston per stroke is $12^2 \times .7854 \times 18 = 2{,}035.8$ cubic inches. Then, the clearance is $\frac{128 \times 100}{2{,}035.8} = 6.3$ per cent. The clearance may be as low as $\frac{1}{2}$ per cent. in Corliss engines and as high as 14 per cent. in very high-speed engines.

52. Theoretically, there should be no clearance, since the steam that fills the clearance space does no work except during expansion; it is exhausted from the cylinder during the return stroke and represents so much dead loss. This is remedied to some extent by compression. If the compression were carried up to the boiler pressure, there would be very little, if any, loss, since the steam would then fill the entire clearance space at boiler pressure, and the amount of fresh steam needed would be the volume displaced by the piston up to the point of cut-off, the same as if there were no clearance. It is not practicable to build an engine without any clearance, because it is necessary to allow for lost motion and adjustment in the joints of the connecting-rod and because it is also necessary to allow for the formation of water in the cylinder due to the condensation of steam, particularly when starting the engine. As water is practically incompressible, some part of the engine would be broken when the piston reached the end of its stroke, provided there were no clearance space to receive the water; usually the cylinder heads would be blown off. Neither is it practicable to compress to boiler pressure, as a general rule, for that causes too great strains on the engine. Automatic cut-off, high-speed engines of the best design, with shaft governors, usually compress to about half the boiler pressure and have a clearance of from 5 to 14 per cent. Engines that do not have a high rotative speed, say not over 100 revolutions per minute, have very little compression and very small clearance. Such are the Corliss and other releasing-gear engines.

REAL AND APPARENT CUT-OFF.

53. It is customary, in speaking of the point of cut-off, to say that the engine cuts off at $\frac{1}{2}$ stroke, $\frac{1}{4}$ stroke, etc. By this is meant that the steam is cut off when the piston has completed $\frac{1}{2}$ or $\frac{1}{4}$ of its stroke, as the case may be. For example, if the stroke is 48 inches and the steam is shut off from the cylinder when the piston has moved 18 inches, the cut-off is $\frac{18}{48} = \frac{3}{8}$. The cut-off thus spoken of is the **apparent cut-off.**

54. The **real cut-off** takes account of the clearance space. It is the ratio between the volume of steam in the cylinder and clearance space when the piston is at the cut-off point and the volume of steam in cylinder and clearance space when the piston is at the end of the stroke. For example, let the volume of steam between the valve and piston when the latter is at cut-off be 4 cubic feet. Suppose that when the piston is at the end of its stroke the volume of steam in the cylinder and clearance space is 9 cubic feet. Then, the real cut-off is $\frac{4}{9}$.

55. The **relation between the apparent and real cut-offs** may be shown graphically as follows: Let the

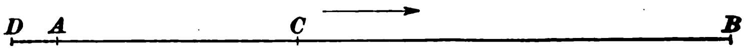


FIG. 24.

length AB, Fig. 24, represent the stroke of the engine. Suppose that the piston is moving in the direction of the arrow and that the steam is cut off when the piston has reached the point C.

Then, according to the above, the apparent cut-off is $\frac{A\,C}{A\,B}$. It is clear that, since $A\,B$ represents the stroke of the piston, it will also represent, to some scale, the volume swept through by the piston. Now, to the same scale, lay off $A\,D$ equal to the volume of the clearance. Then, from the above definition, the real cut-off is $\frac{D\,C}{D\,B} = \frac{A\,C + A\,D}{A\,B + A\,D}$. Let s represent the apparent cut-off, k the real cut-off, and i the clearance expressed as a per cent of the stroke. Then, in Fig. 24, $s = \frac{A\,C}{A\,B}$ and $i = \frac{A\,D}{A\,B}$.

Rule 1.—*To find the real cut-off, add the clearance volume, expressed as a per cent. of the stroke, to the apparent cut-off, expressed in per cent., and multiply the sum by 100; divide the product by 100 plus the clearance volume, in per cent.*

Or,
$$k = \frac{(s + i) \times 100}{100 + i}.$$

EXAMPLE.—In a 13" × 18" engine, the steam is cut off when the piston has moved over 8 inches of its stroke. The clearance is 8 per cent. of the volume displaced by the piston. Find the apparent cut-off and real cut-off.

SOLUTION.—The apparent cut-off is $\frac{8 \times 100}{18} = 44.4$ per cent. Applying rule **1**, the real cut-off is found to be $\frac{(44.4 + 8) \times 100}{100 + 8} = 48.5$ per cent. Ans.

56. The **ratio of expansion,** also called the **number of expansions,** is the ratio between the volume of steam in the cylinder and clearance when the piston is at the end of its stroke and the volume in the cylinder and clearance when the piston is at the cut-off point. That is, in Fig. 24, the ratio of expansion is $\frac{D\,B}{D\,C}$. Since $\frac{D\,B}{D\,C} = \frac{1}{\frac{D\,C}{D\,B}} = \frac{1}{k}$, it follows that the ratio of expansion is the reciprocal of the real cut-off. For example, if the volume of steam behind the piston when at the end of its stroke is 15 cubic feet and when at cut-off is 5 cubic feet, the real cut-off is $\frac{5}{15} = \frac{1}{3}$. The ratio of expansion is $\frac{15}{5} = 3$; in ordinary language, the steam would be said to have 3 expansions.

When the *real* cut-off is given in per cent., the ratio of expansion is found by dividing 100 by the real cut-off in per cent. Thus, if the real cut-off is 25 per cent., the ratio of expansion is $\frac{100}{25} = 4$.

ILLUSTRATIVE EXAMPLE.—Let it be required to find the clearance, the actual cut-off, and the ratio of expansion of a 12″ × 24″ engine under the following conditions: When the engine is on its center, the water from a vessel which with the water weighed 5 pounds was poured into the end of the cylinder. After pouring in just enough water to fill the clearance space, the vessel and water were weighed and found to weigh $1\frac{3}{4}$ pounds; consequently, the weight of water poured out of the vessel was $5 - 1\frac{3}{4} = 3\frac{1}{4}$ pounds. The weight of 1 cubic inch of water is .03617 pound. The number of cubic inches poured into the cylinder is, therefore, $\frac{3.25}{.03617} = 89.85$ cubic inches, nearly, which is the volume of the clearance space. The area of the 12-inch piston is $12^2 \times .7854 = 113$ square inches, very nearly; consequently, the piston displacement is $113 \times 24 = 2{,}712$ cubic inches. The clearance volume is, therefore, $\frac{89.85 \times 100}{2{,}712} = 3.31$ per cent. of the piston displacement; in other words, we say that the clearance is 3.31 per cent.

The cut-off takes place when the piston has moved 15 inches of its stroke. The apparent cut-off is, therefore, $\frac{15 \times 100}{24} = 62.5$ per cent. of the stroke. In accordance with rule **1**, the real cut-off is $\frac{(62.5 + 3.31) \times 100}{100 + 3.31} = 63.7$ per cent.

In accordance with Art. **56**, the ratio of expansion is $\frac{100}{63.7} = 2$, very nearly.

EXAMPLES FOR PRACTICE.

1. What is (*a*) the clearance volume in cubic inches, and (*b*) the clearance in per cent. of a 16" × 24" engine, if 7¼ pounds of water are required to fill the clearance space?

Ans. { (*a*) 200.4 cu. in. (*b*) 4 per cent.

2. If the engine in example 1 cuts off when the piston has made 6 inches of its stroke, what is (*a*) the apparent cut-off, (*b*) the real cut-off, and (*c*) the ratio of expansion?

Ans. { (*a*) ¼, or 25 per cent. (*b*) 27.9 per cent. (*c*) 3.59, nearly.

THE BILGRAM VALVE DIAGRAM.

57. Graphic Method of Determining the Effect of Change in Proportion of Valves.—The action of the valve of a plain slide-valve engine when operated by an eccentric can be readily analyzed by means of a diagram that has been designed by Mr. Hugo Bilgram. This diagram is extremely useful not only in the analysis of an existing slide-valve gear, but as it also exhibits in a graphic form the effects of any change in the proportions of the valve, it is invaluable in the design of a new valve.

The valve diagram and its application to an existing valve gear is shown in Fig. 25. In a case of this kind, the outside and inside lap of the valve, the travel of the valve, and the stroke of the engine are known; and the lead, if not known, may be assumed. With these data, the amount that the steam ports are opened (the port opening), the point of cut-off, the point of release, the point of exhaust closure, and the angle of advance of the eccentric can be determined.

58. To any convenient scale, draw on the line $a\,b$, with o as a center, the semicircle $a\,d\,b$ having a radius equal to that of the crank (one-half the stroke). About o as a center and with a radius equal to one-half the valve travel, describe the semicircle $a'\,o'\,b'$. Draw a line $g\,h$ parallel to $a\,b$ and at a distance from it equal to the lead. With a radius equal to the outside lap of the valve and with a center o' on the semicircle $a'\,o'\,b'$ describe a circle r that is tangent to gh. The position of the center o' is easiest found by trial. About o' as a center and with a radius equal to the inside lap of the valve, describe a circle s. Next draw the straight lines od, ol, and om tangent, respectively, to the outside lap circle r and the inside lap circle s. Through o

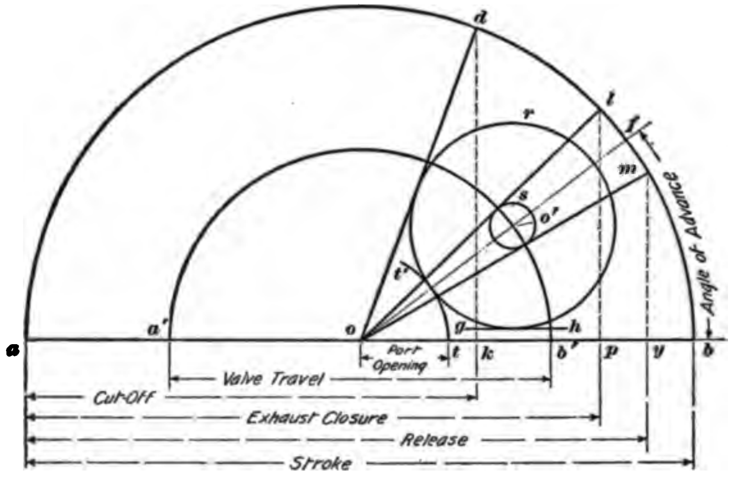


FIG. 25.

and o' draw the straight line of. From the points of intersection d, l, and m of the lines od, ol, and om with the semicircle adb, drop perpendiculars, as dk, lp, and my on the straight line ab. About o as a center describe an arc tangent to the outside lap circle and intersecting ab in t.

In the diagram just drawn, the distance ot represents the port opening to the same scale to which the diagram was drawn; the distance ak shows the piston movement up to the point of cut-off; the distance ay shows the piston movement up to the point of release, and the distance ap shows the piston movement up to the point where the exhaust port is closed, i. e., up to the point where compression begins. The angle fob is the angle of advance.

59. When the valve has no inside lap, no inside lap circle can be drawn; in that case, drop a perpendicular from the intersection point *f* of the line *o f* with the semicircle *a d b* to the line *a b*. The distance between *a* and the point of intersection of this perpendicular with the line *a b* will then represent the piston movement up to the points of compression and release.

60. A careful study of the diagram will show the effects of changes in the valve proportions and valve setting. Thus, suppose that, the valve proportions remaining the same, it has been decided to give more lead. Then, the lead line *g h* being at a greater distance than before from *a b*, the center *o'* will be higher up and farther to the left; in consequence, the intersection points *d*, *l*, and *m* will also be to the left of their former positions and the intersections of perpendiculars dropped from these points on *a b* will be nearer *a* than before. In other words, the increase of lead causes the different events to take place earlier. Since *o'* will occupy a different position than formerly, the angle *f o b* (the angle of advance) will now be greater; this shows that, with the valve proportions remaining the same, increasing the lead is impossible without increasing the angle of advance.

61. Suppose that the inside lap is *decreased*, all other valve proportions and the lead remaining the same. Then, since the inside lap circle *s* will be smaller, the point *l* will be nearer *f* than before, and, consequently, the intersection of a perpendicular dropped from *l* on *a b* will be to the *right* of *p*; that is, the exhaust closure will take place later. The intersection point *m* will also be closer to *f*, and, obviously, the intersection of a perpendicular dropped from it on *a b* will be to the *left* of *y*; that is, the release will take place earlier.

Suppose that the outside lap is *increased* in order to get an earlier cut-off and that the lead is to remain as before. Then, owing to the greater diameter of the circle *r*, its center *o'* will be farther to the left, and, consequently, the tangent lines *o d*, *o l*, and *o m*, and their perpendiculars *d k*, *l p*, and *m y* will also move to the left, thus indicating that all the events will take place earlier.

62. Assume that while the outside lap was increased, no change was made in valve travel. Then, since the lap circle *r* is larger than before, it follows that *o t* will be smaller; that is, an increase of outside lap not accompanied by an increase of the valve travel will cause a decrease in the port opening. From this it is evident that in order to keep the port opening constant, an increase of outside lap must be accompanied by an increase of the valve travel.

By studying in the manner indicated in the preceding articles the effect on the valve diagram of any change in the valve proportions, in the angle of advance, etc., the effects on the steam distribution can be noted easily.

63. In designing a new valve for an engine, the port opening, the point of cut-off, and the stroke of the engine are known; the lead must be fixed upon. With these data, the valve proportions, the valve travel, and the angle of advance are readily determined by means of the Bilgram valve diagram.

On the line *a b* and from *o* as a center, describe the semicircle *a d b* to represent the path of the crankpin, using any convenient scale. Draw the lead line *g h* parallel to *a b* and at a distance equal to the lead from it. With a radius equal to the port opening, describe an arc *t t'* about *o* as a center. On *a b* lay off *a k* equal to the desired cut-off. At *k* erect a perpendicular, and from its point of intersection *d* with the semicircle *a d b* draw the straight line *d o*. Now, by trial, find the radius and the position of the center *o'* of a circle that will be tangent to the lines *o d* and *g h* and tangent to the arc *t t'*. The radius of this circle represents the outside lap required, while the distance of the center *o'* from *o* represents one-half the valve travel. By drawing a straight line, as *o f*, through *o* and *o'*, the angle of advance is determined. The question of how much, if any, inside lap to give is one that each designer must answer for himself, remembering that the giving of inside lap makes release later and exhaust closure earlier.

SIZE OF STEAM PASSAGES.

64. The average practice of steam-engine builders is to proportion the steam passages so that the steam will flow at a velocity of 6,000 feet per minute through the main steam pipe, 6,000 feet per minute through steam ports that are relatively long and tortuous, as the steam ports of plain slide-valve engines, 7,500 feet per minute through very short and direct steam ports, 4,000 feet per minute through the exhaust ports, and also through the exhaust pipe.

65. With these velocities as a basis, the following rules for proportioning the steam passages have been deduced. In the case of steam pipes and exhaust pipes, the commercial size of pipe whose area is nearest the calculated area should be selected. In case either the steam pipe or the exhaust pipe, or both, is very long, say above 200 feet, it may be advisable to select a pipe one size larger than the one whose area is nearest the calculated area. In case the steam pipe is but poorly or not all protected by covering, the use of a larger size of pipe is especially necessary.

66. Rules and Formulas for Calculating the Sizes of Steam Passages.—

Let a = area of steam port in square inches;
b = area of exhaust port and pipe in square inches;
c = area of steam pipe leading to engine;
d = piston speed in feet per minute;
e = area of piston in square inches.

Rule 2.—*To find the area of the steam pipe leading to an engine or pump, multiply the area of the piston by the piston speed in feet per minute and divide the product by 6,000.*

Or,
$$c = \frac{d\,e}{6,000}.$$

EXAMPLE.—A 14" × 36" engine is to run 100 revolutions per minute. What size should the steam pipe be ?

SOLUTION.—The piston speed in feet per minute is $\frac{36}{12} \times 100 \times 2 = 600$ feet. The area of the piston is $14^2 \times .7854 = 153.938$, say 154 square inches. Applying rule **2**, we get

$$c = \frac{154 \times 600}{6,000} = 15.4 \text{ square inches.}$$

The nearest commercial size of pipe is $4\frac{1}{2}$ in. nominal diameter, whose internal area is 15.939 sq. in. Ans.

67. Rule 3.—*To find the area of the exhaust pipe or exhaust port for an engine or steam pump, multiply the area of the piston by the piston speed in feet per minute and divide the product by 4,000.*

Or,
$$b = \frac{d\,e}{4,000}.$$

EXAMPLE.—What should be the area of the exhaust port and what should be the size of the exhaust pipe of the engine mentioned in the example given in Art. **66**?

SOLUTION.—Applying the rule just given, we get

$$b = \frac{154 \times 600}{4,000} = 23.1 \text{ square inches}$$

as the area of the exhaust port. Since a 6-inch pipe has an actual internal area of 28.889 square inches, while the next smaller commercial size of pipe, viz., 5-inch, has an area of but 19.99 square inches, experienced engineers would select the 6-inch pipe, in order not to cramp the exhaust. Ans.

68. Rule 4.—*To find the area of the steam port, multiply the area of the piston by the piston speed in feet per minute; divide the product by 7,500 if the port is short and direct, and by 6,000 if the port is long and tortuous.*

Or,
$$a = \frac{d\,e}{7,500} \text{ for a short port,}$$

and
$$a = \frac{d\,e}{6,000} \text{ for a long port.}$$

EXAMPLE.—What should be the area of the steam port for the engine given in the example of Art. **66**, if the port is short?

SOLUTION.—Applying rule 4, we get

$$a = \frac{154 \times 600}{7,500} = 12.32 \text{ sq. in.} \quad \text{Ans.}$$

THE ROTARY ENGINE.

69. Since the time of Watt, it has been the aim of many inventors to produce an engine in which the piston has a rotary motion, thus dispensing with the connecting-rod and crank. Innumerable designs have been proposed and patented, many of which have been actually tried; except for special service where economy in the use of steam is but a minor consideration, they have all proved commercial failures. It is a very simple matter to design a rotary engine that will turn (run); it is an entirely different matter, however, to have a rotary engine develop in constant and extended service a horsepower on the same steam consumption as a reciprocating steam engine.

70. Rotary engines have been constructed in a great variety of forms, many of which can only be characterized as freaks. The remainder usually consist either of a rotary piston of some suitable form, bearing against a rolling, sliding, or swinging abutment, or a design of interlocking pistons similar to that found in the Root blower. Abutments, no matter how carefully designed, made, and fitted, cannot be kept steam-tight for any length of time, and generally will cause bad steam leakage into the exhaust in a short time; rotary engines of the interlocking piston pattern either commence to leak badly after short service or are very noisy. Some rotary engines are valveless and very simple, but extremely costly to operate. Some have eccentric pistons; these rapidly wear the cylinder walls and bearings owing to the difficulty of counterbalancing them properly for high speeds. On the whole it can be safely stated that the rotary engine, owing to inherent irremovable constructive difficulties, will never be a serious commercial competitor of the reciprocating engine.

THE STEAM TURBINE.

71. Two eminent engineers, Mr. Parsons and Mr. Laval, have developed a steam engine working on an entirely different principle than the ordinary steam engine. Instead of making use of the pressure of steam, they utilize the kinetic energy contained in a mass of steam moving with a very high velocity, jets of steam impinging against the blades or vanes of a wheel fitted inside of a suitable casing and thus rotating the wheel at a high speed. This kind of an engine, from its similarity to the turbine, is called a **steam turbine**. A limited number of steam turbines are in use, the latest designs giving an economy about equal to that of the reciprocating steam engine.

THE INDICATOR.

INDICATORS AND REDUCING MOTIONS.

CONSTRUCTION OF INDICATOR.

GENERAL CONSTRUCTION.

1. The **indicator** is an instrument that can be readily applied to a steam engine for the purpose of obtaining a

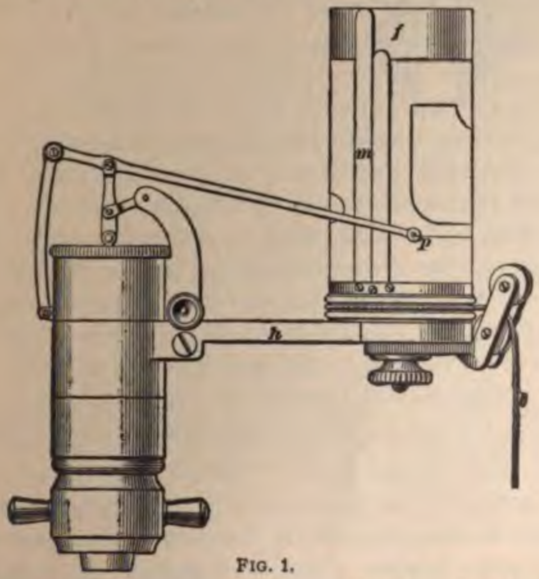


FIG. 1.

diagram of the pressures in the cylinder. It is made in a variety of forms that differ, however, only in minor details;

the general principles involved in all will readily be understood by reference to Fig. 1, which shows the general appearance of an indicator, and Fig. 2, which shows one in

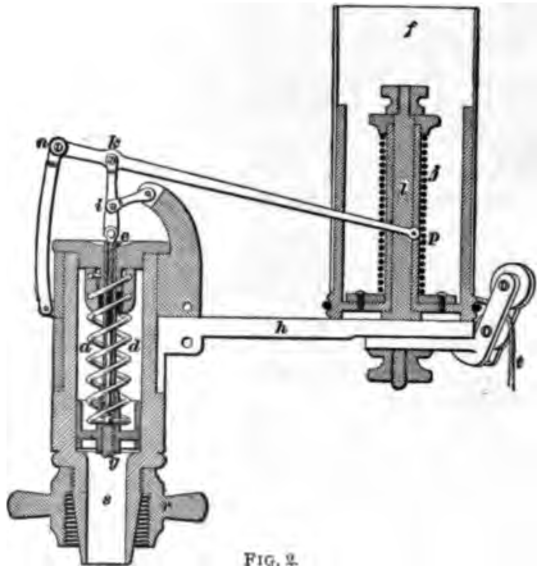


FIG. 2.

section. The instrument consists essentially of a cylinder a, Fig. 2, containing the piston g and the spring d. By turning a cock connected to the small pipe to which the indicator is attached, steam may be admitted to, or shut off from, the cylinder of the indicator at pleasure. When steam is admitted through the channel s, its pressure causes the piston g to rise. The spring d is compressed and resists the upward movement of the piston. The height to which the piston rises should then be in exact proportion to the pressure of the steam, and as the steam pressure rises and falls, the piston must rise and fall accordingly.

2. To register this pressure, a pencil might be attached to the end of the piston rod c, the point of the pencil being made to press against a piece of paper. It is desirable, however, to restrict the maximum travel of the piston to about $\frac{1}{2}$ inch, while the height of the diagram may advantageously be 2 inches or more. To obtain a long pencil movement combined with a short travel of the piston, the pencil is attached at *p* to the long end of the lever *n k p*. The fulcrum of the lever is at *n*. The piston rod is connected to it at *k* through the link *i k*. The pencil motion is thus $\frac{p\,n}{n\,k}$ times the piston travel. This ratio $\frac{p\,n}{n\,k}$ is, for most indicators, either 4, 5, or 6. The point *p* is forced to move in a vertical straight line by the arrangement of the links and joints *i*, *e*, *n*, and *k*, forming what is called a **parallel motion.**

DETAILS OF INDICATOR.

3. The Spring.—The height to which the piston will rise under a given steam pressure depends upon the stiffness of the spring. Indicators are usually furnished with a number of springs of varying degrees of stiffness, which are distinguished by the numbers 10, 20, 30, 40, etc. These numbers indicate the pressure per square inch required to raise the pencil 1 inch and are called the **scale of the spring.** Thus, if a 40 spring is used, a pressure of 40 pounds per square inch raises the pencil 1 inch, and the vertical scale of the diagram is, therefore, 40 pounds per inch; that is, the vertical distance in inches of any point on the diagram from the atmospheric line, multiplied by 40, gives the gauge pressure per square inch at that point. The scale of the spring chosen should not be less than one-half the boiler pressure. For example, we would choose a 40 spring for a steam pressure of 75 pounds per square inch.

4. The Paper Drum.—The indicator must not merely register pressures, but it must register them in relation to the position of the piston. To accomplish this object, a cylindrical drum *f*, Figs. 1 and 2, is provided. This drum can be revolved on its axis *l* by pulling the cord *t*, which is coiled around it. When the pull is released, the spring *j* rotates the drum back to its original position. If, now, the cord *t* be attached to some part of the engine that has a motion proportional to the motion of the piston, the motion of the drum will also be proportional to the motion of the piston.

CONNECTING INDICATOR TO ENGINE CYLINDER.

5. To attach the indicator to the engine, a hole is drilled in the clearance space of the cylinder and tapped for a ½-inch nipple. If this hole is in the top of the cylinder, the indicator cock may be screwed directly into it, or, if more convenient, a nipple and coupling may be used. If the cylinder is tapped at the side, a nipple and elbow may be used so as to bring the indicator into a vertical position; since, however, it is desirable to keep the connections to the indicator as short and direct as practicable, some engineers prefer to omit the elbow and attach the indicator in a horizontal position. The indicator is attached directly to the cock by the nut *r*, Fig. 2, which wedges the conical projection *s* of the indicator tightly into the cock and thus prevents leakage of steam. On account of the resistance offered by the pipe and elbows to the flow of steam to the indicator, it is preferable to have an indicator at each end of the cylinder, but if that is not convenient, one indicator may be connected with both ends of the cylinder by means of a three-way cock.

6. Most cylinders of the better class of engines are now provided with bosses having holes tapped in them for the convenient application of the indicator; in many old engines, however, no special provision for the indicator has been made. In such cases care must be taken to drill holes that will not be covered with the piston when it is near the end of the stroke. If the hole cannot be tapped directly into the clearance space, a passage must be chipped to the clearance space in order that the steam can reach the indicator.

7. The indicator connection should never be on the side of the cylinder directly opposite to the steam ports; the current of entering steam would strike against the opening leading to the indicator, and the pressure shown by the diagram would thereby be considerably increased. With many engines that have not been provided with special indicator attachments, it will be found that holes for this purpose may be conveniently drilled and tapped in the cylinder heads.

INDICATORS FOR SPECIAL PURPOSES.

8. Gas-Engine Indicators.—An indicator specially adapted to gas engines, and one that is sometimes applied to steam engines, pumps, and hydraulic machinery when high pressures are used, is shown in Fig. 3. The cylinder has two bores *a* and *b*. The larger bore *a* is ½ square inch in area (the size usually employed when testing a steam engine), and the area of the smaller bore *b* is ¼ square inch. The piston *c* is fitted to the smaller bore and is that used when indicating a gas engine or a very high pressure steam engine. It gives but half the movement of the pencil given by the larger piston used in *a*; so, if the spring used is stamped 40, the calculations, when using the smaller piston, must be made as if an 80-pound spring had been used. The pencil movement is also of special design, the moving parts being stronger and more rigid than those used on regular steam-engine patterns. With the regulation piston of ½ square inch area in chamber *a*, the indicator may be used for steam engines with ordinary pressures.

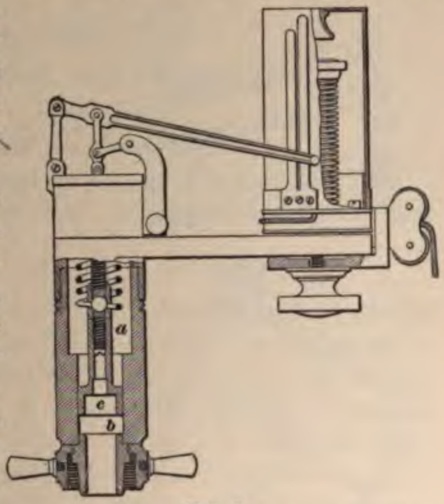


FIG. 3.

9. Ammonia Indicators.—For use on the ammonia cylinders of refrigerating machines, it is preferable to use a special indicator the working parts of which are made of steel instead of brass, because ammonia has no effect on steel, but rapidly corrodes brass. In case it is not possible to procure an ammonia indicator, an ordinary steam-engine indicator will answer the purpose, provided the piston is removed after every set of cards is taken and both cylinder and piston are wiped dry and well covered with oil. This will prevent the ammonia gas from attacking the portions of the indicator made of brass.

REDUCING MOTIONS.

PANTOGRAPH MOTIONS.

10. Purpose of a Reducing Motion.—The motion of the paper drum is nearly always taken from the crosshead. However, since the stroke of the crosshead is longer than the circumference of the drum, it is necessary to arrange a form of mechanism some point of which will copy to a reduced scale the stroke of the piston. Such a mechanism is called a **reducing motion.**

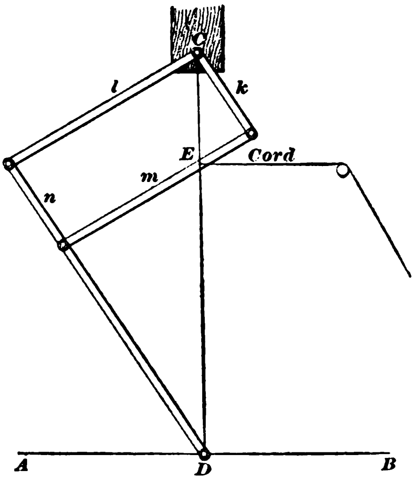


FIG. 4.

11. The **pantograph,** Fig. 4, is an excellent form of reducing motion. It consists of four links joined together in the form of a parallelogram. One of the links *n* is prolonged and is pivoted at the end to the crosshead *D*. The opposite corner of the parallelogram is pivoted to the fixed point C. The cord is attached to the point E on the link m, which point must be on the straight line connecting C and D. AB represents the length of the stroke. Letting h represent the length of the indicator diagram, we have the following proportions: $AB : h = CD : CE$, or $\frac{AB}{h} = \frac{CD}{CE}$.

12. The **lazy tongs,** Fig. 5, is a modified form of the pantograph that is much used as a reducing motion. It

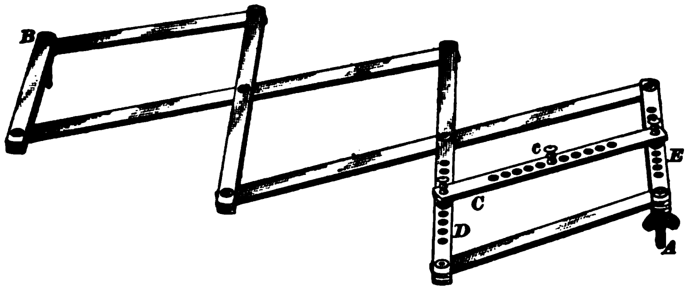


FIG. 5.

consists of a series of bars joined together in such a manner as to form a flexible frame. The joint A of the frame is attached to any convenient stationary point on the engine or its surroundings and B is attached to the crosshead. The bar C is provided with a number of holes, in one of which is placed a pin c to which the indicator cord is attached. The bars D and E to which C is joined are provided with a series of holes, and C may be placed in any position in which its ends are attached to holes similarly located in the two bars. For each position of C, one of the holes in it will be in line with the two joints A and B at the extremity of the frame; this is the hole in which the pin c is to be placed for that position of C.

13. The ratio of the length of the diagram to the length of the stroke is equal to the ratio of the distance Ac to the distance AB; this is true for any distance between the points A and B. To find the correct position of the bar C

for a given length of card, when the length of stroke is known, set the points *A* and *B* at some convenient distance apart; multiply this distance by the desired length of the card and divide the product by the length of the stroke; the quotient so obtained will be the distance from *A* at which to set the pin *c*, keeping *A* and *B* at the distance apart to which they were set at the beginning of the operation. A very convenient method of locating *C* is to make the distance *A B* equal to the length of the stroke and then locate *C* so that the distance *A c* is, as nearly as possible, equal to the desired length of the diagram. For example, let it be desired to take a diagram $3\frac{1}{4}$ inches long from an engine having a stroke of 32 inches; make the distance *A B* 32 inches and then attach *C* to the holes in *D* and *E* that will make the distance *A c*, as near as may be $3\frac{1}{4}$ inches.

14. The pantograph and lazy tongs are theoretically correct reducing motions; that is, the motion imparted to the indicator cord is exactly proportional to the motion of the crosshead. The point to which the cord is attached moves in a straight line parallel to the direction of motion of the crosshead. The fixed point of either the pantograph or lazy tongs may be at any place that will enable the cord to be led to the indicator in the shortest and most direct manner; it is not necessary, as is sometimes assumed, to locate the fixed point on a line equidistant from both ends of the stroke. In locating the point of attachment, however, it is important to guard against striking the joints of the frame at the ends of the stroke; neglect of this precaution may result in breaking the reducing motion. A disadvantage of the pantograph is the danger of lost motion due to wear in the joints.

SWINGING-LEVER REDUCING MOTIONS.

15. A lever with one end pivoted at some convenient fixed point and the other attached to the crosshead of the engine forms one of the simplest reducing motions; and if the device is correctly designed and carefully constructed, it can be made to give as accurate results as are obtainable in any way. Two common forms of swinging-lever reducing motions are those illustrated in Figs. 6 and 7. The form shown in Fig. 6 is called the **slotted swinging lever.**

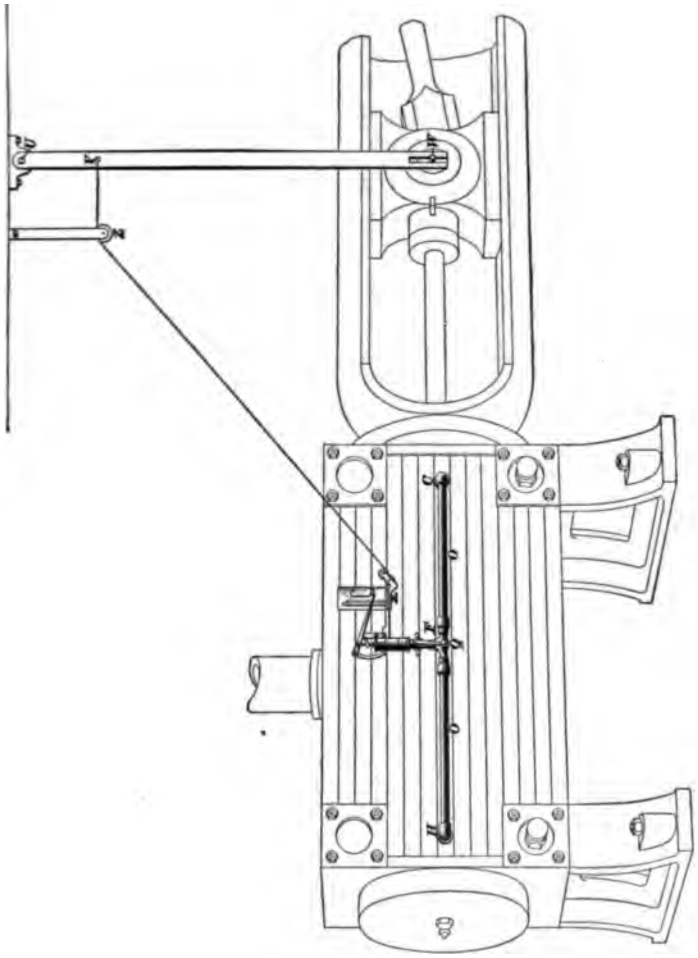


FIG. 6.

The lower end of the lever is slotted and fits over a pin in the crosshead; the other end of the lever is pivoted at a fixed point U and the cord is attached at V. The cord is guided by a pulley Z so that it will leave the point V in a direction parallel to the line of motion of the crosshead.

In the device shown in Fig. 7, the lever is connected to the crosshead by a link WD. The cord is attached to the circumference of a grooved sector EVF, called a **Brumbo pulley.** The center of the circle of which the sector forms a part lies in the center line of the lever pivot U. The sector may be attached directly to the lever or it and the lever may be keyed to a short shaft in such a manner that the cord can be led directly to the indicator. Instead of the sector, the cord may be attached to a pin, as in the motion shown in Fig. 6; in this case a guide pulley similar to the pulley Z, Fig. 6, would be required.

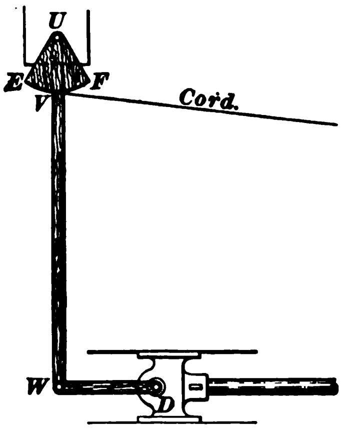


FIG. 7.

16. Error of Swinging-Lever Motions.—The types of swinging-lever motion illustrated in Figs. 6 and 7 are imperfect, from the fact that the motion imparted to the cord is not exactly proportional to the motion of the crosshead. In the slotted swinging lever, Fig. 6, the distance from the pivot U to the center of the pin in the crosshead is variable, while the distance from U to the point V to which the cord is attached is constant; in other words, the length of the long arm of the lever varies, while the length of the short arm remains constant. This results in a variation in the relative motions of crosshead and cord for different parts of the stroke, and the diagram obtained is, in consequence, distorted.

With the motion illustrated in Fig. 7, the link WD acts like a connecting-rod to transmit the straight-line motion of the crosshead to the end of the lever that moves in a circular arc; the link thus has an angular motion that has a disturbing effect on the ratio of the cord movement to that of the crosshead. The result is a diagram whose proportions are not perfect.

17. Methods of Reducing the Errors.—For ordinary work with the indicator, the amount of distortion with carefully made swinging-lever motions is not serious and may be ignored. To secure good results, the levers should always be suspended from such a point that when the crosshead is at the middle of its stroke, they will be perpendicular to its line of motion. The accuracy of the motion will, in general, be increased by increasing the length of the levers; for most purposes it will be sufficient to use a lever whose length is twice that of the stroke, and in some cases a lever even shorter than this is used. The accuracy of the motion with the connecting link, Fig. 7, is also increased by increasing the length of the link; for ordinary work, a link whose length is equal to one-third of the length of the stroke may be used. The lever in this motion should be so suspended that the extreme positions of *W* above and below the line of motion of the point *D* are about equal.

18. Theoretically Correct Motions.—The errors of the swinging-lever motions that were noted in Art. **16** can be neutralized and a theoretically correct reduction obtained by the methods illustrated in Figs. 8 and 9. In each of the figures the cord is attached to a sliding bar *S* whose line of motion is parallel to that of the crosshead. In Fig. 8 the bar *S* is provided with a pin that works in a slot in the swinging lever. By this arrangement the ratio of the distances from the pivot *U* to the pins *W* and *V* is constant for all positions of the crosshead, and the motion of the bar *S* is exactly

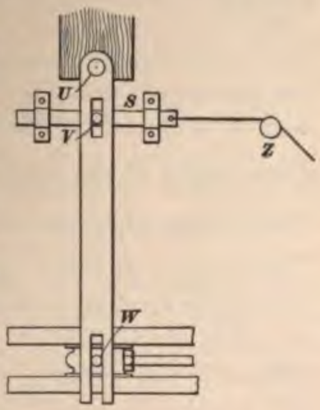


FIG. 8.

proportional to that of the crosshead. In Fig. 9 the bar S is connected to the swinging lever by a short link $V C$. In order to secure a theoretically correct reduction, the length of this link must be such that the ratio of the distance $V C$ to the distance $W D$ is equal to the ratio of $U V$ to $U W$, and the bar S must be so located that the center lines $V C$ and $W D$ of the two links are parallel. When these conditions are fulfilled and the points U, V, and W lie in the same straight line, the center of the joint C lies on the straight line joining U and D.

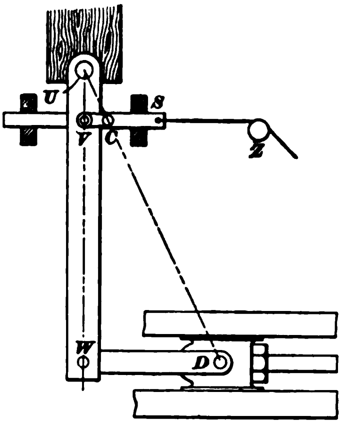


FIG. 9.

The cord must be guided by pulleys, as Z, so that it will leave the bars S, Figs. 8 and 9, in a direction parallel to the line of motion of the bars.

19. Rule for Proportioning Swinging-Lever Motions.—With any swinging-lever reducing motion, the ratio of the length of the diagram to the length of the stroke is equal to the ratio of the distance from the pivot to the point of attachment of the cord to the distance from the pivot to the pin by means of which the lever is connected to the crosshead; thus, with either of the motions shown in Figs. 6 to 9, let l represent the length of the diagram and L the length of the stroke, then $\frac{l}{L} = \frac{U V}{U W}$. In accordance with this principle, we have the following

Rule.—*To find the distance from the pivot at which to connect the cord, or to find the radius of the Brumbo pulley when the length of the stroke, the length of the diagram, and the distance from the pivot of the lever to the point where it is connected with the crosshead are known, multiply the length of the diagram by the distance from the pivot to* *the point of the lever at which it is connected with the crosshead and divide the product by the length of the stroke.*

Let l = length of diagram;
L = length of stroke;
d = distance from pivot to point of attachment of cord (see $U\,V$, Figs. 6 to 9);
D = distance from pivot to point where lever is connected to crosshead (see $U\,W$, Figs. 6 to 9).

Then,
$$d = \frac{D\,l}{L}.$$

EXAMPLE 1.—The stroke of an engine is 28 inches; the length $U\,W$ of the lever is 6 feet; what must be the distance $U\,V$ to give a diagram 3½ inches long?

SOLUTION.—Applying the rule just given, we have

$$d = \frac{72 \times 3\frac{1}{2}}{28} = 9 \text{ in. Ans.}$$

EXAMPLE 2.—In Fig. 7 find the radius $U\,V$ of the arc $E\,F$ in order that the diagram may be 3½ inches long, the stroke of the engine being 38 inches and the length $U\,W$ being 5 feet 5 inches.

SOLUTION.—This example is solved in the same manner as the preceding one. The effective length of the lever is 5 feet 5 inches = 65 inches. Applying the rule, we have

$$d = \text{radius of arc } E\,F = \frac{65 \times 3.5}{38} = 6 \text{ in., nearly. Ans.}$$

REDUCING WHEELS.

20. Reducing wheels form a very convenient and theoretically accurate method of reducing the motion of the crosshead to the required value for the paper drum. These wheels are furnished in a variety of forms, some of which are designed to be attached directly to the indicator, while others are provided with means for clamping to some point on the engine bed. One of the latter type is illustrated in Fig. 10. The cord a is attached to a bar or rod fastened to the crosshead in such a manner that the cord will lead from it to the wheel in a line parallel to the line of motion of the crosshead, and the cord *b* is attached to the indicator drum. The smaller pulley can be removed and replaced by one of several others of different sizes. The proportions of the two pulleys can thus be varied so as to secure the desired length of diagram. Thus, if the stroke of the engine is 12 inches and the desired length of the diagram is 3 inches, the diameter of the larger pulley should be four times that of the smaller. The hub of the larger wheel contains a spring that is wound up when the cord *a* is unwound from the wheel by the outward motion of the crosshead; when the crosshead makes its return stroke, the spring turns the wheel and winds the cord on again.

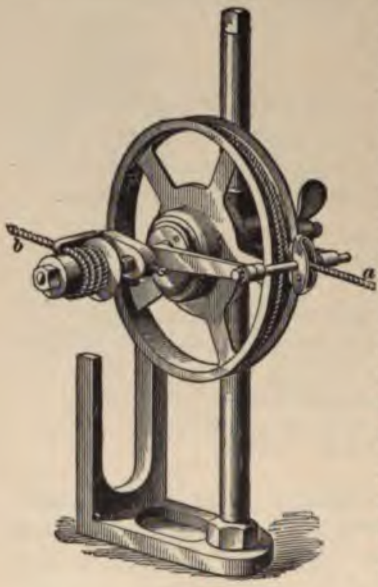


Fig. 10.

REDUCING MOTIONS FOR HIGH SPEEDS.

21. When an engine has a high rotative speed, the quick changes in direction of motion set up severe stresses in a reducing gear. In a pantograph or a swinging-lever motion, these stresses are likely to cause a springing of the parts that will distort the diagram and lead to erroneous results. The shocks and stresses also tend to wear the joints rapidly; the lazy-tongs motion, with its great number of joints and moving pieces, is on this account poorly adapted for high-speed work. A swinging lever, if made of stiff and light wood with joints bushed and neatly fitted, will give good results at nearly any speed of rotation at which it is practicable to run an engine. Instead of bushing the joints, they may be made adjustable as shown in Fig. 11. The end of the bar is split by a saw kerf passing through the center of the hole forming the pivot bearing and extending far enough into the bar to permit the two parts to be drawn up tight against the pivot by a wood screw *S*. Instead of the screw a somewhat more substantial method is to use a small bolt passing through the end of the bar. A joint made in this way and fitted to a turned pin, if well lubricated, gives the best satisfaction and lasts almost indefinitely.

FIG. 11.

22. With reducing wheels the quick reversals of the direction of rotation that take place at high speeds make it necessary to use a stiff spring to overcome the inertia of the wheel. To reduce the inertia, the wheels are made as light as is practicable ; with many reducing wheels, lightness is secured by the use of aluminum. The better class of reducing wheels can now be successfully used for nearly any speed of rotation likely to be met with.

INDICATOR CONNECTIONS.

23. Indicator Cords.—In order to transmit the motion from the reducing motion to the paper drum with as little loss or distortion as possible, it is necessary to use a cord that will stretch but little. To meet this requirement, indicators are generally supplied with a special braided cord that will give good results for most purposes. In the case of large engines, where long cords are required, the amount of stretch with the best cord obtainable is considerable and may result in a distortion that would be undesirable for accurate tests. For such cases a fine copper or steel wire may be used to advantage. It is always best to so arrange the reducing motion and indicator that the cord may be led to the paper drum in the shortest and most direct practicable line. When the cord is attached to a pin on the reducing motion, it must be guided so as to leave the pin in the line of its motion, as is illustrated in Figs. 6, 8, and 9; the use of guide pulleys should, however, be avoided as much as it is practicable.

24. Stop Motions.—Various means are used to stop the motion of the paper drum when it is required to change the paper or when the indicator is not in use. A common method is to have a short cord attached to the paper drum with a hook *a*, Fig. 12, on the end; the cord from the reducing motion has a loop into which the hook may be fastened when it is desired to operate the paper drum. The length of the cord from the reducing motion can readily be made adjustable by the use of a loop *l*, formed as shown in Fig. 12. The thin strip of wood or metal *b* provides a very ready means of changing the length of the loop and of tying it securely in any position. To prevent the cord leading from the reducing motion from being thrown about and getting tangled when it is unhooked from the cord leading to the paper drum, it is well to have a rubber band fastened in a convenient position and provided with a hook into which the loop *l* may be secured. The elasticity of the rubber band can thus be made to keep the cord stretched and to prevent it from being tangled and broken.

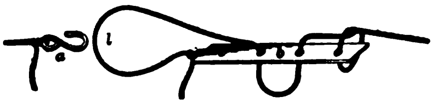


FIG. 12.

25. Paper drums are sometimes provided with a stop motion that will hold them in place and prevent the cord from being wound on; this merely has the same effect as lengthening the cord, but is open to the objection that at high speeds the loose cord is apt to make trouble by flying about and getting caught. In addition to the stop motions above noted, indicator manufacturers have designed a number of very useful devices, some of which absolutely prevent any trouble with the cord and make it easy to start and stop the paper drum.

SPECIAL ATTACHMENTS.

26. Simultaneous Diagrams.—With an engine having two or more cylinders it is sometimes desirable to take a diagram simultaneously from each end of all the cylinders so as to get a record of what takes place in each cylinder at some particular time. It is difficult for a number of operators to apply the pencils of a set of indicators to the paper all at the same time; and, to overcome this difficulty, a number of devices have been invented by means of which the pencils of all the indicators can be simultaneously operated by one person. Of these devices the simplest and most successful is an electromagnet that is attached to the indicator. When a number of indicators are to be operated simultaneously, the electromagnets of all are connected by a wire; when a current is sent through the wire by pressing a button or key, each electromagnet pulls its pencil against the paper and holds it there until the circuit is opened.

INDICATOR DIAGRAMS.

DIRECTIONS FOR TAKING DIAGRAMS.

27. The makers of indicators furnish very complete instructions for the care and use of their instruments; these instructions should be carefully studied before attempting to use a new indicator or one with which the user is not thoroughly familiar. The following directions for taking diagrams apply to all makes of indicators: Before attaching the indicator to the engine, see that it is clean and in good working order. The piston should move freely. See that the joints of the various levers and links are oiled with fine oil and that they are slack enough to avoid friction, yet not so slack as to allow the pencil to shake. Adjust the pencil so that it just touches the paper and sharpen the point so that it makes a very fine light line. A heavy coarse line on a diagram indicates poor work.

Select a spring that will give a diagram about $1\frac{1}{4}$ or $1\frac{3}{4}$ inches in height. If, upon trial, the spring chosen makes a wavy line, choose a stiffer one. A stiffer spring is required on a fast-running engine than on a slow-running engine when the steam pressure is the same. See that there is no backlash between the piston and spring.

Adjust the length of the cord so that the drum turns backwards and forwards without striking either of the stops at the end of the travel. When it touches one or the other of the stops, the cord is either too short or too long. If it touches both, the travel of the drum is too great, and the cord must be fastened to a point on the reducing motion having less travel.

Keep the drum moving only when taking diagrams. Unhook the cord before putting a paper on the drum. In putting on the card, see that it fits the drum without wrinkles, and fold back the projecting edges over the clips *m*, Fig. 1, so that they will not touch the pencil lever.

28. Before taking the diagram, turn on the steam a minute or so to warm the indicator; then press the pencil lightly on the paper long enough to take a single diagram. Shut the cock and again press the pencil to the paper. Since the indicator piston is then only subjected to atmospheric pressure, the pencil will make a straight line called the atmospheric line. Disconnect the cord and remove the card. Write on the card the scale of the spring used, the speed of the engine, and any other desired particulars.

If one indicator is used for both ends, first open the three-way cock to admit steam from one end. Take the diagram and open the cock to the other end, and take the diagram from that end. Then shut off the steam entirely and take the atmospheric line.

GENERAL FEATURES.

29. Purpose.—In actual practice the imperfections in the construction of the engine and the velocity at which the steam must flow through the pipes and ports combine to modify the pressures in the cylinder and, in consequence, the form of the diagram drawn by the indicator pencil. By a careful study of the peculiar features of the diagram, an experienced engineer is able to determine with a considerable degree of certainty the general type and condition of the engine and the circumstances under which the diagrams were taken. The following general outline of the characteristic features of diagrams taken under different conditions will enable the student to interpret most of the diagrams with which he will meet.

30. Points and Lines of the Diagrams.—In Figs. 13 and 14 are shown indicator diagrams from the crank end

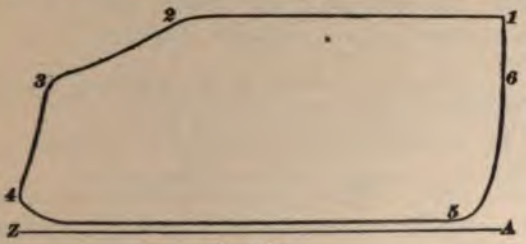


FIG. 13.

FIG. 14.

and head end of an engine. The different *points* of the stroke are plainly shown. They are as follows:

1 is the beginning of the stroke.
2 is the point of cut-off.
3 is the point of release.
4 is the end of the stroke.
5 is the point of compression.
6 is the point of admission.

The *lines* included between any two of these points have received special names, which are as follows:

6–1 is the admission line.
1–2 is the steam line.
2–3 is the expansion curve.
3–4–5 is the period of release.
4–5 is the back pressure line.
5–6 is the compression curve.
A Z is the atmospheric line.

31. The Vacuum Line.—It is sometimes desirable to have the vacuum line (line of no pressure) on the card also. The vacuum line may be drawn below and parallel to the atmospheric line. The distance between them will be $\frac{14.7}{\text{scale of spring}}$ inches. Thus, if the scale of the indicator spring is 30, the vacuum line lies $\frac{14.7}{30} = .49$ inch below the atmospheric line. When great accuracy is desired, the vacuum line should be located in accordance with the indication of the barometer. This is especially desirable when the engine is located at a great altitude above sea level. Then the distance between the atmospheric and vacuum lines may be found by multiplying the reading of the barometer in inches by .49 and dividing by the scale of the spring. For instance, if the barometer stands at 25 inches and the scale of the indicator spring is 30, the vacuum line should be drawn at a distance of $\frac{25 \times .49}{30} = .41$ inch from the atmospheric line.

32. Two Diagrams on a Single Card.—If but one indicator is used, the two diagrams are taken on the same blank, as shown in Fig. 15. With the diagrams placed one over the other, as shown, it is very easy to tell what is taking place in the cylinder at any point of the stroke. On the forward stroke the pencil of the indicator describes the line *A B C D* of the head diagram if the cock is opened to the head end, or it describes the line KLM if the cock is opened to the crank end. Likewise, the lines $GHJK$ and DEF are described during the return stroke.

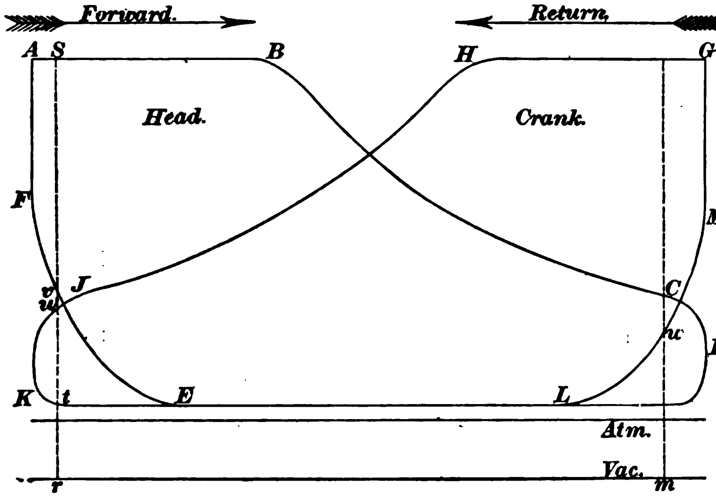


FIG. 15.

Suppose the piston is at a position corresponding to r on the forward stroke; the pressure (above vacuum) urging the piston forwards is rS, while the pressure resisting is rt. Hence, the net pressure on the piston is St. Suppose, now, that the piston is at r on the return stroke; the pressure at the right urging the piston on is ru, while the pressure on the left is rv. The net pressure is, therefore, uv, and is negative; or, in other words, the resistance is greater than the effort.

33. A double diagram of this character tells at a glance what is taking place at either end of the cylinder at any point of the stroke. Thus, when the piston is on the forward stroke, in the position corresponding to m, the steam in the head end is at the point of release C. Draw a line through m perpendicular to the vacuum line. C lies on ABC, and since KLM is described at the same time as ABC, the intersection of the line through C with the line $K L M$ is the point corresponding to C. The intersection w is on the compression line; hence, when release occurs in the head end, compression is taking place in the crank end.

34. Effect of Type and Speed of Engine on Form of Diagram.—The form of a good diagram depends largely

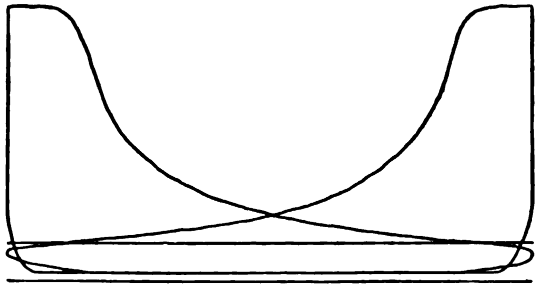


FIG. 16.

on the type of the engine, style of valve, and speed. What would be considered a good diagram from a locomotive or from a high-speed automatic engine would be considered very poor if taken from a Corliss engine. In general, a diagram taken from an engine with releasing gear of the Corliss type will be regular and show but little compression.

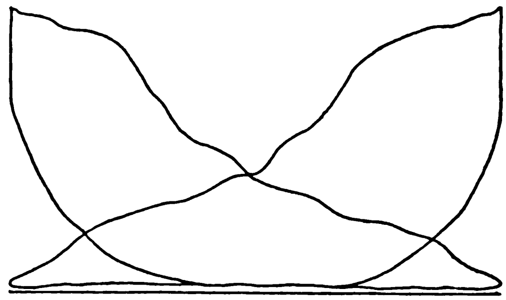


FIG. 17.

The point of cut-off, release, and compression will be sharply marked. The diagram shown in Fig. 16 is what may be expected from this type of engine when the valves are correctly set and in good working order. The fact that the back-pressure line runs below the atmospheric line shows plainly that the engine the card was taken from was condensing. On the other hand, Fig. 17 shows the form of diagram that may be expected from an engine running at 250 to 300 revolutions per minute. On account of the high rotative speed, the lines are irregular, due to the inertia of the moving parts of the indicator. The compression is large, as it should be for engines running at a high speed. The point of cut-off is never very sharply marked.

It is readily seen how totally unlike are the diagrams shown in Figs. 16 and 17, yet each is considered as representing good practice.

FAULTS IN VALVE SETTING REVEALED BY DIAGRAMS.

FAULTS IN STEAM DISTRIBUTION.

35. Some of the most common faults revealed by the indicator diagram are given below. In the diagram following, Figs. 18 to 22,

1 is the admission.
2 is the cut-off.
3 is the release.
4 is the compression.

I. Admission may be too early.
II. Admission may be too late.
III. Cut-off may be too early.
IV. Cut-off may be too late.
V. Release may be too early.
VI. Release may be too late.
VII. Compression may be too early.
VIII. Compression may be too late.

36. Case I. Admission Too Early.—The effect on the diagram of a too early admission is shown in Fig. 18. It is

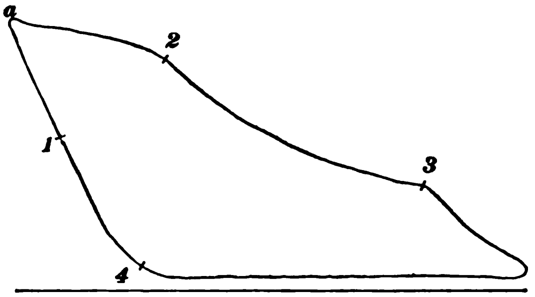


FIG. 18.

seen that the admission line *1a*, instead of being straight and perpendicular to the atmospheric line, as in Figs. 13 to 16, slants backwards. With a single slide valve or piston valve, all the other events, cut-off, release, and compression, are also too early. The remedy is to shift the eccentric on the shaft so as to decrease its angular advance.

In the case of the Corliss engine, the admission may be too early, while the other points are not affected. The fault may then be remedied by adjusting the link rods so as to give the steam valves more lap, and it may not be necessary to shift the eccentric.

The effect of too early admission is to introduce an excessive resistance to the motion of the piston before it reaches the end of the stroke; in consequence, the piston must be pushed to the end of its stroke by the momentum of the fly-wheel acting through the crank and connecting-rod. The result is likely to be pounding at the crosshead, crank, and main bearing.

37. Case II. Admission Too Late.—In this case the admission line *1a* on the diagram slants forwards, as shown in Fig. 19. The remedy is to increase the angular advance until the admission line *1a* becomes perpendicular to the atmospheric line. With a single slide valve in the case of a too late admission, the other events, as *2*, and particularly *3* and *4*, are also too late.

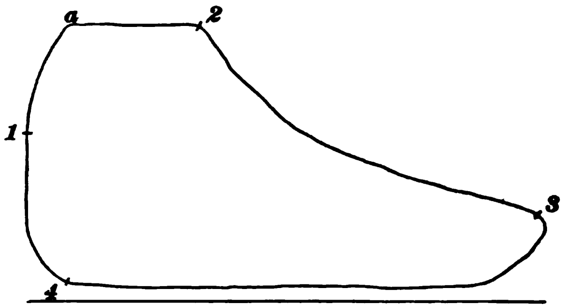


FIG 19.

In case the engine has a Corliss or other releasing gear, the admission may be made earlier by reducing the lap of the admission valves.

The effect of late admission on the running of the engine is not generally as noticeable or severe as is too early admission. It is, however, generally desirable to give the valves enough lead to have the clearance space filled with steam at the boiler pressure just as the piston begins its stroke.

38. Case III. Cut-Off Too Early.—See Fig. 20. Here the steam expands until its pressure is less than the back

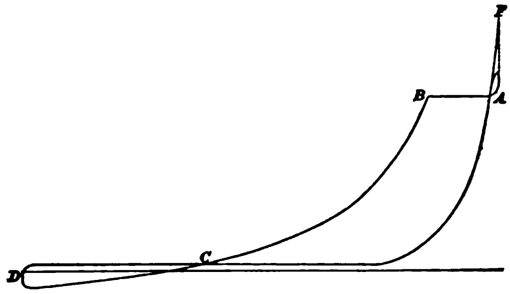


FIG. 20.

pressure; in consequence, the expansion line crosses the back-pressure line, as shown at *C*, and forms a loop. This effect is often observed in automatic cut-off engines working under a light load. It causes a reversal of the pressures on the piston that may result in pounding. The great range in pressure also has a bad effect on the steam consumption.

39. Case IV. Cut-Off Too Late.—See Fig. 21. Here it will be noticed that the terminal pressure is very high.

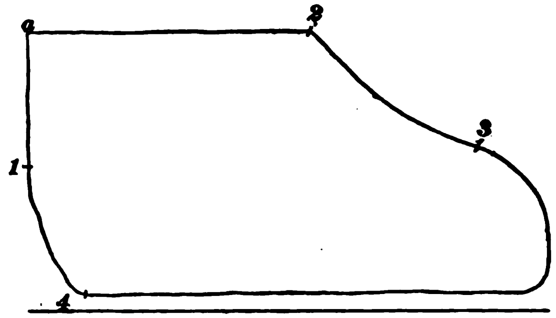


FIG. 21.

When this is the case, a great deal of the benefit of expansion is lost and there is a consequent waste of steam. With an ordinary plain slide-valve engine, the cut-off is always late, it not being practicable to cut off earlier than ½ stroke without seriously affecting the other events.

40. Case V. Release Too Early.—The appearance of the diagram for this case is illustrated in Fig. 18.

41. Case VI. Release Too Late.—This is illustrated in Fig. 19.

42. Case VII. Compression Too Early. — Figs. 20 and 22 show the effects of too early compression. The steam is compressed in the clearance space until its pressure rises above that of the steam in the steam chest; when the steam valve opens there is a flow of steam from the cylinder to the steam chest, as is shown by the loop, until the pressures in the cylinder and steam chest are nearly equal. If the steam valve has no lead, the compression line may rise above the admission line, as shown at *F* in Fig. 20; with lead, the loop will have the form shown in Fig. 22 and at *A* in Fig. 20.

Too much compression is likely to produce an effect on the running of the engine similar to that produced by too early admission; pounding and heating are often to be ascribed to this cause. It also reduces the effective work of the steam.

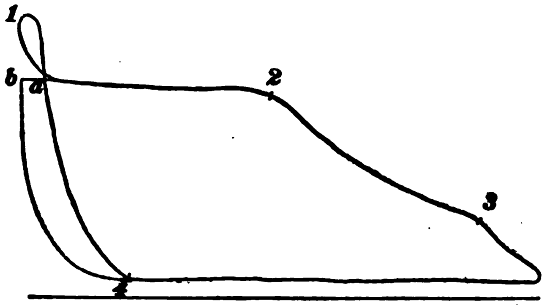


FIG. 22.

With the same cut-off and the proper amount of compression, an amount of work that is represented by the area *a b 4 a*, Fig. 22, included between the line *4 a* and the dotted line *a b 4*, plus the area of the loop, would be gained.

43. Case VIII. Compression Too Late.—This is shown on the diagram by a very short compression curve (see *4–1*, Fig. 19) or by the almost complete absence of such a curve.

REMEDIES FOR FAULTS IN STEAM DISTRIBUTION.

44. Most of the faults enumerated in the preceding articles are due either to incorrect valve proportions or to a fault in the setting; the remedy to be applied in any particular case will, therefore, be determined by a careful consideration of the type of engine and the conditions under which the diagram was taken. With a plain slide valve driven by a fixed eccentric, a change in the angle of advance of the eccentric will have an effect on all the events. Increasing the angle of advance will make admission, cut-off, and release take place earlier and increase the compression; decreasing the angle will have the opposite effect. A general rule is that an increase in the angle of advance of any eccentric with a fixed throw has the effect of making all the events controlled by that eccentric take place earlier, while a decrease in the angle will make them take place later. This rule applies to all types of valves and gears.

It is generally desirable to set the valves so that the work done will be equally divided between the two ends of the cylinder. If an indicator diagram shows a material difference in the work done in the two ends of a cylinder with a slide valve, the fault can be remedied by changing the length of the valve stem so as to make the cut-off take place earlier on the end doing the greater amount of work and later on the other. With an engine having a separate steam valve for each end of the cylinder, either valve may be adjusted so as to make it cut off earlier or later without affecting the other.

45. Remedy for Too Early Cut-Off.—The fault illustrated in Case III and Fig. 20 cannot be remedied by a change in the valve. It is found only in the case of automatic or adjustable cut-off engines working with a light load and a high boiler pressure. The cause is a steam pressure too great for the work to be done with the given size of cylinder and piston speed. The remedy is either to throttle the steam, lower the boiler pressure, or run the engine at a slower speed. That any of these remedies will have the desired effect will be made clear by a consideration of Fig. 23. The solid line *a b c* represents the expansion line of a diagram when cut-off is so early that the steam expands below the back-pressure line *d c*. The work done is represented by the area *a b f c* minus the area *f d c*. Now let the steam be throttled or the boiler pressure be lowered so that the initial pressure will rise only to *a'*. In order that the work done in the cylinder may be the same with this

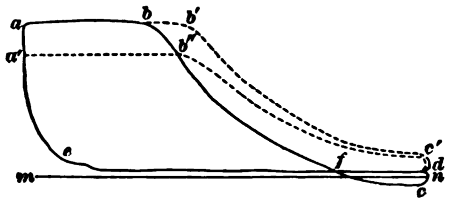


FIG. 23.

pressure as it was with the higher pressure, it is evident that cut-off must take place enough later to raise the expansion line $b'' c'$ far enough above the expansion line $b\ c$ to give the diagram $a'\ b''\ c'\ d\ e\ a'$ an area equal to the net area represented by the full-line diagram. On account of the later cut-off, the terminal pressure does not fall below the back-pressure line and no loop is formed.

If the number of revolutions per minute is reduced, the total work done by the engine remaining constant, it is evident that the work done during each stroke must be increased; this will require the admission of steam during a greater portion of the stroke, so as to produce a diagram having a greater area; expansion will begin later and the expansion line will be prevented from falling below the back-pressure line, as is indicated by the dotted line $b'\ c'$.

46. Remedy for Too Late Cut-Off.—With an *adjustable cut-off*, the fault illustrated by Fig. 21 can be remedied either by raising the boiler pressure so that the same area of diagram will be obtained with an earlier cut-off or by increasing the number of revolutions per minute so as to do the same work with a smaller average pressure. By either method the cut-off will be made to take place earlier in the stroke, and the expansion line will, in consequence, fall nearer to the back-pressure line.

47. Release Too Early or Too Late.—If the release is too early, there is danger of loss of pressure due to the escape of the steam too early in the stroke; on the other hand, if the release is too late, the escape of the steam will be so much retarded that the back pressure at the beginning of the return stroke will be excessive. Either of these will represent a loss of work. The valve should be so designed and set that the drop from the expansion line to the back-pressure line will occur as nearly as possible at the end of the stroke. If the engine is provided with separate steam and exhaust valves, this condition will best be fulfilled by setting the exhaust valves so that one-half of the fall in pressure occurs before the piston begins its return stroke;

the release line will then have a form that is well shown in Fig. 15. With a single-valve engine it is often very difficult to secure a satisfactory release line without seriously affecting the other events controlled by the valve.

48. Rules for Compression.—The best indication that the amount of compression is satisfactory is a quiet- and cool-running engine. At the end of the stroke the reciprocating parts must have their direction of motion changed; a force must act to stop them and reverse their direction of motion. By closing the exhaust before the end of the stroke, a part of the steam that would otherwise escape from the exhaust pipe and be lost is retained in the cylinder and acts as a cushion that helps to stop the reciprocating parts quietly. The energy given up by the reciprocating parts in being brought to rest, instead of being wasted in the production of knocks that would result in heating and rapid wear in the bearings, is stored in the steam compressed in the clearance space. The clearance space is thus filled with steam at a pressure more nearly equal to the boiler pressure, and the quantity of steam that must be taken from the boiler is correspondingly reduced.

If there is too little compression, the reciprocating parts will not be satisfactorily cushioned; if there is too much compression, the energy due to the motion will be absorbed before the end of the stroke; the piston must then be pushed by the crank. In either case the effect will be a sudden reversal in the pressures on the bearings that will produce shocks and heating.

49. No simple rule for determining the exact amount of compression to use can be given; however, it may be stated that the amount of compression required to secure quiet running varies with the speed of the engine, but in no case should the compression line extend above the initial or boiler pressure.

It is average practice to compress to about $\frac{9}{10}$ the initial pressure with high-speed engines, $\frac{5}{10}$ with medium-speed engines, and from $\frac{2}{10}$ to $\frac{3}{10}$ with slow-speed engines.

OTHER FAULTS REVEALED BY DIAGRAMS.

50. Introduction.—In addition to faults in valve design and setting, indicator diagrams furnish useful indications of the condition of the piston, cylinder, and valves; insufficient port opening, a steam pipe too small for its purpose, a cramped exhaust passage, or an obstructed exhaust pipe also have an effect on the appearance of the diagram that will generally make it comparatively easy to locate the fault. For a location of some of these faults, it is necessary to draw in the theoretical expansion line, or equilateral hyperbola.

51. Constructing the Theoretical Expansion Line. The general method of using the equilateral hyperbola for testing the expansion line of an actual indicator diagram is illustrated in Fig. 24, where the diagram $E\,C\,K\,L$ and the

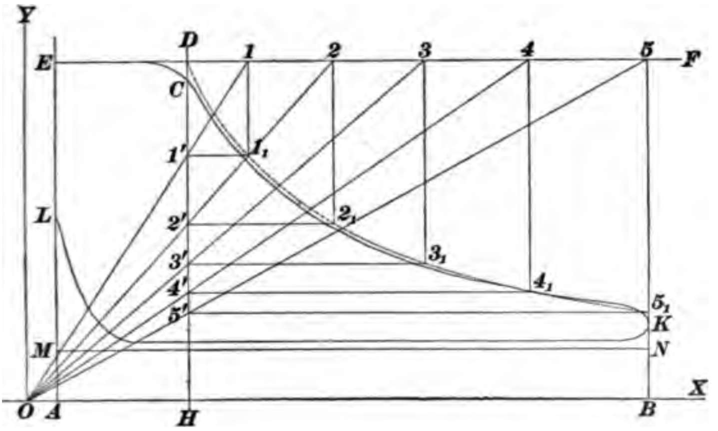


FIG. 24.

atmospheric line $M\,N$ represent the lines drawn by the indicator. To draw the theoretical expansion line on this diagram, first draw the vacuum line $O\,X$, as explained in Art. **31.** Perpendicular to $M\,N$ and $O\,X$ draw the two lines $A\,L$ and $B\,K$, just touching the two ends of the diagram. Measure the length $A\,B$ between these two perpendiculars, and this will give the length of the diagram. Multiply the length so found by the clearance volume of the end of the cylinder from which the diagram was taken, expressed in per cent., and divide by 100; lay off from A a distance $A\ O$ equal to the quotient. From O draw the perpendicular $O\ Y$; this is the **clearance line.** Through the highest point E of the steam line draw the horizontal line $E\ F$. Locate, as nearly as may be done by inspection, the point of cut-off C, and through this point draw the perpendicular $D\ H$. The point O where the vacuum line $O\ X$ (the line of no pressure) and the clearance line $O\ Y$ (the line of no volume) intersect represents the point of no pressure and no volume; the distance $A\ E$ or $H\ D$ represents the original absolute pressure; and $O\ H$ represents the original volume of the steam. To obtain points on the theoretical expansion curve, draw lines as $O1$, $O2$, $O3$, $O4$, $O5$ at random from O to the line $E\ F$. From the points of intersection of these random lines with the line $D\ H$, as the points $1'$, $2'$, $3'$, $4'$, and $5'$, draw lines parallel to the atmospheric line $M\ N$. Then, from the points of intersection 1, 2, 3, 4, and 5 of the random lines drawn from O with the line $E\ F$, drop perpendiculars intersecting the lines drawn from $1'$, $2'$, $3'$, $4'$, and $5'$ in 1_1, 2_1, 3_1, 4_1, and 5_1. These intersections are points on the theoretical expansion line; consequently, through them, by means of an irregular curve, trace the line D-1_1-2_1-3_1-4_1-5_1.

52. Relation Between the Theoretical and the Actual Expansion Lines.—Numerous tests have shown that when the piston and valves are tight so as to prevent leakage of steam to or from the cylinder after cut-off takes place, the actual expansion line will agree very closely with the theoretical. It will generally be found that the actual line falls somewhat lower than the theoretical, especially towards the end of the stroke, and then raises above the theoretical expansion line, as is shown in Fig. 24 by the crossing of the two lines near 4_1. This is thought to be due to the phenomena known as **cylinder condensation** and **reevaporation,** which may be explained as follows: During the period of exhaust, the cylinder walls are cooled by contact with the relatively cool low-pressure exhaust steam. When the hot steam from the boiler is admitted to the cylinder, a part of it condenses and gives up its latent heat to the cold walls and thus heats them again nearly to the temperature corresponding to the initial pressure; the water formed by this process of condensation is deposited in a thin film on the walls of the cylinder. After cut-off takes place and expansion begins, the pressure of the steam in the cylinder falls until its corresponding temperature is lower than the temperature of the cylinder walls; the walls then give up some of their heat and reevaporate some of the water. The steam thus formed towards the end of the stroke prevents the pressure from falling as fast as it otherwise would and has the effect of raising the actual expansion line somewhat above the theoretical.

53. Leaks Indicated by Expansion Line.—If the actual expansion line departs considerably from the theoretical, it is to be inferred that steam either enters or leaves the cylinder after cut-off takes place. Thus, in Fig. 25,

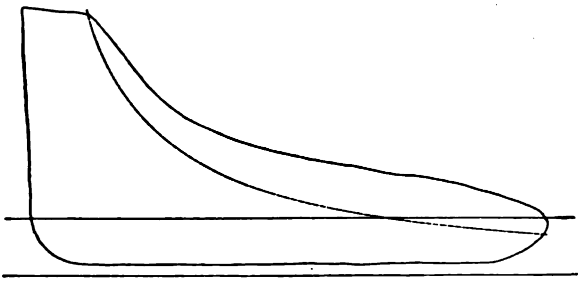


FIG. 25.

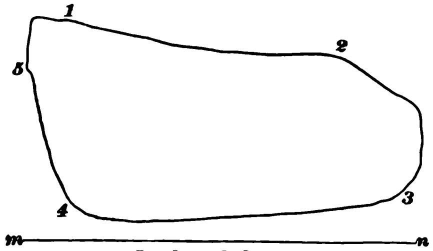
where the expansion line rises very markedly above the theoretical curve, it is evident that the valve leaks and allows steam to enter after cut-off. Similarly, if the expansion line fell below the theoretical curve, the inference would be that steam was escaping from the cylinder through a leaky exhaust valve or past an imperfectly fitting piston. An expansion line that closely follows the theoretical curve is not, however, conclusive evidence that the valves and piston are tight; steam may leak into and out of the cylinder at the same time and at such rates that the expansion line will appear to be quite satisfactory.

54. Determining the Point of Cut-Off.—It is sometimes very difficult to determine exactly the point of cut-off from the indicator diagram, especially when the engine has a high rotative speed. The most general method of determining it is to prolong the expansive line upwards by means of an irregular curve and note where it leaves the actual line of the diagram; then take the point of cut-off at or very near the point of deviation (see Figs. 24 and 25).

Instead of locating the point *D*, Fig. 24, at which to begin the theoretical curve by the method just explained, a method sometimes recommended is to prolong the expansion line by means of an irregular curve until it intersects the horizontal through the point representing the initial pressure. The point of intersection is then taken as the point through which to draw the vertical line *D H*, Fig. 24, to represent the volume at point of cut-off.

The rounding of the diagram near the point of cut-off is caused by the slowness of the valve movement at cut-off. On the Corliss and other releasing-gear engines, the valve cuts off very suddenly, the rounding is very slight, and the point of cut-off is very easily located.

55. Leaks Indicated by Compression Line.—If the piston and valves are tight, the compression line will generally curve quite regularly in one direction until it meets the admission line, as is shown by the diagrams in Figs. 15, 16, and 17. It often happens that, as shown at *5* in Fig. 26, the curvature of the compression line changes as the piston nears the end of the stroke; this change sometimes becomes so pronounced as to form a



Scale of Spring-50.

FIG. 26.

hook shown at *a*, Fig. 27. A compression line of this form indicates that steam is escaping from the compression

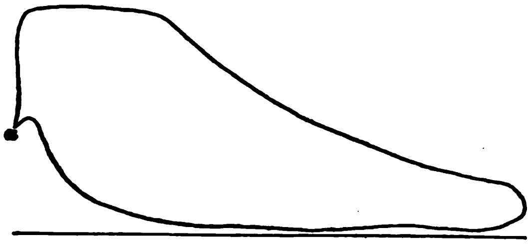


FIG. 27.

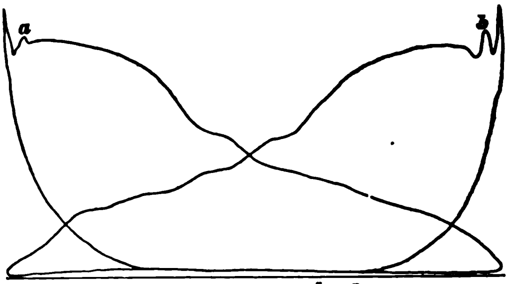
space; the loss may generally be ascribed to a leak either through the exhaust valve or around the piston.

56. Effect of Throttling.—When the steam pipe and its connections are of ample size and the steam ports are well opened, a nearly horizontal steam line whose height above the atmospheric line represents a pressure nearly equal to the boiler pressure may be expected, as is shown in Figs. 13 to 16. Any restriction in the passage leading from the boiler to the cylinder has the effect of preventing the flow of steam fast enough to keep the cylinder filled at boiler pressure up to the point of cut-off. This effect is shown on the diagram by a steam line that gradually falls as the piston advances. Fig. 26 is a diagram from an engine with a throttling governor; the effect of the governor in checking the flow of steam to the cylinder is shown by the drop in the steam line between the point *1* and the point *2*, where cut-off takes place. A long steam pipe or a pipe that is too small for its purpose, a partly closed throttle valve, or any similar obstruction will produce a drop in the steam line similar to the one shown in Fig. 26.

A high rotative speed generally results in a drop in the steam line, as is shown by the diagrams in Fig. 17. With shaft-governor engines, especially, the valve opening is often restricted and steam cannot follow the piston fast enough to keep the pressure up to that at the beginning of the stroke.

57. A high back pressure is caused by some obstruction that prevents the free escape of the exhaust. The exhaust from the engine from which the diagram shown in Fig. 26 was taken was discharged into a system of pipes for heating the building, and considerable pressure, as is shown by the height of the back-pressure line *3–4* above the atmospheric line *m n*, is required to force the steam through the coils of pipe. Somewhat similar results will be produced by a choked exhaust port or an exhaust pipe that is very long or too small.

58. Wavy lines on a diagram are generally due to vibrations of the pencil motion when there is a sudden change in pressure, such as takes place when steam is admitted to the cylinder of a high-speed engine or during the period of expansion. The indicator piston and the pencil motion are quickly set in motion and their inertia carries them beyond the point they would reach if the pressure were



50 Spring, 210 R.P.M., 12"x18" Engine.

FIG. 28.

gradually applied. This effect is well illustrated by the wavy lines at *a* and *b*, Fig. 28, which were produced by vibrations set up by the action of the steam at admission. The expansion lines of the same diagrams show a similar, but less violent, vibration. These effects are common with diagrams from quick rotative speed engines. They are an indication that the indicator piston is in good condition and working freely.

59. Sticking of the indicator piston is suggested by an expansion line that drops by a series of steps somewhat resembling notches (see Fig. 29). These steps or notches

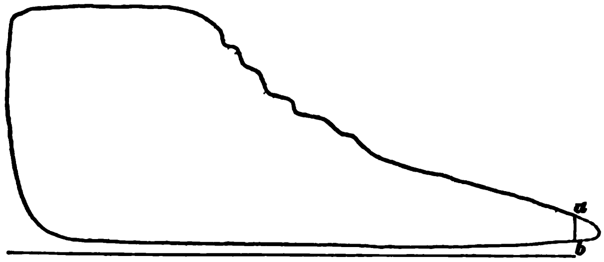


FIG. 29.

can generally be distinguished from the wavy lines produced by the inertia of the parts of an indicator with a free working piston by their angular appearance.

60. Striking of the paper drum against the stops is readily detected by the appearance of the release end of the diagram, which, instead of being rounded, as shown in Fig. 27, will have angular corners with a nearly vertical line connecting them, as shown in Fig. 29, where the full line *a b* shows the appearance of the end of the diagram when the drum struck the stop; the dotted line shows the change effected by giving the drum its full range of motion.

ENGINE TESTING.

THERMODYNAMICS OF THE STEAM ENGINE.

INTRODUCTION.

1. Thermodynamics is that branch of physical science that treats of the relation between heat and mechanical work.

2. A **heat engine** is a device by means of which energy, in the form of heat, developed by the combustion of fuel or derived from any other source is transformed into mechanical motion in such a manner that it can be made to do useful work.

3. The Steam Engine a Heat Engine.—The steam engine is in reality a heat engine, steam being merely a vehicle by means of which the heat energy developed by the combustion of the fuel in the furnace is transferred to the moving parts of the engine. From these moving parts the energy is transmitted by such vehicles as shafting, pulleys, belts, and electric currents to the point where it can be made do the required work.

The general principle on which the action of nearly all heat engines is based is the production of an expansive gas or vapor in a confined space. With the steam engine, water is heated in a closed boiler and changed to an expansive vapor—steam—whose pressure depends on the temperature to which it is heated. Other easily vaporized liquids, for example, naphtha, are sometimes used instead of water, and the action of their vapors depends on exactly the same principles as govern the action of steam.

4. Work Done by Expansive Force of a Gas.—All gases possess the property of expansibility, by virtue of which they expand and fill the space in which they are confined, no matter how great that space may be. This tendency to expand causes the gas to exert a pressure, called the **tension** of the gas, on the walls of the confining vessel. Keeping this principle in mind, let us consider a given volume of gas confined in a cylinder fitted with a movable piston. The gas in the cylinder tends to expand and thus exerts a pressure on the piston. If the force that resists motion is less than this pressure, the piston will be pushed outwards against the opposing force; the expansive force of the gas overcomes a resisting force, and in so doing does work.

5. Heat the Source of Work Done During Expansion. Careful experiments have shown that there is a fixed relation between work and heat and that heat can be changed into work and work into heat. A study of the effect on the gas of its expansion in the cylinder under such conditions that it does work will show that the work has really been done by heat.

To show that heat is the force that moves the piston, let the cylinder be so made that no heat can get to the gas as it expands; under these conditions a thermometer in the gas would show that it gets colder as it expands and pushes the piston along. The work has been done at the expense of a part of the heat of the gas and its temperature falls. In accordance with the theory of heat, the fall in temperature means that the molecules of the gas move slower; part of the kinetic energy represented by their rate of motion at the beginning of expansion has been expended in doing the work of pushing the piston against the resisting force.

6. If the cylinder is so arranged that enough heat can be added to the gas during expansion to keep its temperature *constant*, and a careful measurement of the heat added and the work done is made, it is found that the quantity of heat added is exactly equal to the heat represented by the work. For example, if the piston is pushed 4 feet by the expanding gas against an average resistance of 5,000 pounds, the work done is $4 \times 5{,}000 = 20{,}000$ foot-pounds. Since 778 foot-pounds of work is equivalent to 1 B. T. U. of heat, it follows that $20{,}000 \div 778$, or 25.707 B. T. U. of heat must be added to the gas to make up for the heat expended in pushing the piston and to keep the temperature constant.

7. Adiabatic Expansion.—When a gas expands and does work at the expense of its own heat—no heat being added to it from an outside source—the *expansion* is said to be **adiabatic.**

8. Isothermal Expansion.—When heat is added to a gas so as to keep the temperature constant during expansion, the *expansion* is said to be **isothermal.**

9. Compression of a Gas.—If we have a quantity of gas in a cylinder and push the piston inwards, so as to compress the gas and make it occupy a smaller space, we must do work in overcoming the expansive force of the gas; this work represents a certain amount of energy that is transferred to the gas. If the compression takes place under such conditions that no heat can leave the gas during the change in its volume, the energy represented by the work done on it will appear as heat and the temperature of the gas will be raised; under these conditions we have **adiabatic compression.**

If the compression takes place under such conditions that the heat represented by the work done is removed from the gas so as to keep its temperature constant, the *compression* is **isothermal.**

10. Relation Between Expansion and Compression. The quantity of work that must be done in compressing a gas adiabatically or isothermally from a given volume to a smaller one is exactly equal to the work that the gas can do when expanding, in the same way in which it was compressed, from the smaller volume to the original. Also, the rise in temperature during adiabatic compression and the quantity of heat that must be abstracted when the compression is isothermal are, respectively, equal to the corresponding fall of temperature and the quantity of heat that must be added during adiabatic and isothermal expansion.

11. Relation Between Work and Heat During Expansion or Compression.—In practice, it is seldom that the expansion is purely adiabatic or isothermal. No cylinder can be so made as to absolutely prevent the transfer of some heat to or from the gas, and it is difficult to impart or abstract heat so as to keep the temperature uniform. In any case, however, it is always found that there is a definite relation between the work done and the sum of the quantities of heat represented by the change in temperature of the gas and the heat imparted to or abstracted from it. This relation shows conclusively that the work done by an expanding gas is always a change of heat to work.

12. Expansion Diagrams.—The relation between the pressure and the volume of a gas during expansion may be represented by means of a graphical diagram. To illustrate, consider a cylinder *A*, Fig. 1, in which a piston *P* fits. The cylinder is attached to a reservoir *R* by a pipe *T* that permits air from *R* to enter the space *S* when the valve *V* is opened. A gauge *G* graduated so as to indicate absolute pressures, that is, so that the pointer stands at zero when there is a perfect vacuum in the space *S*, shows the pressure in the cylinder; a cock *C*, when opened, permits any air in the cylinder to escape when the piston is pushed back. Now, with the valve *C* open, push the piston clear back to the end of the cylinder, thus forcing out all the air; then close *C* and open *V*, so as to admit air from *R*, in which there is a constant pressure. Permit the piston to move slowly to the left with *V* open, and the gauge shows a constant pressure of the air in the space S. When the piston has moved a certain distance to the left, close V, so as to stop the admission

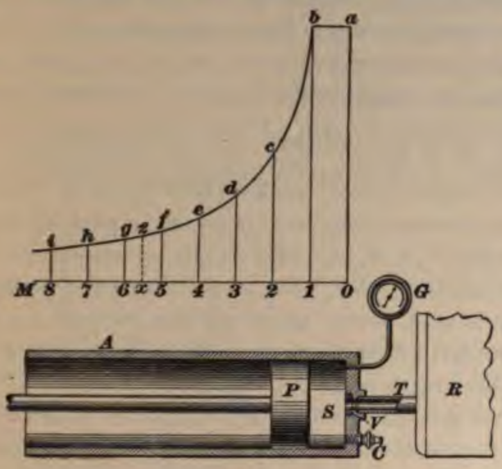


FIG. 1.

of air from the reservoir. Now, as the piston is permitted to move farther to the left, the gauge shows that the pressure falls. If the temperature in the cylinder is kept constant, it is found that when the piston is twice its original distance from the end and the air has expanded to twice its original volume, the pressure, in accordance with Mariotte's law, is only one-half the original pressure. When the volume is three times as great as the original volume, the pressure is found to be one-third the original pressure. When the volume has increased four times, the pressure is one-fourth as great, etc.

13. To represent this action graphically, draw a line OM to represent the piston motion and divide this line into a number of equal parts, O–1, 1–2, 2–3, etc., each of which, to some convenient scale, represents a motion of the piston through a distance equal to that through which it moved while V was open. Since the volume of air in the cylinder is proportional to the distance of the piston from the end of the cylinder, each of the sections O–1, O–2, etc. represents a volume equal to the original volume of air admitted to the cylinder from the reservoir, and the distances *O–1*, *O–2*, *O–3*, etc. represent the volume of the air in the cylinder for piston positions corresponding to the points *1*, *2*, *3*, etc.

From *O* draw a vertical line *O a*, and, to some convenient scale, make its length represent the pressure at the beginning of the piston stroke. Draw other vertical lines from the points *1*, *2*, *3*, etc., and, to the same scale as that to which *O a* was drawn, make their lengths represent the pressures corresponding to the piston positions represented by the points *1*, *2*, *3*, etc. and to the volumes represented by the distances *O–1*, *O–2*, *O–3*, etc. Since the pressure, when the piston is at *1*, is the same as the pressure at the beginning of the stroke, the length of the perpendicular *1–b* is the same as the length of *O a*. At *2* the volume is *O–2*, twice the original volume, and if the expansion is isothermal, the pressure is one-half the pressure at *1*; consequently, the length of the line *2–c* is one-half the length of *O a* or *1–b*.

Any desired number of points *c*, *d*, *e*, *f*, etc. can be located and a curve drawn through them. The distance of any point *x*, on the line *O M*, from the point *O* represents, to the scale of volumes, the volume of air in the cylinder when the piston is in the position corresponding to this point; likewise, the vertical distance *x z* from the point *x* to the curve represents, to the scale to which the pressures were laid off, the pressure for the corresponding piston position and volume.

14. The Isothermal Expansion Line, or Equilateral Hyperbola.—The curve that represents the relation between the pressure and volume when the temperature is constant is called the **isothermal expansion line.** This curve follows the law of the curve known in mathematics as the **equilateral hyperbola**; it is, therefore, often called by that name.

15. The Adiabatic Expansion Line.—If no heat is added to the air as it expands, that is, if the expansion is adiabatic, the gauge *G*, Fig. 1, shows a more rapid drop in pressure as the piston advances; each vertical line representing the pressure in the cylinder after expansion begins is shorter than the corresponding line of the isothermal curve; the curve drawn through their upper ends will, therefore, fall below the isothermal curve. The curve representing adiabatic expansion is called the **adiabatic expansion line.**

16. Comparison of the Isothermal and Adiabatic Expansion and Compression Lines.—In Fig. 2. the

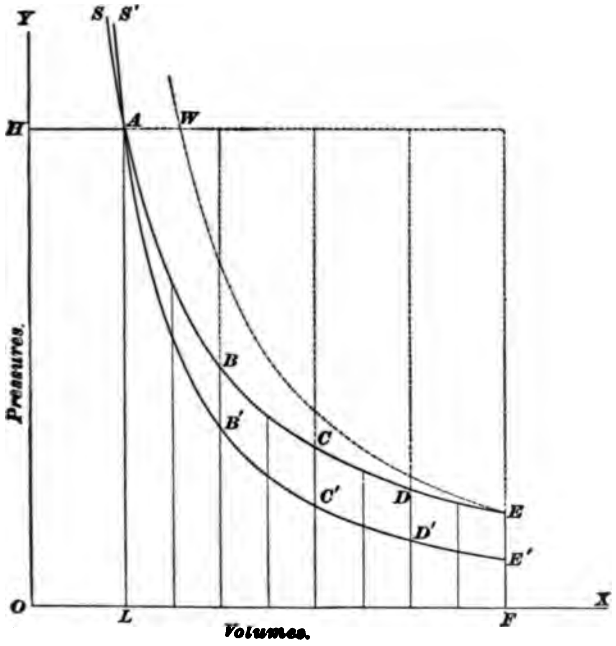


FIG. 2.

curve $A\,B\,C\,D\,E$ represents the isothermal, and $A\,B'\,C'\,D'\,E'$ the adiabatic, expansion line of a quantity of air whose original volume and pressure are, respectively, represented by the distances $O\,L$ and $O\,H$ or $L\,A$. If the air were compressed isothermally from the volume $O\,F$ and the pressure $F\,E$, the pressure would rise as the volume decreased, and the curve $E\,D\,C\,B\,A$ would show the relation between the volume and the pressure. If, with the same original volume and pressure, the air were compressed adiabatically, the curve representing the relation between the volume and the pressure would rise above the isothermal compression curve, as is shown by the dotted line $E\,W$. If a quantity of air, whose volume is represented by $O\,F$ and whose pressure by $F\,E'$, is compressed adiabatically, the curve representing the relation between the volumes and the pressures during the process of compression will be $E'\,D'\,C'\,B'\,A$, which is the same curve that represented the relation for adiabatic expansion from the volume $O\,L$ and the pressure $L\,A$.

17. Expansion of Steam.—When steam expands and does work, there is the same relation between heat given up and work done as has been explained for gas. Owing, however, to the properties of saturated steam, by virtue of which the pressure depends solely on the temperature and is independent of the volume, the relation between volume and pressure is not as simple as is the case with a perfect gas. For example, if a given weight of dry saturated steam expands adiabatically, a part of it will be condensed; while if the expansion is isothermal, the steam will be superheated during its expansion. If there is a mixture of steam and water, that is, if there is water in the vessel in which the steam expands, the relation between volume and pressure during expansion depends on the proportion of water in the mixture. As long as there is water present, the steam will be saturated and the pressure during isothermal expansion will be constant. This will be evident if we consider the fact that the pressure of saturated steam (steam in contact with water) depends solely on the temperature; if the temperature is constant, the pressure must also be constant, no matter what the volume may be. During isothermal expansion, the heat that is added merely changes some of the water to vapor, which fills the increased space, and there is no change in the pressure of the original steam.

If water is present during the adiabatic expansion of steam, it will give up some of its heat to assist the steam in doing its work; in consequence of the heat derived from the water, the temperature and pressure of the steam will fall slower during expansion as the quantity of water from which heat can be derived is greater.

18. Expansion Curve of Steam.—A consideration of the above outline of the effect of water on the expansive action of steam will make it clear that an innumerable variety of curves, depending on the quantity of water present and the conditions under which expansion takes place, will correctly represent the relation between the pressures and volumes for the expansion of saturated steam. It has, however, been found that under the conditions generally existing in the cylinder of a steam engine, the curve that most nearly represents the relation between pressures and volumes is the **equilateral hyperbola,** which is the curve that shows the relation between the pressures and the volumes of a perfect gas when it expands according to Mariotte's law.

CALCULATING THE WORK DONE ON A MOVING PISTON.

19. Net or Effective Pressure.—A piston that is being pushed through a cylinder by the expansive force of a gas or vapor acting on one side must generally overcome the resisting force of a gas or vapor on its opposite side. Thus, in Fig. 3, let the space in the cylinder at the left of the piston be in communication with the steam space of a boiler in which there is an absolute pressure of 100 pounds per square inch, while the space at the right is open to the atmosphere and, in consequence, is filled with vapor at a pressure of about 14.7 pounds

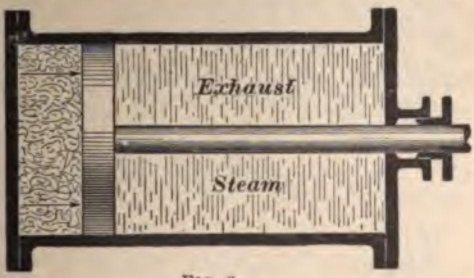


FIG. 3.

per square inch. It is evident that, neglecting the friction of the piston in the cylinder, the force that can be transmitted through the piston rod, and so made do work in overcoming some outside resistance, is the difference between the total pressure of the steam on the left and the total pressure of the air on the right of the piston. This difference is called the **net** or **effective pressure** on the piston. Since the pressure of the atmosphere is 14.7 pounds per square inch, and if the area of the piston is 100 square inches and the absolute pressure of the steam 100 pounds per square inch, the net pressure on the piston in Fig. 3 is $(100 - 14.7) \times 100 = 8{,}530$ pounds.

20. Rule for Calculating Work When Net Pressure and Piston Displacement Are Known.—The work done as the piston moves from one end of the cylinder to the other may be found as follows:

Let P = the net pressure per square foot exerted on the piston;
A = area of piston in square feet;
L = distance in feet moved over by the piston.

Then, the total net pressure on the piston is $P \times A$ pounds, and the distance through which this pressure acts is L feet. The work done is the force multiplied by the distance, or $PA \times L = PAL$ foot-pounds. But AL equals the area of the piston multiplied by the length of the stroke, which equals the volume displaced by the piston during its movement from one end of the cylinder to the other. Let V represent this volume expressed in cubic feet. Then, letting W represent the work in foot-pounds, we have $W = PAL = PV$.

It is usually more convenient to express pressures in foot-pounds per square inch instead of pounds per square foot. Let p represent the net pressure on the piston in pounds per square inch.

Then, $$P = 144\,p,$$

and $$W = PV = 144\,p\,V.$$

Rule 1.—*To find the work done by a piston moving in a cylinder, multiply 144 by the net pressure on the piston in pounds per square inch and by the volume displaced by the piston expressed in cubic feet. The result will be the work in foot-pounds.*

The same result will be obtained by multiplying the pressure in pounds per square inch by the volume displaced by the piston in cubic inches and dividing the result by 12.

The volume displaced by a piston during a single stroke or a given period of time is often called the piston displacement for the stroke or the given period.

EXAMPLE.—The piston of an engine is acted upon by a net pressure of $32\frac{1}{2}$ pounds per square inch. The volume swept through by the piston at each stroke is $5\frac{1}{2}$ cubic feet. (*a*) How much work is done at each stroke? (*b*) If the engine makes 80 strokes per minute, what horsepower does it develop?

SOLUTION.— (*a*) According to rule **1**, the work

$$W = 144 \times 32\frac{1}{2} \times 5\frac{1}{2} = 25,740 \text{ ft.-lb. Ans.}$$

(*b*) The number of foot-pounds per minute is $25,740 \times 80$, and the horsepower developed is, therefore,

$$\frac{25,740 \times 80}{33,000} = 62.4 \text{ H. P. Ans.}$$

21. Work Diagrams.—The work done by a moving piston may be represented by a diagram similar to the diagrams used to represent the relation between the volumes and pressures of an expanding gas or vapor. For example, in Fig. 4, two lines OX and OY are drawn at right angles, the line OX being horizontal and the line OY vertical. Suppose that the area of the piston is 2 square feet and that the distance moved by it is 6 feet. Then, when the piston moves 1 foot, it displaces a volume of 2 cubic feet. On the line OX lay off a distance *O–1*, and let this distance represent a piston travel of 2 feet. Then, the distance *O–2*, which is twice *O–1*, represents a piston travel of 2×2 feet $= 4$ feet, and, similarly, the distance *O–3*, which is three times the distance *O–1*, represents a travel of 3×2 feet $= 6$ feet.

Since the piston area does not change, the volume swept through is proportional to the piston travel; therefore, *O–1* may be taken to represent the displacement when the

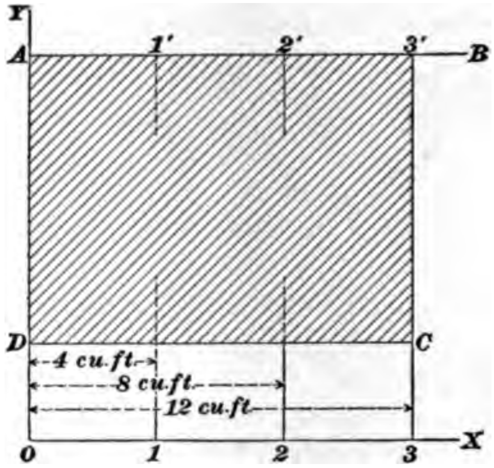


FIG. 4.

piston has traveled 2 feet. That is, *O–1* represents a volume of $2 \times 2 = 4$ cubic feet, *O–2* represents 8 cubic feet, and *O–3*, 12 cubic feet. The piston is supposed to be moving from left to right, that is, in the direction *O X*.

When the piston is at the beginning of its travel, that is, at the position represented by *O Y*, lay off on the line *O Y* a distance *O A*, which, to the scale selected, represents the pressure on the left side of the piston. Suppose the pressure is 60 pounds per square inch. Then, if *O A* is 2 inches, the scale is $\frac{60}{2} = 30$ pounds; that is, a vertical height of 1 inch represents 30 pounds per square inch pressure. Suppose the pressure to be the same throughout the stroke. Then, when the piston is at the point represented by *1*, the pressure is represented by the distance *1–1'*, which is equal to *O A*. Likewise, when the piston is in positions *2* and *3*, the distances *2–2'* and *3–3'*, respectively, represent the pressures at those points. In brief, the pressure upon the left side of the piston at any position may be found by measuring the vertical distance between the lines OX and AB at that point and multiplying by the scale, 30 pounds per inch of height. In a similar manner, lay off on the line OY a distance OD, which, to the scale already used, represents the pressure of the atmosphere on the right of the piston, and is, therefore, equal to $\frac{14.7}{30} = .49$ inch. Since this pressure on the right of the cylinder is constant throughout the stroke, the distance from any point on the line OX to the line DC parallel to OX represents the opposing pressure on the piston when it is at the corresponding point of its stroke.

22. The net pressure on the piston is represented by the distance DA $(= OA - OD)$. We have shown that, to the scale selected, $O\text{-}3 = DC$ represents the piston displacement. According to rule **1**, the work done by the piston is proportional to the net pressure multiplied by the volume. Now, on the diagram of Fig. 4, DA represents the net pressure and DC the volume. But $DA \times DC =$ area $A3'CD$. Hence, the area $A3'CD$ must, to some scale, represent the work done by the piston.

AO is 2 inches and OD .49 inch; hence the distance $DA = OA - OD$ is $2 - .49 = 1.51$ inches; DC equals 2 inches. Therefore, the area of the diagram is $1.51 \times 2 = 3.02$ square inches. The scale of pressure adopted was 1 inch equals 30 pounds per square inch. Hence, $p = 30 \times DA$. Since DC (= 2 inches) represents 12 cubic feet of volume, the scale of volumes must be $\frac{12}{2} = 6$ cubic feet per inch of length. Hence, $V = 6 \times DC$. Then, from rule **1**, the work is

$$\begin{aligned} W &= 144\,p\,V = 144 \times (30 \times DA) \times (6 \times DC) \\ &= 144 \times 30 \times 6 \times (DA \times DC) \\ &= 144 \times 30 \times 6 \times 3.02 = 78{,}278.4 \text{ foot-pounds.} \end{aligned}$$

23. The diagram may be used in another way. The distances $O\text{-}1$, $O\text{-}2$, and $O\text{-}3$ may represent the distances moved through by the piston instead of the volumes displaced by it. Then, DC represents the stroke of the piston, in this case 6 feet, and since $DC = 2$ inches, the horizontal scale is $\frac{6}{2} = 3$ feet of piston travel = 1 inch of length.

The work is

$$W = 144\, p\, A\, L.$$

As before,

$$p = 30 \times D\,A,$$
$$L = 3 \times D\,C,$$

and $A = 2$ square feet.

Hence,

$$\begin{aligned} W &= 144 \times (30 \times D\,A) \times 2 \times (3 \times D\,C), \\ &= 144 \times 30 \times 2 \times 3 \times (A\,D \times D\,C) \\ &= 78{,}278.4 \text{ foot-pounds.} \end{aligned}$$

The latter method is the one usually employed in calculating the horsepower of an engine by means of the indicator diagram.

24. Diagrams for Varying Pressures.—The diagram of Fig. 4 is very simple, because the pressure on both sides of the piston is constant throughout the stroke, thus making the diagram a rectangle. Suppose the pressure decreases uniformly throughout the stroke, as shown in Fig. 5. Here the net pressure at the beginning of the stroke is represented by the distance DA, and at the end of the stroke by the distance CB. To calculate the work, it is necessary to find the average net pressure throughout the stroke. In this case the diagram is a trapezoid; the average pressure is, therefore, represented by the line $HG = \frac{1}{2}(DA + CB)$. This distance HG is called the **mean ordinate** of the diagram $A\,B\,C\,D$. It has such a length that, being multiplied by the distance $D\,C$, it will give the area of a rectangle $E\,F\,C\,D$ that will be equal to the original area $A\,B\,C\,D$. The work is found by multiplying this mean ordinate by the length $D\,C$, then by the scales of pressures and volumes, and by 144.

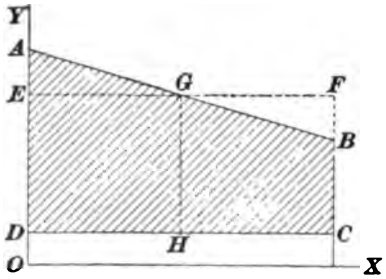


FIG. 5.

25. In Fig. 6 diagrams taken from both sides of the piston of an actual steam engine are shown on the same

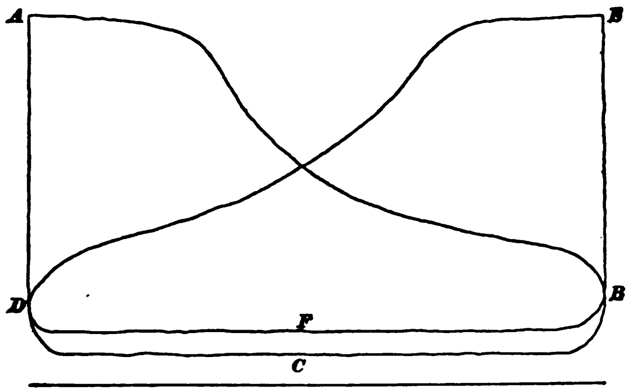


FIG. 6.

card. The line *A B* represents the varying steam pressure during the forward stroke and the line *B C D* of the crank-end diagram represents the back pressure opposing the motion of the piston during the forward stroke. Hence, the net pressure at any piston position is given by the vertical distance between the line *A B* of one diagram and the line *D C B* of the other diagram at the point representing the piston position. Likewise, the net pressure for any point of the return stroke is given by the vertical distance between the line *E D* of the crank-end diagram and the back-pressure line *B F D* of the head-end diagram. The net work done by the piston, as in the preceding cases, is given per stroke by the area *A B C D* for the forward stroke and the area *E D F B* for the return stroke. It will be noticed that the area *B C D F* has been taken from one diagram and added to the other diagram.

Now, to find the average work per stroke of a double-acting engine, the sum of the areas representing the work done during the forward and return stroke is divided by 2. Evidently, the sum of the areas will be the same whether we add the areas *A B C D* and *E D F B* or add the areas of each diagram, as $A B F D$ and $E D C B$. Hence, the *average work* will be correctly given by considering the area of each diagram as representing the work done on the side of the piston the diagram was taken from and dividing the sum of the areas by 2. While the assumption that the area of each diagram represents the net work done on its side of the piston of a double-acting engine is not entirely correct, it is, nevertheless, a very convenient assumption to make, and will *not* cause any error in finding the *average pressure* per stroke when both diagrams are considered. The convenience of making the assumption just explained is best exemplified in case of diagrams taken on separate cards; in that case it would be necessary to very carefully transfer the back-pressure lines from one card to the other in order to get the correct area representing the work done on each side of the piston. This is a tedious operation calling for considerable skill in the use of drawing instruments; the necessity for this operation is obviated by making the assumption stated.

In a single-acting engine, which takes steam on one side of the piston only, the other side of the piston being open to

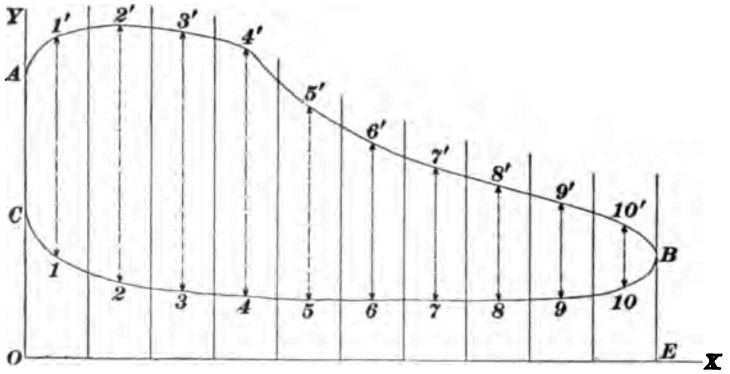


FIG. 7.

the atmosphere, the area of the diagram represents correctly the work done during the revolution. This can readily be seen by a consideration of Fig. 7, where $O E$ is the atmospheric line. The work done by the piston during the forward stroke is given by the area $OABE$, and the work that must be done on the return stroke to make the piston return to the beginning of the working stroke is given by the area $OCBE$. Hence, the net work is equal to the difference of the areas $OABE$ and $OCBE$, which is the area ABC (the area of the diagram).

26. To find the area of an indicator diagram, we must find its mean ordinate. This may be done approximately in the following manner: Divide the length OE of the diagram (see Fig. 7) into a number of equal parts (10 or 20 parts are most convenient) and through each division draw a vertical line. Half way between these vertical lines draw the lines 1–$1'$, 2–$2'$, 3–$3''$, etc., extending between the lines AB and BC. These vertical distances between the two curves are called **ordinates**. As shown in the figure, there are ten of these ordinates equally distant from each other. If their lengths are all added together and the sum divided by the number of ordinates, the result is the average distance between the lines, or the mean ordinate.

This ordinate multiplied by the distance OE gives the area of the diagram. Usually both the ordinate and OE will be measured in inches; the area will then be expressed in square inches. The area being found, the work is calculated by rule **1.** That is, multiply the area by the vertical scale of pressures, by the horizontal scale of volumes, and by 144. The result is the work in foot-pounds.

EXAMPLE.—The area of a diagram like that shown in Fig. 7 is found to be 7.34 square inches. The vertical scale of pressure is 36 pounds equals 1 inch, and the horizontal scale of volumes is $2\frac{1}{4}$ cubic feet equals 1 inch. What is the work per stroke of piston ?

SOLUTION.—Multiply the area by the horizontal and vertical scales, and by 144, or work $= 7.34 \times 36 \times 2\frac{1}{4} \times 144 = 95{,}126.4$ ft.-lb. Ans.

27. Work Diagram for Expanding Steam.—In connection with the diagram of Fig. 4, the piston area was taken as 2 square feet and the length of stroke as 6 feet. Fig. 8 shows the pressure diagram on the supposition that steam from the boiler is shut off when the piston has reached one-third its stroke. Up to that point steam has entered from the boiler at a constant pressure, shown by the line *A B*. The volume of steam in the cylinder at this point is 4 cubic feet. As the piston moves forwards, the pressure begins to fall. When two-thirds the stroke is completed, the steam that previously occupied 4 cubic feet now occupies 8 cubic feet; that is, its volume is doubled, and assuming the expansion to follow Mariotte's law, its pressure should be one-half what it was originally, or $\frac{1}{2}$ *b B*. When the piston reaches the end of the stroke, the steam occupies 12 cubic feet, or three times its original volume. Therefore,

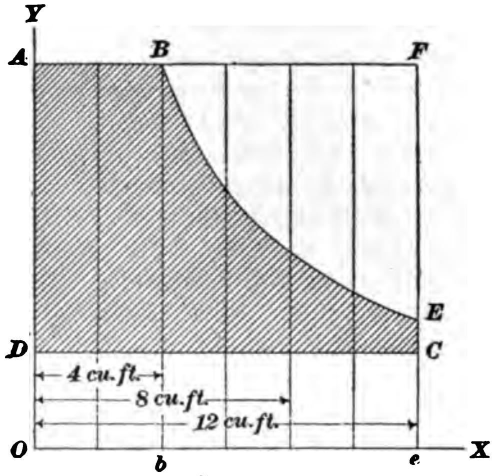


FIG. 8.

its pressure is one-third the original pressure; that is, $e E = \frac{1}{3} b B$. The line *B E* shows the fall of pressure during the last two-thirds of the stroke. When steam is shut off from the boiler in this manner and does work at the expense of its own heat, it is said to be used expansively.

28. Gain in Work by Using Steam Expansively.—It was found that the area of the diagram of Fig. 4, which represents the work done when the steam followed the piston full stroke, was 3.02 square inches, and the work done per stroke of piston was 78,278.4 foot-pounds. The area of the diagram *A B E C D*, Fig. 8, which represents the work done when the steam supply to the cylinder was cut off when one-third of the piston stroke was completed, is nearly 1.82 square inches. The work done per stroke is, therefore, $1.82 \times 30 \times 6 \times 144 = 47,174.4$ foot-pounds.

In the first case, a cylinder full of steam, 12 cubic feet, was taken from the boiler, and the work obtained from each cubic foot was, therefore, $\frac{78,278.4}{12} = 6,523.2$ foot-pounds. In the second case only 4 cubic feet of steam was taken from the boiler. Consequently, the work done by each cubic foot of steam used was $\frac{47,174.4}{4} = 11,793.6$ foot-pounds, or nearly twice as much as was done by a cubic foot when the steam followed the piston for the full stroke.

EXAMPLES FOR PRACTICE.

1. The mean ordinate of a diagram similar to that shown in Fig. 7 is 1.2 inches long. The vertical scale of pressure is 1 inch = 40 pounds per square inch, and the horizontal scale of distances is 1 inch = 10 inches. The length of the diagram is 3 inches, and 1 foot of actual length of the vessel that contains the steam represents a volume of 452 cubic inches. What is the work done in one stroke of the piston? Ans. 4,520 ft.-lb.

2. The mean ordinate of a diagram is .89 inch; the length of the diagram, 3.2 inches; the vertical scale of pressures, 1 inch = 50 pounds per square inch; the horizontal scale of volumes, 1 inch (diagram) = .56 cubic foot. Find the work done in 12 strokes. Ans. 137,797.6 ft.-lb.

HORSEPOWER OF STEAM ENGINES.

INDICATED HORSEPOWER AND NET HORSEPOWER.

29. The relation between the pressures on the two sides of a moving piston and the work done on the piston was explained in Arts. **19** to **26**, and the student is advised to carefully review the explanation there given in conjunction with his study of this section. When the work done in a given period of time is known, the corresponding horsepower is obtained as follows: *Having the work given in foot-pounds per minute, to find the horsepower divide by 33,000; if the work is given in foot-pounds per second, the horsepower is found by dividing by 550.* Horsepower is often abbreviated to H. P.

30. Indicated Horsepower.—The indicator furnishes the most ready method of measuring the pressures on the piston of a steam engine and, in consequence, of determining the amount of work done in the cylinder and the corresponding horsepower. The power measured by the use of the indicator is called the **indicated horsepower.** It is the total power developed by the action of the net pressures of the steam on the two sides of the moving piston. The indicated horsepower is generally represented by the initials I. H. P.

31. Friction horsepower is the part of the indicated horsepower that is absorbed in overcoming the frictional resistances of the moving parts of the engine. If the engine is running light—with no load—all the power developed in the cylinder is absorbed in keeping the engine in motion, and the friction horsepower is equal to the indicated horsepower. This principle furnishes a simple approximate method of finding the friction horsepower of a given engine; since, however, the friction between the surfaces increases with the pressure, the power absorbed in overcoming engine friction will be greater as the load on the engine is increased.

32. Net horsepower is the difference between the indicated and the friction horsepower. It is the power the engine delivers through the flywheel or shaft to the belt or the machine driven by it, and is sometimes called the **delivered horsepower.** Since the power an engine is capable of delivering when working under certain conditions is often measured by a device known as a *Prony brake*, the net horsepower is also called the **brake horsepower.**

33. The **mechanical efficiency** of an engine is the ratio of the *net horsepower* to the *indicated horsepower;* or it is the percentage of the mechanical energy developed in the cylinder that is utilized in doing useful work.

To find the efficiency of an engine, when the indicated and net horsepowers are known:

Rule 2.—*Divide 100 times the net horsepower by the indicated horsepower.*

EXAMPLE.—The indicator diagrams taken from an engine running under full load show the I. H. P. to be 238.5. The diagrams taken when the engine is running at the same speed under no load show a horsepower of 39.7. (*a*) What is the approximate net H. P. developed by the engine? (*b*) What is the efficiency of the engine?

SOLUTION.— (*a*) Approximate net H. P. = I. H. P.—friction H. P. =

$$238.5 - 39.7 = 198.8. \quad \text{Ans.}$$

(*b*) By rule **2**, the efficiency is

$$\frac{100 \times \text{net H. P.}}{\text{I. H. P.}} = \frac{100 \times 198.8}{238.5} = 83.4 \text{ per cent.} \quad \text{Ans.}$$

The mechanical efficiency of a good engine is from 75 to 90 per cent.

MEASURING THE INDICATED HORSEPOWER.

34. In accordance with the principles explained in Arts. **19** to **26,** when the net pressure on the piston and the piston displacement for a given period of time are known, the work done during the given period can be calculated. The usual period of time considered when calculating the power of an engine is 1 minute; since 33,000 foot-pounds of work per minute is equal to 1 horsepower, the horsepower is obtained by dividing the work done in one minute by 33,000.

FINDING THE M. E. P.

35. The **mean effective pressure,** or M. E. P., is defined as the average pressure urging the piston forwards during its entire stroke in one direction, less the pressure that resists its progress.

36. The mean effective pressure may be found in three ways:

1. The area of the diagram in square inches may be measured by an instrument called the *planimeter;* the M. E. P. is then found by dividing the area of the diagram in square inches by the length of the diagram in inches and multiplying by the scale of the spring.

EXAMPLE.—The area of the diagram is 4.2 square inches and the length is 3.5 inches; a 40 spring being used, find the M. E. P.

SOLUTION.— $\frac{4.2}{3.5} \times 40 = 48$ lb. per sq. in., M. E. P. Ans.

2. A special form of planimeter may be used by means of which the M. E. P. may be measured directly.

3. Where a planimeter is not available, the M. E. P. may be found with a fair degree of accuracy by multiplying the length of the mean ordinate by the scale of the spring.

37. The Planimeter.—A common form of this instrument is shown in Fig. 9. It consists of two arms hinged

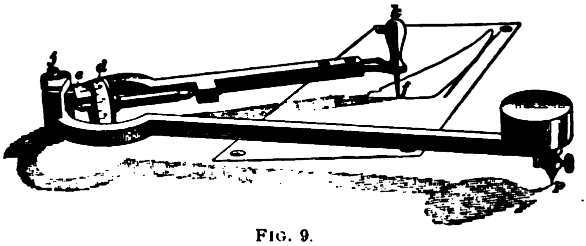


FIG. 9.

together by a pivot joint at *j*. One arm carries a recording wheel *d*, which rolls on the surface to which the card is fastened, while the outline of the diagram is being traced by the point *f*. The needle point *p* is fixed in the paper or drawing board, and remains stationary during the operation.

The indicator card should be fastened to a smooth table or drawing board that has been previously covered with a piece of heavy unglazed paper or cardboard. The point *p* should be placed far enough from the card to enable the wheel to roll on the unglazed paper without touching the card, as it will slip if rolled over a smooth surface. Set the zero of the wheel *d* opposite the vernier *e*; then, with the tracing point *f*, follow the line of the diagram carefully, *going around the diagram in the direction of the hands of a watch*, and stop exactly at the starting point.

38. Reading the Vernier.—The area is read from the recording wheel and vernier as follows: The circumference of the wheel is divided into 10 equal spaces by long lines that are consecutively numbered from 0 to 9. Each of these spaces represents an area of 1 square inch and is subdivided into 10 equal spaces, each of which represents an area of .1 square inch. Starting with the zero line of the wheel opposite the zero line of the vernier and moving the tracing point once around the diagram, the zero of the vernier will be opposite some point on the wheel; if it happens to be directly opposite one of the division lines on the wheel, that line gives the exact area in tenths of a square inch. The zero of the vernier, however, will probably be between two of the division lines on the wheel, in which case write down the inches and tenths that are to the left of the vernier zero, and from the vernier find the nearest hundredth of a square inch as follows: Find the line of the vernier that is exactly opposite one of the lines on the wheel. The number of *spaces on the vernier* between the vernier zero and this line is the number of hundredths of a square inch to be added to the inches and tenths read from the wheel. An example is presented in Fig. 10, where the 0 of the vernier lies between the lines on the wheel representing 4.7 and 4.8 square inches, respectively, showing that the area is something more than 4.7 square inches. Looking along the vernier it is seen that there are three spaces between the vernier zero and the line of the vernier that coincides with one of the lines on the wheel; this shows that .03 square inch is to be added to the 4.7 square inches read from the wheel, making the area

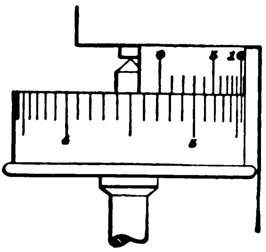


FIG. 10.

4.73 square inches, to the nearest hundredth of a square inch.

39. Modifications of Planimeter.—Planimeters are made in a number of different forms, some of which differ considerably from that shown in Fig. 9. One of the most important modifications in the general form is found in the **Lippincott** and the **Willis** planimeters. In these instruments the wheel, in addition to its rotary motion, slides in the direction of the axis of its spindle, and the area is indicated by the amount of this sliding motion as measured by a scale parallel to the axis. The **Coffin averaging instrument** is another modification, in which the end of the bar to which the wheel and vernier are attached is guided along a straight line by a slot instead of being jointed to another bar.

40. Measuring the M. E. P. Directly.—With the planimeter illustrated in Fig. 9, the M. E. P. is found by dividing the area as measured by the instrument by the length of the diagram and multiplying the quotient by the scale of the spring. Many planimeters, however, including those mentioned in the last article, can be used to measure the M. E. P. directly, no calculation being required. For this purpose, special adjustments and scales are provided by means of which the instrument can be set to correspond to the length of the diagram and the scale of the spring. The makers furnish complete instructions for the use of each of these special attachments.

41. Hints for Use of Planimeter.—It is well to so place the fixed point (*p*, Fig. 9) of the instrument that, as the tracing point moves around the diagram, the arms will swing about equally on each side of a position at right angles with each other. A slight dot is generally made with the tracing point to mark the point at which its motion around the diagram begins; when the tracing point reaches this dot in the paper, the operator knows that the motion around the diagram has been completed. The direction of motion of the tracing point must always be the same as that of the hands of a watch; motion in the opposite direction will move the wheel in the wrong direction and give a negative reading for the area.

When measuring diagrams with loops, like Fig. 11, move the tracing point so that it will follow the outline of the loops in a direction opposite to the direction of motion of the hands of a watch, as is indicated by the arrowheads on the diagram, Fig. 11. This will cause the instrument to automatically subtract the areas of the loops from the area of the main part of the diagram.

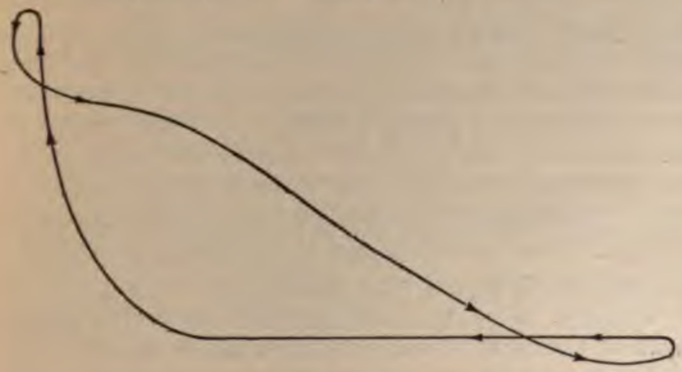


FIG. 11.

An excellent check on the work is to start with the recording wheel at zero and pass the tracing wheel around the diagram two or three times, noting the reading of the wheel each time the tracing point returns to the point of starting. Each reading of the wheel divided by the number of times the outline of the diagram has been traced should give, very nearly, the value of the first reading; if there is a considerable difference between the first reading and the value obtained by dividing the second reading by 2 or the third reading by 3, it is an indication that an error has been made, and the work should be repeated. If the difference is small, the work may be assumed to be satisfactory and the value to be used for the area or the M. E. P. may be taken as the average found by dividing the last reading by the number of times the tracing point passed around the diagram.

42. Finding the M. E. P. by Ordinates.—This operation may be performed by the aid of two triangles, a scale, and a hard lead pencil; if two triangles are not available, a single triangle and a straightedge will suffice. Lines perpendicular to the atmospheric line and tangent to the two ends of the diagram must first be drawn; the perpendicular distance between these tangents will be the length of the diagram, and this length must be divided into some number of equal parts (10 or 20 parts are the most convenient, but any other number may be used). Midway between any of the points of division draw a line parallel to the two tangents; the part of this line included between the lines of the diagram is the middle ordinate of its corresponding space. The sum of the lengths of all of these middle ordinates divided by the number of spaces is the mean ordinate and gives, approximately, the average height of the diagram. The length of the mean ordinate should agree very nearly with the value obtained by dividing the area of the diagram—as measured by a planimeter—by the length of the diagram. The M. E. P. is found by multiplying the length of the mean ordinate by the scale of the spring with which the diagram was taken.

If a scale graduated to correspond with the scale of the spring is available, the M. E. P. may be obtained by measuring the ordinates in pounds instead of in inches; the sum of the lengths of the ordinates as so measured divided by their number gives the M. E. P. of the diagram. For example, let the scale of the spring be 40, then each $\frac{1}{40}$ inch in the length of an ordinate represents a pressure of 1 pound per square inch, and by measuring the length of an ordinate with a scale graduated in fortieths of an inch, the number of pounds pressure represented by that ordinate is found.

43. A convenient method of finding the sum of the lengths of the ordinates of a diagram, and one that is especially to be recommended when a decimal scale is not available, is the following: Take a strip of paper having a straight edge a little longer than the sum of the lengths of the ordinates. Lay this strip along the first ordinate. From the point on the strip representing one end of the first ordinate lay off the length of the next ordinate. In the same way lay off on the strip the length of each of the ordinates in succession. The length of the strip included between the extreme, or first and last, points so marked will be equal to the sum of the lengths of the ordinates, and this length divided by the number of ordinates will give the length of the *mean ordinate*.

EXAMPLE.— (*a*) The lengths between the extreme points on a strip of paper on which has been laid off successively the lengths of the 10 ordinates of an indicator diagram is $12\frac{5}{16}$ inches. What is the length of the mean ordinate to the nearest .001 inch? (*b*) The diagram was taken with a 20 spring; what is the M. E. P.?

SOLUTION.— (*a*) Reducing the fractional parts of the sum of the lengths of the ordinates to a decimal, we have $\frac{5}{16}$ inch = .3125 inch. The length of the mean ordinate is, then, $12.3125 \div 10 = 1.23125$ inches, or to the nearest .001, 1.231 inches. Ans.

(*b*) Multiplying the length of the mean ordinate by the scale of the spring, the M. E. P. is $1.231 \times 20 = 24.62$ lb. per sq. in. Ans.

44. Locating the Ordinates.—The length of the diagram will seldom be divisible into equal parts that can readily be laid off by a scale, and to divide the length into equal parts by a cut-and-try process will be found very tedious. These difficulties may, however, be overcome by an application of a simple geometrical principle, in the manner illustrated in Fig. 12. The tangent lines $a\,b$ and $c\,d$ are first drawn perpendicular to the atmospheric line $m\,n$. A scale is then selected so graduated that when the *0* mark is placed on the line $a\,b$ and the scale lies diagonally across the diagram, the desired number of spaces will be included between the *0* mark and a mark that will fall on the line $c\,d$. In Fig. 12 it was desired to divide the diagram so as to get 10 ordinates. The length of the diagram is a little less than 5 inches; a scale graduated in inches can, therefore, readily be placed with the *0* mark on the line $a\,b$ and the 5-inch mark on the line $c\,d$. Lines drawn parallel to $a\,b$ and $c\,d$

through each of the inch and half-inch marks from *0* to *5* would evidently divide the diagram into 10 spaces of equal

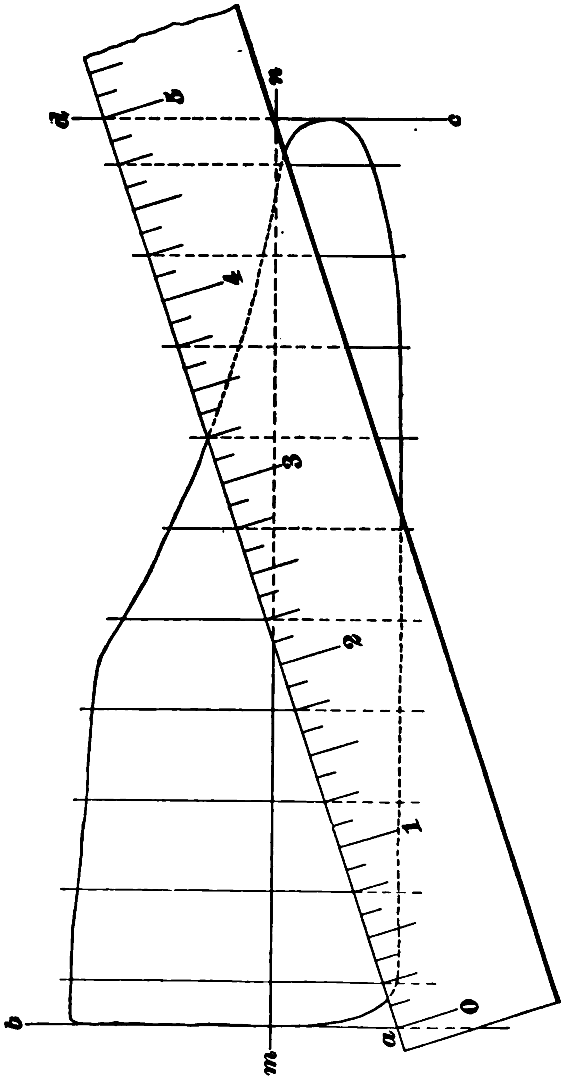


FIG. 12.

width, and since the ordinates are to be drawn through the middle of these spaces, we see that to locate the ordinates it is only necessary to make a mark on the diagram opposite each of the quarter-inch marks on the scale, and draw parallels to *a b* and *c d* through these marks.

45. Mean Ordinate of a Diagram With Loops.—To find the mean ordinate of a diagram with loops (see Fig. 11), subtract the sum of the lengths of the ordinates of the loops from the sum of the lengths of the ordinates of the main part of the diagram and divide by the total number of ordinates. In order to get reasonably accurate results with a diagram of this kind, it will generally be necessary to use a greater number of ordinates than are required for a more simple form of diagram.

46. Approximate Determination of M. E. P.—To approximately determine the M. E. P. of an engine when the point of apparent cut-off is known, and the boiler pressure, or the pressure per square inch in the boiler, from which the supply of steam is obtained, is given, and when an indicator diagram is not obtainable, use the following rule:

Rule 3.—*Add 14.7 to the gauge pressure and multiply the number opposite the fraction indicating the point of cut-off in the table, Art.* **46,** *by the pressure. Subtract 17 from the product and multiply by .9. The result is the M. E. P. for good, simple non-condensing engines.*

Cut-off.	Constant.	Cut-off.	Constant.	Cut-off.	Constant.
$\frac{1}{6}$	.566	$\frac{3}{8}$	.771	$\frac{2}{3}$	.917
$\frac{1}{5}$	.603	.4	.789	.7	.926
$\frac{1}{4}$	.659	$\frac{1}{2}$	.847	$\frac{3}{4}$	.937
.3	.708	.6	.895	.8	.944
$\frac{1}{3}$	.743	$\frac{5}{8}$	.904	$\frac{7}{8}$	.951

If the engine is a simple condensing engine, subtract the pressure in the condenser instead of 17. The fraction indicating the point of cut-off is obtained by dividing the distance that the piston has traveled when the steam is cut off by the whole length of the stroke; i. e., it is the apparent cut-off. For a $\frac{5}{8}$ cut-off and 92 pounds gauge pressure in the boiler, the M. E. P. is $[(92 + 14.7) \times .917 - 17] \times .9 = 72.6$ pounds per square inch.

It is to be observed that this rule cannot be applied to a compound engine or any other engine in which the steam is expanded in successive stages in several cylinders.

EXAMPLE.—Find the approximate M. E. P. of a non-condensing engine cutting off at $\frac{1}{2}$ stroke and making 240 revolutions per minute. The boiler pressure is 80 pounds gauge.

SOLUTION.— $80 + 14.7 = 94.7$. Using rule **3** and table, Art. **46**, the constant for $\frac{1}{2}$ cut-off is .847, and $.847 \times$ boiler pressure $= .847 \times 94.7 = 80.21$. M. E. P. $= (80.21 - 17) \times .9 = 56.89$ lb. per sq. in. Ans.

EXAMPLES ON FINDING THE M. E. P.

EXAMPLE 1.—The projection of the head-end diagram shown in Fig. 13 upon the atmospheric line is the distance $A\ Z$, and it is divided,

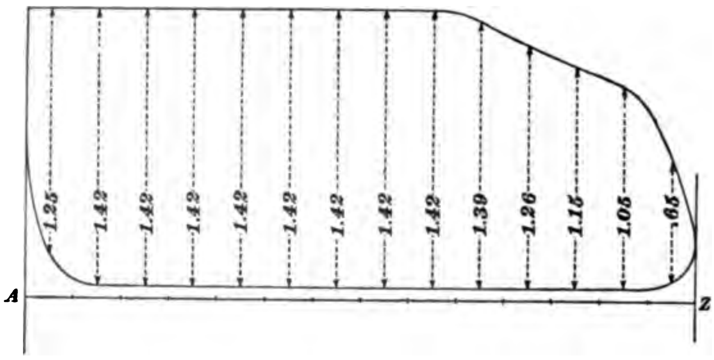


FIG. 13.

in this case into 14 equal spaces. The length of each of the perpendicular lines drawn through the diagram opposite the centers of these spaces is marked on the line itself. The scale of the spring used in obtaining the diagram was 40 pounds. Find (*a*) the length of the mean ordinate and (*b*) the M. E. P. of the diagram.

SOLUTION.— (*a*) The sum of the lengths of the 14 ordinates is 18.11 inches; the length of the mean ordinate is, therefore, 18.11 ÷ 14 = 1.294 in., nearly. Ans.

(*b*) Multiplying the length of the mean ordinate by the scale of the spring, we have M. E. P. = 1.294 × 40 = 51.76 lb. per sq. in. Ans.

EXAMPLE 2.—The projection of the crank-end diagram shown in Fig. 14 upon the atmospheric line is the distance *A Z*, and it is divided

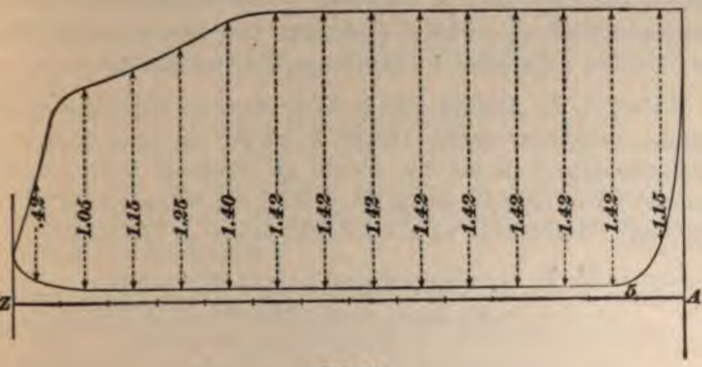


FIG. 14.

in this case into 14 equal spaces. The length of each of the perpendicular lines drawn through the diagram opposite the centers of these spaces is marked on the line itself. The scale of the spring is 40 pounds. Find (*a*) the mean ordinate and (*b*) the M. E. P. of the diagram.

SOLUTION.— (*a*) The sum of the lengths of the 14 ordinates is 17.78 inches; the length of the mean ordinate is, therefore, 17.78 ÷ 14 = 1.27 in. Ans.

(*b*) Multiplying the mean ordinate by the scale of the spring, we have M. E. P. = 1.27 × 40 = 50.8 lb. per sq. in. Ans.

EXAMPLE 3.—What was the average M. E. P. in the cylinder during the revolution represented by the two diagrams in examples 1 and 2?

SOLUTION.—Since the M. E. P. in the head end was 51.76 pounds per square inch and that in the crank end was 50.8 pounds per square inch, the average for the two strokes making up the complete revolution was $\frac{51.76 + 50.8}{2}$ = 51.28 lb. per sq. in. Ans.

CALCULATING THE INDICATED HORSEPOWER.

47. General Rule for Calculating I. H. P.—Knowing the dimensions and speed of the engine and the mean effective pressure on the piston, we have all the data for finding the rate of work done in the engine cylinder expressed in horsepowers. Work is the product of force multiplied by the distance through which it acts. In the case of the engine cylinder, the total force is the M. E. P. per square inch multiplied by the area of the piston; and the distance through which the force acts in 1 minute is the distance the piston moves in 1 minute, which is equal to the number of strokes per minute multiplied by the length of the stroke.

Rule 4.—*To find the indicated horsepower developed by the engine, multiply together the M. E. P. per sq. in., the area of piston in square inches, the length of stroke in feet, and the number of strokes per minute. Divide the product by 33,000; the result will be the indicated horsepower of the engine.*

Let I. H. P. = indicated horsepower of engine;
P = M. E. P. in pounds per square inch;
A = area of piston in square inches;
L = length of stroke in feet;
N = number of working strokes per minute.

Then, the above rule may be expressed thus:

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

In a double-acting engine the number of working strokes per minute is twice the number of revolutions per minute. For example, if a double-acting engine runs at a speed of 210 revolutions per minute there are 420 working strokes per minute. A few types of engines, however, are single-acting; that is, the steam acts on only one side of the piston. Such are the Westinghouse, the Willans, and others. In this case, only one stroke per revolution does work, and, consequently, the number of strokes per minute to be used in the above rule is the same as the number of revolutions per minute.

Unless it is specifically stated that an engine is single-acting, it is always understood, when the dimensions of an engine are given, that a double-acting engine is meant.

48. Piston Speed.—The product LN of rule 4 gives the total distance traveled by the piston in 1 minute. This is called the **piston speed.** It is usual to take the stroke in inches. Then, to find the piston speed, multiply the stroke in inches by the number of strokes and divide by 12, or, letting S represent the piston speed, $S = \frac{lN}{12}$, where l is the stroke in inches. But $N = 2R$, where R represents the number of revolutions per minute. Hence,

$$S = \frac{lN}{12} = \frac{l \times 2R}{12} = \frac{lR}{6}.$$

Rule 5.—*To find the piston speed of an engine, multiply the stroke in inches by the number of revolutions per minute and divide the product by 6.*

EXAMPLE.—An engine with a 52-inch stroke runs at a speed of 66 revolutions per minute. What is the piston speed?

SOLUTION.—By rule 5, $S = \frac{52 \times 66}{6} = 572$ ft. per min. Ans.

The piston speeds used in modern practice are about as follows:

	Ft. per min.
Small stationary engines........	300 to 600.
Large stationary engines........	600 to 1,000.
Corliss engines.................	400 to 750.
Locomotives.....................	600 to 1,200.

49. Allowance for Area of Piston Rod.—It is generally considered sufficiently accurate to take the total area of one side of the piston as the area to be used in calculating the horsepower of an engine. The effective area of one side of the piston is, however, reduced by the sectional area of the piston rod, and if it is important that the power be calculated to the greatest practicable degree of accuracy, an allowance for the area of the piston rod must be made. This is done by taking as the piston area one-half the sum of the areas exposed to steam pressure on the two sides of the piston. Thus, if we have a piston 30 inches diameter with a 6-inch piston rod, the average area is $\frac{30^2 \times .7854 + (30^2 \times .7854 - 6^2 \times .7854)}{2}$ $= 692.72$ square inches. If the piston rod is continued past the piston so as to pass through the head-end cylinder head, i. e., if the piston has a **tail rod**, allowance must be made for the tail rod. Thus, with a piston 30 inches diameter, a piston rod 6 inches diameter, and a tail rod 5 inches diameter, the average area is $\frac{(30^2 \times .7854 - 5^2 \times .7854) + (30^2 \times .7854 - 6^2 \times .7854)}{2}$ $= 682.9$ square inches.

EXAMPLE 1.—The diameter of the piston of an engine is 10 inches and the length of stroke 15 inches. It makes 250 revolutions per minute with an M. E. P. of 40 pounds per square inch. What is the horsepower ?

SOLUTION.—The number of working strokes is $250 \times 2 = 500$ per minute. Applying rule 4, we get

$$\text{I. H. P.} = \frac{40 \times \frac{15}{12} \times (10^2 \times .7854) \times 500}{33{,}000} = 59.5 \text{ H. P.} \quad \text{Ans.}$$

EXAMPLE 2.—In Fig. 15 are shown two indicator diagrams taken from an $18'' \times 20''$ engine, making 200 revolutions per minute. The

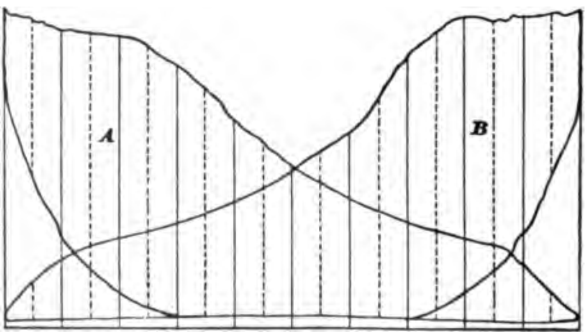


FIG. 15.

scale of the spring is 60. Compute (*a*) the mean effective pressure and (*b*) the horsepower.

SOLUTION.—(*a*) Divide the diagrams into 10 equal parts, as shown by the full lines. Then, as previously directed, draw lines or ordinates (see dotted lines in cut) perpendicular to the atmospheric line through the middle points of each of the 10 equal divisions. Measuring the lengths of all the dotted lines and adding them together, we find the sum of the lengths for diagram *A* is 7.8 inches and for diagram *B* 7.84 inches. Dividing each of these results by 10 and multiplying by the scale of the spring, we have $\frac{7.8}{10} \times 60 = 46.8$ pounds per square inch = M. E. P. for diagram *A*, and $\frac{7.84}{10} \times 60 = 47.04$ pounds per square inch = M. E. P. for diagram *B*. The average M. E. P. for both cards is $\frac{46.8 + 47.04}{2} = 46.92$ lb. per sq. in. Ans.

(*b*) To find the horsepower, the value for the M. E. P. must be substituted for *P* in the formula corresponding to rule 4, $\frac{PLAN}{33,000} = \text{H. P.}$ Reducing the stroke to feet, and substituting the values of *P*, *L*, *A*, and *N*, we have

$$\frac{46.92 \times \frac{18}{12} \times (18^2 \times .7854) \times (200 \times 2)}{33,000} = 241.21 \text{ H. P. Ans.}$$

ENGINE CONSTANTS.

50. An **engine constant** for a given engine is a number obtained by combining into a single factor all the factors of the horsepower rule that are constant for that engine. This factor may then be substituted for the factors that were combined to produce it, and a new rule obtained for that engine, in which the number of unknown quantities is less than in the original rule. The labor involved in calculating the I. H. P. for the engine is thus considerably reduced.

51. Constant for a Uniform Speed of Rotation.—When the speed of rotation of a given engine is uniform, all the factors except the mean effective pressure are constant; the engine constant for this case can, therefore, be found by the following rule:

Rule 6.—*Multiply together the length of the stroke in feet, the area of the piston in square inches, and the number of* *working strokes per minute, and divide the product by 33,000; the quotient will be the engine constant.*

This rule may be expressed by the formula

$$C_u = \frac{L\,A\,N}{33{,}000};$$

in which C_u is the engine constant for the uniform speed of rotation, and L, A, and N have the same meaning as in the formula corresponding to rule **4**. The constant C_u is the horsepower of the engine for a mean effective pressure of 1 pound per square inch.

To find the I. H. P. when the engine constant for a uniform speed of rotation is known, multiply the engine constant by the M. E. P.

EXAMPLE 1.—What is the engine constant for a 16" × 20" engine running at a uniform speed of 200 R. P. M.?

SOLUTION.—The length L of the stroke is $\frac{20}{12}$ feet, the area A of the piston is $16^2 \times .7854 = 201$ square inches, and the number of strokes N is $2 \times 200 = 400$. Substituting these values in the formula corresponding to rule **6**, we have

$$C_u = \frac{\frac{20}{12} \times 201 \times 400}{33{,}000} = 4.06. \quad \text{Ans.}$$

EXAMPLE 2.—What is the I. H. P. of the engine of example 1 when the average M. E. P. for a pair of indicator diagrams is 43.2 pounds per square inch?

SOLUTION.—Multiplying the engine constant by the M. E. P., we have I. H. P. $= 4.06 \times 43.2 = 175.39$. Ans.

52. Constant for a Varying Speed of Rotation.—When the speed of rotation is variable, the engine constant is given by the following rule:

Rule 7.—*Multiply together twice the length of stroke in feet and the area of the piston in square inches; divide the product by 33,000 for a double-acting engine. For a single-acting engine, multiply the length of stroke in feet by the area of the piston in square inches and divide the product by 33,000.*

Or, $$C_v = \frac{2\,L\,A}{33{,}000} \text{ for double-acting engines,}$$

and $$C_v = \frac{L A}{33,000} \text{ for single-acting engines,}$$

where C_v = engine constant.

The value of C_v derived from these formulas is the I. H. P. of the engine for a speed of 1 revolution per minute and a mean effective pressure of 1 pound per square inch. To find the I. H. P., multiply this constant by the number of revolutions per minute and the product so obtained by the M. E. P.

53. Formulas for M. E. P. and I. H. P. in Terms of Area of Diagram.—The fact that the M. E. P. of a diagram is equal to its area in square inches divided by its length in inches and this product multiplied by the scale of the spring enables us to develop a formula by means of which the horsepower can be calculated from the area and length of the diagram and a constant that is obtained by multiplying the engine constant by the scale of the spring. Such a formula will be found convenient when the area of the diagram is measured by a planimeter that cannot be set to measure the M. E. P. of the diagram directly.

Let a = area of diagram in square inches;
l = length of diagram in inches;
s = scale of spring.

Then $$\text{M. E. P.} = \frac{a s}{l}.$$

This value of M. E. P. can be substituted for P in the formula corresponding to rule **4**, giving us the formula

$$\text{I. H. P.} = \frac{\frac{a s}{l} L A N}{33,000}.$$

For a given engine from which a number of diagrams are to be taken, the factors s, L, A, and N will generally be constant; these factors may, therefore, be combined with the factor 33,000 in the same manner as was done in Art. **51**; a constant which we will call C_a may thus be obtained which will be given by

Rule 8.—*Multiply together the scale of the indicator spring, the length of stroke in feet, the area of the piston in square inches, and the number of working strokes per minute. Divide the product by 33,000.*

Or,
$$C_a = \frac{s\,L\,A\,N}{33,000}.$$

The indicated horsepower will then be given by multiplying this constant by the area of the diagram if the engine is single-acting, or the average area of the two diagrams if the engine is double-acting, and dividing the product by the length of the diagram.

54. If the indicator reducing motion is so constructed that the length l of the diagrams is constant, the constant may be made to include this factor. Calling such a constant C_l, we have

Rule 9.—*Multiply together the scale of the indicator spring, the length of stroke in feet, the area of the piston in square inches, and the number of working strokes per minute. Divide this product by the product of 33,000 and the length of the diagram.*

Or,
$$C_l = \frac{s\,L\,A\,N}{33,000\,l}.$$

With this constant, the indicated horsepower can be found by multiplying it by the area of the diagram if the engine is single-acting, or the average area of the two diagrams when the engine is double-acting.

EXAMPLE.—Calculate the value of the constant by which to multiply the area of the diagrams to find the I. H. P. for a 12" × 16" engine running at 250 R. P. M. when the scale of the spring is 50 and the length of the diagrams is $3\frac{1}{4}$ inches.

SOLUTION.—The length L of the stroke is $\frac{16}{12}$ feet, the area A of the piston is $12^2 \times .7854 = 113.1$ square inches, and the number N of working strokes is $2 \times 250 = 500$ per minute. Substituting these values and the given values for the scale of the spring and the length of the diagram in rule **9**, we have

$$C_l = \frac{50 \times \frac{16}{12} \times 113.1 \times 500}{33,000 \times 3\frac{1}{4}} = 32.64. \quad \text{Ans.}$$

55. Formula for I. H. P. in Terms of Total Length of Ordinates.—

Let n = number of ordinates drawn on diagram;
o = sum of the lengths of ordinates in inches;
h = length of mean ordinate in inches;
C_o = constant for calculating the I. H. P. from the ordinates;
s = scale of indicator spring.

In accordance with Art. **43**, the length of the mean ordinate is equal to the sum of the lengths of the ordinates divided by their number; that is,

$$h = \frac{o}{n};$$

and in accordance with Art. **42**, the mean effective pressure is equal to the length of the mean ordinate multiplied by the scale of the spring, or

$$\text{M. E. P.} = s\,h = s\frac{o}{n}.$$

Substituting this value of the M. E. P. for P in the formula corresponding to rule **4**, we have

$$\text{I. H. P.} = \frac{s\frac{o}{n}L\,A\,N}{33{,}000}.$$

56. For the diagrams taken from a given engine running at a uniform rate of speed, the factors s, n, L, A, and N are generally constant. They may, therefore, be combined with the constant factor 33,000 to form a new constant whose value is given by the following rule:

Rule 10.—*Multiply together the scale of the indicator spring, the length of stroke in feet, the area of the piston in square inches, and the number of working strokes per minute. Divide this product by the product of 33,000 and the number of ordinates.*

Or, $$C_o = \frac{s\,L\,A\,N}{33{,}000\,n}.$$

This constant multiplied by the sum o of the lengths of the ordinates in inches for a diagram gives the indicated horsepower for a single-acting engine. For a double-acting engine one-half the sum of the lengths of the ordinates of the two diagrams is to be taken. It is to be observed that the number of ordinates must be the same for each diagram, and that in case of a double-acting engine the *sum* of the number of ordinates of the two diagrams *must not be used.*

EXAMPLE 1.—Calculate the value of the constant C_o for diagrams taken with a 40 spring from a 28" × 42" engine running at 90 R. P. M. when the number of ordinates is 20.

SOLUTION.—The area A of the piston is $28^2 \times .7854 = 615.75$ square inches; the length L of the stroke is $\frac{42}{12} = 3\frac{1}{2}$ feet, and the number N of working strokes is $2 \times 90 = 180$ per minute. Substituting these and the values given for the scale s of the spring and the number n of ordinates in rule **10**, we have

$$C_o = \frac{40 \times 3\frac{1}{2} \times 615.75 \times 180}{33,000 \times 20} = 23.51. \quad \text{Ans.}$$

EXAMPLE 2.—What is the I. H. P. of the engine of example 1, when one-half the sum of the lengths of the 20 ordinates of the two diagrams is $19\frac{3}{16}$ inches?

SOLUTION.—

$$\text{I. H. P.} = 23.51 \times 19\tfrac{3}{16} = 451.1, \text{ nearly.} \quad \text{Ans.}$$

BRAKE HORSEPOWER.

DYNAMOMETERS.

57. Dynamometers are instruments for measuring power. They are divided into two main classes: *absorption dynamometers* and *transmission dynamometers.*

58. The most common form of **absorption dynamometer** is the Prony brake, which consists simply of a friction brake designed to absorb in friction and measure the work done by a motor, or the power given out by a shaft.

59. A **transmission dynamometer** is used to measure the power required to drive a machine or do other work; then, to determine the power required to run the shafting in a mill, a transmission dynamometer would be interposed between the shafting and the source of power, and by suitable belt connections the shafting would be driven through the dynamomoter, from which the power could be determined. Since transmission dynamometers do not enter into the work of the steam engineer, they will not be treated of here.

60. Brake horsepower is a term often applied to the power measured by a Prony brake or other type of absorption dynamometer. The brake horsepower of an engine or other motor working under given conditions is the same as the net horsepower. Since the power measured by an absorption dynamometer is the power a motor delivers at the shaft or flywheel, it is sometimes called the **delivered horsepower.**

61. The Prony Brake.—Fig. 16 represents a simple and common form of Prony brake. It consists of two wooden

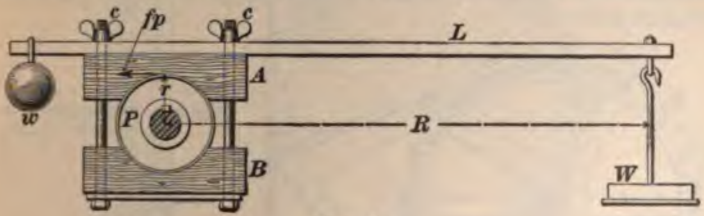


FIG. 16.

blocks A and B that are clamped together upon a pulley P by the bolts and thumbnuts c, c. The same bolts clamp an arm L to the upper block, from which a scale pan bearing a known weight W is suspended. The distance R from the center of the pulley to the perpendicular through the point from which the scale pan is suspended is also known. The counterweight W should be so adjusted as to just balance the extra length of L on the right and the weight of the scale pan.

Suppose the pulley to revolve left-handed and the bolts c, c tightened until, with a weight W in the scale pan, the lever L will remain stationary in a horizontal position. Then the foot-pounds of work absorbed by the brake can be found by multiplying the weight W by the circumference of a circle whose radius is R (in feet) and by the number of revolutions of the pulley.

Rule 11.—*To find the horsepower, multiply the weight in the scale pan by the length in feet of the lever arm about the center of the shaft, by the number of revolutions of the pulley per minute, and by 6.2832. Divide the product by 33,000.*

Or,
$$\text{H. P.} = \frac{W R N \times 6.2832}{33{,}000},$$

where H. P. = number of horsepower absorbed;
R = length in feet of lever arm about center of shaft;
W = weight in scale pan;
N = number of revolutions per minute.

EXAMPLE.—A brake with an arm R 6 feet long was placed on the flywheel of an engine. If the engine ran at 200 revolutions per minute, what power did it develop when the brake balanced with 14 pounds in the scale pan?

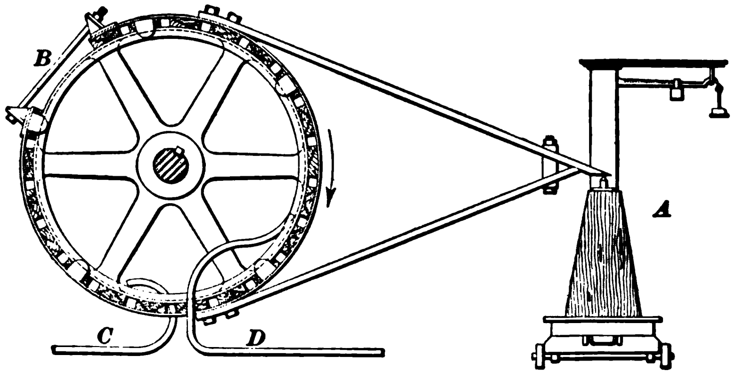


FIG. 17.

SOLUTION.—Applying the rule just given, we get

$$\text{H. P.} = \frac{14 \times 6 \times 200 \times 6.2832}{33{,}000} = 3.198. \quad \text{Ans.}$$

62. Brakes are often constructed of a metal band that extends entirely around the pulley, the rubbing surface being formed of blocks of wood fitted to the inside of the band. A weight arm is attached to one side of the pulley, and the friction is varied by means of a bolt and nut used to connect the two ends of the band.

Instead of hanging weights in the scale pan, the friction may be weighed on a platform scale, as shown in Fig. 17. In this case, the direction of rotation of both pulley and arm is the same. Rule **11** may be used for calculating the brake horsepower, substituting the weight indicated by the scale

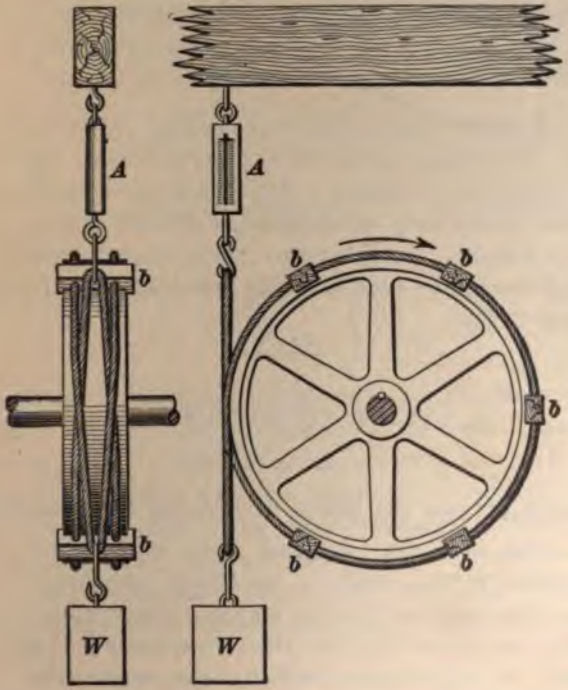


FIG. 18.

for the weight in the scale pan, and taking the length of the lever as the distance between the center of the shaft and the point where the lever presses on the platform. In reading the weight off the scale beam, it must be remembered that the weight to be used in the calculation is the *difference* betwen the weight at which the scale balances when the brake is not applied and when applied.

63. It is essential that Prony brakes should be well lubricated, and for all except small powers, means must be provided for conducting away the heat generated by friction. If there are internal flanges on the brake wheel, water can be run on the inside of the rim, the flanges serving to retain the water at the sides and centrifugal force to keep it in contact with the rim. A funnel-shaped scoop can be used to remove the water. It should be attached to a pipe and placed so as to scoop out the water, which should flow continuously. This arrangement is shown in Fig. 17.

64. A **rope brake,** like that in Fig. 18, will give good results. The figure shows the construction so clearly that no description is necessary. To obtain the brake load, subtract the brake pull as registered by the spring balance from the weight. In this case, the lever arm is equal to the radius of the pulley plus one-half the diameter of the rope, expressed in feet.

STEAM CONSUMPTION.

CALCULATIONS RELATING TO STEAM CONSUMPTION.

65. The indicator diagram also enables us to find approximately the amount of steam consumed by the engine. In referring to the steam consumption, it is customary to take as a unit the *steam consumed per horsepower per hour.* It is to be observed that the expressions "steam consumption" and "water consumption" when applied to a steam engine are synonymous.

Take a point a on the expansion line before the release (see Fig. 19); measure the pressure from the vacuum line, and from column 6 of the Steam Table find the weight of a cubic foot at that pressure. The cubic contents of the cylinder (including the clearance) up to the point *a* multiplied by the weight per cubic foot, must give the weight of steam in the cylinder at this instant. This weight would be the steam consumed per stroke were it not for two circumstances. (1) When the fresh steam from the boiler enters the cylinder, it comes in contact with the cylinder walls, which have been cooled down by the exhaust steam. A

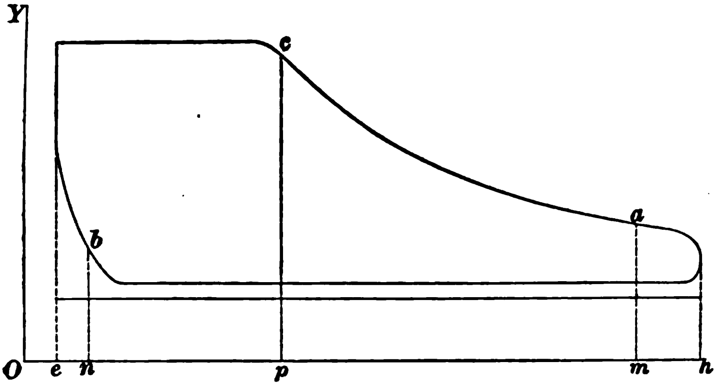


FIG. 19.

glance at the Steam Table shows that the exhaust steam is at a great deal lower temperature than the fresh steam. Consequently, part of the incoming steam condenses, and, of course, the indicator diagram takes no account of this condensed steam. Hence, the steam actually in the cylinder is less than originally entered from the boiler, because part of the original steam has changed to water. (2) On account of the closure of the exhaust port, some steam is compressed and saved.

66. To find the weight of the steam saved by compression, take a point *b* on the compression curve, measure its pressure from vacuum as before, and compute the weight of the steam in the cylinder up to *b*. Subtract this from the weight first obtained, and the difference will be the weight of steam per stroke accounted for by the indicator. Multiply this weight per stroke by the number of strokes per hour and divide by the I. H. P. of the engine. The result will be the steam used per I. H. P. per hour.

EXAMPLE.—Fig. 19 represents an indicator diagram taken with a 45 spring from an engine having an 18" × 24" cylinder, running at 120 revolutions and developing 130 horsepower. The clearance is 5 per cent. Find the steam consumption per I. H. P. per hour.

SOLUTION.—Project the two ends of the diagram perpendicularly upon the vacuum line $O\ h$, as at e and h; $e\ h$ is then the length of the diagram. Lay off $e\ O$ equal to the clearance; that is, equal to 5 per cent. of $e\ h$. Draw $O\ Y$ perpendicular to $O\ h$. Take the point a near the point of release and measure the distances $a\ m$ and $O\ m$. Take the point b somewhere on the compression line and measure the distances $b\ n$ and $O\ n$. The measurements are found to be:

$$a\,m = .71 \text{ inch};$$
$$O\,m = 3.17 \text{ inches};$$
$$b\,n = .6 \text{ inch};$$
$$O\,n = .333 \text{ inch}.$$

The length of the diagram $= e\ h = 3\frac{1}{3}$ inches; the length of the stroke is 2 feet. Hence, each inch of the length of the diagram equals $\frac{2}{3\frac{1}{3}} = .6$ foot of stroke. Since the scale of the indicator spring is 45, the above measurements reduced to pressures in pounds per square inch and feet of stroke become:

$$a\,m = .71 \times 45 = 31.95 \text{ pounds};$$
$$b\,n = .6 \times 45 = 27 \text{ pounds};$$
$$O\,m = 3.17 \times .6 = 1.9 \text{ feet};$$
$$O\,n = .333 \times .6 = .2 \text{ foot}.$$

The area of the piston is $18^2 \times .7854 = 254.47$ square inches $= \frac{254.47}{144} = 1.767$ square feet. Consequently, the volume of steam in the cylinder when the piston is at the point represented by a is $1.9 \times 1.767 = 3.3573$ cubic feet. The volume when the piston is at b is $.2 \times 1.767 = .3534$ cubic foot. The weight of a cubic foot of steam at an absolute pressure of 31.95 pounds per square inch is found from the Steam Table to be .078723 pound; and at a pressure of 27 pounds, the weight is .067207 pound. Hence, the weight of the steam in the cylinder is $.078723 \times 3.3573 = .264297$ pound; while the weight of steam saved by compression is $.067207 \times .3534 = .023751$ pound. The steam used per stroke is, therefore, $.264297 - .023751 = .240546$ pound. To find the amount used per I. H. P. per hour, multiply the weight used per stroke by the number of strokes per hour and divide by the I. H. P. Therefore, the required weight is

$$\frac{.240546 \times 120 \times 2 \times 60}{130} = 26.645 \text{ lb. Ans.}$$

67. Suppose the weight of the steam in the cylinder to be calculated by taking the point c near the point of cut-off. $c p = 1.59$ inches $= 1.59 \times 45 = 71.55$ pounds; $O p = 1\frac{1}{3}$ inches $= \frac{4}{3} \times .6 = .8$ foot of stroke. The volume of steam in the cylinder when the piston is at c is, therefore, $.8 \times 1.767 = 1.4136$ cubic feet. One cubic foot of steam at the pressure of 71.55 pounds, absolute, weighs .168009 pound. The weight of the steam in the cylinder is, therefore, $.168009 \times 1.4136 = .237498$ pound. Subtracting the steam saved by compression, the steam used per stroke is $.237498 - .023751 = .213747$ pound, and the steam per I. H. P. per hour is

$$\frac{.213747 \times 120 \times 2 \times 60}{130} = 23.677 \text{ pounds.}$$

Now, unless the valve leaks, the weight of the steam when the piston is at a can be no greater than when it is at c, since no fresh steam has been allowed to enter; but the calculation shows that there is .264297 pound in the cylinder when the piston is at a, and only .237498 pound when the piston is at c. This shows that $.264297 - .237498 = .026799$ pound has been condensed to water by the time the piston has arrived at c, but has been re-evaporated before the piston arrives at a. Hence, by calculating the water consumption at cut-off and then at release, a good idea of the amount of cylinder condensation may be obtained. If the steam used by the engine be actually caught and weighed and then compared with the weight as calculated from release, an idea may be obtained of the amount of condensation at release. The computed consumption is always less than the actual consumption.

68. Where there is a sufficient amount of compression, the work may be simplified by taking the two points a and b at the same height above the vacuum line, as shown in

Fig. 20. Since the absolute pressure at *a* and *b* is the sam the clearance may be left entirely out of account, and th

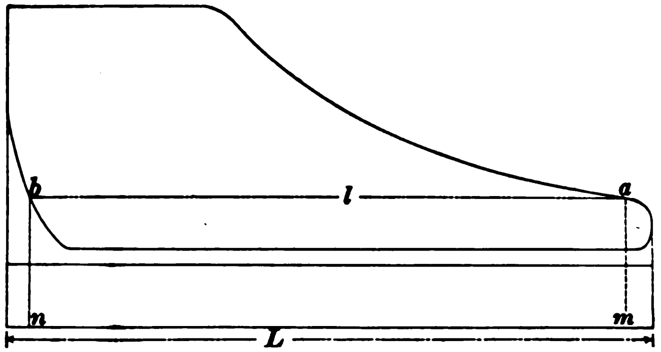


FIG. 20.

volume to be used in the computation will be $\frac{l}{L}$ times the volume of the cylinder, or, in other words, $\frac{l}{L} \times$ length of stroke $\times$ area of piston. When this method is used, the steam consumption may be found directly from the formula

$$Q = \frac{13{,}750\, l\, W}{PL},$$

in which Q is the number of pounds of steam consumed per horsepower per hour; W, the weight of a cubic foot of steam at the absolute pressure a, and P, the M. E. P.

Expressing the formula in words, we have the following rule:

Rule 12.—*Take two points, one on the expansion line and one on the compression line, both equally distant from the vacuum line. Find the pressure of the steam at these points, and from the Steam Table find the weight of a cubic foot of steam at that pressure. Multiply this weight by the distance between the two points and by 13,750. Divide the product by the M. E. P. and by the length of the diagram. The result will be the pounds of steam consumed per I. H. P. per hour, as shown by the diagram.*

EXAMPLE.—From a diagram taken from an $18\frac{1}{4}'' \times 30''$ engine, the following measurements were obtained (see Fig. 20): $a\,m = .667$ inch; $l = 3.08$ inches; $L = 3.5$ inches; M. E. P. = 35 pounds; spring, 45. What is the steam consumption per I. H. P. per hour?

SOLUTION.—The indicator diagram being taken with a 45 spring, the pressure at a is $45 \times .667 = 30$ pounds, absolute. The weight of a cubic foot of steam at this pressure is .0742 pound. Using rule **12**,

$$Q = \frac{13{,}750 \times 3.08 \times .0742}{35 \times 3.5} = 25.65 \text{ lb.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. Size of engine, $12'' \times 20''$; length of diagram L, 3.4 inches; length l, $2\frac{1}{4}$ inches; height $a\,m$, $\frac{3}{4}$ inch; R. P. M., 230; spring, 30; M. E. P., 18 pounds per square inch. What is the steam consumption per I. H. P. per hour? Ans. 25.26 lb. per I. H. P. per hr.

2. Size of engine, $12'' \times 12''$; M. E. P., 51.1; length of diagram L, 2.6 inches; length l, 1.8 inches; height $a\,m$, .7 inch; R. P. M., 350; spring, 70. What is the steam consumption per I. H. P. per hour? Ans. 21.92 lb. per I. H. P. per hr.

3. If, in the above engine, example 2, the pressure at cut-off is 110 pounds, absolute; the clearance is 8 per cent.; the length of the diagram to the point of cut-off is .7 inch; the pressure at a point on the compression curve is 49 pounds, absolute, and the distance of this point from the end of the diagram is .14 inch, what is the steam consumption per I. H. P. per hour at cut-off? Ans. 19.44 lb. per I. H. P. per hr.

SIZE OF STEAM ENGINES.

69. The problem of selecting a size of simple engine that will develop a given indicated horsepower is capable of an infinite number of correct solutions, depending on the conditions present. The factors that determine the indicated horsepower are the mean effective pressure, the length of stroke, the diameter of the piston, and the number of revolutions per minute. Before the diameter of the piston and the length of stroke, which data constitute the size of the engine, can be determined, the boiler pressure, point of cut-off, and piston speed must be chosen. From the boiler pressure and the point of cut-off the mean effective pressure is then estimated in the manner explained in Art. **46.** The area of the piston is then given by the following rule:

Rule 13.—*To find the piston area in square inches, multiply the indicated horsepower by 33,000 and divide by the product of the mean effective pressure and the piston speed in feet per minute.*

Or,
$$A = \frac{33{,}000\,H}{PS},$$

where
A = area of piston;
H = indicated horsepower;
P = mean effective pressure;
S = piston speed.

To find the diameter, divide the result of rule **13** by .7854 and extract the square root of the quotient.

EXAMPLE.—Find the piston diameter for a 25-horsepower engine using steam at 70 pounds gauge pressure, cutting off at $\frac{2}{3}$ stroke, and to have a piston speed of 300 feet per minute. Engine is non-condensing.

SOLUTION.—By rule **3**, Art. **46**, the mean effective pressure is $[(70 + 14.7) \times .937 - 17] \times .9 = 56$ pounds per square inch. Applying rule **13**, we get

$$A = \frac{33{,}000 \times 25}{56 \times 300} = 49.1 \text{ square inches.}$$

The corresponding diameter is $\sqrt{\frac{49.1}{.7854}} = 8$ in., about. Ans.

70. Since the piston speed is the product of the number of strokes and the length of stroke, to find the latter the number of strokes must be assumed. Then, to find the length of stroke in feet, divide the piston speed by the number of strokes.

EXAMPLE.—If the engine in the example given in Art. **69** is to make 120 revolutions per minute, what should be the stroke in inches?

SOLUTION.—Stroke in feet $= \frac{300}{120 \times 2} = 1.25$ feet, or $1.25 \times 12 = 15$ in. Ans.

CONDENSERS.

INTRODUCTION.

EFFICIENCY.

1. The perfect combustion of 1 pound of good coal will produce 14,000 B. T. U. and 1 B. T. U. is equal to 778 foot-pounds.

The mechanical energy stored up in 1 pound of good coal is, therefore, $14,000 \times 778 = 10,892,000$ foot-pounds. Reducing this to horsepower per hour, we have $\frac{10,892,000}{33,000 \times 60} = 5.5$ horsepower, nearly. This means that if all the heat made available by the burning of a pound of coal was transformed into work, it could be done at the rate of 5.5 horsepower per hour.

2. We know from practical experience that 1 pound of coal never has produced 5.5 horsepower per hour, about the best record ever having been made giving 1 horsepower per hour for 1 pound of coal. This comparison shows what an imperfect apparatus a steam engine really is when considered as a heat engine.

3. We will now endeavor to trace out what becomes of all this heat that is produced by the combustion of the fuel.

In the first place, a very considerable part of the heat produced by the combustion of the fuel escapes in various ways from the boiler, the largest part of which passes up the stack with the gases of combustion. All this heat is not actually wasted, however, as a large part of it is necessary to produce draft, but it is lost so far as doing work in the cylinder is concerned. Then, the losses by radiation, blowing off with the blow-off valves, the safety valve, and the gauge-cocks, and various other small losses make up the total loss from the boiler.

This loss varies greatly in different boilers, but a fair average may be taken at 4,000 B. T. U. This leaves 10,000 B. T. U. in the steam that goes to the engine and represents the part of the heat available for doing work in the cylinder.

4. As each unit of heat in the steam is capable of doing a definite amount of work, it is obvious that the more heat that is available for this purpose, the greater the amount of work can be performed, and the doing of this work by the steam is accompanied by a reduction of temperature. From this it is easily seen that the amount of work which can be done by steam depends on the decrease in temperature.

5. According to the now universally accepted theory, heat energy consists of the motions of the molecules of the hot body. In order, therefore, to change into work *all* the heat energy contained in the steam, it will be necessary to take from the steam all its molecular motion; in other words, to lower its temperature until its molecules will be at rest.

From experiments it has been concluded that the temperature at which a body will be in such a state is about 460° below zero, Fahrenheit. This point is called the **absolute zero** of temperature. Temperatures measured from this zero point are called **absolute temperatures.**

Absolute temperatures are obtained by adding 460° to the ordinary temperature. That is,

Absolute temperature = ordinary temperature + 460°.

For example, the ordinary temperature of steam at atmospheric pressure is 212°. The absolute temperature is 460 + 212 = 672°. This means that if steam or other gas at 212° could be cooled down 672°, the molecules would cease moving and there would be no heat in the body.

It is at once evident that it is impossible practically to cool the steam leaving the engine cylinder to even approximately so low a temperature; long before reaching the absolute zero, the steam would be changed to ice. This explains why it is impossible for an engine to obtain 778 foot-pounds from each B. T. U. conveyed to it.

6. Let T_1 denote the absolute temperature of the steam entering the engine cylinder and let T_2 represent the absolute temperature of the steam leaving the cylinder. If we take the amount of energy contained in the entering steam as proportional to the absolute temperature T_1, then it may be proved that the amount of work extracted by the engine is proportional to $T_1 - T_2$; or,

$$\text{Useful work : total energy} :: T_1 - T_2 : T_1.$$

The efficiency of an engine is the ratio of the useful work to the total energy; therefore,

$$\text{Efficiency} = \frac{T_1 - T_2}{T_1}.$$

The above reasoning may perhaps be made clearer by comparing the temperature of steam to the "head" of a water-power. If a source of water is 1,500 feet above the sea level, the work or potential energy of the water is represented by the head of 1,500 feet. But possibly the water-wheel to which the water is led is itself 1,400 feet or more above the sea level; in this case, only 100 feet of the 1,500 is available head, and the efficiency of the arrangement cannot exceed $\frac{1,500 - 1,400}{1,500} = \frac{1}{15}$. Absolute zero is the "sea level" of temperature, but as far as steam is concerned, it is utterly impossible to lower its temperature to the absolute zero by reason of its changing into ice at 32° F., corresponding to 492° absolute. About the lowest temperature to which the steam can be lowered in practice is 562° absolute. This temperature corresponds to an absolute pressure of 1 pound, which is probably the lowest pressure attainable.

The above expression for the efficiency applies equally well to steam engines, hot-air engines, gas engines, or any other form of heat engine.

CONDENSATION.

7. It is plain that the only two ways of increasing the efficiency of the steam engine is either to raise the temperature T_1 of the live steam or to lower the temperature T_2 of the exhaust; T_1 may be raised by increasing the boiler pressure; T_2 may be lowered by using a **condenser.**

In non-condensing engines, that is, engines that are not supplied with a condenser, the steam is exhausted into the atmosphere, and therefore the exhaust steam must have, at least, the pressure of the atmosphere; in practice, the back pressure of steam in a non-condensing engine is scarcely ever less than 16 pounds above vacuum, and is oftener 17 pounds or more. In good condensing engines the back pressure is often as low as 2 pounds above vacuum.

Suppose the boiler pressure of the steam is 80 pounds absolute (above vacuum); the temperature corresponding to the pressure is, from the Steam Table, 311.9° F., and the absolute temperature is, therefore, 460° + 311.9° = 771.9° F. The absolute temperature corresponding to a pressure of 17 pounds is 460° + 219.5° = 679.5° F., and corresponding to a pressure of 3 pounds is 460° + 141.7 = 601.7° F. The thermal efficiency of the engine, if non-condensing, is $\frac{T_1 - T_2}{T_1}$

$= \frac{771.9 - 679.5}{771.9} = 12$ per cent., nearly; if condensing to 3 pounds (absolute), the efficiency is $\frac{T_1 - T_2}{T_1} = \frac{771.9 - 601.7}{771.9}$ $= 22$ per cent.

8. The increase of economy by the use of the condenser may be shown in another manner. Let $A\ B\ C\ D\ E\ F$, Fig. 1, be an indicator diagram from a non-condensing

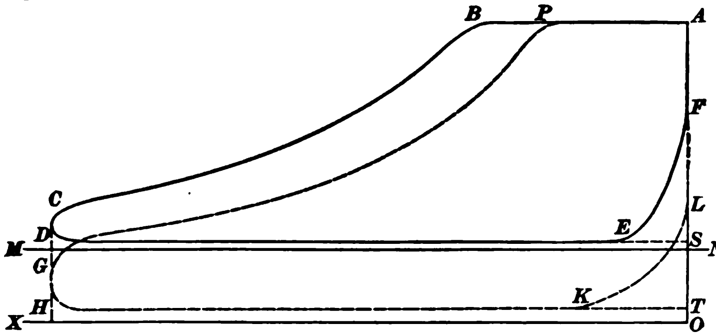


FIG. 1.

engine. $M\ N$ is the atmospheric line and $O\ X$ the vacuum line. The back pressure, as shown by the diagram, is $O\ S$. The area of the diagram represents, to some scale, the work done per stroke. Now let a condenser be attached to the engine. The back pressure will be lowered to $O\ T$, the line $H\ K$, instead of $D\ E$, now being the lower line of the diagram, and $A\ B\ C\ H\ K\ L$ will be the new diagram, its area, as before, representing the work done per stroke. Hence, by adding a condenser to the engine, the work per stroke has been increased by an amount represented by the area $F\ E\ D\ H\ K\ L$, the steam consumption remaining the same. Suppose the steam to be cut off at a point P, making the area of the diagram $A\ P G\ H K\ L$ equal to the area of the original diagram $A\ B\ C\ D\ E\ F$. Then, the work per stroke is the same in both engines, but the condensing engine uses an amount of steam per stroke represented by the length $A\ P$, while the non-condensing engine uses an amount represented by $A\ B$. Either case shows the economy of the condenser.

THEORY OF THE CONDENSER.

FORMATION OF VACUUM.

9. If a cubic inch of water is converted into steam at the atmospheric pressure, it will occupy 1,646 cubic inches of space, and, conversely, if 1,646 cubic inches of steam at the atmospheric pressure are condensed into water, it will occupy but 1 cubic inch of space; hence, if a closed vessel is filled with steam at the atmospheric pressure and that steam is condensed to a cubic inch of water, $\frac{1645}{1646}$ of the space will be theoretically devoid of air or any other known substance and a perfect vacuum would be the result. This is not strictly true in practice, however, from the fact that the feedwater of the boilers always contains a small quantity of air, which passes into the condenser with the exhaust steam and is released there when the steam is condensed; more or less air also finds its way into the condenser through leaks around the piston rod and valve stems, and in the case of the jet condenser and the induction condenser, the air contained in the condensing water is also released in the condenser under the influence of the partial vacuum. There is still another obstacle in the way of producing a perfect vacuum in the condenser, which may be explained as follows: Water in a vacuum emits a certain amount of vapor, and vapor is also formed in a jet condenser and in an induction condenser by the heat in the exhaust steam being imparted to the condensing water; therefore, if the condenser were successively filled with steam and the steam were condensed at each filling, the air and vapor, unless they were removed, would accumulate from these various sources until the vacuum would be entirely destroyed.

10. Air and vapor differ from water in that they are expansible, while water is not. To illustrate this, suppose that two closed vessels *A* and *B*, Fig. 2, are filled, *A* with water and *B* with air or vapor; now pump out, say, one-half the water in A, as shown at C. The space a above the water is a vacuum; but if one-half the air or vapor in B is pumped out, the air or vapor will still fill the vessel, but at a lower pressure. The air or vapor will have become attenuated, or thin, by expansion. In obedience to the law that the different pressures of a gas are inversely to their volume, the pressure in D will be just one-half what it was in B before one-half the air or vapor had been withdrawn. If the original pressure in B was 15 pounds absolute per square inch, it will now be $7\frac{1}{2}$ pounds absolute per square inch in D, or since $7\frac{1}{2}$ pounds pressure correspond closely to 15 inches of mercury, we may say that a vacuum of 15 inches exists in the vessel. Should three-fourths of the air and vapor in B be pumped out, the pressure will be one-fourth the original pressure, that is, $3\frac{3}{4}$ pounds, and $15 - 3\frac{3}{4} = 11\frac{1}{4}$ pounds of the pressure have been removed. Since a vacuum gauge indicates not the pressure that exists in the vessel, but the pressure that has been *removed* from the vessel, counting from the pressure of the atmosphere, a vacuum gauge would now indicate $11\frac{1}{4} \times 2 = 22\frac{1}{2}$ inches of vacuum, nearly, because a pressure of 1 pound per square inch corresponds very nearly to 2 inches of mercury indicated by the vacuum gauge.

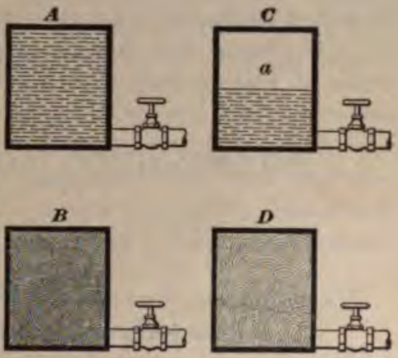


Fig. 2.

11. The object of the condenser is to remove a large part of the back pressure on the exhaust side of the piston of a steam engine when exhausting into the atmosphere, which back pressure, obviously, cannot be less than the pressure of the atmosphere. By making the engine exhaust into a condenser, the back pressure will be lowered to the pressure existing in the condenser, and, consequently, with the pressure on the steam side of the piston remaining the same as before, the net pressure on the piston will be increased by the use of a condenser. As previously explained, air and vapor will collect in the condenser and if not removed will destroy the vacuum. To get rid of this air and vapor, the condenser is fitted with an air pump, or is provided with other means by which the air and vapor are removed from the condenser along with the condensed steam and condensing water. This operation restores and maintains a constant vacuum.

LOSSES IN CONDENSING ENGINES.

12. A condensing engine requires more watchful attention on the part of the running engineer than an engine exhausting into the atmosphere. Without this extra attention the condenser may be the source of a positive loss, so far as the consumption of steam is concerned. Leaky steam valves and leaky pistons are largely responsible for this loss. Leaks of this nature are not so easily detected in the condensing engine as they are in the non-condensing engine, with which the exhaust steam is usually in view as it leaves the exhaust pipe. A continuous stream of steam being emitted from the exhaust pipe indicates leaky steam valves or leaky piston or both; the only sure way to discover such leaks in a condensing engine is by means of the indicator card.

13. There may be another source of loss in a condensing engine, when independent air and circulating pumps are used, in the extravagant use of steam in the engine or engines that operate these pumps. The steam cylinders of the air and circulating pump should be fitted with indicator gear and cards taken, and the steam valves should be set with the same care that is bestowed upon the main engine. In large plants a considerable saving of steam may be effected by compounding the steam cylinders of independent air and circulating pumps, or in the case of multiple-expansion engines, to lead the exhaust steam from the engines of independent air and circulating pumps into the receiver. By so doing, the energy remaining in the exhaust steam may be converted into work in the low-pressure cylinder of the main engine.

CONSTRUCTION OF CONDENSERS.

CLASSIFICATION.

14. Condensers may be divided into two general classes, which are *jet condensers* and *surface condensers.*

15. A **jet condenser** may be defined as a condenser in which the exhaust steam and the condensing water mingle and where the steam, consequently, is condensed by direct contact with the water.

16. In a **surface condenser** the exhaust steam and the condensing water do not mingle; the exhaust steam is condensed by coming into contact with metallic surfaces that are kept cool by cold water constantlv flowing over them.

JET CONDENSERS.

TYPES.

17. There are three types of jet condensers, viz.: *The common jet condenser*, *the siphon condenser*, *and the induction condenser.*

THE COMMON JET CONDENSER.

18. The **common jet condenser** consists of an air-tight vessel of sufficient strength to sustain the pressure of the atmosphere from without. The exhaust steam flows into this vessel from the cylinder after its available energy has been spent in driving the piston to the end of its stroke. The steam on entering the condenser is met by a jet or spray of water forced through small apertures in a pipe placed in the pathway of the incoming steam. The steam and spray are thus mechanically mixed and the steam, surrendering much of its heat to the sprayed water, is condensed into water, which falls to the bottom of the condenser along with the injection water. From the condenser it flows through the foot-valve into the air pump, by which it is pumped into the hotwell. So much of the water as may be required for feeding the boilers is taken from the hotwell by the feed-pump and forced into the boilers, the remainder of the water in the hotwell flows through the discharge pipe into the sewer or elsewhere.

19. The form of a common jet condenser is immaterial, but it is important that it be placed so that the water cannot be drawn into the cylinder from it in case of flooding.

20. The common jet condenser in its simplest form is shown in Fig. 3, in which A is a cylindrical or rectangular

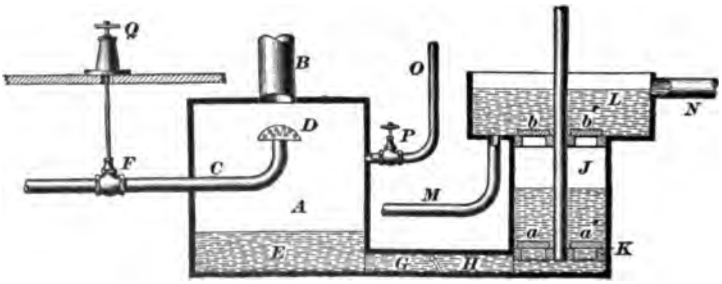


FIG. 3.

vessel forming the **body** of the condenser; B is the exhaust pipe from the engine through which the exhaust steam enters the condenser; C is the **injection pipe**; D is a perforated enlargement on the end of the injection pipe, called a **rose**, whose duty it is to scatter, or spray, the injection water that is forced into the condenser by the pressure of the atmosphere on the surface of the water supply outside.

The spray coming into direct contact with the incoming steam deprives the steam of a part of its heat and condenses it into water, which, together with the injection water, falls to the bottom of the condenser, as shown at E; F is the **injection valve,** which regulates the flow of the injection water into the condenser.

21. The handle or wheel Q of the injection valve of a jet condenser is always placed within reach of the engineer's station, as the valve must either be opened or closed at the same moment that the engine is, respectively, started or stopped; otherwise the condenser either will be flooded with water or else it will get hot.

Should the condenser become flooded there is danger of the water being drawn into the cylinder and cause serious damage by blowing out the cylinder head or breaking the piston; besides, the condenser being full of water when flooded, there is no room for the exhaust steam to enter and a vacuum cannot be formed until the air pump clears the condenser of its superfluous water, which will require several strokes of the bucket to accomplish. On the other hand, should the condenser be deprived of injection water the incoming steam cannot be condensed, and it will accumulate in the condenser until the pressure is equal to or greater than the pressure of the atmosphere outside; the condenser is then said to be hot. When this occurs, the injection water cannot be forced into the condenser by the pressure of the atmosphere alone. To provide for such a contingency, jet condensers of this type are usually fitted with a water pipe connected with one of the auxiliary pumps, by which cold water is forced into the condenser and a vacuum is produced thereby. This pipe is shown at O, Fig. 3, and is provided with a valve P, which should be kept closed when not in actual use, to exclude any air that might leak into the condenser through the auxiliary pump. If no such pipe is provided, it will be necessary, in case of the condenser becoming hot, to deluge the outside of it with cold water from a hose or buckets.

22. A **foot-valve** G is placed in the channel way H through which the water flows from the condenser into the air pump; J is the **air-pump barrel**; K is the **air-pump piston,** or **bucket,** as it is commonly called. The bucket is perforated with apertures that are fitted with flat valves a, a', which open upwards; when the bucket descends the valves open, permitting the air, vapor, and water in the lower, or receiving, chamber of the air pump to pass through the openings in the bucket into the barrel of the pump. The bucket, in ascending, forces the air, vapor, and water through the **delivery valves** b, b' into the **hotwell** L. Whatever quantity of water may be required for boiler feeding is taken from the hotwell by the feed-pump through the pipe M; the remainder of the water in the hotwell runs off through the **discharge pipe** N.

If the injection water is so impure as to be unfit for boiler feedwater, all the water in the hotwell is allowed to run to waste and pure feedwater is supplied from another source; this is a great loss of heat, however, as the temperature of the feedwater must be raised to at least that of the water in the hotwell to make it of equal value as feedwater; to do this, extra heat is required from the fuel, unless the exhaust steam from the auxiliary engines is utilized in a feedwater heater for that purpose, in which case the assistance of the vacuum is lost to the auxiliary engines. This statement applies to all condensers in which the injection water comes in direct contact with and is mixed with the exhaust steam.

23. While the construction of a jet condenser shown in Fig. 3 is practically obsolete at present, it clearly shows all the essential parts, and its simplicity renders its operation easily understood. With this fundamental knowledge of its operation, it is comparatively easy to understand the action of later designs of jet condensers.

24. The jet condenser now generally used differs somewhat in form from the one just described. It occupies less space and the air pump used in connection therewith is usually a double-acting, horizontal pump operated by an independent engine, but the principles involved are practically the same in both. As an example of this type of condenser a description of one of this class is given below.

In Fig. 4 is shown a section of a Worthington independent jet condenser. The cold water enters the condenser at *b*,

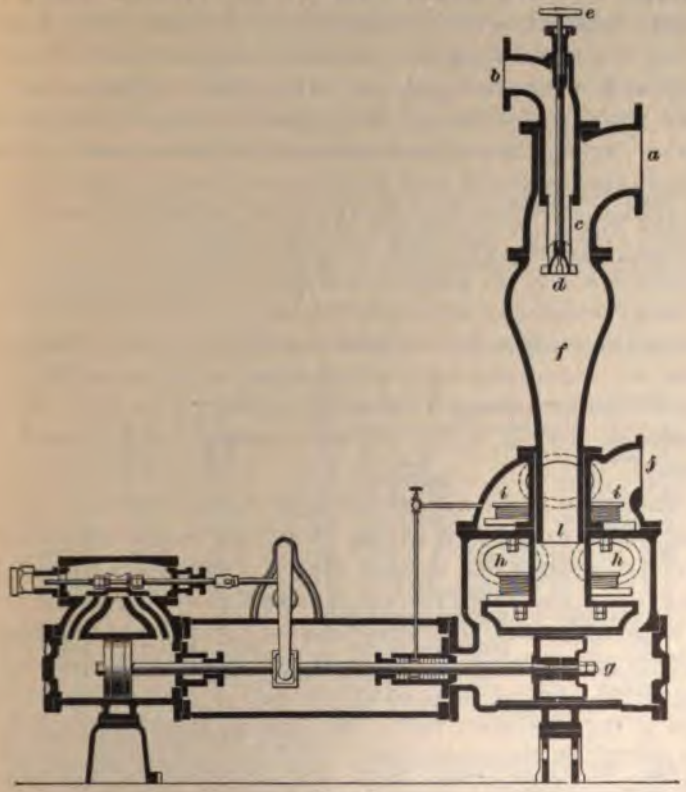


FIG. 4.

passes down the spray pipe *c*, and is broken into a fine spray by the cone *d*. The exhaust steam in the meantime comes in at *a*, and, mingling with the spray of cold water, is rapidly condensed. The velocity of the entering steam is imparted to the water, and the whole mixture of steam, water, uncondensed vapor, and air is carried with a high velocity through the cone *f* into the air-pump cylinder *g*, whence it is forced by the pump through the discharge pipe *j*.

25. The above described condensing apparatus is operated as follows: The air pump having been started, a vacuum is formed in the condenser, the exhaust pipe, the engine cylinder, and injection pipe; this causes the injection water to enter through the injection pipe attached at *b* and to flow through the spray pipe *c* into the condenser cone *f*. The main engine being then started, the exhaust steam enters through the exhaust pipe attached at *a*, and, coming into contact with the cold water, is condensed. The velocity of the steam is communicated to the water and the whole passes through the cone *f* and through the receiving valves *h*, *h* into the pump *g* at a high velocity, carrying with it in a commingled condition all the air and uncondensable vapor that enters the condenser with the steam. The mingled air, vapor, and water are expelled by the pump through the discharge valves *i*, *i* and the delivery pipe at *j* before sufficient time or space has been allowed for separation to occur.

The spray pipe *c* has at its lower end a number of vertical slits through which the injection water passes and becomes spread out in thin sheets. The spray cone *d*, by means of its serrated surface, breaks the water passing over it into fine spray and thus insures a rapid and thorough admixture with the steam. This spray cone is adjustable by means of a stem passing through a stuffingbox at the top of the condenser and is operated by the handle *e*.

26. Exhaust steam from an engine enters a vacuum with a velocity of about 1,900 feet per second, and water, under atmospheric pressure, with a velocity of 47 feet per second.

In the common jet condenser, the injection water and the condensed steam fall to the bottom of the condensing chamber and come to rest there before entering the air pump;

thus the momentum acquired by water rushing into a vacuum is lost; whereas, in the class of jet condenser now under consideration, this force is utilized to assist the pump by accelerating the velocity of the falling water. This is accomplished by contracting the lower end of the condenser cone into a neck, or throat, as shown at *l*, Fig. 4. Moreover, the air and vapor being intimately mixed with the water, the load on the pump is more steady and regular than is the case in the common jet condenser.

The danger of the water working back into the engine cylinder, in case of flooding, is less with this form of jet condenser than with the common jet condenser.

27. The injection opening of a jet condenser must not be more than 20 feet above the surface of the water supply, and it is highly important that the injection pipe be entirely free from air leaks.

28. In Fig. 5 is shown a jet condenser in connection

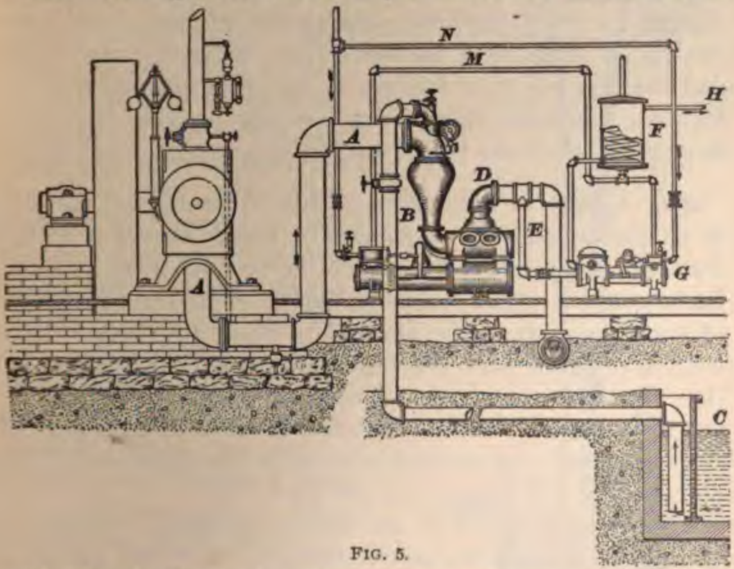


FIG. 5.

with the boiler and engine. The exhaust pipe *A* leads directly to the condenser. The injection pipe *B* draws water from the reservoir *C*. After the steam is condensed, the mixture of exhaust steam and injection water is discharged through *D* into the sewer. A portion of this discharge, however, flows through *E* to the feed-pump *G*, which forces it through the coil in the heater *F* to the pipe *H* leading to the boiler. The exhaust from the two pumps is discharged into the feedwater heater through the pipe *M*. It will be noticed that water from the overflow pipe *D* enters the feed-pump under a slight head. This is because the water is heated by the exhaust steam, and hot water cannot be raised by a pump like cold water. A pipe *N* leads from the boiler and supplies steam for both pumps.

THE SIPHON CONDENSER.

29. The **siphon condenser** differs from the common jet condenser in that no air pump is required to remove the air, uncondensed vapor, and water, but a circulating pump or a head of water is needed to supply the injection water when the lift is more than 20 feet. The vacuum is generated and maintained by a column of water flowing downwards through a vertical pipe of not less than 34 feet in length, having its lower end immersed in the water of the hotwell.

30. It will be remembered that a column of mercury 30 inches in height or a column of water 34 feet in height will just balance the atmosphere at the sea level when the barometer stands at 30 inches, but if an additional amount of water or mercury be allowed to enter the upper end of the water pipe or mercury tube, the equilibrium between the column of water or mercury and the column of air outside will be disturbed, and an amount of water or mercury corresponding to that allowed to enter at the upper end of the pipe or tube will flow out at the lower end.

This is the principle of the siphon condenser. So long as the proper amount of water continues to flow into the upper end of the pipe and a corresponding amount flows out at the lower end, the air and vapor in the condenser will be carried out by the descending water and a vacuum will be formed and maintained therein. If the area of the pipe is contracted into a neck, or throat, the velocity of the falling water will be accelerated and the action of the condenser will be improved thereby.

It is important that the stream of injection water entering the condenser should have a steady and continuous flow, and there must be no air leaks in the exhaust pipe or condenser. The siphon condenser is often, but wrongly, called the **injector condenser.**

FIG. 6.

31. An illustration and a description of an example of this type of condenser, known as the **Baragwanath condenser,** is here given.

Fig. 6 represents a sectional view, in which *a* is the exhaust pipe; *b* is the injection pipe; *d* is the long discharge pipe, or **tail-pipe,** and *e* is the hotwell. The operation of this condenser is as follows: The steam enters through the exhaust pipe *a* and flows through the exhaust nozzle *f* into the condensing chamber *g*. Here it is met and condensed by the injection water that enters from the water-jacket *h* into the condenser in a thin conical sheet, flowing through the annular opening between the exhaust nozzle *f* and the prolongation of the shell of the condenser forming the inverted cone *i*. A vacuum is now formed in the condensing chamber *g* by the condensation of the steam and by the air and uncondensed vapor being entrapped and carried out of the chamber by the cylindrical stream of water. The injection water and the water of condensation flow from the condensing chamber *g* down through the throat *j* with such velocity as to carry with them the air and vapor that passed over with the steam and the injection water.

The exhaust nozzle *f* is adjusted by means of the wheel and screw spindle *k*, and can be set so as to admit just the right quantity of injection water. An automatic atmospheric relief valve *l* is fitted for the purpose of discharging any excessive accumulation of air, steam, or vapor that may collect in the exhaust pipe into the atmosphere. A hotwell overflow or discharge pipe *m* is always fitted to the hotwell.

Although the condenser is placed at a height of 34 feet above the hotwell, the vacuum assists the circulating pump in a proportionate degree, so that with a vacuum of 24½ inches the actual height that the water is forced by the pump is but 7 feet.

32. It is sometimes the practice with this type of condenser to place a tank about 15 feet below the condenser from which the injection water is forced into the condenser by the pressure of the atmosphere upon the surface of the water in the tank, or *siphoned in*, as it is termed. The tank is supplied with water by an ordinary tank pump or from the street main. This arrangement has two advantages, viz.: (1) It insures a steady flow of water into the condensing chamber, which is an important consideration, and (2) a lower priced circulating pump may be used. In this case, however, there must be a cross-pipe connection between the injection pipe and the discharge pipe at the water-supply level, for the purpose of charging the siphon and generating a vacuum in the condenser. This is done by opening the valve in the cross-pipe connection and admitting water to the discharge pipe; the water in falling down the discharge pipe draws the air from the condensing chamber and a partial vacuum is formed, which induces the injection water to enter. As soon as the siphoning action through the condenser is established, the valve in the cross-pipe connection is closed.

33. If the level of the injection-water supply is not more than 20 feet below the injection inlet to the condenser, the water will be siphoned over as soon as a vacuum is formed, and a circulating pump may then be dispensed with.

34. The flow of water, impelled by the incoming steam and by its own gravity, pours through the throat *j* continuously in one direction and with such high velocity that the possibility of the condenser becoming flooded and the water working back into the engine cylinder is very small.

The exhaust nozzle being adjustable admits of close regulation of the injection-water supply, which still maintains the conical form of the stream of water as it enters the condensing chamber and exposes the same area of condensing surface; even when nearly shut off there are no bare spots on the sides of the condenser, the conical sheet of water simply being made thinner.

THE INDUCTION CONDENSER.

35. The operation of the **induction condenser** is based on the same principle as that of the steam injector, used so largely for boiler feeding, and it may properly be called an **injector condenser,** although it is not given this name by the trade.

Fig. 7 represents a partial sectional view of a condensing apparatus of this type. It is known as the **Korting universal exhaust steam induction condenser** and is manufactured by L. Schutte & Co., Philadelphia, Pennsylvania.

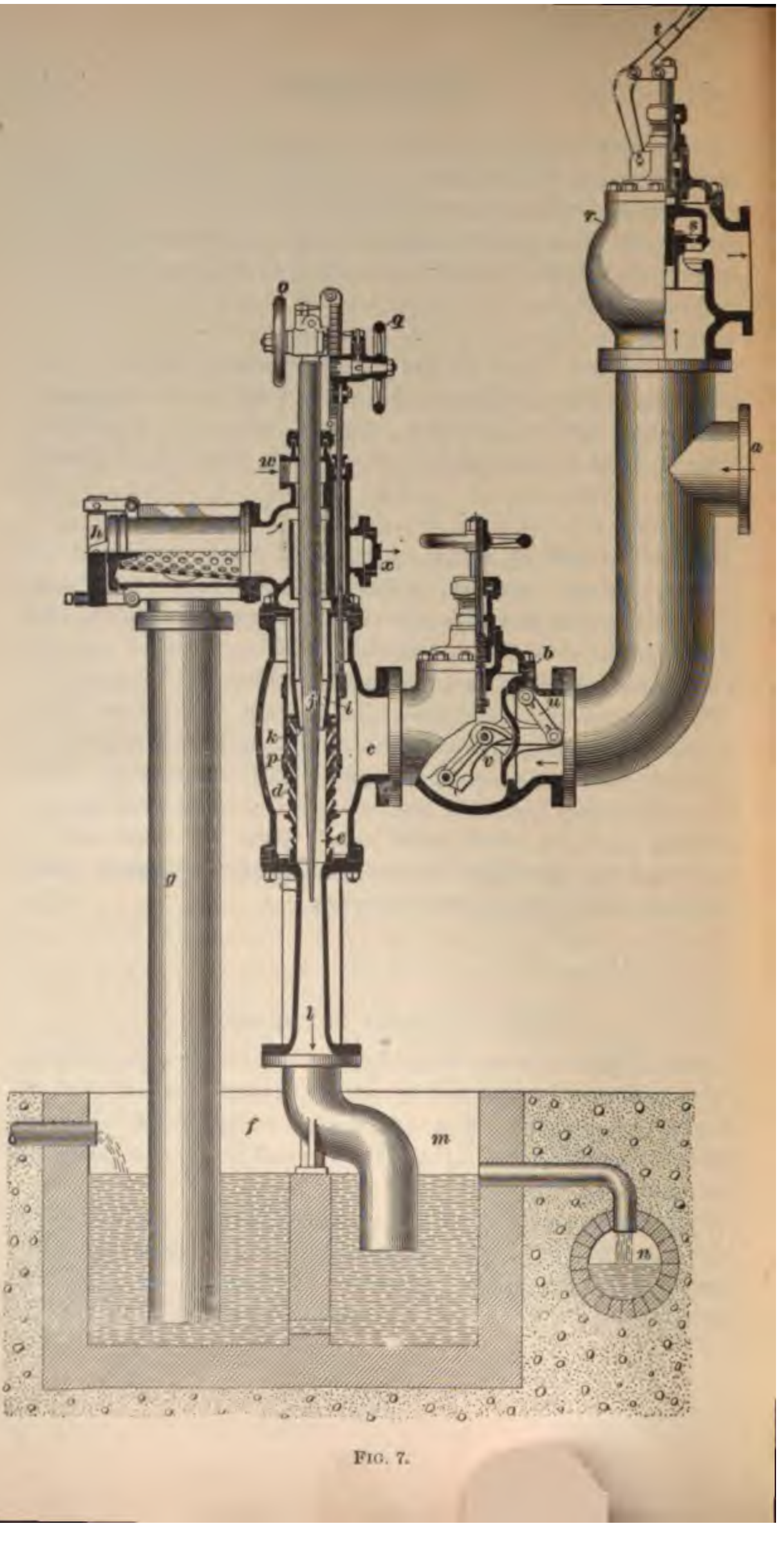


FIG. 7.

Referring to the figure, the exhaust steam enters at *a*, and after passing through the balanced horizontal check-valve *b* enters the water chamber *c*; it then passes through the inclined openings in the tube *d* into the condensing chamber *e*, where it is met by the injection water and is condensed, forming a partial vacuum—the condenser having first been started by a supplementary jet of steam or stream of water—as will be explained hereafter. The vacuum in the chamber *e* induces the injection water to be siphoned into the condenser from the supply reservoir *f* through the injection pipe *g* and the strainer *h*, from whence it flows into the annular space *i* around the ram *j*, passing into the condensing chamber *e* through the annular opening *k*, where it meets the exhaust steam, which is then condensed. Here the injection water and the water of condensation intermingle and with the air and vapor are carried down the discharge pipe *l* into the hotwell *m*, the surplus water flowing into the sewer *n*.

36. To obtain the best results under the varying quantities of steam it may be called upon to handle, this condenser requires that it shall be adjustable. This is accomplished by the ram *j* being made tapering and capable of being raised and lowered at will, which operation varies the size and capacity of the annular opening *k* and controls the volume of water admitted to the condensing chamber *e*. The ram is adjusted by the hand wheel *o* acting through a rack and pinion. The area of opening required by the steam that enters the condenser is regulated by the sleeve *p*, which covers more or less of the openings in the tube *d*, as may be required. This sleeve is raised or lowered by the hand wheel *q*, which also acts through a rack and pinion. It will be observed that this apparatus is capable of very fine adjustment.

37. Like all condensers, this one requires a valve that opens into the atmosphere to relieve it of any accumulation of steam, air, or vapor that may collect in it. This is provided in the **automatic free exhaust valve** *r*. This valve closes automatically when the condenser contains a vacuum and opens automatically when the vacuum is destroyed. It is fitted with a piston *s* to prevent the valve hammering. If it should become necessary or desirable to cut the condensing apparatus out of service and to run the engine non-condensing, the free exhaust valve may be locked open by turning the hand lever *t* to the left.

The operation of the check-valve *b* is as follows: The inclined suspension bar *u* has a tendency to open the valve, while the inclined supporting bar *v* has a tendency to close it. Thus the valve is balanced by its own weight, which is so distributed that there is always a slight excess of closing tendency. The object of this valve is to prevent the water in the condenser being siphoned back into the steam cylinder.

38. When the injection water is supplied to the condenser under pressure, as from an elevated tank or from the street main, instead of being siphoned up, the openings *w* and *x* are blanked.

When the injection water is siphoned into the condenser and water under pressure is used for starting, the starting water enters at the opening *w* and the opening *x* is blanked.

When the injection water is taken under high suction and steam is used for starting, the steam enters through the opening *w*, a check-valve is attached at the opening *x*, and an overflow pipe is connected with the check-valve to discharge free or into the discharge pipe *l*.

39. It will be noted that this condensing apparatus requires neither air pump, circulating pump, nor tail-pipe, as is the case with the siphon condenser; therefore, all the power that is gained by working the engine condensing is made available for useful work, instead of being offset by the power required to work the air and circulating pumps, as in other types of condensers.

SURFACE CONDENSERS.

PURPOSE.

40. When the injection water is so impure as to be unfit for feedwater, the condensed exhaust steam may be saved for feeding purposes by keeping it separate from the injection water. It is the purpose of the surface condenser to do this.

The surface condenser differs from the jet condenser in that the steam is condensed by coming into contact with the cold surfaces of numerous tubes through which cold water is being pumped, instead of coming into direct contact with the injection water. In this way the water of condensation and the circulating, or injection, water, are kept entirely apart, which permits the water of condensation to be used as boiler feedwater. Condensed steam is pure water, therefore it is in the best possible condition for feedwater.

PUMPS REQUIRED.

41. The surface condenser requires two pumps, viz., an air pump and a circulating pump. The duty of the circulating pump is to force the injection water through the tubes of the condenser; the air pump removes the air, vapor, and water of condensation from the condenser. There being two pumps connected with the surface condenser to do the work that is performed by one pump in the case of the jet condenser, they can be much smaller than when only one pump is used.

ADVANTAGES.

42. Though the surface condenser is more complicated, its first cost greater, and it requires more attention on the part of the attending engineer than does the jet condenser, the value of pure feedwater for the boilers more than compensates for these disadvantages. It is far better to keep impurities out of the boilers entirely than to dose them with patent compounds, chemicals, kerosene oil, etc., or to allow the impurities to accumulate in the boilers in the form of scale, mud, etc. to the great detriment of the boilers, incurring loss of fuel and time and the expense of scaling and cleaning them.

CONSTRUCTION.

43. Construction of the Condenser.—Fig. 8 represents a sectional view of a surface condenser, without the

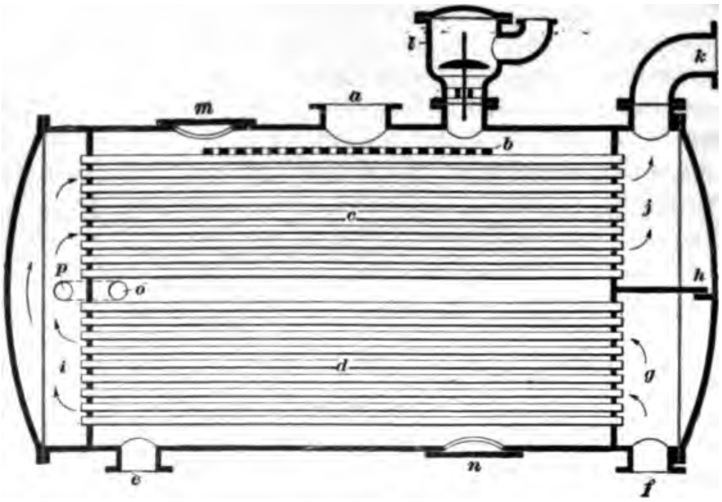


Fig. 8.

pumps. The body of the condenser is an air-tight vessel of either cylindrical or rectangular form, it matters not which. The exhaust pipe from the engine is connected at *a*. The steam, on entering the condenser, strikes the perforated scattering plate *b*, which distributes the steam over the tubes more evenly than otherwise would be the case, and it also protects the upper tubes from the damaging effects of the exhaust steam. The steam then circulates around the tubes of the upper nest of tubes *c*. The instant the steam comes into contact with the cold surfaces of the tubes it is condensed into water, which falls in the form of rain upon the lower tubes *d*, around which it circulates, and is further cooled as it falls to the bottom of the condensing chamber and flows out of the nozzle *e* to the air pump.

The injection water passes from the circulating pump through a pipe attached at *f* to the receiving chamber *g*; it is here impeded in its course by the division plate *h*, which deflects the water into the lower tubes *d*, through which it passes to the water chamber *i* at the opposite end of the condenser; from thence it returns through the upper tubes *c* into the delivery chamber *j* and then out of the discharge nozzle *k*. An automatic atmospheric relief valve, sometimes called the **snifting valve,** is shown at *l*. This valve opens outwardly and is for the purpose of relieving the condenser of any excess of steam, air, or vapor that may accumulate within it. It would also be useful in the event of the breaking down of the air pump or of its becoming inoperative from any cause, as the engine could then be run non-condensing by exhausting through this valve into the atmosphere.

44. Making Up for Loss of Water.—Part of the steam that is generated in the boilers is lost in various ways, such as blowing off with the safety valve, opening the gauge-cocks, and blowing the whistle, also through the cylinder and other drain cocks, by leakage, etc.; therefore, after all the exhaust steam from the engine is condensed into water there will not be enough of it to keep up the supply of feedwater; this deficiency is sometimes made up by drawing a corresponding amount of water from the circulating side of the condenser. For this purpose a U-shaped by-pass pipe is fitted around one of the tube-sheets and provided with a cock or valve; this allows a communication to be opened up between the steam and water sides of the condenser; the openings for this pipe are shown at *o* and *p*.

45. Though it is sometimes necessary to make up the deficiency of feedwater from the injection supply, it is objectionable to do so, for the reason that, sooner or later, the water in the boilers will become almost as impure as if injection water only had been used as feedwater, and the object of the surface condenser will be defeated. It will then be necessary to blow off some of the very impure water from the boilers and to replace it with a corresponding amount that is less impure. This will cause a serious loss of heat, as the water blown out has already been heated to the temperature corresponding to the pressure of steam carried, while the water pumped in to take its place must also be heated to the same temperature before it will be in the proper condition to be converted into steam. To prevent this loss, it is best to make up the loss of water with purified water or with distilled water, if circumstances permit this to be done.

46. Effects of Grease.—In course of time the tubes of a surface condenser become coated with grease on the steam side, carried over from the cylinder by the exhaust steam. Grease being a non-conductor of heat, the efficiency of the condenser is seriously impaired when the tubes are thickly coated with it, and to restore its usefulness the grease must be removed. This was comparatively an easy matter when animal or vegetable oils only were used for lubricating the pistons. The condenser was fitted with a special cock, called variously an **impermeator, alkali cock,** or **soda cock,** by which caustic soda or caustic potash were injected into the steam side of the condenser upon the grease-covered tubes. The alkali coming into contact with the grease converted it into soap, saponifying it, so to speak. The soap, being soluble, would be dissolved and washed out through the drain cock. This operation was assisted and hastened by having a small live-steam pipe enter the condenser near the bottom. The steam side of the condenser being filled with clean water up to and covering the top row of tubes and the alkali introduced, steam was let into the condenser through the small pipe until the water boiled. This was called boiling out the condenser and it was a very efficient way to get rid of the grease. But since mineral oils have almost entirely superseded animal and vegetable oils for cylinder lubrication, the boiling-out process is no longer feasible, because alkalies have no effect upon mineral oils; therefore, the grease deposited upon the condenser tubes from mineral oils must be removed by hand. In order to be able to clean out a surface condenser by hand, the tubes must be removed. In drawing the tubes, the greater part of the grease is stripped off the tubes by the tube-sheet; this necessitates a man going inside the condenser to scrape the grease off the tube-sheet. Manholes *m* and *n*, Fig. 8, are provided for this purpose.

47. It is not an easy task to clean out a surface condenser having several thousand tubes; it is one of the most disagreeable duties a practical engineer has to perform. This grease is very sticky, resembling tar, and is very difficult to remove not only from the tubes and tube-sheets, but also from the skin and clothing of the operator. It can generally be softened by a liberal application of kerosene oil or gasoline, which facilitates its removal. If gasoline is used, it must be remembered that the vapor given off is highly inflammable, and hence no naked light should be used near the condenser.

48. In view of the loss of efficiency in the condenser and the loss of time and the labor of cleaning out the grease, it is most desirable that the grease should be kept out of the condenser. This may be accomplished by introducing an efficient grease extractor in the exhaust pipe. The grease extractor is an apparatus that is coming largely into use and which contributes greatly towards keeping condensers and boilers free from grease. It should be found in every well-equipped condensing steam plant as a matter of economy.

49. The Double-Tube Surface Condenser.—Another form of surface condenser that is now extensively used is the **Wheeler double-tube surface condenser,** which is manufactured by the Wheeler Condenser and Engineering Company, of New York City.

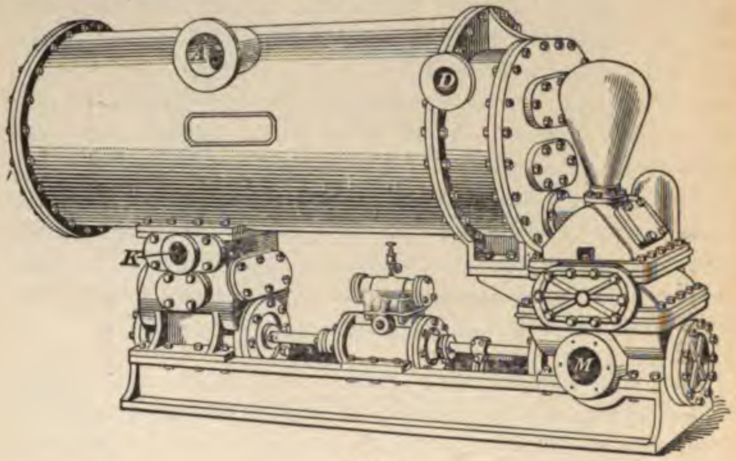


FIG. 9.

Fig. 9 is a perspective view and Fig. 10 a sectional view of the same. The injection water enters at M, Fig. 9, and

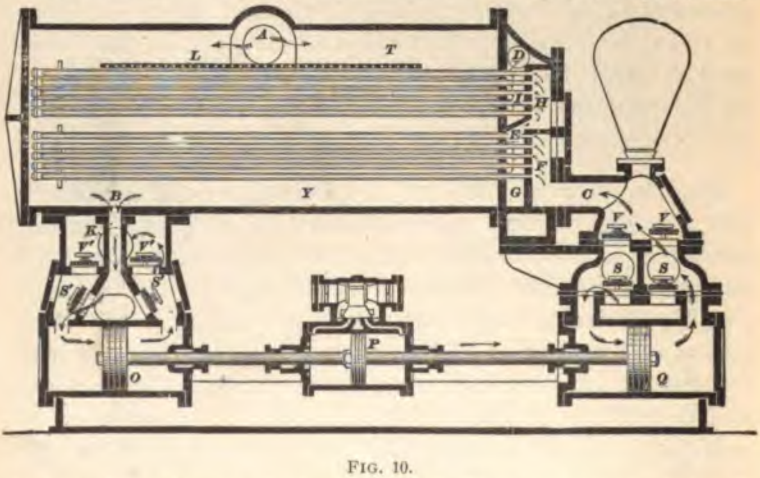


FIG. 10.

is forced by the circulating pump Q, Fig. 10, into the inlet C of the condenser. From C the water is forced into the chamber F and flows, as indicated by the arrows, through the inner tubes of the lower nest of double tubing to the left. Having passed through their entire length, the water returns through the annular space between the outside of the inner tubes and the inside of the outer tubes into the chamber G. Fig. 11 shows more clearly the arrangement of this double tubing. From G, Fig. 10, the circulating water passes through E to H and from H to I through the upper

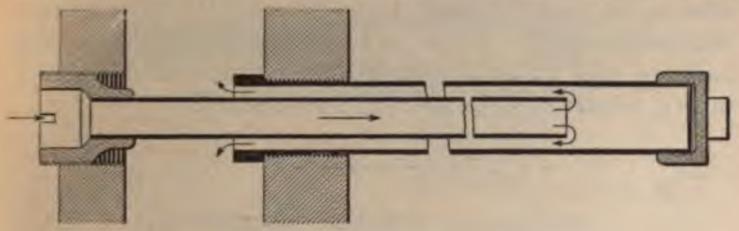


FIG. 11.

nest of double tubing, as has already been explained. From I it is discharged through the nozzle D, carrying with it all the heat it has taken from the exhaust steam while passing through the two nests of double tubing.

The nozzle A is connected with the exhaust pipe of the steam cylinder of the engine. The movement of the air-pump piston O draws the air, vapor, and water of condensation out of the condenser through the opening B and discharges them through the valves V', V' and nozzle K in the manner indicated by the arrows.

The valves S' and V' are opened and closed automatically by the pressure of the air beneath them and by the pressure of the air and springs above them. A partial vacuum is generated in the condenser Y by the air pump drawing the exhaust steam from the engine cylinder into the condenser, where it is condensed and a normal vacuum formed.

As the exhaust steam enters the condenser through the nozzle A, it first comes into contact with the perforated scattering plate L, which distributes the steam over the tubes and also protects the upper rows of tubes from the damaging effects of direct contact with the exhaust steam.

The steam then comes into contact with the cold tubes through which the cold circulating water is being pumped and is immediately condensed. As soon as this occurs, the water of condensation collects at the bottom of the condenser and flows through the opening *B* into the air-pump barrel, from which it is discharged by the piston of the air pump through the nozzle *K* into the hotwell or feed tank.

In this condenser the air and circulating pumps are operated by the independent steam cylinder *P*. The tubes of this condenser are secured to the tube-sheet at one end only and are free to expand and contract without danger of injury, rendering tube packing unnecessary.

DETAILS OF SURFACE CONDENSERS.

50. Condenser Tubes.—The latest practice regarding condenser tubes is to make them of a composition consisting of copper, 70 per cent.; zinc, 29 per cent.; tin, 1 per cent. They are $\frac{5}{8}$ inch outside diameter, No. 18 B. W. G. in thickness, and are spaced $1\frac{1}{16}$ inch between centers. They should be carefully tinned within and without, and when they are 6 feet or over in length, they should be supported by one or more supporting plates. The tube-sheet should be 1 inch thick and made of the same composition as the tubes, with smoothly finished holes for the tubes and proper means provided for packing them.

51. Condenser-Tube Packing.—There is a number of methods of packing condenser tubes. The one generally used at the present time is shown in Fig. 12, in which *a* represents a part of the tube-sheet in section; *c* is a screw gland that screws down on the packing *d*. The packing may be either cotton tape, cotton lamp wicking, or a rubber ring. The screw glands are counterbored, forming the shoulder *e*, the purpose of which is to prevent the tubes crawling; the glands are slotted, as shown at *b*, to admit a tool for screwing up.

52. Supporting Condenser Tubes.—Condenser tubes are subjected to great changes in temperature and, in consequence, to extremes of expansion and contraction. If they were rigidly fixed in both tube-sheets, they would become warped, bent, and distorted, and would thereby soon be destroyed; hence the necessity of giving them end play.

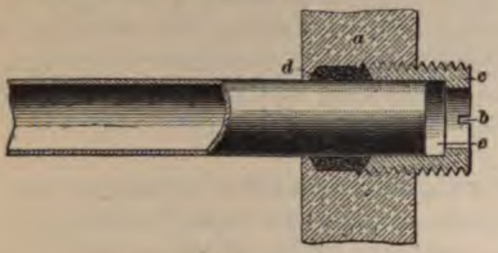


FIG. 12.

But the end play must be restricted, else the tubes will crawl by successive expansion and contraction until they become detached from either one or the other tube-sheet; this is obviated by the shoulder *e* in the gland. The space between the end of the tube and the shoulder is provided to give the tube room to expand or for the end play mentioned above.

LEAKAGE OF SURFACE CONDENSERS.

53. Water Leaks.—Owing to unequal expansion and contraction, the tubes of a surface condenser are subjected to very severe strains, and it is quite a common occurrence for them to split, causing leaks which if permitted to continue, will defeat the object of the surface condenser by admitting the injection water through the leaks to the steam side. The tube packing also will give out in time, which is another source of leakage. A serious leakage may make itself known by a noticeable increase in the feedwater, but these derangements must be looked for and the condenser tested for leaks occasionally or whenever the engineer in charge has reasons to believe it to be necessary. The test is made by removing the condenser bonnets and filling the steam side of the condenser with water; both ends of every tube should then be carefully examined. If water flows from any of the tubes, it proves that those tubes are split, and they should be drawn out at once and new tubes inserted. If the time cannot be spared just then to put in new tubes, the leaky ones may be plugged up at both ends with dry white-pine plugs, as a temporary repair; but it is always best to make a permanent repair whenever possible. Defective tube packing should be looked for at this time and any packing that leaks should be renewed.

54. Air Leaks.—Air leaks in a condenser will be revealed by the vacuum falling, all other conditions being right. They can generally be traced by the whistling sound the air makes while being drawn into the condenser or they may be located by holding a lighted candle along the joints; the flame will be drawn inwardly at the leak. An air leak in a condenser can usually be stopped by setting up on the nuts of the bonnet or gland; if this does not stop the leak, plaster it over with red-lead putty containing a small proportion of litharge; this cement will harden in a short time and effectually stop the leak.

GALVANIC ACTION IN SURFACE CONDENSERS.

55. One of the impurities frequently found in water is sulphuric acid. This acid is derived from the decomposition of iron and copper pyrites (sulphurets of iron and copper), which are widely distributed throughout the earth, especially in mineral regions. The sulphuric acid—in connection with the copper in the condenser tubes and the iron of the condenser casing, feedpipes, and boilers—completes a galvanic battery, which sets up a galvanic current that attacks the iron or steel, causing rapid corrosion that is especially injurious to the boilers. To arrest the generation of this galvanic current, the condenser tubes are tinned both inside and outside. The coating of tin prevents the acidulated water from coming into direct contact with the copper in the tubes.

56. As a further precaution against the deteriorating effects of galvanic action, plates of zinc are often suspended in the air-pump channel ways and in the hotwell or feed tank. The galvanic current attacks the zinc in preference to the iron or steel of the boilers and its destructive energy is expended in disintegrating the zinc. The zinc plates being gradually destroyed, it is evident that they must be renewed occasionally.

The metal straps by which the zinc plates are suspended should be filed bright where in contact with the zinc and in contact with the metal from which they are suspended. After the straps are bolted in place, the outside of the joints should be made water-tight by covering them with paint or cement to insure a good electrical contact.

The zinc plates are enclosed in wire baskets to prevent the larger particles of zinc that become detached from the plates during the progress of disintegration being carried into the boilers and lost by being blown out through the bottom blow.

CONDENSER ACCESSORIES.

COOLING TOWERS.

INTRODUCTION.

57. The great economy of using a condenser in connection with the steam engine has become so generally recognized by steam users, as it has been by engineers from the time of Watt, that it has been found desirable, as well as feasible, to install them in localities where the water supply is inadequate to operate a condensing apparatus in the ordinary manner. This demand has called into existence various devices for the purpose of cooling the discharge water from a condenser and using it over again as injection water, which operation may be repeated indefinitely with the same water, needing only a small supply of extra water to replace that which is evaporated during the cooling process.

The principle on which all these cooling devices operate is that the evaporation of a part of the water undergoing the cooling process extracts the heat from the remaining part.

58. There are two distinct methods of cooling liquids; the first is by absorbing and carrying off the heat by a liquid previously cooled, as, for instance, occurs in the surface condenser, in which the heat of the exhaust steam is absorbed and carried off by the cold injection water; while the second method is effected principally by evaporation.

59. Evaporation is the conversion of a fluid into a vapor. When water is freely exposed to a current of air, the air in immediate contact with the water soaks up, as it were, more or less of the water and the air becomes charged with vapor. As evaporation proceeds only from the surface of liquids, the quantity of liquid evaporated depends on the extent of the surface exposed. Dry air absorbs moisture very rapidly at first, but more and more slowly as the process proceeds, until it ceases altogether and the air is then surcharged with moisture; it is then said to be saturated, which means that the air has soaked up all the moisture that it can hold. It is evident, then, that the more air that is brought into contact with the liquid to be cooled, the more rapidly will the evaporation proceed.

60. The capacity of air for absorbing moisture increases rapidly with the temperature. Dry air at 100° F. will take up a much greater quantity of moisture than air at 60°. One cubic foot of water vapor at a temperature of 60° F. weighs 5.745 grains, at 100° it weighs 19.766 grains, and at 110° it weighs 26.112 grains. If saturated air at 60° is heated to 100°, it will no longer be saturated, and will then be capable of taking up 14.021 grains more moisture per cubic foot.

61. During the process of evaporation a great amount of heat is absorbed by the vapor, or, as it may be said, consumed in forming the vapor, and becomes latent. 966 B. T. U. are required to convert 1 pound of water into vapor at the atmospheric pressure. As this heat is derived or taken from the water that remains after a portion of it has been converted into vapor, the remaining water is cooled thereby.

CONSTRUCTION OF COOLING TOWERS.

62. Advantage has been taken of the cooling effect of evaporation to cool the discharge water of a condenser so that it may be used as injection water again. The application of the principle is usually through the medium of an apparatus called a **cooling tower,** which consists substantially, as its name implies, of a tower-like structure, usually about 30 feet high, to the top of which the warm water is pumped and there liberated and allowed to fall to the bottom of the tower against a current of ascending air which is impelled upwards by a fan blower. The water in descending is broken into spray or thin sheets by coming into contact with obstructions placed in its path for that purpose, thus presenting the greatest possible area of evaporating surface to be acted upon by the air.

There is a number of cooling towers in use, and while they differ somewhat in construction and material, they all are governed by the same principle.

63. The **Worthington cooling tower,** shown in Fig. 13, consists of a cylindrical steel shell *a* open at the top, supported upon a suitable foundation, and having fitted at one side a fan *b*, the function of which is to circulate a current of air through the tower and its filling. This filling consists of layers of cylindrical tubular tiling *c*, *c'*, *c''*, which rest upon a grating *d* supported by a brick wall *e* extending around the circumference of the tower. The warm discharge water from the condenser enters the tower through the pipe *f*, passes up the central pipe *g*, and is delivered on the upper layer of tiling and over the whole cross-section of the tower by the distributing device *h*, which consists of four pipes, radiating from the central pipe *g*, which are caused to revolve about the central pipe by the reaction of jets of water issuing from perforations on one side of each pipe. The water thus delivered spreads over the outside and inside surfaces of the walls of the tiling and forms a continuous sheet, which is presented to the action of the air.

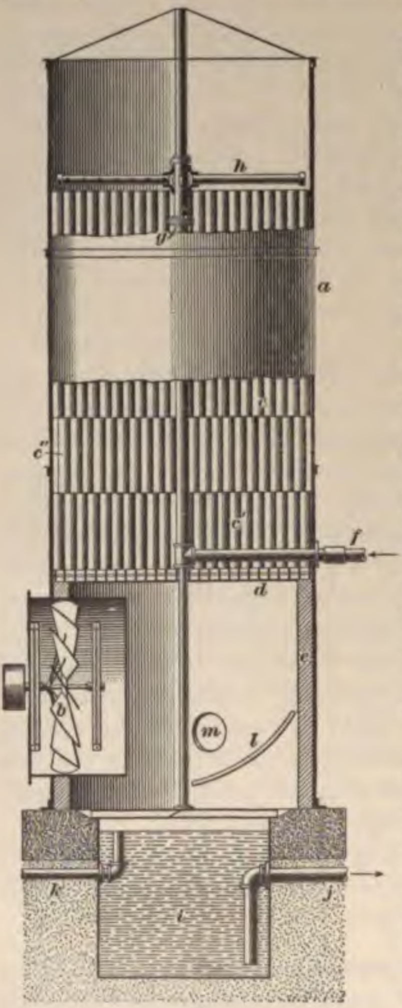


FIG. 13.

The tiling is placed on end in horizontal layers, one on the other, and is packed as closely as possible, the walls of each individual tile of each successive layer being disposed so as to come opposite the air space of the next lower layer, breaking joints, as it were; the object of this disposition is to break up both the currents of air and water so that the most extended contact will take place.

If there are ten layers of tiling in a tower, then there are nine places in addition to the original spreading at the top at which there is a complete redistribution of the water. It will be seen that each tile must rest on at least two, and possibly three, in the next lower layer. Assuming, however, that each tile in the upper row of tiles rests on only two others, a given quantity of water placed on any one tile in the top layer will be divided over at least two tiles in the second layer, three in the third, four in the fourth, and so on until it becomes spread over fifty-four tiles in the bottom layer on the grating.

The air is distributed in a similar manner, but in a reverse direction, to the flow of water, and there is a large free area for its passage upwards over the entire cross-section of the tower. The warm water falling through the tower is cooled by three processes: first, by radiation of heat from the sides of the tower; second, by contact with cool air; and third, by evaporation. The latter is by far the most effective and important, for the reason that the evaporation of a pound of water in this way carries off 966 B. T. U. The cooled water falls from the grating into the reservoir *i* at the bottom of the tower, and from there is forced into the condenser, either by atmospheric pressure or by a circulating pump through the injection pipe *j*, to perform condensation again.

The current of air is impelled through the tower by the circulating fan *b*, driven either by a small steam engine, an electric motor, or by belting from the main engine or line shaft, as the case may be.

A deflecting plate *l* is used to direct the current of air upwards through the tower. A manhole *m* affords access to the interior of the tower for examination, cleaning, and repairs, and *k* is the overflow pipe from the reservoir.

64. The **Barnard-Wheeler water-cooling tower** is represented in Fig. 14. The tower casing is usually constructed of steel plates. Within the tower are hung a number of mats made of special steel-wire cloth, galvanized

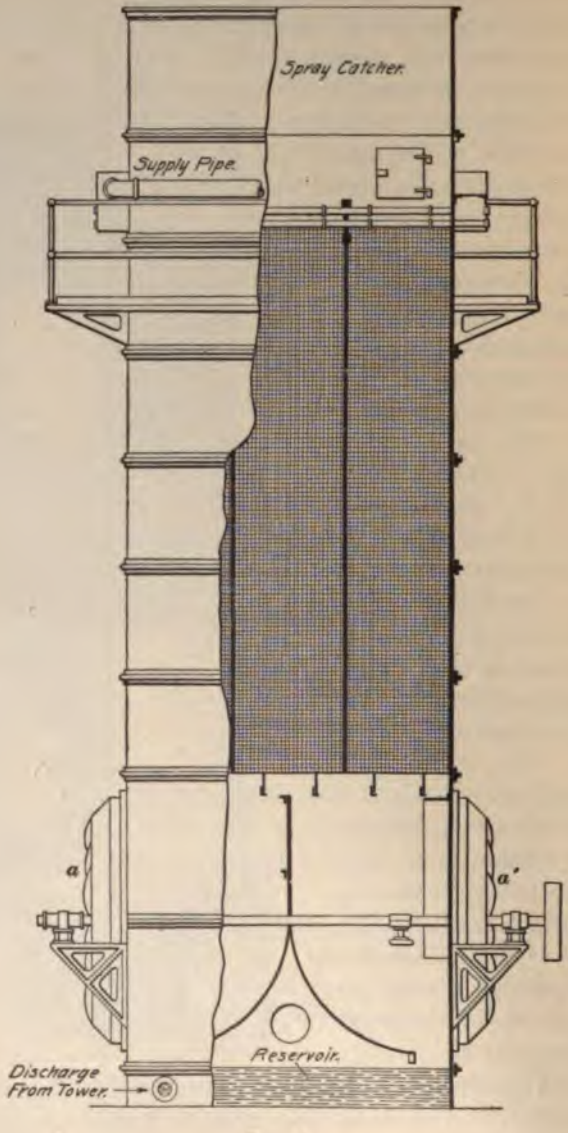

Spray Catcher
Supply Pipe
a
a'
Reservoir
Discharge
From Tower

FIG. 14.

after weaving. The pump discharge is led to the top of the tower and the warm water is there distributed by a suitable system of piping to the upper edges of the mats, over the surface of which it spreads in thin films, compelling a partial interruption of the flow and continuously bringing new portions of the water to the surface, thereby exposing it to the evaporating and refrigerating effects of the air-currents.

The mats are practically metallic sponges capable of holding a large quantity of water in suspension, which accumulates and drips off into the supply reservoir at the bottom of the tower. To assist the cooling action, the air in immediate contact with the water is set in rapid circulation by means of the fan blowers *a*, *a'*, Fig. 14, which force air into the lower part of the tower and upwards between the mats.

65. Barnard's fanless self-cooling tower is shown in Fig. 15. In this device the use of mechanical means for circulating air for cooling the water is dispensed with, thus avoiding the wear and tear and the expenditure of power that are always associated with moving parts. The hot circulating water discharged from the condenser is pumped up through the central stand pipe *a*, from which it is led to the trough *b* and distributing pipes *c*, *c*, causing a constant flow of thin films of water over the meshes of the wire mats *d*, *d'*, and finally draining into the tank or reservoir *f* forming the foundation of the tower, from whence the cooled water is returned through the injection pipe *e* for use again in the condenser.

The mats are placed radially and are entirely exposed to the atmosphere; they are so arranged as to permit the air to come into contact with the descending films of water by natural circulation, and the consequent evaporation is carried far enough to reduce the temperature of the injection water to a sufficiently low degree for condensing purposes.

66. The **Dean refrigerating tower** is rectangular in form and the cooling surfaces consist of metal tubes placed horizontally, over which the water to be cooled slowly flows, dripping from tube to tube and exposing it in thin films to the refrigerating effect of an upward current of air kept in circulation by a fan blower.

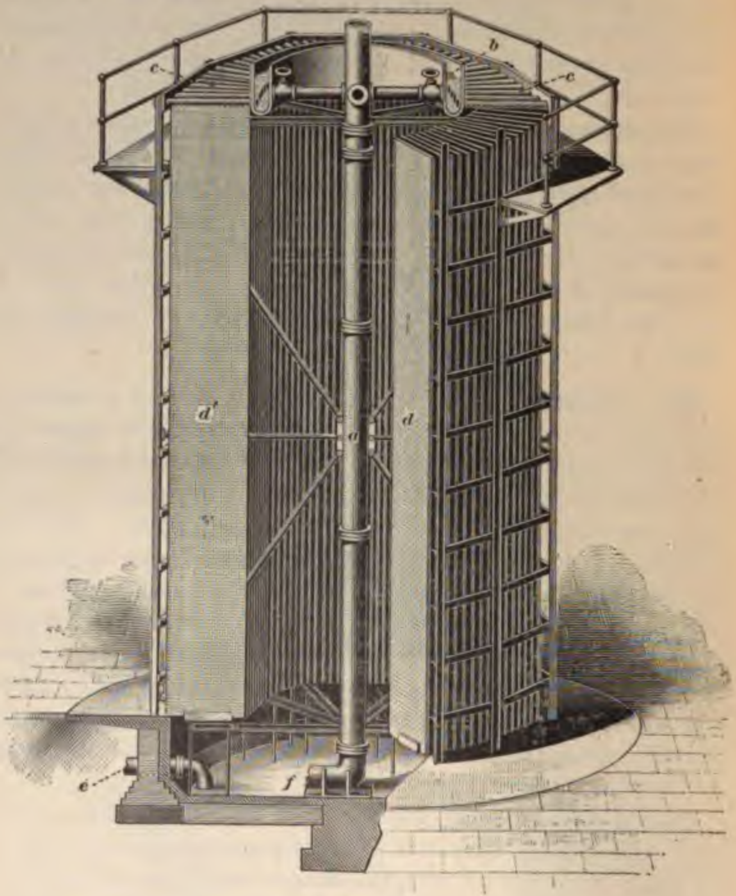


FIG. 15.

67. The **Stocker cooling tower** is a structure built of wood, steel, or bricks, according to circumstances. The cooling surfaces are built up of checkerwork or crosspieces of boards in horizontal layers set at right angles to one another. At the intersections are placed upright partitions diagonally across the square openings between the boards. The water is pumped to the top of the tower and trickles down over these surfaces in thin films, which are broken up in falling at each intersection of the boards, and the water is thus brought into contact with the current of air that is forced upwards through the tower by fan blowers.

68. Cooling towers may be located on the roof of a building, if space for them is not available upon the ground, and they may be used in connection with any type of condenser. It is to be observed that there is a possibility of the water in a cooling tower freezing during very severe weather.

VARIOUS COOLING SYSTEMS.

69. There are other methods of cooling discharge water from condensers for the purpose of using it over again as injection water besides by cooling towers.

70. The **Schutte system** of cooling discharge water consists of a series of centrifugal spray nozzles that project the warm water from the condenser into the open air by a whirling action in the form of spray, under a pressure of 15 pounds or more to the square inch. A tank, pond, or other catch basin is provided to retain the cooled water.

71. The **Linde system** of cooling discharge water consists of a number of sheet-iron cylinders immersed about one-third of their diameters in the water of the condenser. These cylinders are revolved slowly, thereby carrying up thin films of water adhering to their surfaces, which, on coming into contact with a current of air, are evaporated, producing the cooling effect. The condenser is merely a tank containing submerged pipes connected with the engine exhaust. The water in the tank is kept in constant motion by an agitator.

LOSS OF WATER BY EVAPORATION.

72. When injection water is recooled, the loss of water by evaporation is proportional to the amount of heat carried off and depends on the difference between the temperature of the water before and after cooling. For example, if 200,000 pounds of water is cooled down from 120° to 60° in a given time, 60 B. T. U. are absorbed from every pound of water that is evaporated; therefore, 200,000 × 60 = 12,000,000 B. T. U. are carried off with the vapor. Now, as it requires about 1,000 B. T. U. to vaporize 1 pound of water, $\frac{12000000}{1000}$ = 12,000 pounds of the water, or 6 per cent., have been converted into vapor and lost, which amount must be restored to the volume of available injection water from some original source of supply.

CONDENSER FITTINGS.

73. Certain fittings are required on a condenser to make it complete. A vacuum gauge, as a matter of course, is a necessity on all condensers, and it should be placed in a conspicuous part of the engine room, where the engineer may consult it at any time.

74. A surface condenser should be provided with standard thermometers, one each for the injection water, the feedwater, and the discharge water. The thermometers should be permanently connected to the condenser by means of small pipes, through which streams of the various waters mentioned should be kept circulating around the bulbs of their respective thermometers so that the temperature of the waters may be noted at a glance. Inasmuch as the best results can be obtained only by having the temperature of the feedwater as high as possible and the temperature of the discharge water as low as possible, and as these temperatures can only be accurately determined by means of standard thermometers, their utility is obvious. The careful engineer that keeps the condenser temperatures at their best points will save many a ton of coal, and to enable him to do this the thermometers ought to be supplied, as a means of economy. The condenser temperatures are regulated by the injection valve or by the speed of the circulating pump, if independent. By these means more or less circulating water is passed through the condenser tubes, as required.

75. It may happen in winter that the injection water is very cold and that not enough of it is required for condensation to fill the barrel of the circulating pump, which may cause slamming of the valves. If the circulating pump is independent of the main engine, the difficulty may be remedied by slowing down the pump; but if the pump is attached to and operated by the main engine, this cannot be done. To meet this contingency, a reverse air valve is fitted to the barrel of the circulating pump, which will allow a certain amount of air to enter the pump at each stroke and relieve the slamming. The amount of lift of the air valve is controlled by a screw stem and hand wheel; the valve may be permanently closed by the same means.

76. Another method of accomplishing the same purpose is to provide a communication, by a pipe or channel way, between the discharge and the receiving chambers of the circulating pump, fitted with a valve, which, on being opened, will permit a part of the circulating water to return to the receiving chamber of the pump instead of passing through the condenser tubes. This valve is called the **regurgitating valve,** because it permits the water to regurgitate back towards its source. By regurgitating is meant the flowing back and forth of a liquid.

77. Surface condensers are usually provided with an **alkali cock,** by which caustic soda or caustic potash may be injected into the condenser for the purpose of dissolving the grease that collects upon the tubes from the exhaust steam. The alkali cock is simply an ordinary plug cock screwed into the top of the condenser over and near the center of the upper nest of tubes, and having a cup or funnel-shaped top into which the alkaline solution may be poured. As an accompaniment to the alkali cock, a small live-steam pipe is run into the lower part of the condenser for the purpose of boiling the alkaline water as an aid to dissolving the grease.

78. A by-pass pipe containing a cock or valve is fitted around one of the tube-sheets of a surface condenser, whereby any deficiency of feedwater from the condensed steam may be supplied from the circulating water, provided there is no other source of supply.

79. The discharge water and feedwater being intermingled in a jet condenser, only two standard thermometers are required—one for the injection water and the other for the feedwater.

80. All condensers should be provided with a sufficient number of drain cocks so located as to completely drain all the water out of them. They should also have manholes and handholes wherever practicable and wherever necessary to afford access to the interior for examination, cleaning, and repairs.

81. The hotwell, or feed tank—which is practically the same thing and constitutes a part of the condensing apparatus—should be provided with a glass water gauge so that the height of water within may be observed at all times. It is by observing the fluctuations of the water level in this gauge that the engineer is informed as to the sufficiency or insufficiency of the feedwater supply.

82. A **vacuum breaker** is a necessary adjunct of a jet condenser. This is a device that will automatically break the vacuum in the condenser in case of a sudden breakdown of the air pump, and will thus prevent the filling up of the condenser and the subsequent filling of the cylinder with water, which is coupled with the danger of wrecking the cylinder. A vacuum breaker is usually constructed in the form of a float-operated valve attached to the condenser and opening towards the atmosphere; it is so placed that when the water in the condenser exceeds a certain height, it will lift the float, open the valve, and admit air, thus destroying the vacuum and preventing a further inrush of injection water.

QUANTITY OF WATER REQUIRED FOR CONDENSATION.

83. The number of pounds of condensing water required to condense a pound of steam depends on the initial and final temperature of the steam and the initial and final temperature of the condensing water.

84. A pound of steam after being condensed to water having the temperature at which it leaves the condenser contains a number of B. T. U. above 32° F. that is given by subtracting 32 from its temperature. But a pound of steam before condensation contains a number of heat units above 32° corresponding to its pressure. Consequently, the number of B. T. U. that must be abstracted is given by subtracting from the total heat of the steam the difference between the final temperature of the steam after condensing and 32.

Each pound of condensing water in passing from its initial to its final temperature absorbs a number of B. T. U. equal to the difference in the initial and final temperatures. Therefore, the number of pounds of water required will be the number of B. T. U. to be abstracted from a pound of steam divided by the number of B. T. U. absorbed by a pound of condensing water. Hence the following rule:

Rule 1.—*To find the weight of condensing water per pound of steam, from the total heat of a pound of exhaust steam subtract the difference between the final temperature of the* *condensed steam and 32; divide the remainder by the difference between the temperatures of the entering and departing condensing water.*

Or,
$$W = \frac{H - (t_3 - 32)}{t_1 - t_2},$$

where W = number of pounds of water required to condense a pound of steam;
H = total heat of vaporization of 1 pound of steam at the pressure of the exhaust. This may be obtained from column 5 of the Steam Table;
t_1 = the temperature of departing condensing water;
t_2 = the temperature of entering condensing water;
t_3 = the temperature of the condensed steam upon leaving the condenser.

The pressure of the exhaust is to be taken as the pressure at the point of release in the cylinder. In a jet condenser the final temperatures of the cooling water and the condensed steam are the same; that is, $t_3 = t_1$.

EXAMPLE 1.—The steam exhausts into a surface condenser at a pressure of 4 pounds absolute. The temperature of the condensing water on entering is 60° and on leaving is to be 100°. The temperature of the condensed steam on entering the air pump is 140°. How many pounds of condensing water are required per pound of steam?

SOLUTION.—From the Steam Table, the total heat of 1 pound of steam at a pressure of 4 pounds absolute is 1,128.641 B. T. U. Then, applying rule **1**, we have

$$W = \frac{1{,}128.641 - (140 - 32)}{100 - 60} = 25.52 \text{ lb. Ans.}$$

EXAMPLE 2.—Steam exhausts into a jet condenser at a pressure of 2 pounds absolute. The temperature of the condensing water is 60° and the temperature of the mixture as it enters the pump is 135°. How much condensing water is used per pound of steam?

SOLUTION.—Applying rule **1**, we have

$$W = \frac{1{,}120.462 - (135 - 32)}{135 - 60} = 13.566 \text{ lb. Ans.}$$

CAUSES OF AN IMPERFECT VACUUM.

85. An imperfect, i. e., a low, vacuum is due to one or more of three causes, which are the amount of condensing water supplied may be insufficient; the air pump may be out of order; or there may be air leaks.

86. The probable cause may be ascertained easily if a *log* of the performance of the engine is kept. In this log the temperature of the hotwell, the temperature of the discharged condensing water, the initial temperature of the entering condensing water are entered, say, every hour. Should the vacuum in a surface condenser be found imperfect, the temperature of the hotwell and the discharged condensing water should be ascertained. If the temperature of both is found to be more than has been noted down in the log book for a more perfect vacuum, it would tend to show that not enough condensing water is supplied. If an increased supply of condensing water fails to improve the vacuum, although it lowers the temperature of the hotwell and of the discharged condensing water, the indications are that an air leak exists somewhere about the condenser or engine. If no air leak is found, the air pump must be examined. Usually one or more of the air-pump valves will be found either broken or in poor condition. If the temperature of the hotwell and of the discharged condensing water are the same as in the case of a more perfect vacuum, the indications are that there is either an air leak or that the air pump is partially disabled.

87. In a jet condenser the temperature of the hotwell and that of the condensing water is always the same, since the exhaust steam and condensing water mingle. Hence, the hotwell temperature remaining the same, as in the case of a more perfect vacuum, either an air leak or a partially disabled air pump is indicated. If the hotwell temperature is greater, it indicates an insufficient quantity of injection water.

88. The maximum vacuum attainable, theoreticall
never be more, in inches of mercury, than the height
column of the barometer at the same time and place. (
to mechanical imperfections of the condensing appa
about the best vacuum that may be attained in pr
will be within 2 inches of the theoretical vacuum.

COMPOUND ENGINES.

ADVANTAGES OF COMPOUNDING.

HISTORICAL.

1. In 1781 Jonathan Hornblower constructed and patented an engine, similar to the present "compound," that had two cylinders of different sizes. Steam was first admitted into the smaller cylinder and then passed over into the larger, doing work on a piston in each. In Hornblower's engine the two cylinders were placed side by side, and both pistons acted on the same end of a beam overhead. The use of this early compound engine was abandoned on account of the suit brought by a Birmingham firm for infringing their patent, which applied to the use of a separate condenser and air pump. At the beginning of the nineteenth century the compound engine was revived by Woolf, with whose name it is often associated, and who expanded the steam to six and even to nine times its original volume in two separate cylinders.

ECONOMY IN USE OF STEAM.

2. The compound engine has a mechanical advantage over the single-cylinder engine with an equal ratio of expansion and of equal power, but it is doubtful whether this advantage was clear to either Hornblower or Woolf when they designed their compound engines. This advantage of the compound engine over the simple expansive engine is that the pressure on the bearings is more even throughout the stroke and not so high at the beginning of the stroke. However, a far more important merit of the compound engine lies in a fact that neither of them discovered for many years, and which is that by dividing the whole range of expansion into two or more parts and performing the work in separate cylinders, an economy in the use of steam was effected.

3. If an attempt is made to obtain a very high ratio of expansion in a single-cylinder engine, we are confronted with the fact that the gain due to the high ratio of expansion is lost entirely or even overbalanced by the loss due to the initial condensation of the entering high-pressure steam, this loss being occasioned by the cooling of the cylinder walls during exhaust.

4. Let the temperature of the entering steam be 753° absolute and the temperature of the exhaust steam 613° absolute. Then, the fall in temperature during expansion will be $753° - 613° = 140°$. Now, the cylinder walls, especially in engines having a very slow piston speed and having the cylinder poorly covered, will cool off to nearly the temperature of the exhaust steam, and hence the incoming steam must give up part of its heat in order to heat the cylinder walls again. As a matter of course, the condensed steam represents nearly a total loss of heat. We say *nearly* because, owing to the fact that the cylinder walls do not cool *entirely* down to the temperature of the exhaust steam and also on account of the reduction of pressure, some of the steam condensed on entering is evaporated again near the end of the stroke.

5. Suppose, now, that the temperature of the incoming steam is 842° absolute and that the temperature of the exhaust steam is again 613°; that is, that we have a higher ratio of expansion. We now have a temperature range of 842° − 613° = 229°. A little thought will show that with this enlarged temperature range, a good deal more initial condensation will occur in order to raise the temperature of the cylinder walls to 842° again. From this the conclusion may be drawn that increasing the range of temperature increases the loss due to cylinder condensation.

6. Having seen that a high ratio of expansion, i. e., a high temperature range in a single cylinder, means a great initial condensation and subsequent loss in economy, let us see what the effect will be if the expansion of steam takes place in successive stages in several cylinders through the same temperature range. Then, the total temperature range being the same, it is evident that the temperature range in each cylinder will only be a fraction of it, and this has been found to reduce the condensation loss considerably below that occurring in one cylinder having the same total temperature range. This fact has been established beyond doubt by numerous experiments and has led successively to the adoption of compound engines, then triple-expansion engines, and lately quadruple-expansion engines.

7. We will now consider some facts showing the reason why a division of the temperature range between several cylinders tends to reduce cylinder condensation. It takes an appreciable period of time to change the temperature of the cylinder walls from the temperature of the exhaust steam to that of the entering steam. In most cases, if not in all, the time during which the entering hot steam is in contact with the cylinder walls is so short that the incoming steam cannot heat them to its own temperature. Likewise, the time during which the cylinder walls are in contact with the exhaust steam is usually so short that they do not cool entirely down to its final temperature. Consequently, the fluctuation of the temperature of the cylinder walls is less than the temperature range of the incoming and exhaust steam. It is a well-known fact that the *rate* at which heat passes from a hot body (the entering steam in this case) to a cold body (the cylinder walls in the case under consideration) depends on the difference in temperature between the two bodies. When the temperature difference is great, the rate of heat transmission is much larger, proportionally, than it would be if the temperature difference were small. In other words, the rate at which the transfer of heat takes place increases much faster than the temperature difference. It follows from this principle, when the difference in temperature between that of the incoming steam and that of the cylinder walls, previously cooled by the exhaust, is small, that less heat, in proportion to the temperature difference, will be given up to reheat the cylinder walls than would be the case for a larger temperature difference. Similarly, if the temperature of the exhaust steam is but little below that of the cylinder walls, less heat in proportion to the temperature difference will be lost by the cylinder walls. Now, by expanding steam successively in several cylinders, we not only decrease the temperature range in each cylinder, but also greatly decrease the fluctuation in temperature of the cylinder walls of each. Then, as the rate of heat transmission is proportionally much smaller for a small fluctuation of temperature, it follows that the sum of the condensation losses in the several cylinders will be smaller than the condensation loss occurring in a single cylinder having the same temperature range between the initial and final temperature.

8. Another probable cause of reduction of cylinder condensation by compounding is that the steam condensed in the high-pressure cylinder and reevaporated during the exhaust period enters the low-pressure cylinder as *steam;* hence, the heat given up in the high-pressure cylinder by the entering steam in heating the cylinder walls is not a total loss, as would be the case if, instead of exhausting into a second cylinder, exhaust took place into the atmosphere or into a condenser.

MECHANICAL ADVANTAGES OF COMPOUNDING.

9. So far we have considered only the saving effected in the cylinders of the engine, as it is there that the greatest difference is shown; but there are other advantages incidental to the use of the triple-expansion and compound engines that deserve attention and recognition. The pressures, and consequently the strains, in a triple-expansion or a compound engine are more evenly distributed throughout the stroke than they are in a simple engine, and the turning moment on the shaft is also more nearly equal; therefore, lighter working parts can be used with the same margin of safety and lighter flywheels to give the same regularity of speed. These lighter weights and more even strains mean less friction in the engine, and this in turn means a saving of coal.

10. In order to show how this is possible, we will consider two engines of equal horsepower, the one a simple

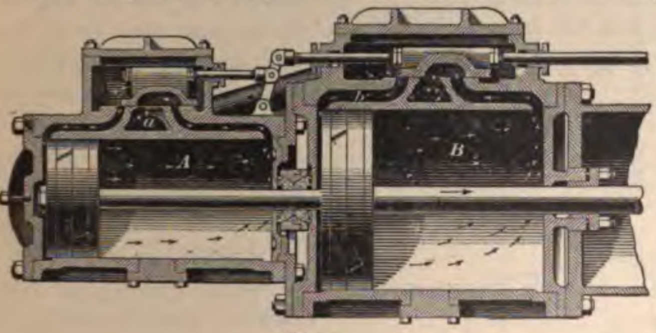


FIG. 1.

expansive engine and the other a so-called **tandem compound engine** that has two cylinders in line with each other and a piston rod common to both cylinders. Obviously, both pistons are carried on the same rod, as shown in Fig. 1. The cylinder *A* is called the **high-pressure cylinder** and the cylinder *B* is known as the **low-pressure cylinder.** The exhaust from the high-pressure cylinder passes through the exhaust passage *a* into the steam chest of the low-pressure cylinder.

The simple expansive engine has the following dimensions and data: Area of piston, 420 square inches; stroke, 30 inches; revolutions per minute, 90; steam pressure, 105 pounds gauge. The mean effective pressure of this engine, as found from the indicator card *A*, Fig. 2, is 48.66 pounds per square inch.

The tandem compound engine has the following dimensions and data: Area of high-pressure piston, 200 square inches; of low-pressure piston, 600 square inches; stroke, 30 inches; revolutions per minute, 90; steam pressure, 105 pounds gauge. The mean effective pressure, as found from the indicator card shown at *B*, Fig. 2, is 51.1 pounds in the high-pressure cylinder and 17.03 pounds in the low-pressure cylinder, as found from the indicator card shown at *C*, Fig. 2. The two engines will practically be of the same horsepower, since the mean impelling force on the piston of the simple engine is 420 × 48.66 = 20,437.2 pounds and the mean impelling force on the two pistons of the tandem compound engine is 200 × 51.1 + 600 × 17.03 = 20,438 pounds, and their respective strokes and number of revolutions are equal.

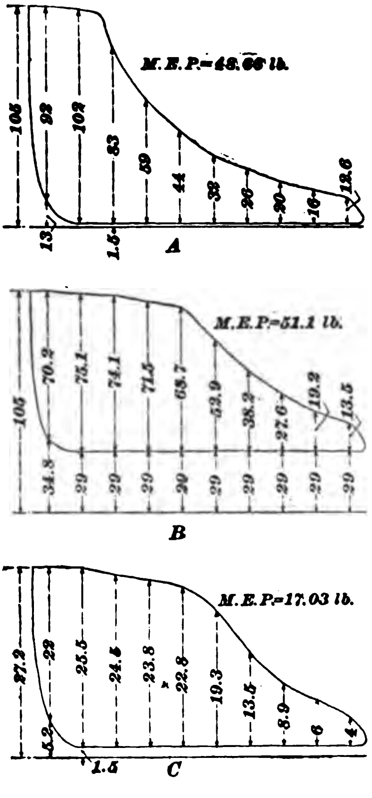


Fig. 2.

11. For convenience, we shall consider the load on the respective piston rods when the engines have made $\frac{1}{20}$ the stroke. Then, from the diagram *A*, Fig. 2, we find the net pressure per square inch on the piston to be 92 pounds, the back pressure at this point being 13 pounds. It is to be understood that the ordinates, by means of which the mean effective pressures are found, are erected at points on the diagram corresponding to $\frac{1}{20}$, $\frac{3}{20}$, $\frac{5}{20}$, etc. of the stroke. Since the area of the piston is 420 square inches, the load on the piston rod at this point of the stroke is $420 \times 92 = 38,640$ pounds. In the tandem compound engine, by diagram *B*, Fig. 2, the pressure on the high-pressure piston is 70.2 pounds net; the net pressure on the low-pressure piston, by diagram *C*, Fig. 2, is 22 pounds. The reason these given pressures are the net pressures follows from what has been stated regarding the action of steam in the cylinders of a compound engine. Then, the load on the high-pressure piston is $200 \times 70.2 = 14,040$ pounds and the load on the low-pressure piston is $600 \times 22 = 13,200$ pounds, making a total load on the low-pressure piston rod of $14,040 + 13,200 = 27,240$ pounds. In the same manner, as shown above, we may calculate the loads on the pistons at $\frac{3}{20}$, $\frac{5}{20}$, $\frac{7}{20}$, etc. of the stroke and then arrange them in the form of a table, as given below:

Stroke.	Load on Piston Rod.	
	Simple.	Compound.
$\frac{1}{20}$	38,640	27,240
$\frac{3}{20}$	42,840	30,320
$\frac{5}{20}$	34,860	29,520
$\frac{7}{20}$	24,780	28,580
$\frac{9}{20}$	18,480	27,420
$\frac{11}{20}$	13,440	22,160
$\frac{13}{20}$	10,920	15,740
$\frac{15}{20}$	8,400	10,860
$\frac{17}{20}$	6,720	7,440
$\frac{19}{20}$	5,292	5,100
	204,372	204,380

From this table we see that while in the compound engine under discussion the maximum load on the piston rod is only 30,320 pounds, it is 42,840 pounds in the simple engine, thus showing that if the simple engine were converted into a compound of equal horsepower, retaining all parts except the cylinder and piston rod, the maximum load on the various parts, and hence the maximum stresses, would be less. An inspection of the table will also show that there is less variation in the loads in case of the tandem compound than in case of the simple engine; hence, the tandem compound will run steadier. The above shows that if conditions allow it, an engine giving trouble through weakness of its component parts and insufficient bearing surface may be remodelled and will probably give good satisfaction when changed to a tandem compound of equal power.

12. Having shown that the loads are smaller with a tandem compound engine, it will be seen upon reflection that an engine rigid in all its parts may have its horsepower increased by making it a tandem compound without exceeding the maximum loads the various parts sustained when it was a simple engine. Thus, if we changed the simple engine under discussion to a tandem compound giving indicator cards like *B* and *C*, Fig. 2, and having an area of 250 square inches in the high-pressure cylinder and 750 square inches in the low-pressure cylinder, the load on the low-pressure piston rod would be $250 \times 75.1 + 750 \times 25.5 = 37,900$ pounds at $\frac{3}{20}$ the stroke, or considerably less than if the engine were a simple expansive one. We will now compute the horsepower of both engines. That of the simple engine is $\frac{420 \times 48.66 \times 2.5 \times 2 \times 90}{33,000} = 278.7$ horsepower, nearly. The horsepower of the compound engine having cylinders of 250 and 750 square inches area is $\frac{250 \times 51.1 \times 2.5 \times 2 \times 90}{33,000} + \frac{750 \times 17.03 \times 2.5 \times 2 \times 90}{33,000}$ $= 348.4$ H. P., nearly. We thus see that the horsepower of the engine is increased by $348.4 - 278.7 = 69.7$ horsepower and that the maximum loads and stresses are still less than in case of the simple expansive engine here considered.

13. At one time or other, nearly every manufacturing concern finds that it needs more power. The common way of accomplishing this is to put in a larger cylinder, which, unless a very generous allowance of bearing surface has been made in the engine as originally constructed, leads, in a great many cases, to hot bearings and various other troubles. It would certainly seem to be the better plan to convert the engine into a tandem compound if conditions allow it. This is not always feasible, however; if the boiler pressure is less than 80 pounds gauge, it is hardly ever advisable to change to a compound. No general rule can be given as to when it is advisable to make the change; each case must be treated on its own merits.

14. So far only the tandem compound and the simple expansive engine have been compared, and it has been shown that the tandem compound engine possesses mechanical advantages over the simple engine of equal horsepower. Many compound engines are built as **cross-compounds,** one of which is shown in plan in Fig. 3.

The cross-compound engine possesses a mechanical advantage not only over the simple expansive engine, but also over the tandem compound engine of equal horsepower. In a cross-compound engine the high-pressure cylinder *a* and low-pressure cylinder *b* are placed parallel to and alongside of each other, as shown in Fig. 3. Both cylinders are connected to a common crank-shaft *c*, which carries the flywheel *d* between the bearings and has cranks *e* and *f* at each end. These cranks are placed at right angles to each other, so that when one crank is on a dead center, the other is midway between the centers. In consequence of this there is a much more uniform turning moment on the crank-shaft than in a tandem compound engine, and hence the cross-compound engine will run steadier than the tandem compound, which runs steadier than the simple expansive engine. In comparing the steadiness of running, it is assumed that the flywheels are equal for engines of equal horsepower. From this the conclusion may be drawn that

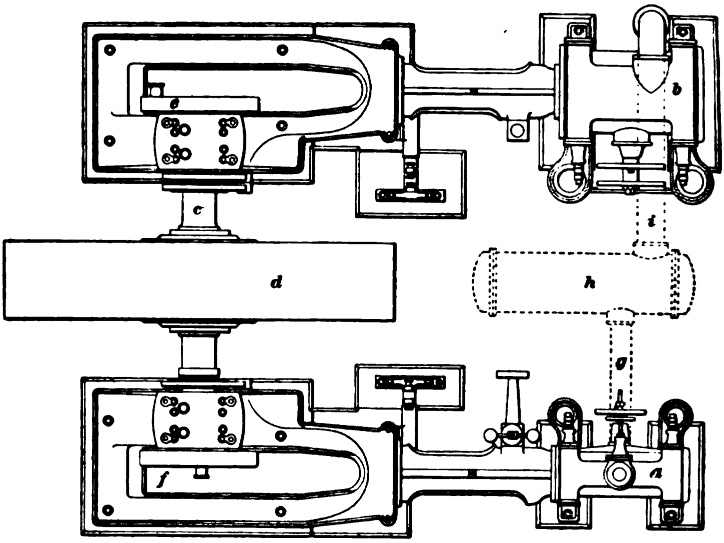


FIG. 3.

for equal degrees of steadiness the cross-compound will have the lightest flywheel, a heavier one is needed for the tandem compound, and the simple expansive engine needs the heaviest wheel.

In a cross-compound engine the exhaust from the high-pressure cylinder passes through the high-pressure exhaust pipe *g* (see Fig. 3) into the **receiver** *h* and thence through the receiver steam pipe *i* into the low-pressure steam chest. The exhaust from the low-pressure cylinder passes either into the atmosphere or into a condenser. The purpose of the receiver will be explained later.

CONSTRUCTION OF COMPOUND ENGINES.

CLASSIFICATION.

15. Types.—Compound engines may be divided into two general types, which are the *Woolf compound type* and the *receiver compound type*.

16. The Woolf Compound.—In the Woolf compound type the pistons of each cylinder commence and complete the stroke at the same time, and, consequently, the high-pressure cylinder can exhaust directly into the low-pressure cylinder. In Woolf compound engines either both pistons operate directly upon one crank, as in a tandem compound engine, or they operate upon two or more cranks on the same shaft, which are either in line or 180° apart. The absence of a receiver, or vessel destined to receive the high-pressure exhaust before it passes to the low-pressure cylinder, is the distinguishing feature of the Woolf compound engine. While most tandem compound engines are of the Woolf compound type, it must not be inferred that all of them belong to that type. A tandem compound engine does not need a receiver, as far as the steam distribution is concerned, but some of them, especially in very large sizes, are fitted with reheating receivers, where an additional amount of heat is supplied to the high-pressure exhaust in order to reduce the cylinder condensation in the low-pressure cylinder.

17. The Receiver Compound.—In the receiver compound type the high-pressure cylinder exhausts into a separate vessel, chiefly in order that the cranks may be placed at any desired angle other than 0° (in line with each other) or 180° apart. This is not feasible without the employment of a receiver, which, however, need not be a separate vessel, but may take the form of a very large exhaust pipe from the high-pressure to the low-pressure cylinder, or which may be formed by an exceptionally large low-pressure steam chest.

18. The reason why a receiver is needed when the cranks are at any other angle than 0° or 180° with each other may be explained as follows: Assume a cross-compound engine with cranks 90° apart and cutting off at ½ stroke in both cylinders. Let the high-pressure piston just have commenced its stroke. Then, the low-pressure piston will be just past its mid-stroke position and the low-pressure steam port is closed. But while the high-pressure piston is taking live steam on one side, it is exhausting on the other side, and there being no space for all the high-pressure exhaust steam to go into, the exhaust steam will be compressed to an extent depending on the volume of the low-pressure steam chest and the passage leading to it. This compression represents a waste of work, which is avoided by the use of a receiver.

19. Definitions.—The number of stages in which the expansion of the steam is carried on is denoted by calling the engine a *compound*, *triple-expansion*, *quadruple-expansion*, or *quintuple-expansion engine*. In a **compound engine** the steam is expanded in two separate stages, but not necessarily in two cylinders; in a **triple-expansion engine** the steam is expanded in three separate stages, but not necessarily in three cylinders, etc.

The expressions *compound*, *triple-expansion*, *quadruple-expansion*, etc. must not be taken to infer that the steam is expanded to twice, three times, four times, etc. its original volume, or that the engine has two, three, four, five, etc. cylinders. They refer solely to the number of different stages in which the steam is expanded. Thus, compound engines have been built with three cylinders and three cranks, there being one high-pressure cylinder and two low-pressure cylinders. In such an engine one-half of the exhaust steam from the high-pressure cylinder passes to each low-pressure cylinder. Such an engine is distinctly a *compound* engine, and may be more fully denoted as a *three-cylinder*, *three-crank*, *compound engine*.

20. All engines in which the expansion of the steam takes place in more than one stage are called **multiple-expansion engines,** to distinguish them from the single-cylinder or **simple expansive engine.** In giving the size of a multiple-expansion engine it is customary to give the sizes of their cylinders in inches, commencing with the smallest cylinder, and to write the stroke in inches last. Thus, a compound engine whose high-pressure cylinder is 11 inches in diameter, low-pressure cylinder 20 inches in diameter, and stroke 15 inches, would be designated as a 11″ × 20″ × 15″ compound engine. In the same manner, the designation a 14″ × 22″ × 34″ × 18″ triple-expansion engine would mean that the diameters of the cylinders were 14 inches, 22 inches, and 34 inches, and that they had a common stroke of 18 inches.

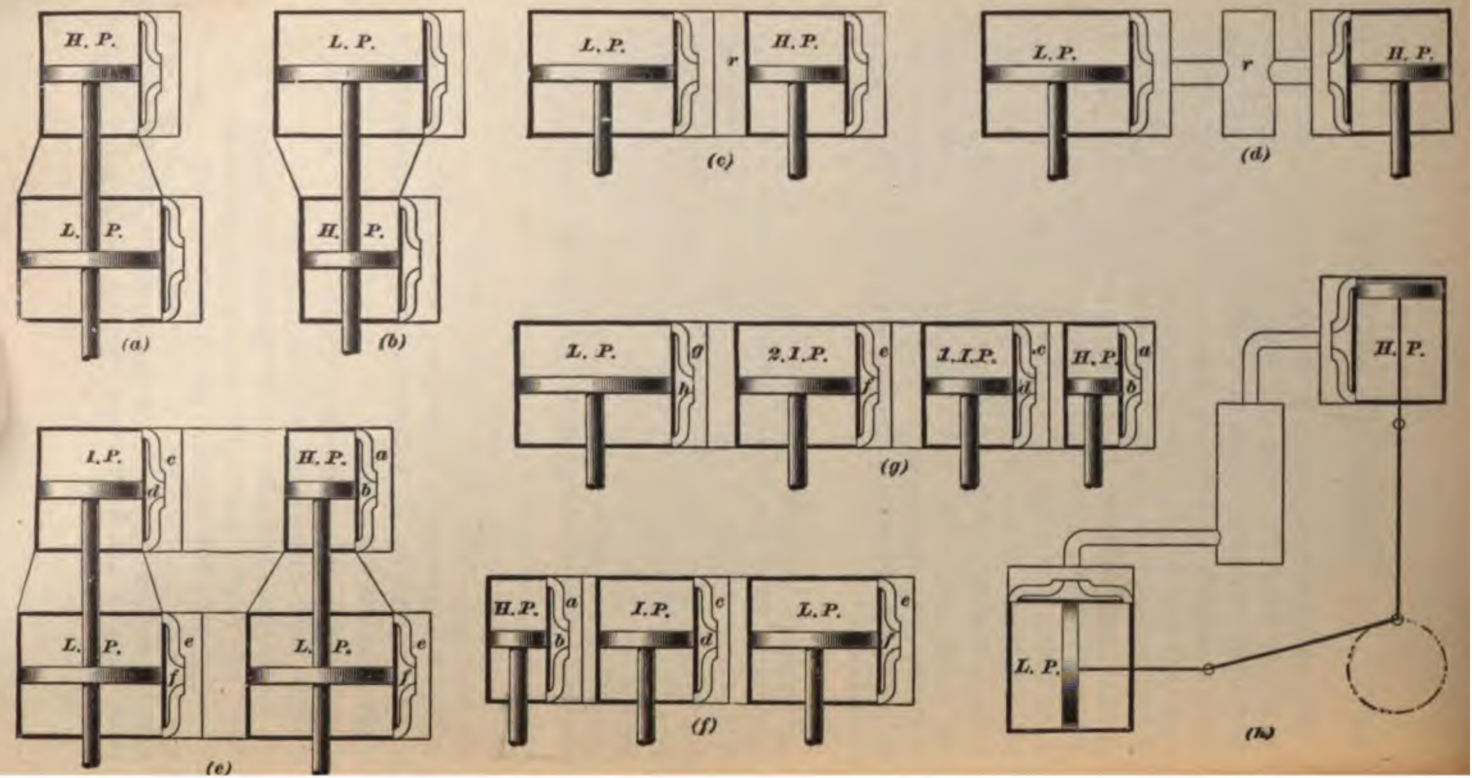
21. In a triple-expansion engine there is at least one cylinder between the high-pressure and low-pressure cylinders. This cylinder is called the **intermediate-pressure cylinder.** In a quadruple-expansion engine there are at least two cylinders through which the steam must pass after leaving the high-pressure cylinder and before reaching the low-pressure cylinder. These are called the **first** and **second intermediate-pressure cylinders,** respectively.

CYLINDER ARRANGEMENTS.

22. It is customary to designate the cylinders and other component parts of compound and multiple-expansion engines as follows:

H. P. = high-pressure cylinder;
I. P. = intermediate-pressure cylinder;
1 I. P. = first intermediate-pressure cylinder;
2 I. P. = second intermediate-pressure cylinder;
L. P. = low-pressure cylinder.

In the description of the various cylinder arrangements, these designations have been adopted, and in Fig. 4 the


H. P.
L. P.
(a)
L. P.
H. P.
(b)
L. P.
r
H. P.
(c)
L. P.
r
H. P.
(d)
I. P.
c
d
H. P.
a
b
L. P.
e
f
L. P.
e
f
(e)
H. P.
a
b
I. P.
c
d
L. P.
e
f
(f)
L. P.
g
h
2. I. P.
e
f
1. I. P.
c
d
H. P.
a
b
(g)
H. P.
L. P.
(h)

cylinders have been lettered to correspond. In English books, *M. P.* designates the intermediate-pressure cylinder. The most common forms of compound, triple-expansion, and quadruple-expansion engines are illustrated in diagrammatic form in Fig. 4. The most common form of tandem compound engine is shown in Fig. 4 (*a*). Here the *L. P.* cylinder is next to the crank-shaft. In Fig. 4 (*b*) the low-pressure cylinder is furthest from the crank-shaft. Tandem compound engines of either design may be vertical or horizontal; when vertical they are often called **steeple compound engines.**

Fig. 4 (*c*) shows the cylinder arrangement of a cross-compound, high-speed engine. In these engines there are generally two flywheels placed outside of the cranks, the latter being at right angles with each other. The cylinders being close together, the receiver *r* is generally in the form of a jacket that ties the cylinders together. This arrangement of cylinders greatly lessens the floor space required. Fig. 4 (*d*) shows the cylinder arrangement of cross-compound engines of large power. It will be noticed that this is the same as shown in Fig. 3. Here the cylinders are separated in order to allow the flywheel to be placed between the cranks. The receiver *r* is generally a separate vessel. Cross-compounds may be either vertical or horizontal.

Fig. 4 (*e*) shows the cylinder arrangement of a two-crank, four-cylinder, triple-expansion engine. Steam enters the *H. P.* cylinder from the steam chest *a*; it exhausts through *b* into the intermediate receiver *c*, whence it passes into the *I. P.* cylinder. The latter exhausts through *d* into the low-pressure receiver *e e*, common to both *L. P.* cylinders. Through *f f* the exhaust steam from the *L. P.* cylinders passes into the condenser. The cranks are placed 90° apart. For large powers the cylinders may be separated, similar to the cross-compound engine shown in Fig. 4 (*d*). In that case the low-pressure receiver would most likely be a separate vessel. Fig. 4 (*f*) shows the cylinder arrangement of a three-crank, three-cylinder, triple-expansion engine with the cylinders placed close together. In this case the flywheel is placed on one end of the crank-shaft. The cranks are usually placed 120° apart. The steam from the boiler enters the *H. P.* steam chest *a* and is exhausted from the *H. P.* cylinder through *b* into the intermediate receiver and passes thence into the *I. P.* steam chest *c*. The steam exhausts from the *I. P.* cylinder through *d* into the low-pressure receiver, whence it passes into the *L. P.* steam chest *e*, and finally exhausts through *f* into the atmosphere or condenser.

Fig. 4 (*g*) illustrates the cylinder arrangement of a four-crank, four-cylinder, quadruple-expansion engine. Steam enters the *H. P.* cylinder from the steam chest *a*; it exhausts through *b* into the first intermediate receiver *c*, whence it passes into the *1 I. P.* cylinder. The exhaust from the latter passes through *d* into the second intermediate receiver *e*, and thence into the *2 I. P.* cylinder. This cylinder exhausts through *f* into the low-pressure receiver *g*, whence the steam passes into the *L. P.* cylinder and exhausts through *h* into the condenser. The cranks are placed 90° apart.

Fig. 4 (*h*) shows a cylinder arrangement for a compound engine that is beginning to find much favor with designers of late for large powers. The two cylinders are placed at right angles to each other, and both pistons are connected to the same crankpin. Sometimes two sets of such engines are placed alongside of each other, with the flywheel between them. A compound engine arranged in this manner will occupy much less floor space than a cross-compound engine, and will run just as steadily as the latter, owing to the uniformity of the turning moment on the crank-shaft.

EXAMPLES OF COMPOUND ENGINES.

23. Tandem Compound.—An elevation of a tandem compound, high-speed engine is shown in Fig. 5. In the illustration, *a* is the high-pressure cylinder, which in high-speed tandem compound engines is generally placed behind the low-pressure cylinder *b*. In large medium-speed engines the high-pressure cylinder is often placed nearest the crank-shaft, it being claimed that it is then easier to remove the pistons and to examine the cylinders in case of repairs. When the high-pressure cylinder is behind, as in Fig. 5, it must generally be removed entirely in order to get at the

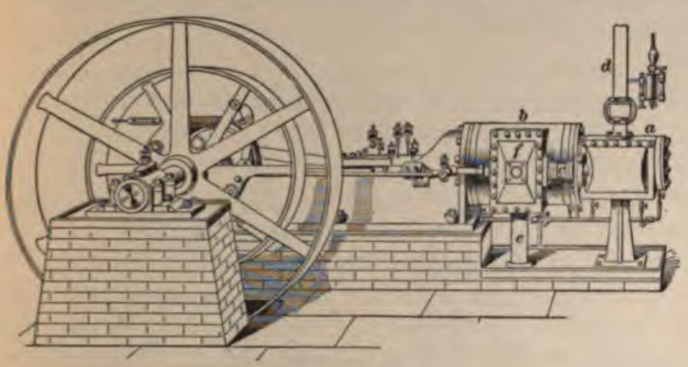


FIG. 5.

inside of the low-pressure cylinder. Steam is conducted to the high-pressure steam chest by the steam pipe *d*; after the steam has expanded in *a* it is discharged through the

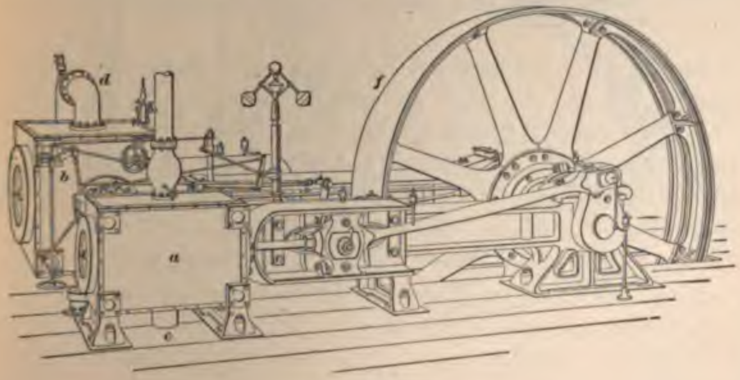


FIG. 6.

connecting pipe *c* into the steam chest *f* of the low-pressure cylinder *b*, and is finally exhausted into the condenser or atmosphere through the exhaust pipe *e*. As in nearly all high-speed engines, a shaft governor is used. The engine is of the side-crank type; that is, the crank is on the end of the crank-shaft. Many tandem compound engines are built, however, with a center crank, which means that the crank is in the center of the crank-shaft. In that case there are usually two flywheels, one on each side.

24. Horizontal Cross-Compound.—A perspective view of a cross-compound, horizontal engine, with Corliss valve

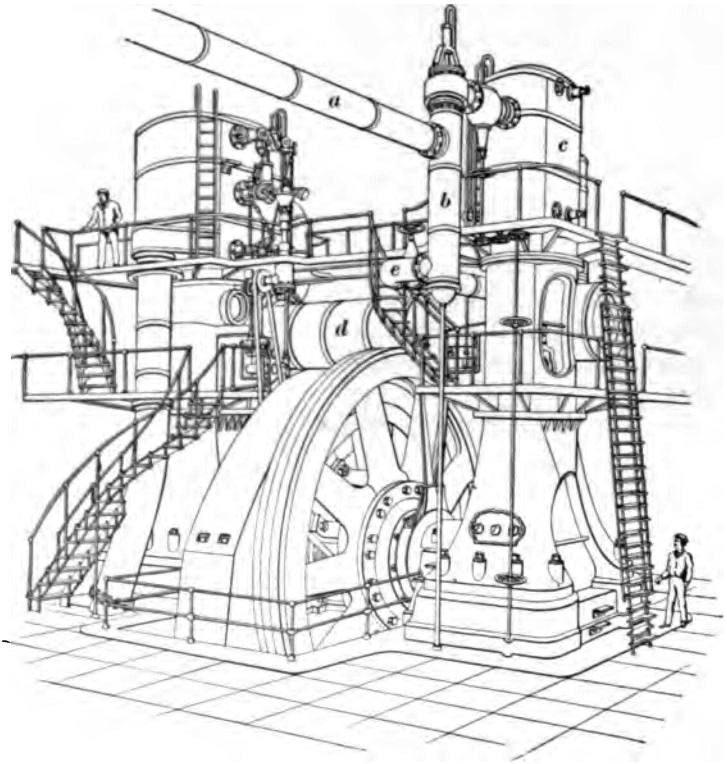


FIG. 7.

gear, is given in Fig. 6. In the illustration, *a* is the high-pressure cylinder and *b* the low-pressure cylinder. The receiver between the two cylinders is placed beneath the floor, the pipe *c* leading to it from the high-pressure cylinder and the pipe *d* leading from it to the low-pressure steam chest. The cranks are placed 90° apart; owing to the fact that the high-pressure crank *e* is on its upper quarter, the low-pressure crank is hidden by the frame. The flywheel *f* serves as a belt wheel and is placed between the cranks.

25. Vertical Cross-Compound.—Fig. 7 shows a perspective view of a vertical cross-compound engine direct-connected to a dynamo and having Corliss valve gear. The steam is led to the engine through the main steam pipe *a*, and before passing into the high-pressure steam chest passes into a separator *b*, which removes the entrained water. The exhaust steam from the high-pressure cylinder *c* passes into the reheating receiver *d*, where the exhaust steam is superheated by live steam taken from the bottom of the separator through the pipe *e*. The dynamo is placed between the bearings, its armature being keyed directly to the crank-shaft.

The particular engine shown is about 4,500 horsepower; it has cylinders 46 and 86 inches diameter, a stroke of 60 inches, and a speed of 75 revolutions per minute.

26. Duplex Vertical and Horizontal Compound.—Fig. 8 is a perspective view of an 8,000 horsepower duplex compound engine with the two cylinders of each engine at right angles to each other. The two engines are duplicates of each other and are so arranged that either engine can be uncoupled. As shown, the pistons of the high-pressure cylinders *a*, *a* and low-pressure cylinders *b*, *b* of each engine are connected to a common crankpin, as *c*. The cranks of this engine are placed 45° apart, so that the crank-shaft receives eight impulses during each revolution, which gives such a uniform turning effect that the flywheel is dispensed with, its place, to some degree, being taken by the revolving field *d* of the dynamo. The steam coming from the high-pressure cylinders passes through reheating receivers *e*, *e*. The engine has high-pressure cylinders 44 inches in diameter, the low-pressure cylinders are 88 inches in diameter, the stroke is 60 inches, and the speed is 75 revolutions per

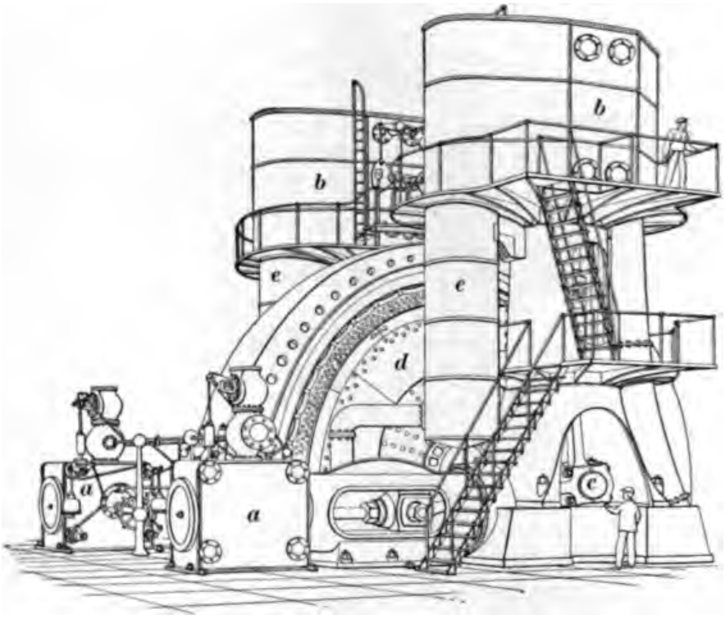


FIG. 8.

minute. The engine can be worked up to 12,000 horse-power.

27. Triangular Connecting-Rod Engine.—Fig. 9 is an end view of one side of a quadruple-expansion, two-crank engine having the so-called **triangular connecting-rod** patented by John Musgrave & Sons, Bolton, England. The engine is of the vertical type and is fitted with Corliss valve gear. The four cylinders are disposed in pairs on each side of the flywheel or rope drum, the high-pressure and first intermediate cylinders being on one side of the drum and the second intermediate and low-pressure cylinders are on the other side. The crossheads of each pair of cylinders are connected by means of a pair of links *a*, *a* and a triangular connecting-rod *b* to a single crank *c*; the two cranks of the engine are opposite each other, so that the weights of the two sets of reciprocating parts mutually balance each other.

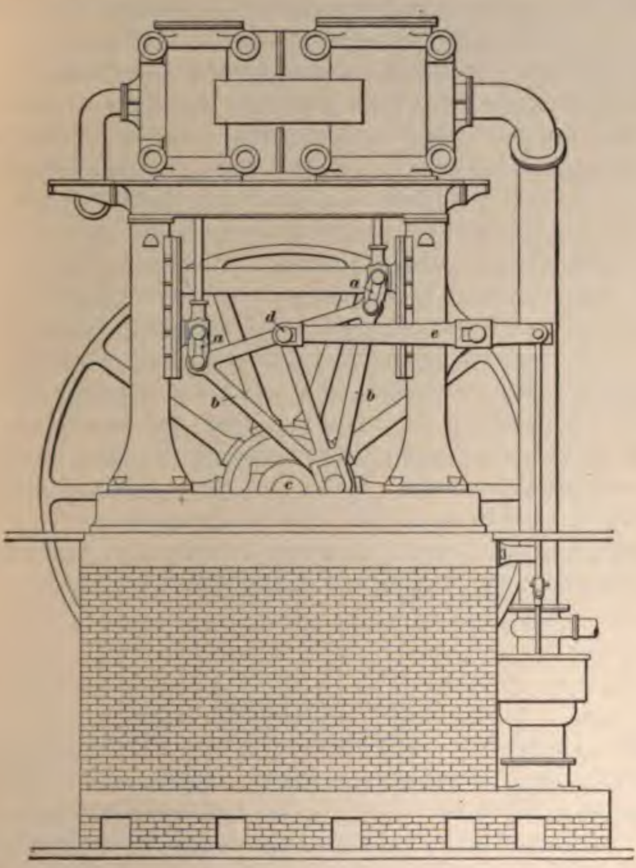


FIG. 9.

The action of the triangular connecting-rod is such that there are no dead centers to the engine, as although the crossheads of both cylinders are connected to one crank, they are never at the ends of their respective strokes at the same time, and the turning effort is the same as if they were connected to cranks set nearly at right angles to each other; that is, when one piston is at the end of its stroke the other is nearly in its central position and has a very effective leverage to turn the crank. But a further and very important effect of this arrangement is that the strains on the crank are gradually changed *around* the crankpin from one side to the other; they are never suddenly reversed, as is the case with an ordinary engine, and in consequence these engines may be run at very high speeds without any noticeable jar or vibration. The triangular connecting-rod mentioned vibrates on a pin *d* in the ends of a pair of levers, as *c*, which latter swing on a fixed center just outside of the right-hand frame. Extensions of these levers outside of the frames are made use of to work the air pump. The motion of the ends of the triangular connecting-rod, to which the crossheads are connected, is a vibrating one, due both to the arc formed by the swinging lever and to the circular path of the crankpin. In consequence of this combined movement, these ends of the rod move up and down in nearly vertical lines, so that the side pressure on the guides is very small, in fact it is much less than in an ordinary engine with a connecting-rod of the usual length.

THE RECEIVER.

28. In practice the receiver is often given such dimensions that it has a volume equal to that of the low-pressure cylinder. When its volume is that large, the pressure within it varies but little, so long as the weight of steam supplied by the high-pressure cylinder in one revolution of the engine is equal to that drawn out by the low-pressure cylinder. Assuming that the pressure in the receiver is constant and neglecting the resistance of the passages connected with the cylinders, the back pressure in the high-pressure cylinder will be the same as the initial pressure in the low-pressure cylinder. In each cylinder, then, the steam is admitted, cut off, and expanded, just as if there were two independent engines, each of which uses the same weight of steam per revolution.

29. The pressure existing in the receiver can and does exert a marked influence on the engine, both in regard to its operation as a machine and as a heat engine. The receiver being a vessel having a constant volume, the pressure within it depends on two things which are the rate at which steam is discharged into it and the rate at which it is drawn out. When the volume of steam drawn out is greater than the volume of steam in the high-pressure cylinder at the moment release begins in that cylinder, it will be apparent that the exhaust steam rushing into the receiver must expand to the volume drawn from the receiver. In expanding, the pressure of the exhaust steam drops, so that the pressure in the receiver, and of course the back pressure on the high-pressure piston, is less than the pressure at the time of release. This difference in pressure at the time of release and in the receiver is termed **drop.**

30. When saturated steam is allowed to expand without doing any work, as is the case when the receiver pressure is less than the pressure at the point of release, the steam becomes superheated to an extent depending on the drop. The effect of this superheating is to partially or entirely evaporate any water in the steam, thus making it drier and tending to reduce the initial condensation in the low-pressure cylinder. It may be incidentally remarked here that the expanding of steam without doing any work is termed **free expansion.**

31. The most economical receiver pressure for a given engine can only be found by experiment, regulating the pressure and noting the effect upon the steam consumption. The receiver pressure is regulated in several ways. First, by making the cut-off later in the high-pressure cylinder and leaving the low-pressure cut-off as before. Since there is now a larger volume of steam entering the receiver, the receiver pressure will be higher. The initial pressure will also be higher in the low-pressure cylinder, and the cut-off being the same as before, the mean effective pressure will be greater. Since the mean effective pressure in the high-pressure cylinder is also greater, it is seen that the effects of a later cut-off in the high-pressure cylinder is an increase of the power of the engine.

32. Making the cut-off earlier in the high-pressure cylinder means a lower pressure at release and a lower receiver pressure; consequently, the mean effective pressures in the two cylinders, and hence the power of the engine, are decreased by this means.

33. The amount of drop can only be regulated by means of the low-pressure cut-off. Thus, leaving the cut-off of the high-pressure cylinder as before, the pressure at release will be the same. But if the cut-off is made later in the low-pressure cylinder, a larger volume of steam will be drawn from the receiver, and, consequently, the receiver pressure will be less and the drop more. Conversely, if the low-pressure cut-off is made earlier, there will be less steam drawn from the receiver, and hence the receiver pressure will be higher and the drop less.

GOVERNING A COMPOUND ENGINE.

34. Obviously a change in the receiver pressure due to a manipulation of the low-pressure cut-off will change materially the amount of work done in each cylinder, although it will leave the total work done by the two cylinders the same as before. Thus, if the receiver pressure is lowered by making the low-pressure cut-off later, the back pressure obviously will be less in the high-pressure cylinder, and hence the mean effective pressure will be more, so that the high-pressure cylinder will now be doing more work. Since the total amount of work done remains the same as before, it follows that the low-pressure cylinder, in spite of its later cut-off, will now be doing less work. Conversely, if the receiver pressure is raised by making the low-pressure cut-off earlier, less work will be done in the high-pressure cylinder and more in the low-pressure cylinder.

35. From the foregoing statements it will be plain that in order to change the power of the engine, the high-pressure cut-off must be changed, while in order to change the distribution of work in the two cylinders, the low-pressure cut-off must be changed. This will explain why the governing mechanism of many compound engines operates upon the high-pressure cut-off, which manner of governing is open to one serious objection, however. When the governor makes the high-pressure cut-off later in responding to an increase of load, the larger share of the total work will be done in the low-pressure cylinder; conversely, if the governor makes the cut-off in the high-pressure cylinder earlier, the larger share of the work will be done in the high-pressure cylinder, and at very early cut-offs the low-pressure cylinder may then actually be a drag on the engine.

36. In order to overcome the objectionable features of controlling only the high-pressure cut-off, many compound engines have the governor operate both the high-pressure and the low-pressure cut-offs. In that case the governor operates the high-pressure cut-off in order to change the power of the engine and the low-pressure cut-off in order to equalize the work done in the two cylinders. That is, when the governor makes the cut-off later in the high-pressure cylinder, it also makes it later in the low-pressure cylinder; conversely, when the load is lighter, the governor makes the cut-off earlier in both cylinders.

RECEIVER FITTINGS.

37. Receivers should always be fitted with gauges registering from zero to the maximum pressure in one direction and from zero to 30 inches of vacuum in the other direction, i. e., with compound gauges. The receiver should also be fitted with pop safety valves of large capacity, so that in case the low-pressure cut-off should be set too early, or, if a releasing gear is used, if the gear should fail to open the low-pressure admission valves, the steam would be relieved. Otherwise, the high-pressure cylinder, aided by the momentum of the flywheel, would pump the receiver full of steam and cause it to explode.

SETTING THE VALVES OF MULTIPLE-EXPANSION ENGINES.

38. In setting the valves of horizontal, compound, and triple-expansion engines, considerable more lead should be given in the low-pressure cylinder. The lead for a 48-inch, low-pressure, condensing cylinder could be from $\frac{1}{4}$ to $\frac{5}{16}$ inch, and if the back pressure is very low the lead may be somewhat larger. The valves of each cylinder of a compound and triple-expansion engine must be set in relation to the crank on which that cylinder is doing its work. For vertical engines it is customary to give a little more lead on the bottom than on the top, for the reason that the force required to accelerate the reciprocating parts is greater on the up stroke than on the down stroke, due to the weight of the reciprocating parts. In many of the valve gears now in use, the action of the valve is so quick at the instant of opening that many engineers now measure the lead not in the amount that the valve is open when the engine is on the center, but by the position of the crank when the valve begins to open. The practice with some engineers is to have the valve begin to open when the crank is yet 15° from the dead center. This angle, however, may be regarded as a limit rather than as an average. Perhaps 7° to 10° would be the average.

EQUALIZING WORK DONE IN THE CYLINDERS.

39. The question of whether or not it is necessary that both cylinders of a compound engine should do an exactly equal amount of work is at present in a somewhat unsettled state as far as cross-compound engines are concerned. From the builder's standpoint it is more desirable to have the initial loads on the piston rods, crossheads, connecting-rods, and crankpins equal in order that these parts may be duplicates and yet of equal strength. The initial load is given by multiplying the area of the piston by the initial steam pressure. Obviously, it is independent of the cut-off, that is, the initial load is the same whether the cut-off is at $\frac{1}{10}$ or $\frac{7}{10}$ of the stroke, so that it can be readily seen that there is no relation between the initial load and the work done in the cylinder. Consequently, it is entirely possible with an equal division of work to have the maximum stresses on the parts of one side of the engine far in excess of those on the parts of the other side. For this reason it is urged by many engineers that in cross-compound engines, where the parts previously named are in duplicate, to so distribute the work that the initial loads will be approximately equal.

40. Opinions differ as to whether a cross-compound engine will run more satisfactory when the work is equally divided, and a decision one way or the other is best based on an actual trial. The mechanical operation of the engine is here referred to, not its economy.

41. In tandem compounds the work is generally evenly distributed between the cylinders, but even quite a perceptible variation seems to have no harmful effect, as far as the quiet running of the engine is concerned.

42. In practice it is probably the best plan with a given engine, when the design permits it, to distribute the work between the cylinders so as to obtain the smallest possible water consumption compatible with steady and satisfactory running. This object in any case can only be attained by repeated trials, changing the receiver pressures in order to change the work done in each cylinder and noting the water consumption after each change. In other words, it is the best plan to try to secure the highest economy obtainable in conjunction with a satisfactory mechanical operation.

43. In many compound engines, especially in engines of the high-speed type, the distribution of work in the two cylinders has been determined upon by the builder and cannot be changed by any means at the command of the attendant. The valves of both cylinders are generally under the control of the governor, and the angle of advance then being a fixed quantity, which cannot readily be changed, it follows that all the attendant can do is to keep the valves centrally set and the engine in first-class running condition. There is little or no chance to improve the economy, i. e., to lower the steam consumption, at least, as far as the engine itself is concerned.

THE STEAM JACKET.

44. When the cylinder of an engine is surrounded with a casing and the space between the cylinder and casing is filled with live steam during running, the cylinder is said to be **steam-jacketed.** In comparatively slow-speed engines, say up to 150 revolutions per minute, using saturated steam, a properly applied and properly cared for steam jacket will generally produce a distinct gain in economy, while one that is improperly applied or cared for will prove a loss and sometimes a very serious one. If possible it is always better to use steam of a higher temperature in the jacket than that which enters the cylinder. In a compound engine a jacket may be applied to the high-pressure cylinder and the receiver, and in a triple-expansion engine to the first and second cylinders and both receivers, but in neither case need it be applied to the low-pressure cylinder. The object of the steam jacket is to prevent condensation taking place in the cylinder; if the steam that runs the engine can be superheated sufficiently to prevent initial condensation, there will then be no need of a steam jacket, and the engine will be more economical without a steam jacket than with one.

45. In order to gain a clear understanding of why a steam jacket under proper conditions will improve the economy, the action of the steam that actually takes place in a steam cylinder after the engine is fairly started and when using ordinary saturated steam should be understood. This action in an unjacketed cylinder is as follows:

Steam enters at the beginning of the stroke, and as it is hotter than the metal with which it comes in contact, a portion of the entering steam is condensed in heating up the cylinder head, piston, and cylinder to the temperature of the steam. As the piston moves forwards, uncovering fresh surfaces of the cylinder, the condensation continues until after the admission of steam ceases—i. e., until the point of cut-off is reached. The steam then commences to expand and correspondingly lowers in pressure and temperature, and unless the steam is cut off very early in the stroke, no further condensation occurs after the cut-off has taken place (although for some distance further the piston uncovers fresh surfaces of the cylinder that are cooler than the steam) for the reason that, as soon as the temperature of the steam falls through expansion, the head, piston, and walls of the cylinder already heated begin to give up their heat to the steam and prevent further condensation taking place. As the expansion is carried still farther and the temperature of the steam correspondingly lowered, a portion of the water previously condensed is reevaporated into steam by the heat given up by the piston, cylinder, and head, raising the terminal pressure in the cylinder. When the piston arrives at the end of the stroke, the exhaust valve opens and the pressure and temperature of the steam lowers immediately, the remainder of the water previously condensed then being reevaporated by the heat extracted from the cylinder walls, piston, and head, and passes out with the exhaust steam. The exhaust steam is much colder than the metal, and during the whole of the return stroke is absorbing heat from the piston, cylinder walls, and head, lowering their temperature, which has to be raised again by the steam entering for the next forward stroke. The action of the heat above described takes place very quickly and affects the metal of the cylinder and head to a slight depth only, as there is not time for it to penetrate very deeply; but it affects it at each stroke of the engine, and a small loss each stroke aggregates a very large one during the day's run. This action is also greatly increased by water held in suspension or entrained in the steam; it is therefore important that dry steam only should be used.

46. The action that takes place in a steam-jacketed cylinder is quite different, as the following description will indicate: Suppose the jackets to contain steam at the same temperature as the steam entering the cylinder—this being the usual case with the high-pressure cylinder of a compound or triple-expansion engine—and also suppose the jackets to be so arranged that neither air nor water can collect and interfere with their usefulness. At the beginning of the stroke the cylinder and head will be at the same temperature as the entering steam, and none of the entering steam will be condensed by them, as in the case of an unjacketed cylinder; the piston itself will be in the same condition as before, and a portion of the steam will be condensed and reevaporated by it. After the admission of steam ceases and the temperature of the steam inside the cylinders becomes less through expansion, heat is transferred from the cylinder walls to this steam, reevaporating the small amount of water condensed by the piston. The walls, in turn, absorb heat from the steam in the jacket, condensing some steam in the jacket, but maintaining the cylinder at nearly an even temperature. When the exhaust valve opens and the pressure and temperature of the steam in the cylinder become lower, there is no water to reevaporate; but as the steam inside the cylinder is then much cooler than the walls of the cylinder, it absorbs heat from them during the whole of the return stroke, which heat, in turn, is absorbed from the steam in the jacket, and this heat is practically wasted. The actual gain effected by the use of a steam jacket on a cylinder is the difference between the saving due to the prevention of condensation in the cylinder at the beginning of the stroke and the loss by heating the exhaust steam during the return stroke; and this gain may in many cases be very slight.

REHEATERS.

47. A **reheater** is a modification of the receiver commonly used in multiple-expansion engines and consists essentially of a receiver containing coils or nests of small pipes, through which high-pressure steam circulates and which are so arranged that the working steam must circulate around them thoroughly and become **reheated,** from which fact the apparatus receives its name. Formerly, receivers were simply reservoirs placed between the cylinders. The next step was to provide these receivers with steam jackets, and from noticing the economy obtained from the steam-jacketed receiver, the reheater was developed. It is claimed by advocates of the reheater that a gain of 10 per cent. can be realized by their use under favorable conditions. This, however, depends entirely on circumstances, and a reheater, instead of causing a gain, may cause a positive loss. Furthermore, it can seldom or never be predicted beforehand whether a reheater will be a gain or a loss. However, it can always be determined afterwards whether the action of the reheater is profitable. If a thermometer placed in the low-pressure exhaust shows that the temperature of the exhaust steam is higher than that corresponding to its pressure, the reheater is wasting heat, and the reheating should be abandoned. In proportioning the reheating surface, 40 square feet of surface are generally allowed for every cubic foot of steam swept into the reheater by the high-pressure piston during exhaust.

48. Fig. 10 shows a common form of a reheater. It consists of a cast-iron or wrought-iron shell *a*, having at one end a tube plate *b* secured between the shell *a* and cover *c*. A large number of wrought-iron boiler tubes, as *d*, *d*, are expanded into this tube plate. These boiler tubes are expanded at the other end into the tube plate *e*, which is bolted to the head *f*. It will be noticed that the head *e* and tube plate *f* form a vessel that is independent of the shell of the receiver, so that the tubes are perfectly free to expand or contract. The right-hand end of the reheater is closed by the cover *g* through which the steam pipe *h* and air drain pipe *i* pass. Stuffingboxes are used to make a steam-tight joint where these pipes pass through the head. An automatic air valve is generally attached to the pipe *i*, through which the air is discharged when starting up and which closes as soon as steam reaches it. The pipe *h* serves as a drain and is led to a trap. The drain for the shell is attached at *k* and is also led to a trap.

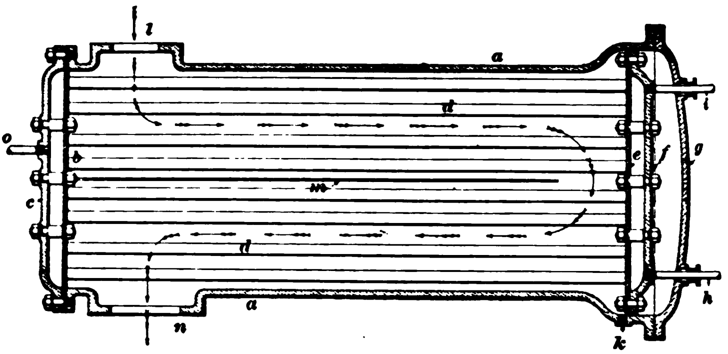


FIG. 10.

The exhaust from the high-pressure cylinder enters at *l* and is compelled by the baffle plate *m* to flow in the path indicated by the arrows over the tubes *d*, *d*, and finally passes through the opening *n* to the low-pressure steam chest. Live steam at boiler pressure is admitted to the tubes through the pipe *o* and leaves through *h*.

49. For very large engines reheaters are often constructed similar to tubular boilers, the tubes being expanded into rigid tube-sheets riveted to the shell. The working steam, then, generally passes through the tubes, while the steam used for reheating circulates outside of them. With such a reheater it is a good plan to take the live steam for the reheater from a connection placed between the throttle and the engine. If this is done, steam will be admitted to both sides of the reheater as soon as the engine is started, and, in consequence, its expansion will be more uniform, which renders it less liable to leakage around the tube ends.

50. A reheating receiver should, in general, never be used when the engine uses superheated steam, for the reason that the steam entering the receiver usually contains enough superheat to prevent initial condensation, and the use of a reheating receiver under these conditions will result in a direct waste of heat.

CALCULATIONS PERTAINING TO COMPOUND ENGINES.

RATIO OF EXPANSION.

RATIO OF EXPANSION BY VOLUME.

51. The ratio of expansion of a multiple-expansion engine is generally given as the ratio between the volume of steam admitted to the high-pressure cylinder and the volume of steam in the low-pressure cylinder at low-pressure release, that is, at the time the low-pressure exhaust valve opens. The ratio of expansion is independent of the low-pressure cut-off or volume of the receiver.

52. The ratio of expansion of a multiple-expansion engine may be found in two different ways. In the formulas corresponding to the following rules:

E = total ratio of expansion;
v = volume of high-pressure cylinder up to the point of cut-off, including clearance;
V = volume of low-pressure cylinder at release, including the clearance;
e = ratio of expansion in the high-pressure cylinder;
v_1 = volume swept through by high-pressure piston, clearance included.

Rule 1.—*To find the total ratio of expansion, divide the volume of the low-pressure cylinder up to the point of release and including clearance by the volume of the high-pressure cylinder up to the point of cut-off and including clearance.*

Or, $$E = \frac{V}{v}.$$

Rule 2.—*To find the total ratio of expansion, multiply the ratio of expansion of the high-pressure cylinder by the volume of the low-pressure cylinder up to the point of release and divide by the volume swept through by the high-pressure piston, clearance included.*

Or, $$E = \frac{e\,V}{v_1}.$$

When applying rules **1** and **2** to a triple-expansion or quadruple-expansion engine, the intermediate cylinders are to be neglected entirely; that is, consider only the first and last cylinders. The ratio of expansion calculated by rules **1** and **2** is known as the **ratio of expansion by volume**, and when the expression "ratio of expansion" is used without any particular qualification, it is always understood to mean *by volume*.

EXAMPLE 1.—The volume of the high-pressure cylinder of a triple-expansion engine, up to the point of cut-off, is 650 cubic inches; the volume of the low-pressure cylinder is 13,650 cubic inches. What is the total ratio of expansion?

SOLUTION.—Applying rule **1**, we have

$$E = \tfrac{13650}{650} = 21. \text{ Ans.}$$

EXAMPLE 2.—The volume of the low-pressure cylinder of a compound engine is 16,548 cubic inches. The volume swept through by the high-pressure piston, including clearance, is 4,234 cubic inches. The ratio of expansion of the high-pressure cylinder being 2.6, what is the total ratio of expansion?

SOLUTION.—Applying rule **2**, we have

$$E = \frac{2.6 \times 16,548}{4,234} = 10.16. \text{ Ans.}$$

RATIO OF EXPANSION BY PRESSURE.

53. The ratio of expansion of a multiple-expansion engine is sometimes given as the ratio between the absolute pressure at the point of cut-off in the high-pressure cylinder and the absolute pressure in the low-pressure cylinder at the point of release. The ratio of expansion by pressure is, for the same engine, always greater than the ratio of expansion by volume, owing to drop in the receiver. The absolute pressures are taken by measurement from indicator diagrams.

HORSEPOWER OF COMPOUND ENGINES.

CALCULATING THE INDICATED HORSEPOWER.

54. The indicated horsepower of a compound or triple-expansion engine is calculated from the indicator diagrams in exactly the same manner as with any simple engine, considering each cylinder as a simple engine and adding the horsepowers of the engines together. In taking the indicator cards from a compound engine, the precaution of taking the cards simultaneously from all cylinders must be observed, especially when the engine runs under a variable load, since otherwise an entirely wrong distribution of power may be shown, and there may also be a great variation between the indicated horsepower really existing and that calculated from diagrams taken at different times.

55. The indicated horsepower of compound engines is sometimes found by referring the mean effective pressure of the high-pressure cylinder to the low-pressure cylinder and calculating the horsepower of the engine on the assumption that all the work is done in the low-pressure cylinder. To do this, the mean effective pressures of the two cylinders are found from indicator diagrams; the mean effective pressure of the high-pressure cylinder is then divided by the ratio of the volume of the low-pressure cylinder to that of the high-pressure cylinder, and the quotient is added to the mean effective pressure of the low-pressure cylinder. The sum is then taken as the mean effective pressure of the engine, and the area of the low-pressure piston as the piston area; with these data, the length of stroke and the number of strokes, the horsepower is computed as for any simple engine. In the case of a triple-expansion engine, the mean effective pressures of the high-pressure and intermediate cylinders are referred to the low-pressure cylinder and added to its mean effective pressure. While this method shortens the labor of computing the horsepower, it obviously does not show the distribution of work between the cylinders, and for this reason is going rapidly out of use.

ESTIMATING THE INDICATED HORSEPOWER.

56. The horsepower of a multiple-expansion engine cannot be estimated with any great degree of accuracy, and in practice it would be considered a satisfactory result if the indicated horsepower and estimated horsepower agree within 10 per cent.

57. The difficulty of determining the horsepower is due to the fact that the mean effective pressure cannot be estimated very closely, and for this reason it is better to take the mean effective pressure from diagrams whenever this is feasible.

58. The horsepower of a multiple-expansion engine is estimated by estimating the horsepower of a simple expansive engine having the same dimensions as the low-pressure cylinder, the same piston speed, the same initial pressure, and the same total ratio of expansion. The problem then resolves itself into finding the probable mean effective pressure of this equivalent simple expansive engine.

59. To estimate the probable mean effective pressure, it is necessary to assume an initial and a terminal pressure in order to determine the probable ratio of expansion by pressure. With a steam pipe of ample size and the throttle wide open, the initial pressure in the high-pressure cylinder will probably average 5 pounds per square inch below the boiler pressure. The terminal pressure, i. e., the pressure at the point of release in the low-pressure cylinder, may be assumed to average 9 pounds absolute in condensing engines and 19 pounds absolute in non-condensing engines. Then, the total ratio of expansion, by pressure, will be the absolute initial pressure divided by the absolute terminal pressure. This determination of one of the factors (the total ratio of expansion), being based on assumptions that may differ considerably from the conditions actually existing, will serve to show why the determination of the mean effective pressure is only approximate at best, and that, hence, the horsepower of the engine cannot be estimated with any great degree of accuracy.

Rule 3.—*To estimate the probable mean effective pressure of a simple expansive engine equivalent in power to a given multiple-expansion engine, multiply the estimated initial absolute pressure in the high-pressure cylinder by the factor opposite the ratio of expansion in Table I. Subtract the estimated back pressure and multiply the remainder by the factor corresponding to the type of engine and given in Table II.*

Or, $$p_m = (p_i c - p_b) \times f,$$

where p_m = mean effective pressure;
p_i = initial absolute pressure in the high-pressure cylinder;
c = factor taken from Table I;
p_b = absolute back pressure;
f = factor taken from Table II.

The absolute back pressure may be estimated at 17 pounds for non-condensing engines and 3 pounds for condensing engines. The values here given are average values and may vary somewhat from the true values that may actually

TABLE I.

FACTORS CORRESPONDING TO RATIOS OF EXPANSION.

Ratio of Expansion.	Factor.	Ratio of Expansion.	Factor.	Ratio of Expansion.	Factor.	Ratio of Expansion.	Factor.
1.32	.966	2.86	.717	7.00	.421	16.0	.236
1.48	.940	3.00	.699	8.00	.385	17.0	.226
1.54	.929	3.33	.661	8.50	.369	18.0	.216
1.60	.918	3.63	.631	9.00	.355	19.0	.207
1.66	.906	4.00	.596	10.00	.330	20.0	.199
1.82	.878	4.44	.561	11.00	.309	21.0	.192
2.00	.846	5.00	.522	12.00	.290	22.0	.186
2.22	.809	5.71	.480	13.00	.274	24.0	.174
2.50	.766	6.00	.465	14.00	.260	26.0	.164
2.66	.744	6.60	.434	15.00	.247	28.0	.155

TABLE II.

FACTORS CORRESPONDING TO TYPE OF ENGINE.

Type of Engine.	Ratio of Expansion.	Factor.
Compound, non-condensing Corliss.....	6–12	.85
Compound, condensing Corliss.........	12–18	.80
Compound, non-condensing slide-valve..	6–12	.75
Compound, condensing slide-valve......	12–18	.70
Triple-expansion, condensing Corliss....	18–27	.72
Triple-expansion, condensing slide-valve	18–27	.60

obtain with a given engine, and thus another source of error is introduced in the estimation of the mean effective pressure.

60. The mean effective pressure having been estimated by rule **3,** the horsepower is calculated by multiplying together the estimated mean effective pressure, the length of stroke in feet, the area of the low-pressure piston in square inches, and the number of strokes per minute, and dividing the product by 33,000. It will be noticed that this is the same rule by which the horsepower of any simple engine is calculated.

EXAMPLE.—Estimate the probable horsepower of a triple-expansion engine having cylinder diameters of 22, 35, and 56 inches; a common stroke of 36 inches; a speed of 140 revolutions per minute; and a boiler pressure of 175 pounds. The engine is condensing and fitted with piston valves throughout.

SOLUTION.—The absolute initial pressure may be estimated at $(175 + 14.7) - 5 = 184.7$, say 185 pounds per square inch. The terminal pressure, by Art. **59,** may be taken as 9 pounds. Then the ratio of expansion by pressure is $\frac{185}{9} = 20.55$. By Table I, the factor for a ratio of expansion of 20 is .199, and for 21 is .192. Since 20.55 is about midway between 20 and 21, the factor to be used may safely be assumed to be midway between .199 and .192, say .195. The back pressure, by Art. **59,** may be estimated at 3 pounds. Since the engine is fitted with piston valves, which are the equivalent of slide valves, the factor to be taken from Table II is .6.

Applying rule **3,** we get

$$p_m = (185 \times .195 - 3) \times .6 = 19.8 \text{ pounds per square inch.}$$

Then, the probable indicated horsepower

$$= \frac{19.8 \times \frac{36}{12} \times 56^2 \times .7854 \times 2 \times 140}{33,000} = 1,241.35, \text{ say } 1,250. \quad \text{Ans.}$$

CYLINDER VOLUMES.

61. The average ratios of cylinder volumes of compound and triple-expansion engines with different steam pressures are given in Table III.

TABLE III.

AVERAGE RATIOS OF CYLINDER VOLUMES.

Initial Steam Pressure, Gauge.	High-Pressure Cylinder.	Inter-mediate Cylinder.	Low-Pressure Cylinder.	Remarks.
100	1		2.60	Non-condensing
100	1		3.60	Condensing
110	1		3.80	Condensing
120	1		4.00	Condensing
130	1		4.15	Condensing
140	1		4.30	Condensing
150	1		4.45	Condensing
160	1		4.60	Condensing
160	1	3.00	7.00	Condensing
185	1	3.33	7.80	Condensing

ENGINE MANAGEMENT.

(PART 1.)

TAKING CHARGE.

1. The first duty of an engineer in taking charge of a power plant is to make a thorough inspection of all parts of it in order to become familiar with every detail of the engines, boilers, pumps, and their appurtenances. He should notice particularly if any parts of the engines, pumps, etc., such as cylinder heads, valve-chest covers, pump or condenser bonnets, connecting or eccentric rods, are disconnected or removed. If any parts have been removed, advantage should be taken of the opportunity to examine them. The condition of the interior parts of the engines should be noted. If the cylinder heads are off, he should examine and try with a wrench the follower bolts and piston-rod nuts. He should look carefully at the walls of the cylinders, and if they are cut, grooved, or pitted, he should make a note of the fact for future reference and comparison. The clearance between cylinder head and piston should be measured and marked on the guides with a center punch in order to ascertain and preserve a record of the space there is to spare for taking up the wear on connecting-rod journals. The cylinders should be wiped out thoroughly with oily waste and a liberal coating of cylinder oil should be applied to the wearing surfaces before closing up the cylinders.

2. If the valve-chest covers are off, the condition of the valves and seats should be noted. The lead of the valves should be examined; this may be done by turning the engines over by hand, having first poured a little oil into the journals. The valve gear should be watched during the operation of turning the engine, in order to discover if there is any derangement of the valve gear; if there is any obstruction to the engine turning freely, it will be revealed during the turning process.

3. If the pump-valve bonnets are off, the valves and valve chambers should be examined. If the valves are of hard, flat rubber and any of them are much worn on their lower faces, they should be turned over; if they are curled from standing dry for some time, or other causes, they should be faced off.

4. If the condenser bonnets are off, examine the interior of the condenser in order to ascertain its condition. If a surface condenser, fill the steam side of the condenser with water and look for leaky tubes; replace with new ones any tubes that may be split and renew all leaky tube packing.

5. Just before replacing any of the covers, bonnets, or cylinder heads, a final examination should be made to see that no tools, waste, or other foreign matter are left inside; look particularly for monkeywrenches, hammers, cold chisels, hand lamps, and pieces of waste, as it is quite a usual occurrence to find one or more of these articles inside of the machinery, left there by careless workmen; it is advisable that the engineer in charge should attend to this duty personally. Prior to replacing covers, bonnets, or cylinder heads, the gaskets should be examined; if they are torn or worn out, new ones should be used, and a thin coating of black lead (graphite) should be applied to them before they are put in place.

6. If the connecting-rod is disconnected at the crankpin end, an excellent opportunity to examine the condition of the crankpin and brasses presents itself; if they are cut or are rough, they should be scraped down with a scraper and finished off with a smooth file, but it must be skilfully done; in connecting up, the crankpin brasses should not be set up too tight upon the pin; better leave them a little slack, to be taken up after the engine has been running for awhile.

7. Examine the piston rod and valve-stem stuffingboxes and put a turn or more of packing in them, as may be required. Examine the cylinder relief valves, if any are fitted to the engine; also examine the drain cocks to see that they are not stopped up.

8. Having put the engine in good order and gotten it ready for steam, the engineer should turn his attention to the feedwater apparatus; he should examine the main feed-pump most carefully. If it is a plunger pump, note the condition of the plunger packing and repack the stuffing-box if necessary; if it is a piston pump, examine the piston-rod packing and put in a turn or two of packing if the box will take it. Examine the petcock and see that it is not stopped up.

9. When satisfied that the pump is in good order, proceed to trace up the pipes. Commence with the suction pipe at the pump; follow it up and examine every foot of it to and from the filter, feedwater heater, or grease extractor, if there be one or the other, to the source of the feedwater supply, wherever that may be, and make sure that there is nothing the matter with this supply. Now start at the pump again and follow up the delivery pipe, tracing it through all its windings, noting every bend and connection, if any; also note where it enters and leaves the feedwater heater, purifier, or economizer, as the case may be, then on through the check-valve and globe valve to the point where it enters the boiler; take out the check-valve and carefully examine it, as well as its seat; wipe off the valve and valve seat with oily waste, and if it is in good condition, replace the valve, but if found in poor order, repair it or replace it with a new one immediately. Try the globe valve in the feedpipe; if it works stiffly, oil the stem and thread and run the valve up and down until it works freely. Treat all globe valves in a similar manner, and if any of them need packing, attend to it at once. Trace up the auxiliary feed-pipes, both suction and delivery, in the same manner and with the same care and attention that was given the main feedpipes.

10. Trace up, from beginning to end, all the auxiliary steam and exhaust piping to and from the various auxiliary engines and pumps, neglecting nothing. Note if the exhaust pipes of the auxiliary system lead to a condenser; if so, locate the valves for changing the auxiliary machinery from non-condensing to condensing and vice versa.

11. If the main engines are condensing, examine the air pump and circulating pump and their valves and trace up all steam pipes and water pipes leading to and from them, whether the pumps are operated by the main engine or independently.

12. Locate all cocks and valves in the piping and ascertain what every one of them is for; locate particularly all those connected with the feedwater supply system, both in the steam pipes and in the water pipes.

13. If the plant is a modern one and is supplied with the latest economical and other appliances and apparatus, such as a separator, grease extractor or filter, feedwater purifier, heater or economizer, evaporator, superheater, reheaters, etc., they must be included in the preliminary inspection and should receive the same care and attention as the other parts of the machinery.

14. In some plants the water of condensation from the steam pipes, valve chest, cylinders, steam jacket, and wherever else it may collect, is saved for use in the boilers;

this is accomplished by leading the various drain pipes into a manifold, from which a single pipe conveys the water to a steam trap, and from thence, by another pipe, to the hot-well or feed tank, where it mingles with the feedwater. These drain pipes should be traced up and examined from beginning to end. Look for leaky or broken joints and split pipes; if any are found, they must be repaired at once. Look also for badly rusted places in the pipes. Note if they are exposed to unusual dampness or dripping water; if such is the case, a coat of thick red-lead paint or paraffin varnish will afford considerable protection. Similar treatment should be accorded to any other iron or steel piping under floors or in places not easily accessible.

15. If during this inspection anything is found that is out of order, it should be repaired at once. Tools and stores for the ordinary repairs are generally provided.

STARTING AND STOPPING ENGINES.

INTRODUCTION.

16. Owing to the great variety of engines and valve gears in use and to the great difference in the sizes and power of steam plants, involving a wide range of appliances and apparatus, it is not possible to give specific directions in detail for starting and stopping each and every one of them. General instructions, with a few examples, can only be given, but it is the intention to make them full enough to enable the intelligent engineer, by using a little judgment and discretion, to apply them to all types of engines.

GENERAL INSTRUCTIONS.

17. Warming Up and Getting Ready.—The engine having previously been put in thorough order and the fires having been lighted in the boiler, it is assumed that the steam pressure is now approaching its working point. About 15 or 20 minutes before starting the engine, raise the stop-valves just off their seats and let a little steam flow into the steam pipe; open the drain cock on the steam pipe just above the throttle. When the steam pipe is thoroughly warmed up and steam blows through the drain pipe, close the drain cock and open the throttle just enough to let a little steam flow into the valve chest and cylinder, or use the by-pass around the throttle, if one is fitted. Open the cylinder relief valves (or drain cocks), also the drain cocks on the valve chest and exhaust pipe, if a non-condensing engine. If cylinders are jacketed, turn the steam into the jackets and open the jacket drain cocks. While the engine is warming up, fill the oil cups and sight-feed lubricator. Squirt a little oil into all the small joints and journals that are not fitted with oil cups. Wipe off the guides with oily waste and squirt some oil over them. By this time the engine is getting warm; if fitted with by-pass valves, use them to admit steam into both ends of the cylinder. Further operations of warming up the cylinder will depend somewhat on the type of engine and valve gear; therefore, additional instructions regarding this matter will be given under their respective headings. In general, however, all cylinders, especially if they are large and intricate castings, should be warmed up slowly, as sudden and violent heating of a cylinder of this character is very liable to crack the casting by unequal expansion.

18. An excellent and economical plan for warming up the steam pipe and the engine is to open the stop-valves and throttle valve at the time or soon after the fires are lighted in the boilers, permitting the heated air from the boilers to circulate through the engine, thus warming it up gradually and avoiding the accumulation of a large quantity of water of condensation in the steam pipe and cylinder. When pressure shows on the boiler gauge or steam at the drain pipes of the engine, the stop-valves and throttle may be closed temporarily, but not hard down on their seats. When this method of warming up the engine is adopted, the safety valves should not be opened while steam is being gotten up.

19. In attending to these preliminary arrangements certain precautions should be taken. For example, stop-valves and throttle valves should never be opened quickly or suddenly and thus permit a large volume of steam to flow into a cold steam pipe or cylinder. If this is done, the first steam that enters will be condensed and a partial vacuum will be formed. This will be closely followed by another rush of steam with similar results, and so on until a mass of water will collect, which will rush through the steam pipe and strike the first obstruction, generally the bend in the steam pipe near the cylinder, with the force of a steam hammer, and in all probabilities will carry it away and cause a disaster. This is called a **water hammer** and has caused many serious accidents.

20. Another precaution that should be taken is the easing of the throttle valve on its seat before steam is let into the main steam pipe; otherwise, the unequal expansion of the valve casing may cause the valve to stick fast and thereby give much trouble. Even if a by-pass pipe is fitted around the throttle, it would be better not to depend on it. Considerable space has been devoted to the subject of warming up and draining the water out of the steam pipe and engine on account of its importance. Water being non-compressible, it would be an easy matter to blow off a cylinder head or break a piston if the engine were started when there was a quantity of water in the cylinder.

21. The last thing for the engineer to do before taking his place at the throttle preparatory to starting the engine, provided he has no oiler, is to start the oil and grease cups feeding. It is well to feed the oil liberally at first, but not to the extent of wasting it; finer adjustment of the oiling gear can be made after the engine has been running a short time and the journals are well lubricated.

22. Cleaning Up.—After an engine has been stopped after a run and everything has been made secure, the machinery should be wiped off before the oil has had time to set. Both the bright work and the painted parts should receive attention in this respect. If there are any rusty spots on the bright work, they should be immediately scoured off with emery cloth. The floors should also be cleaned up and all dirt gathered together and consigned to the ash heap. Cleanliness is essential to a well-kept engine room, and the grade and value of the engineer in charge of it can very readily be determined by a glance of the practiced eye around the engine room. Oil should not be spilled or spattered about; there is no necessity for it and it is a waste of oil. Drip pans should be placed wherever they will do any good and they should be emptied and cleaned out at least once a day. Much saving in oil bills will be effected by the use of an efficient oil filter to filter the drip oil and use it over again instead of throwing it away.

23. When flour emery or emery cloth is used for cleaning bright work, the greatest care should be exercised not to let any of the grit get into the journals; they will be sure to cut if any substance of a gritty nature gets into them. All the oil holes in the small joints and journals that are not fitted with oil cups should be plugged up immediately after the engine is stopped and kept closed until the engine is ready to be started again. Emery should not be used to polish brass or composition; Bath brick is much better for this purpose.

STARTING A SLIDE-VALVE NON-CONDENSING ENGINE.

24. Assuming that the general instructions given in Arts. **17** to **23** have been complied with, the engineer should now take his place at the throttle, having first opened wide the stop-valves. The drain cocks on the steam pipe and engine are supposed to be open and the throttle valve just off its seat. Some steam has been allowed to enter the valve chest and the cylinder is partly warmed up; it is now the duty of the engineer to ascertain if steam has entered both ends of the cylinder and that both ends of it are heated equally. As both steam ports cannot be open to the steam at the same time, the engine, if not provided with a by-pass, should be turned by hand so that both steam ports are opened alternately, thus admitting steam to both ends of the cylinder. In turning the engine, finally stop when the crank is on its upper or lower half center, that being the best point from which to start the engine. When it is evident that all condensation of steam has ceased in the steam pipe, valve chest, and cylinder and all the water has been blown out of them, the engine is ready to be started.

25. A slide-valve non-condensing engine is started by simply opening the throttle; this should be done quickly in order to jump the crank over the first center, after which the momentum of the flywheel will carry it over the other centers. The engine should be run slowly at first, gradually increasing the revolutions to the normal speed. When the engine has reached full speed, the drain pipes should be examined; if dry steam is blowing through them, close the drain cocks; if water is being delivered, let the drain cocks remain open until steam blows through and then close them.

26. The engineer should now make a trip around the engine to ascertain if the journals are running cool. First, try the crankpin end of the connecting-rod by touching it with the palm of the hand; to do this safely, on a high-speed engine, requires some skill and experience, but the art can be acquired by a little practice; the beginner, however, should be very cautious that he does not get his hand caught in the machinery. If the end of the connecting-rod is only blood warm, no harm has yet been done, but it is an intimation that the crankpin may get hot, and requires watching. Assuming that the crankpin is running cool, the next step is to feel the shaft journals and examine the lubricating apparatus. If the journals are running cool, decrease the oil feed gradually and carefully until there is just enough oil fed into the journals to supply the demand without unnecessary waste. It is supposed that the engine is now running satisfactorily, and the engineer may hence turn his attention to a general inspection of his department.

27. It sometimes happens that a cracking noise is heard in the cylinder after the engine has been running for a while. This means "water in the cylinder," and the cylinder drain cocks should be opened promptly. It is also an intimation that the boiler is inclined to prime; this may be checked by closing the main stop-valve just enough to wiredraw the steam a little.

STOPPING A SLIDE-VALVE NON-CONDENSING ENGINE.

28. To stop a slide-valve non-condensing engine, it is only necessary to shut off the supply of steam by closing the throttle, but care should be taken not to let the engine stop on the center. After a little practice, the beginner will be able to stop the engine at any desired point of the revolution. No rule can be laid down for this; it is entirely a matter of experience.

29. After the engine is stopped, shut off the oil feed and close the main stop-valve; be sure that the valve is seated, but without being jammed hard down on its seat. The drain cocks on the steam pipe and engine may or may not be opened, according to circumstances. It will do no harm to allow the steam to condense inside the engine, as the engine will then cool down more gradually, which lessens the danger of cracking the cylinder casting by unequal contraction. All the water of condensation should be drained from the engine before steam is again admitted to it.

30. When an engine is required to run in either direction, in answer to signals or otherwise, as in the case of hoisting engines and locomotives, it is usually fitted with the link-valve motion, which is operated by a system of levers or other apparatus called the **reversing gear.** In warming up an engine fitted with a link, it is only necessary to run the link up and down if a horizontal engine, or back and forth if a vertical engine, to admit steam into both ends of the cylinder; and to start or stop such an engine, either the go-ahead or the backing eccentric, as required, is thrown into action by operating the link by means of the reversing gear.

STARTING A SLIDE-VALVE CONDENSING ENGINE.

31. A steam engine of the condensing type is fitted with either a surface, a jet, or an injector condenser. The function of a condenser is to convert the exhaust steam from the engine into water by condensation, thereby producing a vacuum in the condenser. The pressure of the atmosphere is thus partially removed from the exhaust side of the piston and the net pressure correspondingly increased.

32. If the engine is fitted with a surface condenser, the condenser will be supplied with an air pump and a circulating pump. The air pump removes the air, vapor, and water of condensation from the condenser; it discharges the water into the hotwell or feed tank, while the air and vapor escape into the atmosphere. The circulating pump supplies the condensing water and forces it through the tubes of the condenser.

33. It is sometimes the case that the air pump and the circulating pump are attached to and operated by the main engine; more frequently, however, they are operated by a separate and independent steam cylinder, in which cases the apparatus as a whole, including the condenser is called the **vacuum engine.**

Another arrangement of these pumps is the following: A centrifugal circulating pump is used instead of a reciprocating pump, by which the circulating pump can be operated independently of the air pump, permitting the speed of the circulating pump to be changed without affecting the speed of the air pump. This is desirable, because a greater quantity of injection or condensing water is required in summer than in winter on account of its higher temperature at that season of the year.

34. Before the main engine is started, the air pump and circulating pump should be put into operation and a vacuum formed in the condenser; this will materially assist the main engine in starting promptly, and in cases where the engine is worked to bell signals, such as a hoisting engine in a mine or elsewhere, this is a most important consideration. Prior to starting the air and circulating pumps, the injection valve should be opened to admit the condensing water into the circulating pump; the delivery valve should also be opened at this time. The same course of procedure that is used in warming up and draining the water out of the main engine should be followed with the vacuum engine, and it is started in the same manner, i. e., by simply opening the throttle.

35. After the main engine has been running for a few minutes to equalize temperatures, the speed of the air and circulating pumps and the admission of injection water should be regulated so as to maintain about 26 inches of vacuum in a surface condenser and a feedwater temperature of about 115° F. A higher vacuum than 26 inches, when the barometer stands at 30 inches, will result in a loss of heat from cold feedwater, and it will also cause a high-speed engine to thump while passing the centers through insufficient compression or cushion for the piston; a lower vacuum than 26 inches will cause a loss by too much back pressure. As a rule, there should be about 2 pounds (absolute) of back pressure on the exhaust side of the piston;

this is equivalent to 4 inches on the vacuum gauge and a feedwater temperature of about 115° F.; therefore, the reading of the vacuum gauge should be about 4 inches below the reading of the barometer to get the best results from the engine.

36. If an ordinary jet condenser is used, no circulating pump is required, the water being forced into the condenser by the pressure of the atmosphere. If the air pump is operated by the main engine, which is sometimes the case, a vacuum will not be formed in the condenser until after the engine is started and at least one upward stroke of the air pump is made. In this case the injection valve must be opened at the same moment the engine is started; otherwise the condenser will get "hot" and a mixture of air and steam will accumulate in it and prevent the injection water from entering. When this occurs it is necessary to pump cold water into the condenser by one of the auxiliary pumps through a pipe usually fitted for that purpose; if such a pipe has not been provided, it may be found necessary to cool the condenser by playing cold water upon it through a hose.

37. Jet condensers are sometimes fitted with a valve that automatically opens outwards, called a **snifting valve,** or **snifter,** by which the accumulated steam, vapor, and air may be discharged into the atmosphere. This valve is a disk of metal, similar to a safety valve, that is held to its seat by its own weight and the pressure of the atmosphere. It serves to relieve the condenser of pressure in case of an accident to the air pump.

38. If the engine is fitted with a jet condenser, the course of procedure in starting is similar to that followed in starting an engine with a surface condenser, viz.: The air pump should be started before the main engine is started and thereby form a vacuum in the condenser beforehand, and it should be stopped after the main engine is stopped.

STOPPING A SLIDE-VALVE CONDENSING ENGINE.

39. The operation of stopping a slide-valve surface-condensing engine is precisely similar to that of stopping a non-condensing engine of the same type, with the addition that after the main engine is stopped the vacuum engine is also stopped, and in the same way, i. e., by closing the throttle, after which the injection valve and the discharge valve should be closed and the drain cocks opened.

40. With a jet condenser, the operation of stopping the engine is the same as the above, with the exception that the injection valve should be closed at the same moment that the engine is stopped.

SUMMARY.

41. The instructions given in Arts. **24** to **40** apply to any slide-valve engine, whether vertical or horizontal, and also whether it is fitted with a ball or pendulum governor, a shaft automatic governor, or if it is without any governor at all, from the fact that a governor acts only when the engine is running at or near its normal speed; therefore, while starting or stopping an engine the governor is not in action.

STARTING A SIMPLE CORLISS ENGINE.

42. A simple engine fitted with Corliss valves and valve gear is started and stopped in a somewhat different manner from that practiced with the plain slide-valve engine.

43. In the Corliss engine the eccentric rod is so constructed and arranged that it may be hooked on or unhooked from the eccentric pin on the wristplate at the will of the engineer. The wristplate is provided with a socket for the starting bar; the starting bar may be shipped or unshipped as required.

44. In starting an engine of the Corliss type, after all the preliminaries, such as warming up and draining the water from the engine, starting the oil feed, etc., as heretofore explained, have been attended to, the starting bar is shipped into its socket in the wristplate and the throttle is opened. The starting bar is then vibrated back and forth by hand, by which the steam and exhaust valves are operated through the wristplate and valve rods; as soon as the cylinder takes steam the engine will start. After working the starting bar until the engine has made several revolutions and the flywheel has acquired sufficient momentum to carry the crank over the first center, let the hook of the eccentric rod drop upon the pin on the wristplate. As soon as the hook engages with the pin, unship the starting bar and place it into its socket in the floor.

45. The way to determine in which direction the starting bar should be first moved to start the engine ahead is to note the position of the crank, from which the direction in which the piston is to move may be learned. This will indicate which steam valve to open first; it will then be an easy matter to determine in which direction the starting bar should be moved. After a little practice the engineer will know at a glance which way to work the starting bar.

46. The engine having been started, the engineer should attend to those duties that have been mentioned in the instruction for the slide-valve engine under similar circumstances.

47. If the engine is of the condensing type, the same course of procedure in starting the vacuum engine should be followed as with the simple slide-valve condensing engine, which has been previously explained. In warming up the cylinder of a Corliss engine, it is not necessary to turn the engine to admit steam to both ends; it is only necessary to work the valves by hand with the starting bar.

STOPPING A SIMPLE CORLISS ENGINE.

48. A Corliss engine is stopped by closing the throttle and unhooking the eccentric rod from the pin on the wristplate; this is done by means of the unhooking gear provided for the purpose. As soon as the eccentric rod is unhooked from the pin, ship the starting bar into its socket in the wristplate and work the engine by hand to any point in the revolution of the crank at which it is desired to stop the engine. Then proceed as directed for the simple slide-valve engine. After stopping a Corliss condensing engine, the same course should be followed as with a slide-valve condensing engine in regard to draining cylinders, closing stop-valves, etc.

STARTING A COMPOUND ENGINE.

49. Before starting a compound engine, the high-pressure cylinder is warmed up in the same manner as a simple engine. To get the steam into the low-pressure cylinder is an operation, however, that will depend on circumstances. If the cylinders are provided with pass-over valves, it will only be necessary to open them to admit steam into the receiver and from thence into the low-pressure cylinder. If the cylinders are not fitted with pass-over valves, the steam can usually be worked into the receiver and low-pressure cylinder by operating the high-pressure valves by hand. Sometimes compound engines are fitted with starting valves, which greatly facilitate the operations of warming up and starting. Usually a compound engine will start upon opening the throttle.

50. It sometimes happens that the engine will refuse to start from various causes, viz.: The high-pressure crank may be on its center; there may be too high or too low a steam pressure in the receiver; the engine may be stiff from standing idle a long time, and the oil in the journals has become gummy; the pistons may be rusted fast in the cylinders, or the cylinders may not have been wiped out after the last run and a coating of carbonized oil and rust may have collected on their walls and caused the pistons to stick fast. If the engine is fitted with independent adjustable cut-offs, the cut-offs may be set to cut off too early; or there may be water in the cylinders. There may be some obstruction to the engine turning, although that matter is supposed to have been attended to during the preliminary inspection. In most cases the conditions will suggest the remedy. If the high-pressure crank of a cross-compound engine is on its center and the low-pressure engine will not pull it off, it must be jacked off. If the pressure of steam in the receiver is too high, causing too much back pressure in the high-pressure cylinder, the excess of pressure must be blown off through the receiver safety valve; if the pressure in the receiver is too low to start the low-pressure piston, more steam must be admitted into the receiver. If the engine is stuck fast from gummy oil or rusty cylinders, all wearing surfaces must be well oiled and the engine jacked over at least one entire revolution. If the cut-offs are run up, run them down, full open. If there is water in the cylinders, blow it out through the cylinder relief or drain valves, and if there is any obstruction to the engine turning, remove it.

51. If the crank of a tandem compound engine is on the center, it must be pulled or jacked off. If the high-pressure crank of a cross-compound engine is on the center, it may or may not be possible to start the engine by the aid of the low-pressure cylinder, depending on the valve gear and crank arrangement. When the cranks are 180° apart (which is a very rare arrangement), the crank must be pulled or jacked off the center. When the cranks are 90° apart and a pass-over valve is fitted, live steam may be admitted into the receiver and thence into the low-pressure cylinder, in order to start the engine. When no pass-over is fitted, but the engine has a link motion, sufficient steam to pull the high-pressure crank off the center can generally be worked into the low-pressure cylinder by working the links back and forth. When no pass-over is fitted, but the high-pressure engine can have its valve or valves worked by hand, steam can be gotten into the low-pressure engine by working the high-pressure valve or valves back and forth by hand. If no way exists of getting steam into the low-pressure cylinder while the high-pressure crank is on a dead center, it must be pulled or jacked off.

52. If the air and circulating pumps are attached to and operated by the main engine, a vacuum cannot be generated in the condenser until after the main engine has been started. Consequently, in this case, there is no vacuum to help start the engine; therefore, if it is tardy or refuses to start, it will be necessary to resort to the jacking gear and jack the engine into a position from which it will start. With an independent vacuum engine, however, it is seldom that any such difficulties in starting an engine are encountered. A vacuum having been generated in the condenser beforehand, the pressure in the receiver acting on the low-pressure piston causes the engine to start very promptly, even though the high-pressure crank may be on its center. Notwithstanding differences of opinions among designers in regard to this matter, the value of an independent vacuum engine is fully appreciated by the man at the throttle. In fact, it is almost indispensable with compound condensing engines of high power that are required to run in either direction in answer to bell or other signals, such as hoisting, rolling-mill, and marine engines, where promptness in starting is absolutely essential.

53. Large reversible engines are usually fitted with steam starting and reversing gears, each builder suiting his own fancy in regard to the design; therefore, the engineer should promptly familiarize himself with the mechanism of the particular starting gear that he has to handle.

54. If the engine is fitted with adjustable cut-offs, they should be so manipulated, after the engine has been started, that it will run smoothly and at the lowest steam consumption attainable with the given load. The only way of finding the proper points of cut-off is by experiment, setting the cut-offs where judgment and experience dictate and noting the effect upon the smooth running and steam consumption.

STOPPING AND REVERSING A COMPOUND SLIDE-VALVE ENGINE.

55. Compound slide-valve engines, whether condensing or non-condensing, are stopped by closing the throttle, and, if a reversing engine, throwing the valve gear into mid-position. If the stop is a permanent one, the usual practice of draining the engine, steam chests, and receiver, closing stop-valves, stopping the oil feed, etc. may be followed; or, as before stated, if the cylinders and receivers are complicated castings, as they are apt to be in an engine of this kind, it would be better not to drain them while they are hot, but to let them cool down gradually to avoid the danger of cracking the castings from too sudden and, therefore, unequal contraction.

56. If the engine is intended to run in both directions in answer to signals, as in the cases of hoisting, rolling-mill, and marine engines, the operator, after stopping the engine to signal, should immediately open the throttle very slightly, in order to keep the engine warm, and stand by for the next signal. If the engine is fitted with an independent or adjustable cut-off gear, it should be thrown off, i. e., set for the greatest cut-off, for the reason that the engine may have stopped in a position in which the cut-off valves in their early cut-off positions would permit little or no steam to enter the cylinders, in which case the engine will not start promptly, and perhaps not at all. While waiting for the signal, the cylinder drain valves should be opened and any water that may be in the cylinders blown out. When dry steam blows through the drains, the cylinders are clear of water.

57. When the signal to start the engine is received, it is only necessary to throw the valve gear into the go-ahead or backing position, as the signal requires, and to operate the throttle according to the necessities of the case, for which no rule can be laid down beforehand, as the position of the throttle will depend on the load on the engine at the time. Handling the throttle must be learned by experience on the spot.

58. It is frequently the case that in large plants a working platform is provided on which the reversing gear, throttle-valve lever or wheel, cylinder drain-valve levers, and all other hand gear, gauges, etc., are located and placed within easy reach of the engineer's station. This platform is usually placed in a commanding position, from whence the engineer has a full view of the moving parts of the engine. This is a matter of considerable importance, although an experienced engineer, after he becomes familiar with the various sounds produced by his engine under different conditions, will depend on the ear as much as on the eye in running it. In most cases any derangement of the machinery will give timely warning by making an unusual sound; perhaps it may be only a slight clicking noise, scarcely noticeable among so many different sounds. A careful engineer, however, can detect it as quickly as an expert musician can detect a discordant note, and he should at once proceed to find out the cause, thereby anticipating and preventing a possible breakdown.

STARTING AND STOPPING A CORLISS COMPOUND ENGINE.

59. The operation of starting and stopping a Corliss compound engine is precisely similar to that of starting and stopping a Corliss simple engine; the high-pressure valve gear only is worked by hand in starting, the low-pressure eccentric hook having been hooked on previously. The low-pressure valve gear is only worked by hand while warming up the low-pressure cylinder. The same directions that were given for operating the simple condensing engine apply to the Corliss condensing engine, so far as the treatment of the air pump, circulating pump, and condenser is concerned.

STARTING, STOPPING, AND REVERSING TRIPLE- AND QUADRUPLE-EXPANSION ENGINES.

60. The management of triple- and quadruple-expansion engines is the same as that practiced with the compound engine, with the exception that there is a greater number of moving parts, more journals, more hand gear, and more machinery, in general, to look after, requiring greater activity and alertness on the part of the engineer to care for it.

LINING ENGINES.

INTRODUCTION.

61. Purpose.—The operation of lining an engine is for the purpose of locating the different parts in relation to each other, so that no undue strains will be exerted on any one of the parts and the friction of the moving parts will be reduced to a minimum. Absolute accuracy, though desirable, cannot be attained, and if it could be attained, it could not be maintained. Too much care, however, cannot be exercised in lining an engine, as its future smooth running and efficiency will depend very largely on the accuracy with which this operation is performed.

62. Requirements.—An engine, in order to be in line, must fulfil the following requirements:

1. The center line of the shaft must be at right angles to the center line of the cylinder.

2. The wearing surfaces of the guides must be parallel to the center line of the cylinder. When two guides are used, they must be parallel to each other, and, at least in most designs, equidistant from the center line of the cylinder.

3. The center line of the wristpin must be at right angles to the center line of the cylinder and must lie in the same plane.

4. The center line of the crankpin must be parallel to the center line of the shaft.

5. The center line of the cylinder and of the shaft must both lie in the same plane.

6. The center line of the bore of the brasses at both ends of the connecting-rod must be parallel to one another and must be at right angles to the center line of the connecting-rod.

7. The center line of the piston rod must coincide with the center line of the cylinder.

If the above requirements are fulfilled, the engine may be said to be in line, as far as the machine itself is concerned. In addition to the requirements enumerated above, it is generally necessary that the crank-shaft be level.

LINING UP.

63. Preliminary Conditions.—Let it be understood that a new single-cylinder, horizontal engine is in the course of erection or that an old engine of the same type is receiving a thorough overhauling. In the case of the new engine, it is assumed that the foundation has been built, the bedplate placed in its proper position and secured there by the anchor bolts, and the cylinder has been secured to the bedplate. As the cylinder was fitted to the bedplate in the shop, it may be assumed that it is correctly placed. In the case of an old engine being overhauled, it is understood that all the moving parts have been removed and that the cylinder heads are off. The condition of both engines are now supposed to be the same; therefore, henceforth the course of procedure will be the same for both.

64. Stretching Center Line of Cylinder.—The first step in lining an engine is stretching a line coincident with the center line of the cylinder. This may be done in the following manner:

A strip of board or other convenient material is secured to the head end of the cylinder by means of the stud bolts and nuts, as shown at *a*, Fig. 1. A hole about 1 inch in diameter

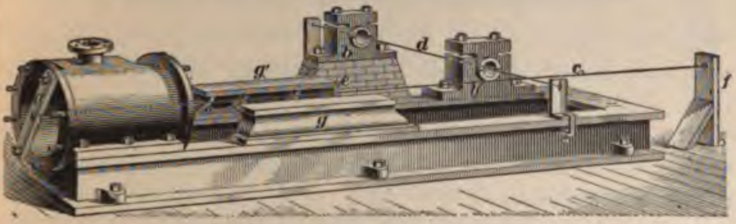


FIG. 1.

is bored through the strip approximately in line with the center of the cylinder. Some form of standard, as *f* for instance, which is pierced similarly to the strip of board *a*, should be erected at the crank end of the bedplate. A very fine braided cord or piece of thin annealed copper wire, as *c*, may now be stretched very tightly through the holes in *a* and *f*. In order to allow of ready adjustment, each end of the line may be fastened to the middle of a piece of, say, $\frac{1}{4}$-inch round iron about 2 inches long, or, better yet, be passed through a hole in the center of a piece of stout sheet tin or other metal. A knot at each end of the line will prevent it slipping through the holes. The pieces of sheet metal may be cut, say, 2 inches square, and the diagonally opposite corners may be turned up at right angles to form handles by which they may be adjusted. If the line used is a fine wire, two holes may be punched in each piece of sheet metal and the end of the wire passed through one of the holes, brought back through the other in the form of a loop, and the end stopped off around the main part of the line.

65. The line should be set central to the bore of the cylinder at the head end by calipering from the inside of the cylinder counterbore to the line. This may be done with a pair of inside calipers, but in most cases it is better to use a light pine stick, like that shown in Fig. 2. The stick *a* should be about 1 inch shorter than the radius of the cylinder counterbore and tapered at each end, with a thin 1-inch

FIG. 2.

wire nail driven into each end, as shown at *b*, making the total length of the stick, including the nails, the exact radius of the counterbore. The advantage of the stick in calipering is that it is lighter and more convenient to use than inside calipers.

If the calipers or stick will just touch the line, no matter from which point on the circumference of the counterbore the measurement is taken, the head-end part of the line will be central to the bore of the cylinder, provided the measurements were carefully and accurately made. If the measurements do not agree, the line that passes through *a*, Fig. 1, must be shifted in the direction shown by the variation until it coincides with the center line of the cylinder. It is considered good practice to make four measurements 90° apart.

66. After adjusting the line at the head end of the cylinder, the crank-end part of the line may be trued up in a similar manner by moving the line at the standard *f*, Fig. 1. The line, now, may or may not be properly adjusted. Hence, to make sure, the alinement of the line at the head end should be tried again. Now, unless the line happened to be very close to the center line of the cylinder before any adjustments were made, it will usually be found to be a shade out of the center, and hence requires readjustment. After adjusting the head-end part of the line, try the crank-end part again and adjust it. Continue this practice, first at one end of the cylinder and then at the other, until no further adjustment is necessary or possible. Then, if the measurements have been carefully made, the line *c* may be considered to coincide with the center line of the cylinder.

67. If the crank-end head is cast solid with the cylinder, as is frequently the case, the line *c* must be trued up from the bore of the piston-rod stuffingbox. This may be done by means of a stick similar to that shown in Fig. 2, but it may be more conveniently done by means of the device shown in Fig. 3. This device consists of a hardwood block *a*, which is turned to just fit into the stuffingbox and has a $\frac{1}{4}$-inch hole *b* bored in the exact center. The face of the block is faced square with the outside, and two center lines *c d* and *e f* are drawn across the face at right angles to each other. By sighting along these lines, it is easy to determine when the line or wire is central with the stuffingbox.

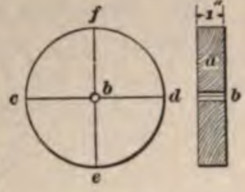


FIG. 3.

68. Stretching Center Line of Shaft.—After completing the adjustment of the line *c*, Fig. 1, another line, as *d*, is stretched to coincide with and to represent the center line of the shaft; it *must* be at right angles to the line *c* and generally *must* be level.

In many engines, the outboard bearing *b*, Fig. 1, is movable to a certain extent and may be adjusted in regard to its relative position with the inboard bearing *b'*. The inboard bearing, however, is usually cast solid with the bedplate; therefore, it is fixed and cannot be moved. Hence, in adjusting the line *d*, the aim must be to stretch the line through the center of the bearing *b* and at the same time to have it at right angles to the line *c*. In order to accomplish this, it will be necessary to erect two standards, as shown in Fig. 1, to which to fasten the line. It is supposed that the top brasses and caps of the main-shaft bearings are in place just as they would be if the shaft itself were in place. The alinement of the line *d* in reference to the bearing *b'* can be tested by calipers, but a better method is to insert a block of wood, made as shown in Fig. 3, into the bore of the bearing. It would be well to fit a similar piece of wood into the bore of the outboard bearing also, as it will greatly facilitate the adjusting of the line *d* by passing it through the holes in the centers of these blocks.

69. Squaring Center Line of Shaft.—We may now proceed to test the an[illegible]een the two lines *c* and *d*, Fig. 1. The line *d* may [illegible]ximately squared with the line *c* by a carpenter's sq[illegible]ssed gently against the two lines. Great care in th[illegible]f the square is necessary, since the lines will yield [illegible]ite an appreciable extent under a very slight pressure. [illegible] crowding of some part of the square against either lin[illegible]ill deflect it and seriously interfere with the test. If the square shows that the lines are not at right angles to e[illegible] other, the line *d* should be shifted until they are, alw[illegible]eping in mind the fact that the line must coincide wit[illegible] center line of the inboard bearing.

70. In lining engines of the larger sizes, the carpenter's square is not accurate enough, since its legs are very short in proportion to the length of the lines. A somewhat different method may then be used, which is based upon the principles of geometry.

Procure a slender strip of wood—6, 8, or 10 feet long, according to the extent of the space to work in. Taper the strip to a point, like a lead pencil, at each end; divide the strip into two exactly equal parts and mark it plainly in the middle. Now hold the strip gently along the line *c*, Fig. 1, with the mark in the middle at the line *d*; mark the line *c* at each end of the strip; then put one end of the strip at one of the marks just made on line *c* and the other end of the strip against the line *d* and mark the place where the end of the strip touches the line *d*; repeat this operation on the other side of the line *c*, and if the end of the strip touches the line *d* at exactly the same spot that it did before, the line *d* is at right angles to the line *c*; but if the end of the strip does not touch the same spot on the line *d* at both trials, the line *d* is not at right angles to the line *c*, and the line *d* must be shifted in the right direction and the whole operation must be repeated until the end of the strip *does* touch the line *d* at the same point at both trials. In shifting the line *d* during these operations, care must be taken that its position through the center line of the inboard bearing *b'* is maintained. A convenient method of marking the lines is to tie a piece of bright-colored thread around the lines at the points that are desired to be marked. The colored threads are easily sighted, and the marks are more sharply defined and cleaner than when made with paint or chalk and they have the advantage of being easily shifted along the lines at will.

71. In order to prevent any displacement of the line while measuring, it is good practice to place blocks of wood or any other convenient material against it at the points marked, in order to steady it, but care must be used not to deflect the line; when the blocks are properly placed, they may be temporarily secured in position.

72. Another test to determine whether the lines are at right angles to each other is to measure from their intersection distances of 6 and 8 feet, one on each line. Then, if the measurement from line to line, measuring in a straight line from the points just laid off, is 10 feet exactly, the lines are at right angles. Instead of using the values 6, 8, and 10 feet, any convenient multiple may be used.

73. Leveling Center Line of Shaft.—The line *d* representing the center line of the crank-shaft may be tested for being level by means of a spirit level, taking great care in applying it not to deflect the line. Another method is to drop a plumb-line from overhead touching the line *d*.

Then, if the line *d* is level, it evidently must be at right angles to the plumb-line, and, consequently, this condition can be tested in the same manner in which the position of line *d* in reference to line *c* was tested.

74. In leveling the line *d*, Fig. 1, it should not be allowed to touch the line *c*, lest it should be deflected; it should pass just over or just under it, say at a distance of $\frac{1}{100}$ inch. If the lines touch each other, any vertical movement of either end of the line *d* will, when in the wrong direction, deflect both lines, thus defeating the primary object of stretching them; viz., that they shall represent the center lines of the cylinder and shaft. After leveling the line *d*, it is well to verify the relative alinement of both lines. When the line *d* is properly adjusted, the fifth requirement will be complied with a degree of accuracy sufficient for practical purposes, providing the lines *d* and *c* will just clear each other where they cross.

75. In lining up a new engine or an old engine with new shaft-bearing brasses, it is generally desirable to have the center line of the shaft lay about $\frac{1}{32}$ inch above the center line of the cylinder, so that as soon as the brasses and journals have worn down to their bearings, the center line of the shaft will be very nearly level with the center line of the cylinder.

76. Shifting Outboard Bearing.—Having proved the correct alinement of the lines *c* and *d*, the outboard bearing may now be shifted until the line *d* coincides with its center line, when it may be secured in position permanently.

77. Testing Alinement of Guides.—We are now ready to test the guides *g*, *g'*, Fig. 1. If the line *d* is level, a spirit level may be used to ascertain if they are in the proper plane in reference to *d*. Their adjustment relative to the center line of the cylinder may be tested in one direction by placing a straightedge successively at each end of the guides and squarely across them. Then, if the measurements from the lower edge of the straightedge down to the line *c* agree at both ends of the guides, they are in line vertically. If not, the same ends of both guides must either be raised or lowered, remembering that raising one end of the guide is equivalent to lowering the other end. By measuring from the inside edges of the guides to the line *c*, it may be ascertained if the guides are parallel to each other and parallel to the center line of the cylinder. To ascertain if the guides are in a horizontal plane, a spirit level may be placed squarely across the guides at both ends.

78. In lining the guides of a Corliss engine a special device similar to that illustrated in Fig. 4 is often used. This consists of a casting *a* that is turned to fit the inside of

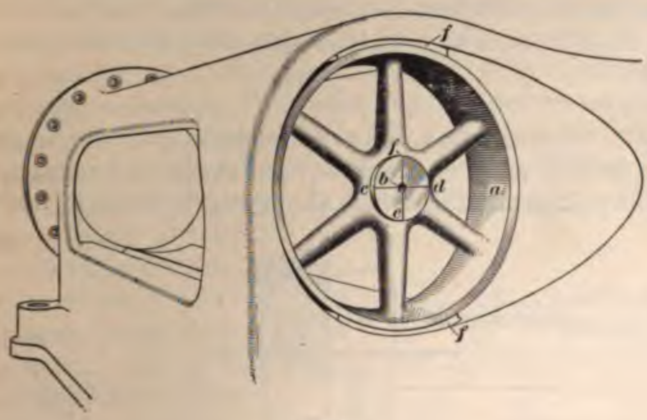


FIG. 4.

the guides. At the center there is a small hole *b* through which the line passes, while the lines *c d* and *e f*, drawn at right angles to each other, serve to locate the center line in its proper position, this being done in a manner similar to that illustrated in Fig. 3 for the piston-rod stuffingbox.

79. Bedding the Shaft.—The line *d*, Fig. 1, having been removed, the crank-shaft, crank, and flywheel are put in place, care being taken not to disturb the line *c*. After the bearings have been adjusted so that the shaft will turn easily in them, the journals of the shaft should be wiped clean and given a coat of red or black marking material. The shaft should then be placed in its bearings, with the lower halves of the brasses in position, and rocked back and forth a few times. The shaft is then lifted out of the bearings and the high spots scraped off with a half-round scraper. This operation is repeated until the shaft shows a good bearing in both the main pillow-block and the outboard bearing. After the lower halves of the boxes are scraped, the upper halves may be put in place and fitted in like manner. The shaft is now lifted from its bearings and the eccentrics and governor-driving device are placed in position, after which the shaft is returned to its place. The crank and flywheel having been fitted to the shaft in the shop, it is to be presumed that they are true with the shaft.

80. Testing Alinement of Shaft.—In order to make sure that the shaft is exactly at right angles to the center line of the cylinder and that it is also level, the following course

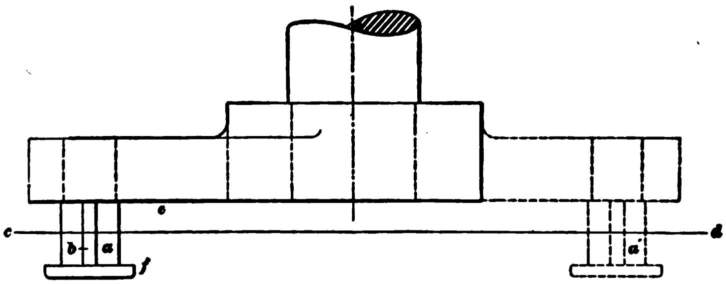


FIG. 5.

may be pursued: The crankpin *a*, Fig. 5, is brought up to the center line *c d* and a piece of wood *b* is fitted between the face of the crank *e* and the head of the crankpin *f*. A mark is made on this piece of wood to coincide with the line *c d*. The shaft is now given a half revolution to bring the crankpin under the line at the other end of its travel, as shown by the dotted lines at *a'*. If the line on the strip of wood *b* again coincides with the center line *c d*, the shaft is at right angles to the center line of the cylinder.

81. Testing Leveling of Shaft.—In order to test the shaft to see whether or not it is level, a fine plumb-line may be suspended vertically before the shaft at the crank end and the crankpin *a* brought into contact with it at the upper half-center and then tested again at the lower half center. If the end of the crankpin just touches the line at both upper and lower half centers, the shaft is horizontal.

82. Lining the Crosshead.—The line *c*, Fig. 1, having been removed, the piston, piston rod, and crosshead are put in place, centering the piston in the cylinder first of all, when the design of the piston is such as to make this adjustment necessary. The next step to take is to ascertain if the center line of the piston coincides with the center line of the cylinder. This may be done as follows: In Fig. 6, let *d* be the upper surface of the guides, which, when properly alined, lies parallel to the center line of the cylinder; the piston having been previously centered, the center line of the piston rod coincides with the center line of the cylinder; therefore, the piston rod should be parallel with the upper surfaces of the guides. This may be readily tested by placing the piston at the forward end of its stroke; then measure downwards from the lower edge of a straightedge laid across the guides at *e* and *f* to the piston rod. If the

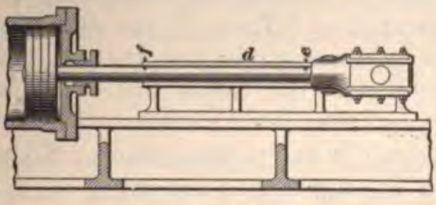


FIG. 6.

two measurements agree, the piston rod is in line; otherwise, the crosshead shoes must be adjusted until the piston rod is in its proper position.

83. Testing the Crankpin.—A method of testing the accuracy of the crankpin is shown in Fig. 7 and it may be put into practice as follows: Connect the connecting-rod to the crankpin and key up the brasses snugly to the pin; then put the crank on or near one of its dead centers, as at c, Fig. 7, and exert a slight pressure against the wristpin end of the connecting-rod in the direction of the arrow x, in order to take up any lost motion in the crankpin brasses. It is understood that the connecting-rod is disconnected from the crosshead, and the latter is pushed back towards the cylinder, so as to be out of the way; then measure the distance a and make a note of it. Put the crank on or near the other dead center c' and measure the distance a'. Then, if the distance a is equal to the distance a', it proves that the center line of the crankpin is parallel to the center line of the shaft in the horizontal plane. But it is necessary that similar conditions should prevail in the vertical plane. To test this, put the crank on the upper half center, as at d, and measure from the wristpin end of the connecting-rod to the inner edge of the guides, as before; then turn the crank to the lower half center d' and measure again from the connecting-rod to the guides. If the measurements agree for both positions of the crank, the crankpin is properly alined. In the figure the crankpin is shown out of line, the connecting-rod then occupying the positions shown in full

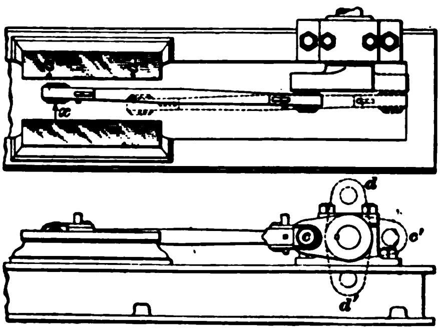


FIG. 7.

and dotted lines, respectively. If the error is very small, it may sometimes be remedied by filing and scraping the crankpin, but if the error is serious, it may require machine work.

84. Testing Bore of Crankpin and Wristpin Brasses. It will generally be advisable to ascertain if the bore of the crankpin brasses is at right angles to the center line of the connecting-rod. This may be done by putting the crank, with the connecting-rod attached to the crank but disconnected from the crosshead, on one of its dead centers, as at *c*, Fig. 7. Then measure the distance from the inner edge of one of the guides to the crosshead end of the connecting-rod, as at *a*, Fig. 7, and make a note of it; then take the rod off the pin and turn the rod half way around on its center line and replace it on the pin. Now measure from *a*, as before; if the two measurements agree, the brasses are correctly bored. To make this test still more satisfactory, the operation may be repeated for several different positions of the crank. The wristpin end of the rod may be tested in like manner; in this case the rod is disconnected from the crankpin and the measurements are taken from the face of the crank or from the collar of the crankpin to the crankpin end of the connecting-rod. If the connecting-rod cannot pass this test satisfactorily, the brasses must be fitted to the crankpin and wristpin by chipping, filing, and scraping until it fills the requirements.

85. Testing Alinement of Wristpin.—The alinement of the wristpin may now be tested, for which purpose the connecting-rod may be used. Key the rod rather snugly to the wristpin, having it disconnected from the crankpin. Then place the crank on or near one of its dead centers and push the crosshead forwards until the end of the connecting-rod just rests on the crankpin. If the center line of the rod is the same distance from both collars of the crankpin, the wristpin is in the correct position in one direction. To prove it for another direction, put the crank on one of its half centers and repeat the above described operation. If the result is the same as before, the wristpin is in its correct position relative to the center lines of the cylinder and shaft.

86. Testing Alinement of Connecting-Rod Brasses in Reference to Each Other.—It still remains to be proved that the center lines of the crankpin brasses and the wristpin brasses lie in the same plane. This is a very necessary condition, because, otherwise, the brasses when keyed tightly to one pin will not fit the other pin, or, as usually expressed, they will bear on one side only. To make this test, a thin coating of Prussian blue or red-lead paint is put on the crankpin and the rod is connected up and adjusted rather snugly to the wristpin and a little less snugly to the crankpin. The crank is then turned through one revolution, when the crankpin brasses may be examined. If they show marking all over, their correct adjustment is assured, but if the coloring matter is rubbed off at either end of the crankpin, it shows that the brasses do not fit the pin and that they must be filed and scraped until they bear equally on all parts of the pin.

87. Order of Operations.—An engineer in lining up his engine or testing his engine for alinement, will do well to perform the various operations in the same order as given here; he should remember that in order to insure correct results, each part of the engine tested *must* be alined before proceeding further. Thus, it is folly to attempt to prove by the methods given here that the center line of the brasses are in the same plane before the correct relative alinement of the wristpin and crankpin are proven.

If a new engine is so far out of line that it cannot readily be adjusted by liners and scraping brasses while the various parts are being assembled, it should be placed in good order by the builder.

POUNDING OF ENGINES.

CAUSES.

88. The causes of pounding in engines are various; they are not always easy to locate in a large engine, owing to the difficulty of locating the exact source of the sounds. These sounds serve as a warning, however, that something is wrong about the machinery, and no time should be lost in ascertaining where and what it is and taking measures to stop it, thereby preventing a possible breakdown.

LOOSE BRASSES.

89. The most frequent cause of pounding in engines is loose journal brasses; the pounding is produced by the journals striking against the sides of the brasses as the cranks are passing the centers and at the instant the change of direction in the motion of the pistons takes place. If the journal-boxes are very slack, the pounding may be so violent as to cause heating of the journal and boxes by the succession of blows they receive; this may especially occur at the crankpins. The remedy for pounding of this nature is obvious. Stop the engine and set up on the brasses gradually, until, after trial, the pounding ceases, taking great care that they are not set up too tight, else they will heat from friction, which may have a more disastrous effect than a moderate amount of pounding. In the case of shaft journals, they may be set up without stopping the engine, provided they can be reached without danger of the engineer being caught in the machinery.

90. It may so happen that the boxes or brasses are worn down until the edges of the upper half and the edges of the lower half are in contact and cannot be set up on the journal any farther; they are then said to be **brass and**

brass, or brass bound. In a case of this kind, the journal must be **stripped,** as it is called, when the cap and brasses are removed from a journal. The edges of the brasses are then chipped or filed off, in order to allow them to be closed in; the amount to be taken off may be determined by trying the brasses on the journal occasionally or by calipering the journal with outside calipers, transferring the measurement to a pair of inside calipers, with which to measure the bore of the brasses as they are being fitted. It is a most excellent plan in practice to reduce the two halves of the brasses so that they will stand off from each other when in place for a distance of $\frac{1}{8}$ inch to $\frac{3}{16}$ inch and fill this space with hard sheet-brass liners, say from 20 to 22 Birmingham wire gauge in thickness each. The object is this: Should the journal become brass bound, the cap may be slacked off and a pair of the liners slipped out without the necessity of stripping the journal, which it is desirable to avoid whenever possible for the reason that it seems to be impossible in practice to put the journal brasses back just where they were before they were disturbed. In large engines it is almost always the case that journals *will* heat after being stripped, and they require a special watch for several days or until they settle down to their proper position relative to the journal.

91. In some instances journal-boxes are fitted with **keepers,** or **chipping pieces,** as they are sometimes called. These consist usually of a cast-brass liner, anywhere from $\frac{1}{4}$ inch to $\frac{1}{2}$ inch in thickness, having ribs or ridges cast on one side, for convenience of chipping and filing. These keepers are sometimes made of hardwood and are capable of being compressed slightly by the pressure exerted upon them during the setting-up process. When the boxes are babbitted, the body of the box is occasionally made of cast iron, in which case iron liners and keepers are used instead of brass ones.

92. The crankpins, being the journals most liable to heat either from pounding or from friction caused by the brasses being set up too tightly, and on account of the comparatively small surface over which the friction is distributed, require the greatest care and need constant watching. The oiling device should be of the best and the oil should never be permitted to stop feeding or the oiling device to get out of order, else there will be trouble.

LOOSE THRUST BEARING.

93. In engines fitted with some types of friction couplings, there is a thrust exerted upon the shaft in the direction of its length. This will necessitate having a **thrust bearing,** or **thrust block,** as it is sometimes called. There is a variety of thrust bearings, but the most common is the collar thrust, which consists of a series of collars on the shaft that fit in corresponding depressions in the bearing. If these collars do not fit in the depressions rather snugly, the shaft will have end play and there probably will be more or less pounding or backlash at every change of load on the engine. This can only be remedied by putting in a new thrust bearing and making a better fit with the shaft collars, unless the rings in the bearing are adjustable, as is sometimes the case, when, of course, the end play may be taken up by adjusting the rings.

WATER IN CYLINDER.

94. Pounding often occurs in the cylinders and is frequently caused by water, due to condensation or carried over from the boilers. This may be a warning that priming is likely to occur in the boilers or has already commenced. The first thing to do at such a time, if the cylinders are not fitted with automatic relief valves, is to open the drain cocks as quickly as possible and to close down the throttle a little to check the priming.

If boilers show a chronic tendency to prime, it is because they are too small for the engine, or they have not steam space enough, or the water may be carried too high in them, which will cause a considerable reduction of the steam space. Unsteady firing, producing great fluctuations in the steam pressure, will also cause both foaming and priming, the result of which is that water will be carried over from the boilers into the cylinders. This is always a source of danger.

Water is non-compressible; therefore, after the clearance space of the cylinder is filled and more water is allowed to enter, if there is no way for it to escape, either the cylinder head will be blown out or the piston broken. Partly closing the stop-valve of the boiler showing a tendency to prime, thereby wiredrawing the steam a little, will generally check priming, if the remedy is applied before the priming becomes violent, after which it is difficult to suppress.

LOOSE PISTON.

95. Another source of pounding in the cylinder is that the piston may be loose on the rod; this is caused by the piston-rod nut or key backing off or the riveting becoming loose, permitting the piston to play back and forth on the piston rod. If due to the nut backing off, the engine should be shut down instantly on its discovery. There is very little room to spare generally between the piston-rod nut and the cylinder head; therefore, it cannot back off very far before it will strike and break the cylinder head. After the engine is stopped and the main stop-valve closed, take off the cylinder head and set up on the piston nut as tightly as possible; there is usually a socket wrench furnished with each engine expressly for this purpose.

96. Although piston-rod nuts seldom work loose and those of vertical engines are less liable to do so than others, still as a measure of safety a taper split pin should in all cases be fitted through the piston rod behind the nut or a setscrew fitted through the nut. If, on examination, this setscrew is found slack, the cause of the nut backing off is thereby explained, and it should be screwed down solid to prevent a recurrence of the trouble.

SLACK FOLLOWER PLATE.

97. A slack **piston follower plate,** or **junk ring,** as it is called by English engineers, will cause pounding in the cylinder. It seldom happens, however, that *all* the follower bolts back out at one time unless they fit very loosely in their sockets, but it is not an infrequent occurrence that one of the follower bolts works itself out altogether and swashes about the cylinder at random. This is a very dangerous condition of affairs, especially in a horizontal engine. If the bolt should get "end on" between the piston and cylinder head, which it surely will sooner or later, either the piston or the cylinder head is bound to be broken. Therefore, if there is any intimation that a follower bolt is adrift in the cylinder, shut down the engine instantly, take off the cylinder head, remove the old bolt, and put in one having a tighter fit.

BROKEN PISTON PACKING.

98. Broken packing rings and broken piston springs will cause a great noise in the cylinder, but it is more of a rattling than a pounding noise, and the sound will easily be recognized by the practiced ear. There is not so much danger of a serious breakdown from these causes as may be supposed, from the fact that the broken pieces are confined within the space between the follower plate and the piston flange. Although they rattle around in the cylinder and make a startling din, they cannot get out or do much harm, aside from causing a leaky piston in the case of the packing rings breaking or possibly slightly scoring the cylinder face. As a matter of course, this should be repaired as soon as possible.

PISTON STRIKING HEADS.

99. There is another source of pounding in the cylinder that is usually confined to old engines; it is produced by the piston striking one or the other cylinder head. One of [illegible] f this is the wearing away of the connecting-rod [illegible] eying up the brasses from time to time has the [illegible] ct of lengthening or shortening the connecting-rod, [illegible] pending on the design, and this change in length destroys [illegible] clearance at one end of the cylinder by an equal amount. [illegible] remedy is to restore the rod to its original length by [illegible] cing sheet-metal liners behind the brasses; this obviously [illegible] ll move the piston back or ahead and restore the clearance. [illegible] may be mentioned here that the length of a connecting-[illegible] d is measured from center to center of the bore of the [illegible] nkpin and wristpin brasses.

It is obvious that the p [illegible] hould never be allowed to [illegible] ike the cylinder head. [illegible] condition generally is not [illegible] ached suddenly; it i [illegible] bout gradually, covering a considerable period of tim [illegible] sufficient to forestall any [illegible] ch occurrence. A rathe [illegible] ase of the piston striking [illegible] cylinder head is due to [illegible] ston rod unscrewing from [illegible] rosshead, in case it is [illegible] d by a thread and check-[illegible] To obviate any dange [illegible] check-nut should be tried [illegible] uently.

[illegible] very reciprocating stea [illegible] ne should have the length o [illegible] the stroke and the clea [illegible] space at each end of the cylinder marked on the gui [illegible]; by this means the relative positions of the piston and cylinder heads can always be seen at a glance.

IMPROPER STEAM DISTRIBUTION.

100. The primary cause of another source of pounding is the improper setting of the steam valve, or possibly its improper design, as it cannot always be accepted as granted that the valve is properly designed. In the case of improper setting of the valve, insufficient compression, insufficient lead, cut-off too early, and late release may all cause pounding on the centers.

101. The manner in which insufficient compression causes pounding may be explained as follows: For practical reasons there must always be some lost motion at the wrist-pin, crankpin, and shaft bearings. Now, in passing the dead centers, the direction of pressure is suddenly reversed, and in consequence the piston rod, connecting-rod, and crank-shaft will be suddenly thrown forwards by the inrushing steam to an extent depending on the lost motion at the pins and shaft bearing. It is this sudden changing of the lost motion from one brass to another, with a violence that may be likened to a blow, that causes an engine to knock in passing the centers when compression is insufficient.

102. The effect of a reversal of pressure is clearly shown in Fig. 8. With the crankpin at *a* and the engine running

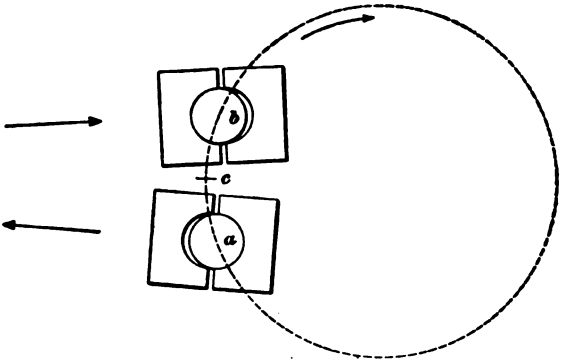


FIG. 8.

over, the connecting-rod is subjected to a pull, but after the crankpin has passed the dead center *c*, the connecting-rod is subjected to a push, in which case the rear brass, as shown at *b*, bears against the crankpin, while in the former case, as shown at *a*, the front brass bears against the crankpin.

By giving a sufficient amount of compression, the lost motion in the pins and journals is transferred gently from one side to the other before the crankpin reaches the dead center, so that by the time the live steam suddenly acts on the piston it cannot throw the rod forwards. If the compression is insufficient to gently take up the lost motion, there will be pounding.

103. Too much compression causes such a great resistance to the motion of the crank that it will tend to slow it down and thus increase the unsteadiness of the engine. Abnormal compression manifests itself by a dull, muffled sound in the cylinder or on an indicator card by the compression line rising above the steam line. It will not cause any pounding at the journals.

104. Insufficient lead is a common cause of pounding; in fact, it is rare to see an indicator card that shows sufficient steam lead. The exact amount of lead to be given to prevent pounding can [illegible] determined by an actual trial; in general, slow-spe[illegible]nes will require less lead than high-speed engines. [illegible]t engines the lead can be readily changed by a pro[illegible]stment of the valve gear. In automatic cut-off, [illegible] engines of the shaft-governor type, however, it is [illegible]ossible, as a general rule, to change the lead by any [illegible]ple adjustment, the lead having been fixed by the [illegible], and a change of it will require an extensive rebuild[illegible] the governor.

105. The reason that i[illegible]ient lead causes an engine to pound is because the pis[illegible]s then little or no cushion to impinge upon as it approaches the end of its stroke, and it is brought to rest with a jerk, as it were. A similar effect will be produced by a late release; the pressure is retained too long on the driving side of the piston. The ideal condition is that the pressures shall be equal on both sides of the piston at a point in its travel just in advance of the opening of the steam port. The position of this point varies with the speed of the piston and other conditions that the indicator card only can reveal; in fact, all conditions dependent on the set of the steam valve can be investigated only by the help of the indicator card. Any departure from the ideal condition above mentioned will produce more or less pounding in an engine.

106. A too early cut-off will expand the steam down too low—even below the back-pressure line sometimes.

This is an abnormal condition, which will cause pounding, and should not be permitted to occur.

A very high vacuum in a condensing engine will sometimes cause pounding by not permitting sufficient cushion for the piston to impinge upon.

POUNDING AT CROSSHEAD.

107. The crosshead is a prolific source of thumping and pounding from various causes, of which the getting loose of the piston rod is one of the most common causes. There are several methods of attaching the piston rod to the crosshead. The rod may pass through the crosshead with a shoulder or taper, or both, on one side of the crosshead and a nut on the other; or the rod may be secured to the crosshead by a cross key, instead of the nut; or the end of the rod may be threaded and screwed into the crosshead, having a check-nut to hold the rod in place. In the first-mentioned case, the nut may work loose, which would cause the crosshead to receive a violent blow, first, by the nut on one side and then by the shoulder or taper on the other at each change of motion of the piston. The remedy is obvious—set up the nut. A similar effect will be produced if the cross key should work loose and back out, the remedy for which is to drive in the key. In the case of the piston rod being screwed into the crosshead and the rod slacking back, the danger is that the piston will strike the rear cylinder head. The check-nut should be closely watched.

108. Another source of pounding at the crosshead is loose wristpin brasses, the remedy for which is to set up on the brasses, but not too tight.

109. In the case of a crosshead working between parallel guides, pounding may be caused by the crosshead being too loose between the guides; in that case the crosshead shoes should be set out.

110. In the case of a slipper crosshead, pounding will result from the wearing down of the shoe, the cure for which is to put a liner between the shoe and the foot of the crosshead or to set it out with whatever means of adjustment are provided.

POUNDING IN AIR PUMP.

111. Pounding in the air pump is generally produced by the slamming of the valves, caused by an undue amount of water in the pump, which will usually relieve itself after a few strokes. The pump piston, however, may be loose on the piston rod or the piston rod may be loose in the crosshead, either of which will cause pounding. A broken valve may also cause pounding in the air pump, all of which must be repaired as soon as detected.

POUNDING IN CIRCULATING PUMP.

112. In a circulating pump of the reciprocating type, pounding may be caused by too much or too little injection water, and the pounding may be stopped by adjusting the injection valve to admit just the right quantity of water. It may so happen, however, that the injection water is very cold, and to admit enough of it to stop the pounding in the circulating pump will make the feedwater too cold. To meet this contingency, should it arise, an air check-valve is often fitted to the circulating pump to admit air into the barrel of the pump as a cushion for the piston; this check-valve may be kept closed, when not needed to admit air, by means of a screw stem above it.

A broken valve, the piston loose on the piston rod, or the piston rod loose in the crosshead will all cause pounding in the circulating pump, the same as in the air pump, and they should all be treated in the same manner as was specified for similar troubles in the air pump.

CONCLUSION.

113. The derangements causing pounding, as well as derangements of machinery in general, produce their own individual sounds, which are easily recognized by the experienced engineer. It is here that the attentive and careful engineer will prove his value, as by taking prompt and judicious action he will prevent a breakdown. He should be able to detect any unusual noise about his engine, though it may be imperceptible to the unpracticed ear. It is almost always the case that any derangement of the parts of an engine will give timely notice by an unusual sound, and if this warning is heeded and promptly acted on by the engineer, a breakdown can generally be prevented. The various sounds produced by an engine while running can be learned only in the engine room by the engineer who is responsible for the proper running of the engine. They cannot be learned in any other way.

114. The engineer, knowing the various causes that produce pounding and thumping in his engine, can prevent them in a great measure by keeping the engine in such good order that they cannot occur.

ENGINE MANAGEMENT.

(PART 2.)

HOT BEARINGS.

CAUSE, PREVENTION, AND CURE.

GENERAL INSTRUCTIONS.

1. Introduction.—Hot bearings are the source of much anxiety and annoyance to the engineer, besides interfering very seriously with the proper performance of the engine.

2. Causes.—The primary causes that lead to the heating of bearings may be enumerated as follows

Newly fitted brasses and journals.
Refitted brasses and journals.
Brasses set up too tightly.
Brasses too loose.
Warped and cracked brasses.
Cut brasses and journals.
Imperfectly fitted brasses.
Brasses pinching the journal at their edges.
Oil feed stopped entirely.
Not enough oil.
Dirty and gritty oils, or oils of bad quality.
Oil squeezed out of the bearings.
Grit from any source in the bearings.

Journal too small, either in diameter or in length.
Overloaded engine.
Engine out of alinement.
ternal heat.
asses fitted too snugly between collars of journal.
Springing of bedplate.
Springing or shifting of pedestal or pillow-block.

3. **Best Form of Bearing.**—The bearing of an engine which the shaft journals run should approximate, as nearly as possible, a hole through ... support. If it were possible, a hole with a bushing of ...itable metal in it would ...rm the best possible beari... ...r a shaft; but since the ...aring, however well de... ...d made, will in course of ...e wear somewhat, it b...omes a necessity that there ...ould be some means of adjusting the brasses, so as to pre...nt the shaft having a side movement when they are worn.

4. **Adjustment of Bearings.**—Some engineers consider ...n error to make bearings adjustable; they say it gives ...portunity for careless men to do mischief through lack ...gment. It is certainly a fact that one of the principal causes of hot bearings is setting them up too tightly. Some persons, as soon as they hear a pound or noise about an engine immediately conclude that some bearing is slack and tighten it up; this propensity is to be deplored. There are numerous other causes of pounding in engines besides slack bearings, and the engineer should be fully convinced that the pound is caused by slack brasses before setting them up. Bearings on an engine that is in line and in good order, if properly adjusted, will run smoothly and noiselessly for months without having to be touched with hammer or wrench, and it should be the object of an engineer to get his engine into that condition as soon as possible and to keep it so.

5. **Watching Bearings.**—Bearings, particularly those of large engines, require constant watching. The engineer or oiler should know at all times the condition of every bearing and oil cup; this will require frequent trips around the engine to examine the oil cups to ascertain if they

are feeding and if they contain sufficient oil and to replenish the oil in the cups whenever necessary. While making his rounds, he should feel with the palm of his hand the brasses of those bearings that have shown a tendency to heat and those that are most liable to heat, particularly the crankpins.

TREATMENT OF HOT BEARINGS.

6. Mixtures for Reducing Friction.—Should any of the bearings show an inclination to heat, as indicated by its temperature rising above blood heat or above the temperature of the surrounding atmosphere, the oil feed should be increased; if the oil does not feed freely, run a wire through the oil tubes. If the bearing continues to get hotter, mix some flake graphite (black lead), flour sulphur, or powdered soapstone with the oil and feed the mixture into the bearing through the oil holes, between the brasses, or wherever else it can be forced in. A little aqua ammonia introduced into a hot bearing will sometimes check heating by converting the oil into soap by saponification, soap being an excellent lubricant. Mineral oils will not saponify.

7. Danger of Increasing Heating.—If, after trying the remedies just mentioned, the bearing continues to grow hotter, say to the extent of scorching the hand or burning the oil, it indicates that the brasses have been expanded by the heat and that they are gripping the journal harder and harder the hotter they get; at this stage, if the engine is not stopped or if the heating is not checked, the condition of the bearing will continue to grow worse as long as the engine is running, and may become so bad as to slow down and eventually stop the engine by excessive friction. By this time the brasses and journal are badly cut and in bad condition generally, and the engine must be laid up for repairs.

8. Remedies for Increasing Heating.—The state of affairs mentioned in Art. **7** should not be permitted to be reached. After the simple remedies given in Art. **6** have been tried and failed to produce the desired result, the engine should be stopped and the cap nuts or key of the hot bearing should be slacked back and the engine allowed to stand until the bearing has cooled off. If necessity requires it, the cooling may be hastened by pouring cold water upon the bearing, though this is objectionable, as it may cause the brasses to warp or crack by unequal contraction. Putting water on a very hot bearing should be resorted to only in an emergency, that is, when an engine *must* be kept running regardless of a spoiled pair of brasses. Water may be used on a moderately hot bearing without doing very much harm. It is quite common in practice, when sprinklers are fitted to an engine, to run a light spray of water on the crankpins when they show a tendency to heat, with very beneficial results.

9. If the engine is not started again until the faulty bearing has become perfectly cool, the cap nuts or key should be set up a little, but not too much, before starting; otherwise, the brasses, having been slacked off, may be too loose, and excessive thumping and pounding will ensue.

10. Dangerous Heating. — Should a bearing become so hot as to scorch the hand or to burn oil before it is discovered or through the necessity of keeping the engine running from some cause, it is imperative that the engine should be stopped, at least long enough to loosen up the brasses, even though it is necessary to start up again immediately, otherwise the brasses will be damaged beyond repair and deep grooves cut into the journals. If the brasses are babbitted, the white metal will melt out of the bearing at this stage. The engine is now disabled, and if there is not a spare set of brasses on hand, it will be inoperative until the old brasses are rebabbitted, if they are worth it, or until a new set is made and fitted. If an attempt is made to rebabbitt a brass while it is in place under the shaft, the chances are that the attempt will result in a failure.

11. Keeping Engine With Hot Bearing Running. If it is absolutely necessary in an emergency to keep the engine running at all hazards while a bearing is very hot, the engineer must exercise his best judgment as to how he shall proceed. After slacking off the brasses, about the best he can do is deluge the inside of the bearing with a mixture of oil and graphite, sulphur, soapstone, etc., and the outside with cold water from buckets, sprinklers, or hose, taking the chances of ruining the brasses and submitting to cutting the journal. Of course, the engine must be stopped as soon as the emergency has passed and the journal then stripped. It is to be expected that the journal will be found to be deeply grooved and the brasses cut and warped. If the brasses were babbitted, most of the white metal will have disappeared and little else but the framework of the brasses will be left. But if the brasses are made of solid composition or bronze, they can be refitted for at least temporary use or until new ones can be procured.

12. Refitting a Cut Bearing.—The wearing surfaces of the brasses and journal must be smoothed off as well as circumstances will permit; but if the grooves are very deeply cut, it will be useless to attempt to work them out entirely, and if the brasses are very much warped or badly cracked, it will be best to put in the spare ones if any are on hand. If not, the old ones must be refitted and used until a new set can be procured, which should be done as soon as possible. As for the journal, it is permanently damaged; temporary repairs can be made by smoothing down the journal and brasses; but at the first opportunity the journal should be turned in a lathe and the brasses properly refitted or be replaced with new ones.

13. After a bearing has once been heated up sufficiently to cut the brasses and journal or to warp or crack the brasses, it is afterwards constantly in danger of heating up again on the slightest provocation; and the engine is thereby rendered unreliable and uncertain in regard to its steady running. No precaution that can be taken to prevent the heating of bearings is too great to be used for the attainment of this end.

CAUSES OF HOT BEARINGS IN DETAIL.

NEWLY FITTED BRASSES AND JOURNALS.

14. Cause of Friction.—The bearings of new engines are particularly liable to heat, due to the wearing surfaces of the brasses and journal having just been machined. Newly worked metal, when viewed through a powerful microscope, presents the appearance of being a mass of fine needle points projecting outwards. When the newly worked surfaces of two pieces of metal are rubbed together under pressure, the needle points of one piece engage with the needle points of the other piece and excessive friction is produced, the result being that the surfaces in contact are cut into grooves, which still further increases the friction; but if the rubbing process is continued in a moderate manner, so that the surfaces in contact do not cut, the needle points will be bent over gradually, each point forming a small hook. Millions of these little hooks side by side form a shell or a hard surface on the rubbing parts, and the needle points can no longer engage with each other, thereby lessening very greatly the danger of heating by friction and eliminating it entirely when properly lubricated.

15. Wearing Down Bearings.—The conditions mentioned in Art. **14** exist with new brasses and the journal of an engine bearing; therefore, if a new engine or one with new brasses is run moderately, in regard to both speed and load, and with rather loose brasses, until the needle points are bent over, there will be little danger of the bearings heating thereafter from this cause if proper attention is given to their adjustment and lubrication. This is what is familiarly termed **wearing down the bearings.** The impression generally conveyed by this expression is that the metal of the brasses and journal is actually worn away; such is not the case, however, as has been explained. If the journal is true and if the brasses are properly fitted to it, there is no necessity for them to be *worn* down; to bend over the needle points is all that is required.

16. Uneven Bearing of Brasses.—Another source of heating of bearings of new engines is the following: For practical reasons there must be a little play between the brasses and their beds; this permits a slight movement of the brasses when pressure is exerted on them by the shaft; and notwithstanding the fact that they may have been most carefully fitted in the shop, they require a certain amount of running to properly adjust and accommodate themselves to their surroundings. This is especially the case with the bearings of large engines, and the same conditions will obtain every time the brasses are removed. It seems almost impossible in practice to put the brasses of a large bearing back again just where they were before removal; it always requires time for them to settle into their old places; therefore, they should not be disturbed unless there is a positive necessity for doing so. The direct cause of the tendency to heat in this instance is that the brasses do not bear evenly on the journal after the several parts of the bearing are assembled. When a bearing runs well, it is not good practice to disturb it; it is better to leave well enough alone.

REFITTED BRASSES AND JOURNALS.

17. The bearings of an engine that has just been thoroughly overhauled and the journals and brasses of which have been refitted are liable to heat. The wearing surfaces of the bearings having been newly worked or machined, the surface of the metal is in the needle stage, and, also, the brasses have not yet had a chance to adjust themselves to the journal and their beds. The engine, therefore, is in about the same condition as a new engine, so far as the bearings are concerned, and should be treated in the same manner, i. e., it should be run moderately, with loose brasses, until the needle points are bent over and a shell has been formed on the wearing surfaces, and until the brasses have accommodated themselves to their surroundings.

BRASSES SET UP TOO TIGHTLY.

18. When the brasses of an engine bearing are set up too tightly, heating is inevitable, and probably more hot bearings result from this cause than any other, and with less excuse. It is often the case that an attempt is made to stop a thump or a pound in an engine by setting up the brasses when the thump could and should be stopped in some other way.

19. The direct cause of heating of bearings when the brasses are set up too tightly is the abnormal friction that is produced by the brasses binding on the journal. The prevention and cure are o[illegible]. The brasses should not be set up too tightly, and if they are, they should be slacked off as soon as possible. As a matter of fact, hot bearings should never occur from [illegible] cause. Only a responsible person should have charge of the bearings and no one else should be permitted to meddle with their adjustment.

BRASSES TOO LOOSE.

20. Bearings may heat on account of the brasses being too loose. The heating is caused by the hammering of the journal against the brasses when the crankpin is passing the dead centers. This derangement is easily remedied, however, by setting up the cap nuts or key. Here the experience and judgment of the engineer is called into play to decide just how much to set up, as it is very easy to overdo the matter and set up too far, with a hot bearing as the result.

21. Most practical engineers have their own particular views regarding the setting up of bearings. One method is to set up the cap nuts or key nearly solid and then slack them back half way; if the brasses are still too loose, they are set up again and slacked back less than before, repeating this operation until the ideal position is reached, that is, when there is neither thumping nor heating. It is important that this desired point be approached very gradually and carefully, else the chances are that it will be overreached and the operation will have to be repeated all over again.

22. Another method of setting up journal brasses is as follows: Fill up the spaces between the brasses with thin metal liners, say from 18 to 22 Birmingham wire gauge in thickness, and a few paper liners for fine adjustment; put in enough of them to cause the brasses to set rather loosely on the journal when the cap nuts or keys are set up solid. Run the engine for a while in that condition and note the effect; then take out a pair of the liners and set up solid again. Repeat this operation until the brasses have reached the ideal point, when there is neither thumping nor heating, and there let them remain as long as they fill the ideal condition. It may require a week or more, and with a large engine longer, to reach the desired point, but it will be all the better to give the needle points time to be bent over and the brasses time to adjust themselves. If this system of treating bearings is carefully carried out, there will be very little danger of their heating. When the proper point is reached, the engine should run a long time without requiring any further adjustment of the bearings. In removing the liners, great care should be exercised not to disturb the brasses any more than is absolutely necessary. A pair of thin, flat-nosed pliers will be found useful in slipping out the liners. This method is preferable to the first one mentioned, because there is not so much danger of setting the brasses up too far.

WARPED AND CRACKED BRASSES.

23. Warped and cracked brasses will cause heating, because they do not bear evenly on the journal, and hence the friction is not distributed over the entire surface, as it should be. The remedy will depend on the extent of the distortion of the brasses. If the distortion is not too great, the brasses may be refitted to the journal by chipping, filing, and scraping; but if they are twisted so much that they cannot, within reasonable limits, be

refitted, nothing will do but new brasses. Warped and cracked brasses are the result of putting water on them while they are very hot, which should be avoided if possible.

CUT BRASSES AND JOURNALS.

24. Brasses and journals that have been hot enough to be cut and grooved are liable to heat up again any time on account of the undue friction produced by the roughness of the wearing surfaces. As long as the grooves in the journal are parallel and match the grooves in the brasses, the friction is not greatly increased; but if a smooth journal is placed between a set of brasses that are grooved and pressure is applied, the journal crushes the grooves in the brasses and becomes brazed or coated with brass, and then the coefficient of friction becomes very high and heating results. The way to prevent heating from this cause is to work the grooves out of the journal and brasses by filing and scraping as soon as possible after they occur.

IMPERFECTLY FITTED BRASSES.

25. Faulty workmanship is a common cause of the heating of crankpins, wristpins, and bearings. The brasses in that case do not bear fairly or sit squarely in their beds, and while they appear all right to the eye, they may not be square in the bearing. A crankpin brass must sit squarely on the end of the connecting-rod and the rod itself must be square. If the key, when driven, forces the brasses to one side or the other and twists the strap on the rod, it will draw the brasses slantwise on the pin and make them bear the hardest on one side or the other, thus reducing the area of the wearing surfaces. The same is true of the shaft bearings. If the brasses do not bed fairly on the bottom of the pillow-block casting or do not go down evenly, without springing in any way, they will not run as they should, and heating will result. Chronic heating of bearings is almost always caused by badly fitting brasses. This is a defect that should be looked for and remedied at once, if found to exist.

BRASSES PINCHING THE JOURNAL AT THEIR EDGES.

26. Brasses, when first heated by abnormal friction, tend to expand along the surface in contact with the journal; this would open the brass and make the bore of larger diameter, if it were not prevented by the cooler part near the outside and by the bedplate itself.

If the brass has become hot quickly and excessively, the resistance to expansion produces a permanent set on the layers of metal near the journal, so that on cooling, the brass closes and grips the journal; it will then set up sufficient friction to heat again and expand sufficiently to ease itself from the journal, and so long as that temperature is maintained the journal runs easily in the bearing. This is why some bearings always run a trifle warm and will not work cool. A continuance of heating and cooling will set up a mechanical action at the middle of the brass, which must eventually end in cracking it, just as a piece of sheet metal is broken by continually bending it backwards and forwards about a certain line.

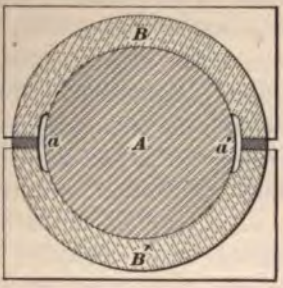


FIG. 1.

27. The cause of heating mentioned in Art. **26** may be prevented by chipping off the brasses at their edges parallel to the journal, as shown at a and a', Fig. 1, in which A is a sectional view of the journal and B, B' represent the top and bottom brasses.

OIL FEED STOPPED.

28. It does not take many minutes for a bearing to get very hot if it is deprived of oil. The two principal causes of a bearing becoming dry are an oil cup that has stopped feeding, either by reason of being empty or by being clogged up from dirt in the oil, and oil holes and oil grooves stopped up with accumulated dirt and gum. Both of these conditions are the direct result of negligence, and their existence can always be prevented by the exercise of reasonable care.

NOT ENOUGH OIL.

29. The effects produced upon a bearing by an insufficient oil supply is similar to that of no oil, only in a lesser degree. Of course it will take longer for a bearing to heat with insufficient oil than with none at all, and the engineer has more time in which to discover and remedy the difficulty. As a rule, however, more oil is used on bearings than is actually necessary, and a waste of oil is the result. A drop of oil at the right time and in the right place is just as good as a quart injudiciously applied. A steady feed, a drop at a time, is what a journal requires.

DIRTY AND GRITTY OILS AND OILS OF BAD QUALITY.

30. Oils containing dirt and grit or deficient in lubricating quality are prolific sources of hot bearings; but it is within the province and power of the engineer to guard against such causes. There is a great deal of dirt in lubricating oils of the average quality, as engineers find who strain it; therefore, all oil should be strained through a cloth or filtered, no matter how clear it looks. All oil cups, oil cans, and oil tubes and channels should be thoroughly cleaned out frequently. Oil may be removed from the cups by means of an oil syringe, with which every engine room should be supplied. All oil removed from the cups and cans should be strained or filtered before using. If the above instructions are strictly followed, all danger of bearings heating from the use of dirty and gritty oils will be eliminated.

31. Bearings heating from the use of oils of bad quality are not so easily disposed of, however; there is such a great variety of lubricating oils on the market whose quality cannot be definitively decided upon without an actual trial that it is a difficult matter to avoid getting a bad lot of oil sometimes. About the only safe way to meet this trouble is to pay a fair price to a reputable dealer for oil that is known to be of good quality, unless the purchaser is an expert in oils. Cheap combination oils, generally speaking, are very deficient in lubricating qualities and hence should be avoided, as also should gummy oils, which choke up the oil channels and glue the brasses and journals together over night.

32. Brasses of very large bearings are often cored out hollow for the circulation of water through them, which assists very materially in keeping them cool.

OIL SQUEEZED OUT OF BEARINGS.

33. Bearings carrying very heavy shafts sometimes refuse to take the oil, or if they do it is squeezed out at the ends of the brasses or through the oil holes, when the journal will run dry and heat. The great weight of the shaft causes the journal to hug the bottom brass so closely that the oil cannot penetrate between them, or, if it does, it is immediately rejected. Large journals require oil of a high degree of viscosity, or heavy oil, as it is popularly called. Oil of this character has more difficulty in working its way under a heavy shaft than a thin oil has, but thin oil has not the body necessary to lubricate a large journal.

This difficulty may be met by chipping oil grooves or channels in the brasses. A round-nosed cape chisel, slightly curved, is generally used for this purpose, taking care to smooth off the burrs made by the chisel; a steel scraper or the point of a flat file will do this. The grooves are usually cut into the brass in the form of a **V** if the engine is required to run only in one direction; if it is to run in both directions, the grooves should form an **X**. In the first instance care must be taken that the **V** is forward of the direction of the rotation of the shaft; that is, the grooves should spread out from their junction in the same direction as that in which the journal turns. The oil grooves may be about $\frac{1}{4}$ inch wide and $\frac{1}{8}$ inch deep and semicircular in cross-section.

GRIT IN BEARINGS FROM ANY SOURCE.

34. Grit is an endless and ever-present source of heating of bearings; it is only by persistent effort on the part of the engineer that he can keep his machinery running cool in a dirty atmosphere. Experience is the best instructor in this matter. The causes of this condition are innumerable, therefore, it is only possible to mention a few of them here. The machinery of coal breakers, stone crushers, and kindred industries is especially liable to be affected in this way. Work done on a floor over an engine shakes dirt down upon it at some time or other; all floors over engines should be made absolutely dust-proof by laying paper between the planks to prevent this. A prolific cause of hot bearings from grit, if the engine room and firerooms communicate, is carelessness in wetting down the ashes and clinkers. If piles of red-hot clinkers and ashes are deluged with buckets of water, which is the common practice, the water is instantly converted into a large volume of steam that rises with a leap, carrying with it large quantities of small particles of ashes and grit that penetrate into every nook and cranny to which it has access, and it will find its way into the bearings sooner or later. Throwing large quantities of water on the hot clinkers and ashes should be stopped; sprinkle them instead and close the fireroom door while the ashes and clinkers are being hauled or wet down or while the fires are being cleaned or hauled.

35. If emery, emery cloth, Bath brick, or other gritty cleaning material is used around a bearing, it is sure to get inside and cause trouble; it is, therefore, better not to use them in too close proximity to a bearing.

36. As a precaution against grit getting into a bearing, all open oil holes should be plugged with wooden plugs or bits of clean cotton waste as soon as possible after the engine is stopped, and should be kept closed until ready to oil the engine again preparatory to starting up. Plaited hemp or cotton gaskets should also be laid over the crevices between the ends of the brasses and the collars of the journals of every bearing on the engine and kept there while the engine is standing still.

37. Bearings are now in use that, it is claimed by their makers, are dust-proof, but their use does not relieve the engineer from the responsibility of taking every precaution possible to keep grit and dirt out of the bearings of his engine.

JOURNALS THAT ARE TOO SMALL.

38. Journals that have insufficient superficial area of wearing surface will heat. In practice only a certain amount of pressure *per square inch* of area can be sustained by a bearing before the friction reaches the point that will cause heating.

The pressure that a bearing will sustain *per square inch* of area of rubbing surface without heating depends on the materials of which the journal and brasses are composed, the fineness of their finish, the accuracy of their fit, the adjustment of the brasses, and the lubricant used.

39. Pressure and friction have a direct relation to each other. Less friction is produced per square inch of surface by a long journal than by a short one of equal diameter with the same total pressure; therefore, a long journal is not nearly so liable to heat as a short one of the same diameter, and a journal of large diameter is not so liable to heat as one of small diameter of equal length. It is the aim of the designer to so proportion the journal that the pressure or friction will not exceed the practical limit that the bearing will sustain. The *total* amount of friction of two bodies in contact depends on the pressure of the one on the other and is nearly independent of the area of the surfaces in contact, hence the necessity of engine journals being large enough to distribute the friction over a sufficient area of surface.

40. There is only one cure for a bearing that heats constantly on account of being too small. This is to make it larger if circumstances permit it to be done. If this is impossible, the best of lubricant must be used, and if necessary, water must be run constantly on the bearing. It is a good idea to have a set of spare brasses in readiness for an emergency.

OVERLOADED ENGINES.

41. The effect produced by overloading an engine is cisely similar to that of the journals being too small. he pressure on the brasses being increased to a point beyond that for which they were designed, the friction exceeds the actical limit and the bearing heats. The only thing to do remedy this difficulty is to reduce the load on the engine within the amount it was intended to stand.

42. In the case of an engine being run at or near its limit ce, or if the journals are too small, especially if a should be incurred by the machinery being shut while new brasses are being made and fitted, it would a wise and economical precaution to have a complete set spare brasses, especially if the brasses are babbitted, on hand ready to slip in when the fatal moment arrives, as it surely will.

ENGINE OUT OF LINE.

43. If an engine is not in line, the brasses do not bear fairly upon the journals. This will reduce the area of the wearing surfaces in contact to such an extent that the friction is in excess of the practical limit, which necessarily will cause heating. If the engine is not very greatly out of line, matters may be considerably improved by refitting the brasses by filing and scraping down the parts of the brasses that bear most heavily on the journal. If this does not answer, the heating will continue until the engine is lined up.

44. The crosshead guides of an engine out of line are apt to heat, and they will continue to give trouble until the defect is remedied. The guides may also heat from other causes; for instance, the gibs may be set up or lined up too much. Of course, if such is the case, they should be slacked off. The danger of guides heating may be very much lessened by chipping zigzag oil grooves in their wearing surfaces and by attaching to the crosshead oil wipers, made of cotton lamp wicking arranged so as to dip into oil reservoirs at each end of guides if they are horizontal, and at the lower end if they are vertical. These wipers will spread a flim of oil over the guides at every stroke of the crosshead, which will keep them well lubricated.

EXTERNAL HEAT.

45. Bearings may get hot by the application of external heat. This may be the case if the engine is placed too near furnaces or an uncovered boiler, or in an atmosphere heated by uncovered steam pipes or other means. The excessive heat of the atmosphere will then expand the brasses until they nip the journals, which will generate additional heat and cause further expansion of the brasses, and so on until a hot bearing is the result.

46. If the engine is placed close enough to a furnace to cause heating from that source, a tight partition should be put up, if possible; this will also prevent dirt and grit from the fireroom getting into the bearings. If the boilers, steam pipes, and cylinders are unclothed, they should be covered with some good non-conducting material; and possibly a ventilating fan could be rigged up to advantage. Other remedies depend on the conditions and require the judgment of the engineer.

BRASSES TOO LONG.

47. If the brasses are too long and bear against the collars of the journal when cold, they will most surely heat after the engine has been running a while; it is hardly possible to run bearings stone cold, they *will* warm up a little and the b[illegible]ses will be expanded thereby, which will cause them to b[illegible] still harder against the collars. This, in turn, will induce greater friction and more expansion of the brasses.

48. The evil may be obviated by chipping or filing a little off each end of the brasses until they cease to bear against the collars while running. A little side play is a good thing for another reason, which is that it promotes a better distribution of the oil and prevents the journal and brasses wearing into concentric parallel grooves.

SPRINGING [illegible]EDPLATE.

49. If the bedplate of a[illegible]ne is not rigid enough to resist the vibration of [illegible]g parts, or if it is sprung [illegible]om the uneven setting [illegible]tability of the foundation, [illegible]e engine will be thrown ou[illegible]ne either intermittently or [illegible]ently, and the bear[illegible]will heat from the causes [illegible] conditions mentioned in A[illegible] 43 and 44; but it will do [illegible] good to refit the brasses u[illegible] the engine bed is stiffened [illegible] some way and leveled up. [illegible]e form of the bedplate and the surrounding conditions generally must suggest the best way to meet this difficulty.

SPRINGING OR SHIFTING OF PEDESTAL OR PILLOW-BLOCK.

50. The effect of the springing or shifting of the pedestal or pillow-block is similar to the springing of the engine bed; that is, the bearing will be thrown out of line, with the consequent danger of heating. As the pedestal is usually adjustable, it is an easy matter to readjust it, after which the holding-down bolts should be screwed down hard. This is one of the few instances where it is permissible for the engineer to put his strength on the wrench. As a rule, a nut or bolt should be set up just solid; with very rare exceptions, a sledge hammer should never be used in driving a wrench, as 3-inch steel bolts have been broken in this way. It is also very bad practice to drive a nut up with cold chisel and hammer, unless the nut is in a position that it is impossible to reach it with a wrench.

If a pedestal is not stiff enough to resist the strains upon it and it springs, measures should be taken to stiffen it. The method to be used can only be determined on the spot and calls for the exercise of judgment on the part of the engineer.

LUBRICANTS, LUBRICATION, AND LUBRICATORS.

LUBRICANTS.

INTRODUCTION.

51. Classification.—Lubricants may be divided into three distinct classes, viz., *animal*, *vegetable*, and *mineral*. The name of each designates its origin. There are also lubricants that are composed of a combination of two or more of the above primary classes, which practically form another, or fourth, class.

52. Properties.—The origin of lubricants is not so important to the engineer as their lubricative properties and their power to resist decomposition, vaporization, and combustion by the application of heat; especially is this the case with cylinder oils and valve oils. The value of a lubricant depends on the amount of greasy particles it contains, or its **viscosity.** Other desirable features of a good lubricant are: It should reduce friction to a minimum. It should be free from acids and free alkalies, or, in other words, it should be neutral, and of uniform constituency. It should not become gummy, rancid, or otherwise altered by exposure to the air and it should be odorless. It should stand a low temperature without solidifying or depositing solid matter. It should be entirely free from grit and all foreign matter. It should be especially adapted to the conditions as to speed and pressure of the rubbing surfaces on which it is to be used; the question of cost is also a consideration. All first-class lubricants possess these properties to a greater or less degree, and each of them is adapted to its own particular class of work. They are also of all degrees of fluidity and solidity—from the thin, light oil used for oiling the indicator down to the thick oils and through the greases to graphite and soapstone for the heaviest journals.

53. Thick or heavy oils [illegible] nerally considered to rank the highest in viscosity; this is not always the case, however. Some oils of high [illegible] c gravity rank lower in [illegible] cosity than others of a [illegible] specific gravity, hence the [illegible] bricative qualities of an [illegible] cannot always be judged [illegible] curately by either its visc[illegible] or specific gravity. Then, [illegible] ferent manufacture[illegible] lubricants have different [illegible] and names for presumedly the same grade of oil. [illegible] nermore, lubricants that [illegible] y be very satisfactory for [illegible] vy journals might not do [illegible] all for light journals, and [illegible] se that answer well for j[illegible] nals and guides would be very objectionable in cylinders and steam chests—all of which goes to show that different lubricants are required for different purposes.

54. Selection of Lubricants. — Though there are numerous tests for determining the various properties and qualities of lubricants, they, as a rule, involve the use of elaborate chemical apparatus and complicated and delicate machines that are entirely beyond the reach of the average engineer in ordinary engine-room practice. Even the reliability of these elaborate tests is questioned and they are a source of dispute between experts. Under these circumstances it is not an easy task to instruct an inexperienced person how to select a lubricant best suited to his particular needs or to enable him to detect adulterants. Some of the simpler tests will be given further on.

55. In a general way, about the best that an engineer who is not an expert judge of lubricants can do is to procure *from a reputable dealer* several samples of oil or grease that in his judgment are best suited to the machinery he has in charge, taking care to select light-bodied oils for light machinery and to grade his selections down accordingly to suit the size and weight of the journals and the work they have to do. Then he should run the machinery for a stated length of time with each oil, carefully noting the results obtained by each. By the time the engineer has reached the end of his experiments with this assortment of oils, he will have discovered by development and observation which is the best one for his purpose, and it will then only be a matter of common sense to hold on to *that* one until he has good reasons to believe that he can get a better one; then, taking the last one as a standard, he might try another lot of *well-recommended* samples in the same way as before, and so on until he finds the best one for his purpose that the market affords, and at the same time he acquires valuable experience with lubricants. It is important, however, that he should confine his experiments to well-known standard brands of lubricants only, otherwise he will waste much valuable time without gaining a corresponding benefit, and when he finds an oil that, after a fair trial, is satisfactory, he should use it and no other.

56. In selecting the samples for trial, the engineer should examine them very carefully in every possible way and compare one with the other; he should note their color and transparency; rub some of each between the fingers and thumb or on the palm of the hand; note if the sample is smooth and oily and contains no grit; pour a few drops on a sheet of tin or a piece of glass and hold the tin or glass at different angles and note how it flows and if it leaves any residue or gum in its track; examine it with a strong magnifying glass for foreign substances; smell it, and if it is rancid or has a very offensive odor, reject it. If the engineer persists in this practice, it will not take him long to learn how to distinguish between the different grades and qualities of lubricants, which will enable him to select the one that will best serve his purpose and, at the same time, add very greatly to his general store of engineering knowledge, thereby enhancing the value of his services.

FLUID LUBRICANTS.

57. Animal Oils.—Animal oils are derived from the fats of animals and fish. Those that are used as lubricants are: *Lard oil, tallow oil, neatsfoot oil, horse-fat oil, sperm oil, whale oil, porpoise oil, seal oil, shark oil.*

Of animal oils, pure lard oil takes the lead as a lubricant for ordinary machinery; but it has the disadvantage of con- ling in cold weather; to ward against chilling to a cer- ain extent, the winter strained oil only should be used when is exposed to a temperature sufficiently low to congeal the linary oil.

low oil is very similar to lard oil.

rse fat is sometimes used in place of tallow, but its r is offensive.

larified neatsfoot oil is an excellent lubricant for light machinery.

Of the fish oils, sperm is the best lubricant, but its scarcity and high price precludes its general use. It is used, principally in a refined state, for oiling indicators and other delicate mechanisms. Whale and porpoise oils are sometimes used in place of sperm oil, but they are inferior to it. Seal and shark's liver oils are used as adulterants. Menhaden fish oil should not be used as a lubricant, as it quickly turns rancid and gums.

58. Vegetable Oils.—Vegetable oils are derived from the fruits, seeds, and nuts of trees and plants. They compare very favorably with animal oils as lubricants, and several of them are excellent for that purpose. The leading vegetable oils that are used for lubricating purposes are: *Olive oil, rape-seed oil, colza oil, cottonseed oil, castor oil, palm oil.*

The olive oil is probably the leading vegetable oil used for lubricating machinery, but all the others in the above list are fairly good for that purpose. Castor oil and cottonseed oils are more liable to gum than pure olive oil. Linseed oil, either raw or boiled, should not be used as a lubricant; it dries quickly and is very gummy. Cocoanut oil (palm oil) soon becomes rancid and in that condition it is not a good lubricant.

59. Mineral Oils.—Mineral lubricating oils are distilled from bituminous shale and from the residuum of crude petroleum after the volatile oils and illuminating oils have been distilled off at various temperatures up to 572° F. The products of the petroleum still, when heated to temperatures above 572° F., are the lubricating oils. These oils are graded according to their specific gravities and are named as follows:

PROPERTIES OF MINERAL OILS.

No.	Name.	Specific Gravity.	Flashing Point.	Burning Point.
1	Solar oil...........	.860 to .880		
2	Mixed oil..........	.880 to .890	370° F.	435° F.
3	Spindle oil, No. 1 ..	.895 to .900		
4	Spindle oil, No. 2 ..	.900 to .906	394° F.	468° F.
5	Machine oil, No. 1..	.906 to .910	426° F.	487° F.
6	Machine oil, No. 2..	.910 to .915		
7	Cylinder oil, pale...	.915 to .920	441° F.	525° F.
8	Cylinder oil, dark..	.920 to .950	and up.	and up.
9	Vulcan oils.........	.910 to .960		

60. Besides the oils given in the table, there are many other mineral lubricating oils on sale under different names, each manufacturer naming his own product to suit himself, but the above list will serve to show the method of grading mineral machine oils in regard to their specific gravities and their flashing and burning points.

All the mineral oils given in the table, if pure, are excellent lubricants, each one being adapted to its specific purpose for light, medium, and heavy machinery and cylinders.

The color of a mineral lubricating oil is not always an indication of its purity or value. A dark-colored oil may be purer than a light-colored one; therefore, in selecting a mineral oil, too much stress should not be laid upon its color.

61. Compounded Oils.—There is a great variety of compounded oils manufactured for all sorts of purposes and at all prices. They are, generally speaking, simply made to sell without regard to merit or value as lubricants. Herein lies the danger of being defrauded in purchasing cheap oils. They are, as a rule, compounded of thin, light ls, which lack the viscosity, or body, for lubrication, and a riety of substances to produce an artificial body that adds hing to their lubricative properties. Most, if not all, of adulterants used for this purpose are of a gummy ure and enemies to good lubrication. If mineral oils are d as the bases of these compounded oils, they are liable to have a low flashing point, which renders them totally unfit for use in cylinders. In fact, these oils had better be entirely ignored by the engineer; but as they are made and doctored to imitate the pure standard oils, they are well calculated to deceive the unwary, as it is not an easy matter to detect the difference between them by mere inspection.

62. A trial on the engine is the best method to test the merit of a lubricant, though some simple tests, as described under the heading "Tests of Lubricants," may be made with beneficial results.

63. Economy.—Lubricants, like everything else that is exposed for barter or sale, are worth just about what is paid for them. A good article must always fetch its price and a poor article is sold cheaply.

There is no economy in buying cheap lubricants; they cost less per gallon, but it takes more gallons to do the required work. Now that excellent oil filters are to be had, enabling the drip oil to be filtered and used over again, there is no necessity for using cheap oils.

64. Greases.—Greases are divided into three classes, viz., *compounded*, "*set*" or *axle*, *boiled* or "*cup*."

65. Compounded greases are made by mixing cheap oils with fats, paraffin, and the various waxes. They soon become rancid, in which state they are unfit for lubrication, being instead friction producers. It is hardly necessary to say that the engineer should avoid these greases, even though they are cheap.

66. Set, or axle, greases are mixtures of low-grade oils and fats converted into grease by the application of lime. They are cheap greases, used principally for lubricating axles of vehicles and the like, and are familiarly known as **cart grease.** These greases are unfit to use in the bearings of engines.

67. Boiled, or cup, greases are those that are well adapted for engine lubrication. They are produced chemically and are not simply mechanical mixtures as are the others. They are perfectly neutral and will remain so indefinitely. They are made by saponifying fats and fatty oils with lime and dissolving the soap in mineral oil.

68. Soaps made by the use of soda or potash are soluble in water, while soaps made by the use of lime are insoluble in water.

There is a series of greases in this class that are made by saponifying the fats and fatty oils by means of caustic soda; the soaps thus made are soluble in water. These greases are good lubricants if properly made, but they are apt to contain either an excess of alkali or an excess of acid; in either case they are liable to be injurious to the bearings. Free acids or alkalies may be detected by the litmus-paper test.

69. Cup and engine greases include: Nos. 1 to 4 cup greases, Nos. 1 to 3 Albany greases, sponge greases, crank-pin greases, gear greases, lubricating packing, plumbago and graphite greases.

SOLID LUBRICANTS.

70. The solid lubricants are: *graphite*, *soapstone*, *sulphur*, *mica*, *metaline*. These may properly be classed under the general head of mineral lub[illegible]s.

71. **Graphite,** called also [illegible]mbago and **black lead,** is used for lubrication either [illegible] form of a powder, flaked, compressed into bushings, [illegible] being mixed with wood fiber and solidified in molds [illegible]essure; this latter is called **fiber graphite.** After bei[illegible]moved from the molds, the [illegible]s are thoroughly dried a[illegible]en saturated with a drying [illegible]after which they are ex[illegible]to a current of hot dry air to oxidize the oil and to har[illegible]e mass. When hard they may be worked the same [illegible]etal. Fiber graphite is claimed to be self-lubricatin[illegible]

72. The powdered and flake graphite are used to mix with greases for heavy journals and also to mix with the ordinary engine oils to cool a hot bearing. When graphite is used as a lubricant, the journal becomes covered with a thin coating of graphite, which reduces friction to a minimum.

73. **Soapstone, sulphur,** and **mica,** in the form of powder, are sometimes mixed with oils and greases to improve their lubricating qualities for heavy and hot journals. Sheets of mica pressed together and held firmly in a casing have been used instead of brasses with fair success.

74. **Metaline** consists of small cylinders of graphite fitted into holes drilled in the surface of the bearing; it is said to require no other lubrication.

LUBRICATION.

75. The object of lubricating the bearings of an engine is to reduce the friction of those parts that rub against one another to a minimum and to prevent the rubbing surfaces becoming hot, which, if the rubbing is continued without lubrication, will ultimately cause seizing, thereby permanently damaging the bearings and rendering the engine inoperative. The lubricant attains its object by interposing itself in the form of a thin film between the rubbing surfaces, either by gravity or pressure, and thus prevents the rubbing surfaces coming into direct contact with one another.

76. Animal and vegetables oils have been used as lubricants for many years, but since the introduction of multiple-expansion engines and high steam pressures, mineral oils have come into very general use, especially for lubricating pistons and slide valves, for the reason that mineral lubricating oils are not carbonized by high-pressure steam as readily as are animal or vegetable oils. Moreover, animal and vegetable oils (called **fatty** oils to distinguish them from mineral, or **hydrocarbon,** oils) are decomposed by the great heat of high-pressure steam and form stearic, palmetic, and oleic acids. These acids when hot readily attack iron, steel, copper, and its alloys; therefore, cylinders, pistons, etc. are eaten away when fatty oils are used for lubricating them.

77. The acids formed by the decomposition of fatty oils are particularly destructive to steam boilers when the exhaust steam is condensed and used as feedwater, as is the case with condensing engines having surface condensers. On the other hand, mineral oils are not affected by alkalies, therefore the old method of saponifying the grease in boilers and surface condensers by boiling them out with soda or potash is ineffectual when mineral oils are used in the cylinders; in that case, if the boiler tubes or condenser tubes become coated with grease, it must be removed by hand. It is far better, however, to keep the grease out of the condenser and boilers entirely by placing an efficient grease extractor in the exhaust pipe between the low-pressure cylinder and the condenser.

78. High-grade cylinder oils only should be used for lubricating pistons and slide valves, and the flashing point should not be lower than 400° F. The higher the temperature of a hot bearing, the less is the lubricating power of the oil or grease used; consequently, a lubricant that may be thoroughly efficient at ordinary temperatures may be ineffectual in reducing the friction of a bearing that has suddenly become heated; hence the practice of mixing graphite, flour sulphur, etc. with the oil to increase its body and lubricative properties.

TESTS [illegible]ICANTS.

[illegible]S.

In giving [illegible] mple tests of lubricants, no pretence [illegible] absolutely accurate in the [illegible] of determining t[illegible] l values of lubricating oils [illegible] greases, but they [illegible] purposes of the engineer very well and assist h[illegible] the selection of his lubricants; besides, they have the merit of being within the reach of every engineer in his ordinary engine-room practice.

80. The pieces of apparatus required to make these tests are few and inexpensive. They consist of an ordinary tin or iron pan 8 or 10 inches in diameter and 3 or 4 inches deep; a metal cup about the size and shape of an ordinary tumbler; a high-grade thermometer that will measure at least 500° F.; a couple of quarts of clean white sand; a half-dozen clear, white glass ½-pint bottles; a large sheet of tin or plate of glass, preferably the latter; a sheet each of red and blue litmus paper; a common thermometer; a quart of gasoline; a few pounds of ice; a pound or two of rock salt and about the same quantity of sal soda (washing soda); a small iron boiler or saucepan; a small quantity of caustic soda or concentrated lye; a pane of glass painted black on one side with a mixture of shellac varnish and lampblack; and a small tin funnel.

81. Test for Acids and Alkalies.—Dissolve a small quantity, say a teaspoonful, of the oil or grease to be tested in five or six times its bulk of boiling water, in which steep a piece of red litmus paper; if the litmus paper remains red after having been soaked in the mixture for a considerable length of time, the oil or grease is *acid*. If the color of the paper turns to dark blue quickly, the oil is *alkali*. If it changes color very gradually to a light blue, the oil is *neutral*. As a check on the above test, try the mixture with a piece of blue litmus paper in the same way. If the color of the paper does not change, but remains dark blue, the oil is alkali. If the paper turns red quickly, the mixture is acid; but if the paper changes very gradually to a pale red, the solution is neutral.

82. Test for Viscosity.—Pour a few drops of each sample of oil upon the large sheet of tin or glass while the sheet is perfectly level, then raise one end of the sheet gently about 1 inch and support it in that position; watch the race of the drops of oil down the inclined plane. The oil that reaches the bottom of the plane *last* ranks highest in viscosity. Of course, this is only a comparative test, but it will enable the operator to select the oil best adapted to his purpose from a number of samples. After making a selection, it would be well to try the precipitation test given in Art. **87** on it for artificial viscosity.

83. Greases cannot be tested for viscosity in the way described in Art. **82**; about the only convenient method for the engineer to do this is by rubbing some of the grease between his fingers and thumb or in the palm of the hand, noting the result. After some practice he will be able to judge approximately the viscosity of the sample.

84. Flashing and Burning Tests.—Pour some of the oil that is to be tested into the metal cup until it is nearly full; place the cup in the pan and surround the cup with sand until the pan is filled with it; place the pan and contents on a hot stove, over a gas jet, or in any other convenient place for heating it; immerse the bulb end of the high-grade thermometer in the oil in the cup and watch the rise in temperature; when it reaches 300° pass a lighted match slowly across the top of the cup; repeat this every two or three degrees rise in temperature until the vapor arising from the oil ignites with a flash, then note the temperature as indicated by the thermometer; it is the **flashing point.** Continue the test until the oil ignites and burns on the surface. When that occurs the reading of the thermometer gives the **burning point.**

85. The Cold Test.—Partly fill the metal cup with a sample of oil; place the cup in the pan; fill the pan around the cup with cracked ice mixed with rock salt and sal soda; cover the apparatus over with a piece of bagging or blanket and keep it covered until the oil in the cup is congealed; then remove the freezing mixture from the pan and fill the pan with hot water; when the oil in the cup commences to melt, immerse the bulb of a thermometer into it and note the temperature; it is the **congealing point.**

86. Saponification Test.—If it is desired to ascertain if animal or vegetable oils are mixed with oil that is represented to be pure mineral oil, it may be determined as follows: Place about a pint of the oil into the small iron boiler or saucepan and add 1 or 2 ounces of caustic soda or concentrated lye; boil the mixtnre for ½ hour and then set it aside to cool. A tablespoonful of chloride of sodium (common salt) thrown into the mixture while cooling will hasten the process. When thoroughly cool, examine the mixture; if the surface is covered with soap, the oil contains animal or vegetable fats; otherwise it is pure mineral oil.

87. Precipitation Test.—The precipitation test is for the purpose of ascertaining if the oil contains paraffin, waxes, gums, etc. Place an ounce of each of the oils in a separate ½-pint bottle, pour 2 ounces of gasoline into each bottle on top of the oil, and shake the bottles until the oil is dissolved by the gasoline; then allow the mixtures to settle. If there is any considerable amount of precipitation or sediment in any of the bottles, it indicates that the oil in them has been treated to produce artificial viscosity and should be rejected.

88. Test for Mineral Oil Mixed With Fatty Oils.—The presence of mineral oil when mixed with animal or vegetable oils may be detected by pouring a drop of the suspected oil upon the sheet of blackened glass and holding the glass at various angles to the light; if it shows rainbow colors, it contains mineral oil.

89. Test to Detect Sulphur in Mineral Oils.—Heat a small portion of the oil to 300° F. in the metal cup and pan of sand and maintain that temperature for about 15 minutes; after cooling, if the sample is considerably darker in color than the original oil, it is unfit to use in cylinders or on hot bearings.

AUTOMATIC LUBRICATORS.

CLASSIFICATION.

90. The devices used for the automatic lubrication of steam engines and similar machinery may, in accordance with their purpose, be divided into two general classes, *bearing lubricators* and *steam lubricators*.

91. A **bearing lubricator** may be defined as one intended for, and only applicable to, the lubrication of bearings. This class is divided into three subclasses, *plain* and *sight-feed* lubricators, and *grease cups*. **Plain** and **sight-feed** bearing lubricators are intended and can only be used for oil; **grease cups**, as implied by the name, are built to use grease.

92. Steam lubricators are intended for the lubrication of the moving parts in contact with the steam; they may be divided into *mechanical*, *water-displacement*, and *hydrostatic* lubricators. A **mechanical steam lubricator** generally has the form of a force pump; it may be operated by hand, n which case its action is intermittent. A hand-operated echanical steam lubricator is generally fitted only as an i icy device, to be used when the automatic lubricator order. When a mechanical lubricator is operated usly by some moving part of the engine, its action automatic. **Water-displacement lubricators** depend for their action on conden of steam in the reservoir containing the oil; the latte g lighter than water floats on top and overflows into a able passage as the water in the bottom of the reser increases. **Hydrostatic lubricators** depend for peration on the pressure generated by a head of water urnished by condensation of steam.

BEARING LUBRICATORS.

93. A **plain lubricator** is the simplest form of a device for automatic lubrication; it generally takes the form shown in Fig. 2. It consists of a body *a* fitted with a central tube *b* and a removable cover *c*. The oil contained in the body *a* is led into the central tube by capillary attraction, a few strands of lamp wick, as *d*, carrying the oil over. The advantage of this oiling device is its simplicity; the disadvantages are its unreliability and its lack of adjustment of the oil feed. The latter can be adjusted to some degree by changing the number of strands of lamp wick; as the flow of oil is not in plain sight, however, there is always some doubt about the action of the lubricator.

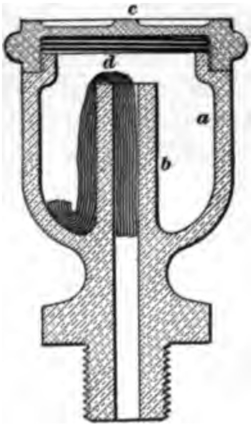


FIG. 2.

94. A **sight-feed bearing lubricator,** as implied by the name, has the oil feed in plain sight. The oil generally is fed by gravity, flowing through an annular opening in the base of the lubricator. The general appearance of this device is shown in Fig. 3. It consists of a glass oil reservoir *a* having a central tube *b* with a valve seat inside of it and at its lower end. A valve *c*, which can be locked in any position by the locknut *d*, serves to regulate the flow of the oil. The oil enters through the hole shown in the lower end of the tube *b*. The drops of oil issuing from the tube *b* show plainly in the sight-feed glass *e*. The upper cover has a hole in it through which the reservoir is filled; a movable cover *f* serves to keep out the dust.

FIG. 3.

95. Various attachments are used for conveying the oil from a stationary sight-feed lubricator to the moving parts, as the crankpin, eccentric, and wristpin. Fig. 4 shows how the oil may be carried to the crankpin by a so-called centrifugal oiling device. The oil from the lubricator *a* flows through the pipe *b* into the ring *c*, which connects to a hole drilled in the center of the crankpin through the fixture *d* that is fastened to the crankpin. The oil entering at *c* passes to the crankpin by the centrifugal force generated by the revolution of the crank and through radial holes out of the crankpin between the surface of the crankpin and the brasses. The main bearing simply carries the stationary lubricator *e*, which discharges directly into the bearing. A separate lubricator *f* may be fitted for the eccentric, discharging into a long trough or funnel *g* fastened to the eccentric strap.

96. Fig. 5 will serve as a suggestion of how automatic lubrication of the wristpin and guides may be obtained. To lubricate the upper guide, the stationary lubricator *a* is used;

a lubricator *b* is placed at a sufficient distance above the level of the lower guide to cause the oil to flow through the channels shown to the guide. To lubricate the wristpin from a stationary cup *c*, a wiping device *d* is attached to the wristpin. This carries the wiper *e*, which is adjusted so as

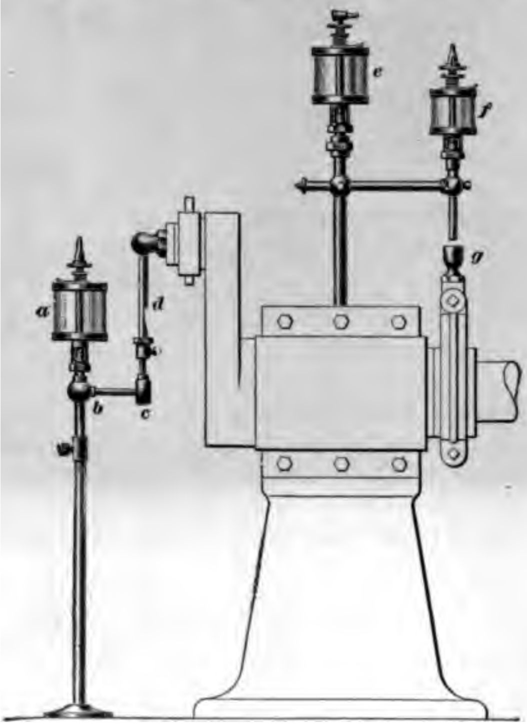


FIG. 4.

to wipe off the drops of oil hanging at the bottom of the nozzle *f* as the crosshead passes back and forth. The oil thus collected flows by gravity through a hole in the center of the wristpin and is delivered through one or more radial holes to the outside of the pin.

A precisely similar wiping fixture may be and often is used for crankpins and eccentrics, using stationary lubricators placed on top of the main bearing.

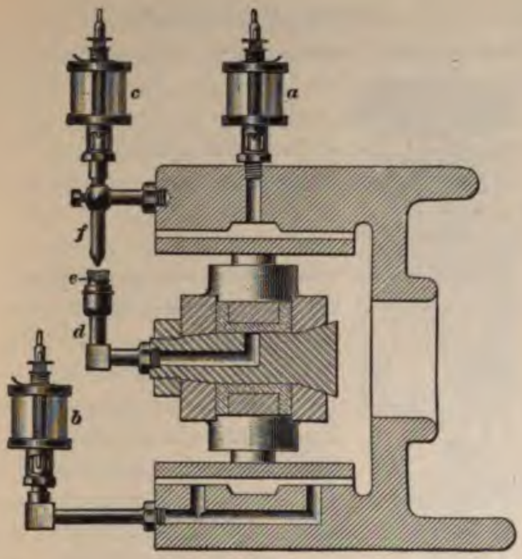


FIG. 5.

97. **Grease cups** are made in various ways and either as **plain** or **compression** cups. In a plain cup the grease only flows down by gravity as the heat of the bearing melts it; to assist the grease, it is a good practice to put a piece of small copper wire in the hole through which the grease leaves the cup. A compression grease cup may be hand-operated or spring-operated; Fig. 6 shows one of the type first named. By screwing the cap down by hand over the base, the grease is forced out.

FIG. 6.

98. Spring-operated compression grease cups have a piston, on top of which is placed a spring that continually forces out the grease. In most of them the rate of flow can be regulated by a suitable valve.

STEAM LUBRICATORS.

99. Mechanical Lubricators.—Hand-operated mechanical steam lubricators are generally small force pumps connected to a suitable oil reservoir and having the discharge pipe connected to the main steam pipe close to the throttle. Their construction and operation is so simple as to require no description.

100. Automatic mechanical lubricators are operated from some moving part of the engine, as some convenient part of the valve gear. Fig. 7 shows one form of such a device, known as the "Rochester automatic lubricator." It consists of a cylindrical oil reservoir *a* containing a piston that is screwed down on the oil by gearing connected to the ratchet wheel *b*. The ratchet wheel is operated by a pawl on one end of the ratchet lever *c*, which is vibrated back and forth by the rocker *d*. This rocker is rocked back and forth by some convenient reciprocating part of the engine. The connection between *c* and *d* is made in such a manner that the arc through which *c* vibrates can be changed so that the pawl will move the ratchet wheel any desired number of teeth within the range of the device. The oil is ejected from the reservoir by the piston and passes through *e* to the engine.

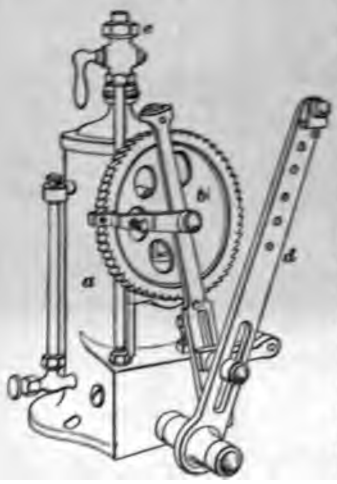


FIG. 7.

101. Water-Displacement Lubricators. — The simplest form of a water-displacement lubricator is shown in Fig. 8. It consists of a cylindrical shell *A* provided with a central tube *a*; a cap *C*, through which the lubricator is filled; and a shank *b* for attaching it in a vertical position to the steam chest or steam pipe. A valve *B* controls the communication between the lubricator and the engine.

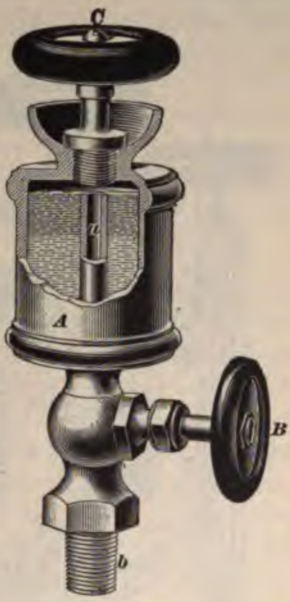


FIG. 8.

102. The operation of the lubricator is as follows: The receptacle is filled with oil and closed. The valve *B* is then opened, thus allowing the steam to pass through the central tube in to the top of the lubricator. The steam, coming in contact with the cold surfaces of the oil and receptacle, condenses. Since water is heavier than oil, bulk for bulk, the drops of condensed steam sink to the bottom of the receptacle. As two bodies cannot occupy the same space at the same time, the drops of water displace a quantity of oil equal in volume to their own; the oil, which has no other means of egress, flows over the edges of the central tube and runs by gravity into the steam pipe.

The objectionable features of this lubricator are that the flow of oil is not readily controlled and that there is no indication of when the lubricator stops working, either for want of oil or otherwise.

103. To overcome the objections mentioned in Art. **102**, **sight-feed** water-displacement lubricators have been designed, one of which is shown in Fig. 9. Its principle of action is the same as that of the lubricator shown in Fig. 8; i. e., it depends on the condensation of the steam and the subsequent displacement of the oil. Its construction is as follows: A cylindrical receptacle *d* is provided with a central tube *a* communicating with the threaded shank *e* and the sight-feed glass *A*. To fill the receptacle, the cap E is provided. The upper end of the lubricator communicates with the sight-feed glass by the passage b. In operation the steam is admitted to the lubricator by means of the valve B, the opening of which admits it to the inside of the lubricator as well as to the sight-feed glass A. The steam, coming in contact with the oil and the top of the lubricator, condenses and displaces the oil, which then flows through the passage b into a conical nozzle, as shown, and issues from the latter either drop by drop or in a thin stream, depending on the position of the regulating valve C. It is apparent that by screwing down the latter, the annular opening between the valve and nozzle is reduced, and hence the flow of oil is checked. Conversely, by screwing up the valve, the rate of flow is increased. The drops of oil issuing from the nozzle flow by gravity through the passage c and thus to their destination. Since the glass tube is transparent, the oil dropping from the nozzle is in plain sight of the attendant. By means of a drain cock, not shown in the figure, the lubricator may be emptied when required. This lubricator uses a **down feed,** which means that the oil is discharged downwards in respect to the feed nozzle. These lubricators are not very reliable in their action, since the oil is not forced through the feed nozzle, but only flows through it by gravity.

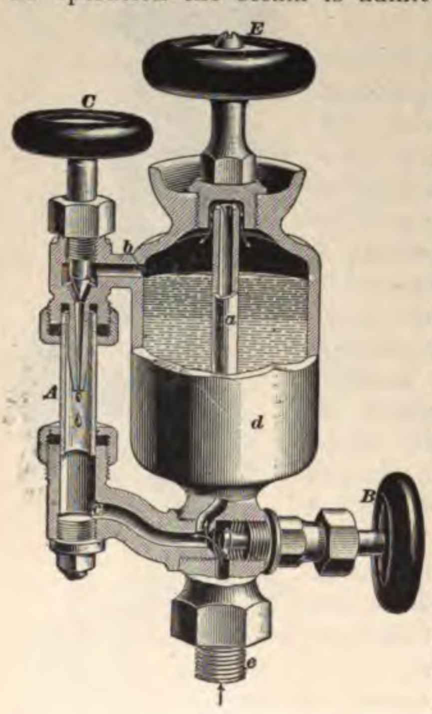


Fig. 9.

104. Hydrostatic Lubricators.—All water-displacement lubricators belong to the **single-connection** type, this meaning that there is only one connection to the steam pipe and, consequently, that the oil must pass through the same passage through which the steam is admitted. Hydrostatic lubricators are made in two styles, *single-connection* and *double-connection*. In a **double-connection** lubricator there are two connecting pipes to the steam pipe, the steam being admitted through one pipe and the oil leaving the lubricator through the other.

105. A typical single-connection hydrostatic lubricator is shown in Fig. 10, (*a*) being a part section and (*b*) a side view. The lubricator is connected to the steam pipe through the nipple *M*. The steam flows through *M* and the pipe *B* into the **condenser** *F*; it also flows through the connection *b* and a passage cored out in *C* to the sight-feed glass *H*. The steam is condensed, both in the condenser and in the sight-feed glass, by radiation. The water in the condenser flows through the pipe *1* into the bottom of the oil reservoir and forces the oil to the top, exerting a hydrostatic pressure on the bottom of the oil, which is transmitted through the oil. The latter flows through the pipe *2* into a nozzle located in the bottom of the sight-feed glass and out of the nozzle into the glass. The drops of oil ascend, by reason of oil being lighter than water, to the top of the sight-feed glass, which, it will be remembered, is filled with water. The oil then flows into the passage within *C* and passes through *b* into the nipple *M* and into the steam pipe.

106. There is an equal steam pressure on top of the water in the condenser and in the sight-feed glass, so that the pressure impelling the oil out of the lubricator is only that due to the hydrostatic head. The rate of flow of the oil through the nozzle in the bottom of the sight-feed glass can be regulated by means of the needle valve *E*; the water can be shut off from the oil reservoir *A* by closing the valve *D*; a drain cock *G* is used for draining the reservoir. A gauge glass *g* shows the amount of oil in the lubricator.

The reservoir can be filled when the filling plug O is unscrewed. A small valve S is closed in order to shut off the steam from the sight-feed glass in case the latter is broken or in need of cleaning. With the valves D and S shut, the gland in which the valve E works is unscrewed;

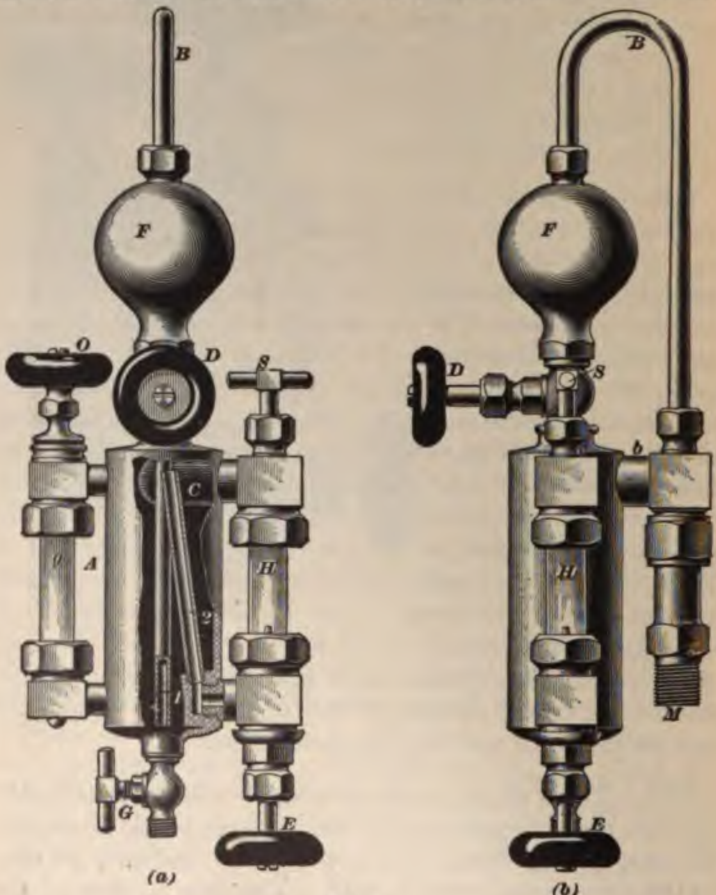


FIG. 10.

the broken or dirty glass tube can be removed and a new one or the cleaned old tube inserted. It will be noticed that this lubricator has an **up feed**; that is, the drops of oil coming from the nozzle flow upwards in the sight-feed glass.

107. To start the lubricator, open the valves *D* and *S*; to stop it, close the valves *D* and *S*. The regulating valve *E* when once adjusted need rarely be disturbed. To drain the lubricator while steam is on the pipe into which it delivers, close the valve *D*, and *E* and *S* being open, open *G*. When not under steam, to drain remove the filler plug *O* and open *G*.

108. A **double-connection lubricator** is shown in Fig. 11, which incidentally shows the mode of attachment to a vertical pipe. The condenser *a* has its independent steam connection; the angle valve *b* admits the steam to the condenser. The water in the condenser passes to the bottom of the reservoir *c* through the pipe *d* and forces the oil upwards into the pipe *e* leading to the bottom of the sight-feed glass *f*. It then flows through an annular opening regulated by the valve *g* up the sight-feed glass and through the pipe *h* and valve *i* into the steam pipe. The pressure impelling the oil forwards is simply the hydrostatic pressure due to the water in the condenser.

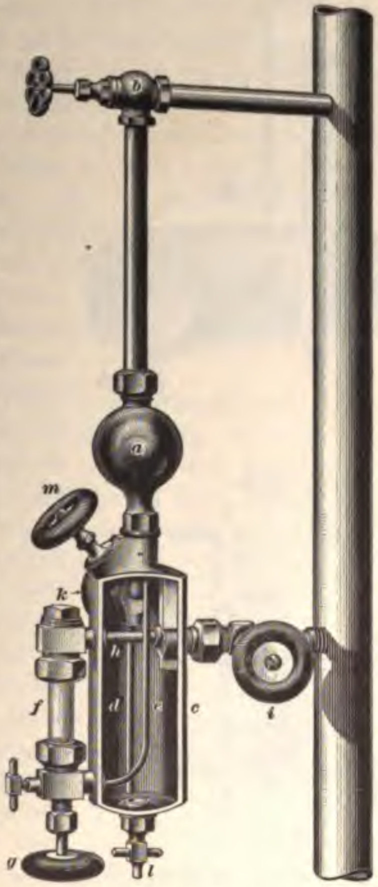


FIG. 11.

109. To fill the lubricator, close the valve *k*, which shuts the condenser off from the reservoir *c*, and also close the valve *g*. Open drain valve *l* and remove filler plug *m*. When water has drained off, close valve *l*, fill with oil, and replace the filler plug. Open valve *k* again and regulate the flow with the valve *g*. To shut off the lubricator temporarily, close the valve *k*; to shut it off permanently, close valves *b* and *i*.

110. Double-connection lubricators are made in which the condenser is an independent vessel; such a one is shown connected up to a horizontal overhead steam pipe in Fig. 12. The receptacle *o*, which may be filled by unscrewing the cap *m*, communicates with the sight-feed glass *c*. The regulating valve *i* controls the flow of the oil. At *j* a valve used for draining the receptacle is shown; the drain pipe may be attached by the union *k*. The valve *l* may be used for closing the passage leading from the bottom of the lubricator to the pipe *a*. The lubricator is connected to the steam pipe *n* by the pipe *a*, which connects *o* to the condenser *p*, which is, in turn, connected to *n* by the pipe *f*. The oil from the lubricator passes to the steam pipe through the pipe *b*. By means of the valves *g* and *h*, the lubricator may be shut off when desired. Its operation is as follows: When starting the cup for the first time, the pipes *a* and *b* and the sight-feed glass *c* are filled with water, the pipe *a* being filled nearly up to *d*. Since the water in the pipe *a* can flow into the bottom of the lubricator, it follows that the oil will

FIG. 12.

be forced through the feed nozzle with a pressure depending on the hydrostatic head *de*.

After passing through the feed nozzle, the drops of oil ascend through the sight-feed glass and up the pipe *b*, the pressure causing the upward flow being due to the difference in specific gravities of the water and oil. To prevent the emptying of the pipe *b* when draining the lubricator preparatory to replenishing the oil supply, a small check-valve *r* is provided. In order to replenish the water that passes from the pipe *a* into the lubricator, the condenser *p* is used. This may be a vessel of any desired shape; it is usually a piece of $1\frac{1}{2}$-inch brass tubing, as shown in the figure. The steam entering from the steam pipe is condensed by coming into contact with the relatively cool surfaces of the condenser; the latter is made large in order to increase the radiating surface. In this style of lubricator the hydrostatic pressure operating the device may be made as great as circumstances will permit by simply extending the loop of the pipe *f* higher up. If this is done, the condenser must also be raised in order to derive the most benefit from the change.

111. Double-connection lubricators should never have one connection attached to the steam pipe between the throttle and boiler and the other between the throttle and engine. If the lubricator is connected in this manner, upon closing the throttle there will be full steam pressure on the condenser and none on the sight-feed glass. In consequence, the lubricator will very rapidly be emptied, the steam pressure forcing all the oil out into the engine. If circumstances require the connection to be made in this manner, a special locomotive double-connection sight-feed lubricator should be selected. Such a lubricator is especially made in such a manner that the oil cannot leave the reservoir when the throttle is closed.

ENGINE INSTALLATION.

COMPARISON OF TYPES OF RECIPROCATING ENGINES.

VERTICAL VERSUS HORIZONTAL ENGINES.

1. Uses of Vertical Engines.—The inverted vertical engine—that is, the engine with the crank-shaft resting in a bedplate placed on the foundation and suitable and appropriate housings containing the guides, the cylinder resting upon and secured to the housings or engine frame—is now the prevailing type for large power-station purposes and many other applications of steam engines. That type of vertical engine in which the cylinder joins the bedplate and has the shaft or beam on top of the engine framing is only used for special and peculiar applications, such as slow-speed pumping and blowing engines, but never for quick-running engines.

2. Controlling Features.—The vertical engine has its distinct application, its advantages, and its disadvantages. The two controlling features that dictate the use of the vertical engine are (1) available floor space for the engine and (2) size of engine. The first reason is self-evident. As to the second reason, in very large horizontal engines, and particularly with the low-pressure cylinder of compound engines, the problem of supporting the weight and preventing the cutting of massive low-pressure pistons running at the speed now common becomes one of great magnitude; in fact, the success is always problematical, even with the most carefully planned constructions. This bad feature of the horizontal engine is entirely overcome by making the engine vertical; the weight of the piston is then borne by the shaft bearings.

3. Supporting Pistons of Horizontal Engines.—Many devices have been tried to support the weight of large pistons, such as tailrods having crossheads running on external guides, but the distance between the points of support or crossheads is usually long, and the allowable deflection can rarely exceed $\frac{1}{32}$ inch, so that this expedient, to be of any service whatever, requires very large rods. Take the case of a 72-inch cylinder having a 72-inch stroke. With a carefully designed cast-iron piston, it would require a piston rod at least 14 inches in diameter having a 5-inch hole through it to support the piston successfully. Pistons having a steam pocket underneath them, into which steam is admitted through a small hollow tailrod, have been used by one very large builder. Forged steel-plate pistons having broad composition shoes riveted to the lower circumference, the shoes projecting into recesses formed in the heads, have been used by an English builder; while very broad pistons in which the weight of the piston does not exceed 3 pounds per square inch of projected area are often resorted to. Many of the devices for supporting pistons have merit, but many engineers believe the vertical engine to be the best solution of the problem.

4. Inaccessibility of Vertical Engines.—The vertical engine is much more inaccessible than the horizontal machine for oiling, inspecting, and repairing; indeed, in some of the very latest American high-grade engine designs, it would be necessary to dismantle the whole machine to remove the crank-shaft, although the specifications usually demand that it be possible to remove the crank-shaft bearings when the shaft is raised $\frac{1}{4}$ inch. The vertical engine costs on an average about 12 per cent. more than the horizontal engine. Generally, the vertical engine will not receive the same degree of care and attention that the horizontal machine will, owing to its inaccessibility and the labor and exertion required to reach its various parts. This should not be the case, but it is so, nevertheless.

5. Comparison of Headroom.—The vertical engine requires quite a high building, not only on account of the design, but also because extra room is needed to draw out the piston and piston rod. If the engine is large, a substantial crane or other means of handling the various parts is necessary. The horizontal engine, where space is available and other conditions do not preclude its adoption, has many practical and commercial advantages over the vertical engine. It is the cheaper engine, is much more accessible for repairs, oiling, and inspection, and can be cared for by men physically incapacitated to handle a vertical engine.

6. Comparison of Floor Space.—The horizontal engine from the nature of its design requires considerable floor space, and in localities where property is valuable, as is often the case along city water fronts or at the center of a large electrical distributing system, and where it is desirable to concentrate as much motive power in as small a space as possible, the horizontal engine must give way to the more expensive and less accessible vertical engine. In very large power plants it is quite customary either to connect the engine galleries, making them continuous throughout the whole plant, or to construct mezzanine galleries around the house on the same level and connecting with all the galleries. If this is done, the various units can be visited for inspection and oiling without descending to the floor each time, thus making easier the labor of attending to this class of engine.

7. Influence of Drainage.—Unless especial care is exercised in the design of the vertical engine to free of water all parts coming in contact with the live steam and to prevent water pockets, its economy will fall below that of the horizontal engine, especially if the latter is of the four-valve type, which type when embodied in the horizontal machine is almost self-draining. The vertical design does not so readily lend itself to drainage and hence requires especial care in this respect.

8. Influence of Balancing.—The mechanical efficiency of the vertical engine is from 2 to 3 per cent. higher than that of the horizontal engine. With an equal measure of care as regards balancing, the vertical engine will operate more smoothly than the horizontal machine; this is due to the fact that the unbalanced vertical force acts vertically through the machine and foundation, while the unbalanced horizontal force is close to the foundation and is counteracted by two heavy masses—the foundation below and the engine above.

9. Combined Vertical and Horizontal Engines.—A type of engine occasionally used is a combination of the horizontal and vertical machine. This engine is usually made a compound, in which the low-pressure cylinder is made vertical, for reasons that have been previously given, while the high-pressure cylinder is placed horizontal. Both engines act on one crankpin, thus making a compact machine having all the advantages of two cranks at right angles. This type of machine has been made in very large units and used for direct-connected electric service and reversing rolling-mill service.

AUTOMATIC CUT-OFF VERSUS THROTTLING ENGINES.

10. Types of Automatic Cut-Off Engines.—There are two distinct types of automatic cut-off engines, the *positive automatic cut-off*, in which the main or cut-off valve closes the port by positive motion derived from some part of the engine, usually an eccentric on the main shaft, and the *releasing-gear cut-off*, in which the main valve or cut-off valve is made to close the admission port by means of vacuum pots, weights, springs, steam pressure, or other sufficient means.

11. Limiting Speeds of Releasing-Gear Engines.—The positive automatic cut-off engine is always used for quick-running engines in preference to the releasing-gear engine. The practicable maximum speed of the latter type may be set at 100 revolutions per minute for engines up to 500 horsepower, 85 revolutions per minute up to 2,000 horsepower, and 75 revolutions per minute up to 5,000 horsepower, although small releasing-gear engines have been run at 150 revolutions per minute. The best builders do not advise speeds higher than those given above. There are a very few builders that run their small and medium-size engines about 20 per cent. faster than the speeds given above, but this does not mean that the same power is obtained at 20 per cent. less investment, for the reason that to run satisfactorily at these high speeds the machines must be especially constructed and heavily built; furthermore, the lack of insurance against shut down due to breakage or heating and the larger quantity of oil required more than offset the item of first cost. For all conditions requiring high speeds and great economy, the positive automatic cut-off engine is usually chosen. These machines can always be run at speeds not determined or limited by the construction or operation of the valve gear.

12. Economy of Automatic Cut-Off Engines.—The economy of the positive automatic cut-off engine with one valve is only about 75 per cent. of that of a releasing-gear automatic cut-off engine of equal grade. There are, however, some positive automatic cut-off engines of the four-valve type in which the cut-off valve is mounted on the main valve and is positively driven by a shaft governor; in such an engine the economy of steam is fully equal to that of the releasing-gear engine. They are capable of much higher speeds than the releasing-gear engine, but owing to some complication of the valve gear, they are not usually run at as high speeds as the one-valve positive automatic cut-off engine. The four-valve positive automatic cut-off engine is somewhat more expensive than the releasing-gear engine, which, in turn, is considerably more expensive than the one-valve automatic engine.

13. Accessibility of Releasing-Gear Engines.—The releasing-gear engine is invariably more complicated than the positive gear and requires closer adjustment, but on the other hand, it is much more accessible for adjustment, even while in motion, than the positive gear, which is unapproachable while the engine is running. The problem of oiling the positive-gear engine is one that cannot be solved too carefully, as the success of the gear largely depends upon the perfection of the oiling devices. The oiling of the releasing-gear engine is an easy problem in comparison. These statements apply not only to the valve gear, but with equal force to the reciprocating and rotating parts of both classes of engines.

14. Comparison of Throttling and Automatic Cut-Off Engines.—The simple throttling engine is the oldest type of engine and is probably the least used at the present time. Its strongest claim to existence is simplicity, and for many purposes and locations the claim is strong. It is much less economical than the automatic cut-off engine, is usually built for slower speeds, and generally there is not much attention paid to the features conducive to economy. One of the defects of the throttling engine is that, for the purposes of regulation, this machine must have from 5 to 20 pounds more steam pressure on the inlet side of the governor than on the outlet side; consequently, the boiler must generate steam from 5 to 20 pounds higher pressure than is actually used in the engine; hence, some waste of heat takes place before the steam arrives in the working cylinder.

15. The automatic cut-off engine adjusts its energy to the resistance by measuring out a supply of steam always at or near the boiler pressure and sufficient to overcome the resistance; the throttling engine always supplies the same volume, but varies the pressure to suit the resistance to be overcome. The automatic cut-off engine is capable of high ratios of expansion; the throttling engine as usually built is not, ⅝ cut-off being the prevailing point, giving only 1⅗ expansions.

16. There is another class of engine, known as the *Meyer valve engine*, belonging to the throttling-engine family, in which a separate expansion valve on the back of the main valve and worked by a separate eccentric is used to effect a cut-off from 0 to ⅝ stroke. This class of engines is capable of high ratios of expansion and hence is quite economical; they are usually well made and provided with a throttling governor. The point of cut-off is adjustable by hand and is set very close to the actual demands, allowing the governor very little range of pressure to adjust the speed of the engine.

SIMPLE ENGINES VERSUS COMPOUND.

17. Influence of Power Required.—Like most engineering problems, the problems relating to the use of compound engines resolve themselves chiefly into problems of finance. The cost of fuel and amount of power required are leading factors in determining the use of compound engines, generally speaking. When the power required is less than 200 horsepower, it will hardly pay to put in a compound condensing engine where the steam pressure is limited to 100 pounds gauge, unless the cost of fuel is very high, say $4, or more, per ton. If 125 pounds of steam can be carried by the boiler, it will be a paying investment. Similar limits apply to the case of a compound non-condensing engine, except that the steam pressures should be changed to 125 pounds gauge pressure and 150 pounds, respectively.

18. As the size of the engine increases, it becomes more important to compound, for the reason that a 1,000-horsepower engine does not cost five times as much as a 200-horsepower engine of similar design and construction. When the price of fuel is low, compounding becomes of less importance, and compound non-condensing engines with a variable load when working with steam pressure not more than 150 pounds are rarely paying investments.

19. Triple- and Quadruple-Expansion Engines.—Triple-expansion condensing engines have shown a real economical advantage over compound engines of 20 per cent. Such engines should not be used with less than 160 pounds steam pressure. Triple-expansion non-condensing engines seldom prove a good investment under ordinary conditions, and the same may be said of quadruple-expansion condensing engines. This statement refers to land engines, but not to marine engines, where quadruple engines are sometimes used not only to secure extreme economy in the use of steam, but also to reduce the vibrations of the ship to a minimum.

20. Steam Consumption.—A good four-valve automatic cut-off engine will consume 24 pounds of dry steam at 100 pounds pressure per horsepower per hour, while a compound condensing engine of similar design, but having a reheating receiver supplied with 50 square feet of tube reheating surface for each cubic foot of steam delivered from the high-pressure cylinder, will consume but 14 pounds of dry steam of 135 pounds pressure per horsepower per hour. A good triple-expansion condensing engine, if supplied with steam at 160 pounds pressure, could accomplish the same work with 11 pounds of dry steam per hour. The above figures as to steam consumption hold only for medium and large engines, say from 500 horsepower up; small engines are not so economical as large engines, which is probably due to the greater ratio of cylinder and port surface to the volume swept through by the piston in small engines.

21. Factors to be Considered.—While the problem of simple versus compound engines is always one of finance, economy of fuel and first cost are not always the determining elements. The compound engine is always more complicated and hence more liable to a breakdown, and if isolated, requires the carrying of more spare parts. It requires a higher degree of skill to maintain it in economical condition and requires better and more expensive boilers, but does not require as many boilers or as large a boiler plant. The question of whether insurance risks may be greater and the facilities for repairs should also be considered in determining the type of engine. The real test in any case is the final influence of the machinery used on the profits of the business.

TANDEM COMPOUND VERSUS CROSS-COMPOUND ENGINES.

22. Cylinder Arrangements of Tandem Compound Engines.—A tandem compound engine is one in which the cylinders are arranged one behind the other, both pistons being on the same piston rod and acting on one crankpin. The cylinders are arranged sometimes with the high-pressure cylinder behind and sometimes with the low-pressure cylinder behind. Both arrangements have their advantages. When the low-pressure cylinder is placed behind and the front low-pressure cylinder head is made on an internal flange, the cylinders and pistons are quite accessible. When the high-pressure cylinder is placed behind the low-pressure, the piston rod must be removed through the front low-pressure stuffingbox and, consequently, can have no projecting collars forged on it to take the thrust of the low-pressure piston. The rod is sometimes fitted with loose steel collars that take the thrust of the low-pressure piston; sometimes the portion of the rod that enters the low-pressure cylinder is made rather large in order to form a sufficient shoulder for the low-pressure piston to bear against. When it becomes necessary to forge a collar on the rod to secure a sufficient bearing shoulder for the low-pressure piston, the stuffingbox throat must be bushed; the collar will then pass through the throat when the bushing is removed. Both pistons should have tapered seats on the rod.

23. Disadvantages of Tandem Compound Engines. The principal objection to the tandem engine is the inaccessibility of the cylinders and pistons for inspection or repair. The cylinders are also liable to get out of alinement if not properly designed and constructed, which occurrence, in turn, reduces the mechanical efficiency of the machine. The loss of alinement is obviated to a considerable extent by making a heavy cast-iron sole plate extend under both cylinders. The front cylinder should be securely bolted to this sole plate, while the rear cylinder should be arranged to slide in suitable ways, which constrain it laterally, but allow it to move longitudinally when it expands and contracts. This feature in large engines is important.

24. Comparison of Spare Parts Required.—The economical performance of the two types of machines are the same. The cross-compound engine has considerably more parts than the tandem, but many of them are exact duplicates, so that in isolated districts the cross-compound engine would probably not require a larger item of spare parts than the tandem. On account of their smaller size, for equal engine power, the first cost of spare parts for a cross-compound engine would be less than for the tandem engine.

25. Comparison of Mechanical Efficiency.—For equal engine power, the mechanical efficiency of the two types of machines should be in favor of the cross-compound engine. This at first thought seems erroneous, but a little consideration will make this fact clear. The frictional resistances of pistons and rods should be the same with both, but in the tandem compound they are much more liable to increase in time, due to its greater liability to get out of alinement. The valve-gear resistances should be practically the same in both types, but usually are slightly in favor of the tandem compound. Considering the resistances at the crossheads and crankpins, while there are twice as many parts in contact in the cross-compound engine, the total force and resultant pressures and the direction and duration of the pressure are the same in both types.

26. The greatest divergency in the frictional resistances of the two types of engine is at the shaft. For equal degrees of unsteadiness of rotation, both engines working at the same economical ratio of expansion and speed, the tandem engine requires a wheel about $1\frac{6}{10}$ times heavier than a cross-compound engine. This, in turn, requires a larger and heavier shaft and bearings and means an increased velocity of the bearing surfaces, and hence more wear and oil; in this respect the cross-compound engine has a decided advantage over the tandem.

27. Comparison of Cost.—The tandem engine has its strongest claim in the matter of first cost; if this, however, is carefully investigated, it will be found that for similar service, economy, speed, pressure, and type, the first cost of the tandem engine will average only about 9 per cent. lower than the first cost of the cross-compound engine. The cost of foundation for a tandem engine will be about 20 per cent. less than that for the cross-compound.

28. Formerly, it was the practice to make the passage of steam from the low-pressure cylinder to the high-pressure cylinder of tandem engines as short and direct as possible, but the prevailing practice at present for equal duty is to give the tandem engine a reheating receiver of a volume equal to that of the receiver of the cross-compound engine, which is usually equal to the volume of the low-pressure cylinder; and it is customary to provide for both types of engines about 50 square feet of tube reheating surface for each cubic foot of steam exhausted by the high-pressure cylinder. Formerly, there was considerable difference of cost between receivers and piping for tandem and cross-compound engines, but as at present constructed, there is no appreciable difference.

29. Comparison of Smoothness of Running.—With equal elaboration to secure smoothness of running, and comparing condensing engines, the tandem engine will generally excel. The reason for this is seen when it is considered that compression is the main factor tending to secure smoothness in turn the dead centers. If the vacuum in the low-pressure cylinder be good, the remaining gas is so attenuated that ordinary means will not secure sufficient compression to absorb the inertia of the reciprocating parts at the end of the stroke, the result being a severe pounding at all journals. To prevent this, extraordinary and expensive means must be used, such as providing separate valve gear to drive the exhaust valves independent of the steam valves. In the tandem engine, both pistons being on the same rod, sufficient ompression can easily be obtained behind the high-pressure and low-pressure pistons to fully absorb the inertia of the ciprocating parts. particularly to releasing lve-gear engines.

SINGLE V LEX ENGINES.

rpose Reversible Engines.—A ngine consists ple engines, usually exact ates in all respects, ting on one crank-shaft; ce, it is similar in arr it to the cross-compound ine. Reversible engin most invariably duplex to ilitate starting the en any possible position at which the cranks may happen to be. Familiar examples of this type of engine are the locomotive, hoisting engines, blooming engines, and barring engines.

31. Purpose of Duplex Non-Reversible Engines.—The duplex non-reversible engine is frequently met with in industrial works, and their existence is usually due to an extension of the industry, where a little forethought has served to save the additional cost of an entirely new engine. In planning and developing an industry, it is reasonably expected to grow and expand; often the exact expansion cannot be predicted with certainty. While a certain amount of surplus power can be provided for in installing the original engine, it is a well-established fact in steam engineering that an underloaded engine is an extremely wasteful and poor paying investment; this fact creates the field for the duplex non-reversible engine. The wheel for the original single engine is made sufficiently large to transmit double the original power, if belt or rope transmission is used; this, however, does not mean that it shall be double the weight or cost, but only 1.4 times the weight for a single engine and about 1.3 times the cost of a wheel for a single engine. Frequently a section of the bedplate containing the shaft bearing is purchased with the original machine, and when the demand for another engine is made, it can be readily attached to the original machine without a shut-down or delay of the works.

32. Methods of Providing for Increased Power.—Other methods are sometimes practiced to accomplish the ends secured by the duplex engine, such as purchasing a larger engine than is required, inserting a thick bushing in the cylinder, and providing a smaller piston, all being so arranged as to be removed when the demand for increased power is made. While this accomplishes the same result, it is done at a sacrifice of economy and ties up a considerable amount of extra capital, due to the first cost of the larger engine, which might otherwise be turned to good investment.

If conditions so change between the time of installing the original engine and such time as increased power is required, the machine intended to be duplex can quite as readily be made into a cross-compound as a duplex by adding a low-pressure cylinder and receiver, and if water be available, a condenser. It might be well to add here that if it is required to obtain the same amount of power with a higher degree of economy, the steam pressure and speed remaining the same, it cannot be obtained by the addition of a high-pressure cylinder, as is very often erroneously assumed. The saving in fuel by compounding and the addition of a condenser should be between 15 and 20 per cent.

33. Comparison of Mechanical Efficiencies.—The mechanical efficiency of the simple engine of equal power compared with the duplex should be a little higher. There should be no appreciable difference in the economical performance of the two types of engines unless the sizes are such as to render the duplex very small engines, in which event the economical efficiency of the duplex engine will suffer a loss. The duplex engine should operate more smoothly than the simple because of the more even turning moment at the crank-shaft. The simple engine of equal power, and if run at the same speed as the duplex engine, to secure the same coefficient of unsteadiness of rotation, will require a wheel 1.6 times heavier than the duplex, and necessarily a heavier shaft and larger bearings; this operates to reduce the mechanical efficiency of the simple engine.

34. First Cost.—For engines of equal power, under the same steam pressure and piston speed, the duplex engine will cost about 1.4 times as much as the simple engine, while the foundations will cost about 1.6 times as much as the foundations for a simple engine. It is to be noted, however, in selecting engines with reference to cost per horsepower that the price will be found to vary on either side of a minimum horsepower, which for ordinary engines will be about 500 horsepower. Owing to this fact, it sometimes happens that a very large duplex engine may be found to cost less than a single engine of equal power.

HIGH-SPEED VERSUS SLOW-SPEED ENGINES.

35. Classification.—The line of demarcation between high-speed and slow-speed engines is not clearly defined when referred to the number of revolutions made per minute or per second, for there is another characteristic that, when considered in connection with the number of revolutions per minute, assigns them to the class in which they belong. This characteristic is the manner in which regulation is secured. Engines with a releasing-valve gear, usually of the four-valve type, and regulated by a pendulum governor are classed as slow-speed engines, barring a few exceptions, while positive automatic cut-off engines, regulated by means of a shaft governor, are invariably classed as high-speed engines. Engines regulated by means of a throttling governor are seldom classed as either high-speed or slow-speed engines; in fact, they usually run at speeds midway between those of the two first-mentioned classes of machine. In considering the relative merits of high-speed and slow-speed engines, the speed refers to the revolutions per minute, and *not to the piston speed*, for, as a matter of fact, the piston speed of modern slow-speed engines often exceeds that of the high-speed engine.

36. Purpose and Advantages of High-Speed Engines.—It may safely be said that the advent of electricity as a medium for the transmission of energy caused the development of the quick-running engine, and it is in the electrical field that it still finds its largest market. It is not confined to the electrical industry, however, and has been applied to nearly every service, either direct-connected or by belting. Its principal merits are comparative low first cost and small space required; its principal objections are wastefulness of fuel and need of constant attention. These objections are not universal, however, as there are some high-speed engines on the market that are quite equal in economy and in all other respects to any of the slow-speed engines. There are also some high-speed engines of the enclosed crank-chamber type that, to a great extent, are self-lubricating and demand very little attention; they usually take the vertical form. The majority of high-speed engines are not very economical machines and must be carefully watched.

37. Comparison of Valves for High-Speed Engines. High-speed engines are almost invariably fitted with a balanced valve, which is frequently a piston valve. It is claimed that this type of valve, if used on a horizontal high-speed engine, will begin to leak about the time the engine is paid for and will not improve with age, notwithstanding the many devices used to adjust the fit of these valves in their liners or casings.

The piston valve applied to the vertical engine has given better results as regards less leakage and resultant economy, but even here it has not been altogether satisfactory, principally because the system of regulation imposes varying travel to the valve and unequal wear on the internal surface of the casing or liner. Other systems of balancing are by means of pressure or cover-plates; these require very careful design and workmanship, but if properly designed and fitted, they are much superior to the piston valve. The clearance with the latter form of valve is usually much less than with the piston valve, but the clearance is generally large in all of them, and it is due to this fact that the periods of compression can be lengthened and the engine be made to operate very smoothly.

The Corliss valve has been used to some extent on high-speed engines, but the result has not been altogether encouraging, and in several instances they have been absolute failures, which was probably due to the fact that if pressure is allowed to remain on Corliss valves sufficiently long to force out the film of oil that is between the valve and the seat, it takes considerable force to move them.

38. Valve Motions.—The valve motion is invariably derived from an eccentric of variable throw and angular advance or from an equivalent crank, so hung as to give a nearly constant lead. The peculiar valve gears of the high-grade slow-speed engine, by virtue of which the valves move but little and very slowly after they have closed the ports, are seldom or never attempted in the high-speed engine.

39. Effects of Large Clearance Volume.—The necessary simplicity and desirable low first cost of high-speed engines has resulted, on account of the types of valves, in large and comparatively long steam ports, which increase the clearance volume and clearance surface. As was previously pointed out, high-speed engines have a high speed as regards the number of revolutions per unit of time; consequently, the stroke must be shortened, which results in a high percentage of clearance volume; the frequency with which this clearance volume or part of it is filled with fresh steam affects the economy of this class of engine to some extent.

40. Effect of High Speed on Regulation.—Aside from the question of economy, one of the leading characteristics of high-speed engines is the regulation. On account of the high rotative speed, the regulator or governor has a much greater opportunity to effect changes in the speed; that is, if the high-speed engine is running 300 revolutions per minute while the slow-speed engine is making 100 revolutions per minute, the high-speed engine may be said to have 600 opportunities to adjust the steam supply while the slow-speed machine has only 200; or, in a unit of time, which may be taken as 1 revolution of the slow-speed engine, the high-speed engine has had 3 opportunities to adjust its speed. Consequently, the regulation of the high-speed engine is much superior to that of slow-running engines, even though they be fitted with governors equally sensitive. As a matter of fact, the better types of high-speed engines, as at present constructed, leave nothing to be desired in the matter of regulation for any possible commercial service.

41. Prevention of Accidents.—The increased risk of wear and the liability to accident due to their rapid motion, and especially when accidents do occur, the seriousness of their nature must be considered in connection with the high-speed engine. The prevailing tendency among builders of this class of engine is to reduce the possibility of accident by selecting higher grades of material, providing liberal wearing surfaces, which are case-hardened or oil-tempered, and using safe and thoroughly tested constructions embraced by massive and well-distributed framings.

42. Lubrication.—High-speed engines require copious lubrication, and unless careful provision be made to collect the excess, great wastes may result in this direction. This is provided against to a great extent by providing splashers, oil guards, drip pans, and in some designs completely enclosing the running parts in oil casings; in some systems provisions are made for draining and collecting all oil in a separate chamber, where it is carefully strained or filtered and automatically returned to the bearings. In this so-called **return system,** a liberal stream or several streams of oil are kept running upon the bearing surfaces.

43. Accessibility of High-Speed Engines.—From the compact, rigid nature of the design of high-speed engines, they are not as accessible as the slow-running machine, but it cannot be argued that they are particularly difficult of access.

44. Influence of High Speed on Weight of Flywheel. Owing to the velocity of the high-speed engine and to the fact that the energy of a flywheel increases as the square of the velocity of the center of inertia, the wheels for high-speed engines can be made very much lighter and still obtain the same degree of unsteadiness as in the slow-running engine. This relieves the bearings of much dead weight and allows the shaft to be made smaller and makes the velocity of its rubbing surfaces much less.

45. Comparison of Economic Performances.—One of the elements in high-speed engines that, no doubt, contributes much to the economy of the machine is the little time allowed for initial condensation of each charge of steam and for the changes in temperature preceding each charge; some of the single-acting very quick-running engines have met with not a little success, their designers attributing it to the fact above mentioned. It must be borne in mind that the very highest duties and efficiencies have been obtained from the slowest running engines, as pumping engines, and many engineers contend that speed is not of vital importance in securing high economy. There is, however, little basis for comparison between the two engines, for slow-speed engines can also be denominated as high-grade engines, while high-speed engines may be classed as low-grade engines; and while there may be no appreciable difference in the economical performance of high-grade engines when run at varying speeds, the economy of a high-speed engine would fall away materially if run at a slow speed.

46. Savings Due to High Speed.—The high-speed engine has it strongest claim over the slow-speed engine in its adaptability to direct-connected work, whether the connection be to electric generators, the shaft of a mill, or any industrial work. There is at once a direct saving not only in the first cost of the engine, but in saving due to the omission of transmission machinery, as jack-shafts, belts, or gearing, bearings and their foundations, and the continuing expense resulting from their attendance, lubrication, and repair. High-speed engines, owing to their greater steam consumption, demand a 20-per cent. larger boiler plant, which is an item of first cost to be considered. While the circulars of high-speed engine builders announce their capacity and willingness to build this type of machine for large powers, they are seldom met with in actual practice; the common range of power is from 60 to 200 horsepower, but they are occasionally built in units as large as 800 to 1,000 horsepower.

47. High-Speed Compound Engines.—High-speed engines are as commonly built compound and triple-expansion as are slow-speed engines, and they are more frequently built compound non-condensing than are slow-speed engines. The compounds are arranged both cross and tandem. With high-speed engines it is quite important to have spare parts on hand.

48. Economy of Slow-Speed Engines.—The slow-speed engine, which at the present time is almost invariably of the four-valve drop cut-off or releasing-valve gear type, is most commonly chosen for all large units where continuous operation is required. In localities where fuel is expensive, even though the steam plant be used as a relay in case of failing water-power, the slow-running economical engine will be found. This condition is generally one requiring very careful study to obtain maximum commercial efficiency. The slow-speed engine admits of many, though practical, complications to the end of securing extreme economy of steam and high mechanical efficiency. The valves are usually so placed as to reduce the clearance volume and clearance surface to a minimum; great care is exercised to free the cylinders of water; steam jacketing of heads and cylinders is common. The polishing of the internal faces of heads and pistons in order to reduce the activity of the metal in receiving and imparting heat to the working steam, and thus reducing initial condensation, is sometimes resorted to; elaborate valve gears to give theoretical steam distribution are possible with the slow-speed engine.

49. First Cost of Slow-Speed Engines.—The slow-speed engine is much the larger, heavier, and more expensive machine, and usually costs $1\frac{3}{10}$ times as much as the high-speed engine of equal power. The foundations are also more expensive, but the boiler capacity need not be so large. The relative complete cost of high-speed and slow-speed power plants is not far from $50 per horsepower for high-speed and $70 per horsepower for slow-speed plants, the engines being simple non-condensing. The economical performance, assuming the engines to be in fairly good condition, should be about 30 pounds of water per horsepower per hour for the high-speed and 24 pounds of water per horsepower per hour for the slow-speed, bearing in mind, however, that the slow-speed engine will for a long time maintain its economical performance, while the high-speed engine will generally lose in efficiency.

50. Direct-Connected Engines for Dynamos.—The prevailing practice in the generation of electricity by steam power is towards direct-connected units, and the cost of the electrical generator is an item constantly urging higher rotative speeds; for this reason the slow-speed engine, a few years ago, was by several builders forced past its practical limits of speed. The impracticability of the move, as made apparent by constant breakage and short life, was soon recognized, and moderate speeds of about 90 to 100 or 110 revolutions per minute were returned to. These speeds are now seldom exceeded, and somewhat slower speeds are advised when long life and immunity from accidents are desired. Reducing the rotative speed of direct-connected engines operates to increase the diameter of the armature or revolving field of the electrical generator, and this, in turn, increases the first cost of the unit; it is here that the high-speed engine obtains its grip in the solution of the problem; but after passing a very moderate power, say 200 to 300 horsepower, the item of economy becomes of such magnitude that the more economical and more expensive slow-running unit establishes its ultimate value to the engineer or purchaser.

51. Attention and Workmanship.—Where the high-speed engine has its disadvantages, corresponding advantages may be found in the slow-running engine, and vice versa; slow-speed engines besides being much more economical do not require the close attention demanded of high-speed engines. With reasonable attention the slow-speed engine will give fair warning in many instances of approaching danger that will be developed too quickly in the high-speed engine to control. The slow-speed engine is more accessible and more readily adjusted; the workmanship need not be so exacting for equal results, the rate of mechanical depreciation is less, and the life and service of the machine is greater. The regulation of the slow-speed engine is not equal to that of the high-speed engine, and they require very heavy wheels for the same degree of unsteadiness.

SELECTION OF ENGINES.

INTRODUCTION.

52. The problem of selecting an engine for a specified service is one demanding all the skill, experience, and forethought of the constructing engineer, for on it rests one of the vital and constant items of expenditure of the industry of which it forms so important a part. The elements determining the selection of an engine may be briefly mentioned here as the influence of the kind of service, the location, first cost, cost of fuel delivered at the boilers, steam pressure available, the duration of service, the facilities for repairs, the kind of labor available, the existing conditions, if additions or renewals, whether the engine shall be condensing or non-condensing, and if one large engine shall be used or whether the power shall be divided into several smaller units; each of these conditions will receive separate consideration.

KIND OF SERVICE.

53. Self-Selecting Service.—Evidently the service to which an engine shall be put is the first determining element in the selection of an engine. In many cases the kind of service at once determines the general type of machine, as, for instance, for hoisting, pumping, blowing, locomotive, and rolling service. There are many lines of service, however, in which the type is not self-selecting. The service may be divided into *continuous running with uniform load*, *continuous running with variable load*, *continuous running with uniform but increasing load due to growth*, and *intermittent running*.

54. Continuous Running With Uniform Load.—When the condition of continuous running with uniform load pertains, the first question arising is the cost of fuel. If this item of expenditure is great, the evident conclusion is that the steam should be worked at as high a ratio of expansion as practicable, since the condition existing most readily permits economical working of steam machinery. Not only should the steam be worked expansively, but it should be done in a high-grade engine. Whether this be high-speed or slow-speed depends on the particular work the machine is called on to do. If the engine is direct-connected to an electric generator of not over 200 horsepower demand and the steam pressure is 125 pounds, while water is available for condensing purposes, maximum efficiency would be obtained by the use of a compound condensing high-speed engine having separate steam and exhaust valves and a governor controlling the admission valve only. Where economy is a desirable item, a high-speed engine having one valve for steam distribution is not a favorite. If the power demanded is large, the slow-speed high-grade engine should be the choice and the cost of fuel should dictate in a great measure the steam pressure, the ratio of expansion, and the refinements in all directions to the end of reducing all expenditures. If no water be available for condensing purposes, the compound non-condensing engine will prove a good investment; but to give good success, the compound non-condensing engine should have at least 140 pounds of steam pressure, and care must be taken that it is not too large for its work. It is better, as far as economy is concerned, to have this type of engine small rather than large for its work.

55. The prevailing practice for terminal pressures is 19 pounds absolute pressure in the United States, while 25 pounds absolute pressure is the practice of some good English builders. If expansion is carried below the atmospheric pressure, the low-pressure cylinder will prove a drag on the engine.

56. Continuous Running With Variable Load.—For continuous running with a variable load, the compound non-condensing engine should be avoided. For this service the compound condensing engine is most suitable, as it works over a wide range of expansion without materially affecting its economical efficiency. The simple condensing engine is well suited for the purpose. If condensing water is not available and if the cost of fuel or demand for power is not sufficient to warrant the use of a cooling tower for condensing water, then the slow-speed non-condensing engine working with a steam pressure of 100 pounds should be the choice.

57. Continuous Running With Uniform But Increasing Load.—For continuous running with a uniform load, which is expected to be increased, however, through extension of the business, we turn naturally to the simple non-condensing engine of high or low grade, depending on the cost of fuel, and arrange to make it into a duplex engine, a condensing engine, or a compound condensing engine, if water is available, as demands are made for increase of power. It would be questionable economy to provide for converting this machine into a compound non-condensing engine on account of the high pressure required to successfully operate this type of machine. In the absence of condensing water, the increased power could be most easily and inexpensively provided by making the engine a duplex.

58. Intermittent Running.—For intermittent running there is much dispute among engineers as to the best type of engine. Here also the cost of fuel enters as a determining element of considerable weight. One of the most familiar examples of intermittent running is the hoisting engine. In the coal fields we find the simplest types of engines with no pretence at economy, while in the Northwestern copper-mining district we find the most elaborate triple-expansion hoisting engines working with 185 pounds steam pressure. Compound condensing hoisting engines are very common in the Northwestern iron-mining district and in the South African gold-mining industry. In both of these localities the cost of fuel is a heavy item; in South Africa it is not only expensive but poor in quality. Condensing water is also very scarce and the prevailing practice is to make the engines compound condensing, using cooling towers to extract the heat from the water, and to use the same water continuously. The contention in respect to high-grade multiple-expansion engines for intermittent work is that if they are more economical in continuous service, the same or nearly the same comparative margin in their favor will result in intermittent work, and it must be conceded that there is reason in the contention.

59. The problem of choosing an engine for certain classes of intermittent work demands the study of local conditions, in which the cost of fuel is probably the most important determining condition. If the cost of fuel is high, say $5 per ton, and the power demanded is large, and the duration of work between stops is 2 minutes, the high-grade multiple-expansion engine should be a paying investment even though the extra expense of providing cooling towers for condensing water be added. If the power required is small, the high-grade engine is very seldom or never used, even though the cost of fuel be large. For reversing rolling-mill service, the simplest and strongest type of engine is used, the high-grade or multiple-expansion engine seldom being used.

60. There is another class of service that might properly be mentioned under the head of intermittent-running engines. This exists in industries that, from their nature, can operate only during a season or part of the year, such as the beet, cane sugar, certain classes of wood fiber, and other industries dependent on season and soil to produce their raw material. There is usually considerable refuse in these industries, having more or less value as a combustible, which, if it cannot be converted into a more valuable byproduct, is generally used as steam-generating fuel. The amount of refuse will quite often determine the type of engine to be used in running the plant. It is usually in excess (except in the beet-sugar industry, where the refuse is a marketable byproduct), and then the simplest, strongest, and cheapest constructed type of engine is chosen.

61. Relay engines can be classified to some extent as intermittent-running engines; but as their term of service is often of considerable duration, especially when they supplement water-power, with the further condition that many water-powers are decreasing in force year by year, the high-grade slow-running engine is generally chosen for this service for large powers and the better type of high-speed engine where smaller power or occasional assistance is required.

INFLUENCE OF LOCATION.

62. Fuel Cost.—The influence of location must be considered in conjunction with other influences, the principal one of these being the price to be paid for fuel; we mention this first because it is the largest and most important influ-ce. It is evident that location is the all-determining uence of fuel cost, as an engine located in the coal-mining may be selected wholly with reference to low first cost, owing to the low fuel cost, while with an engine ated in the iron- and copper-mining districts the first t is a comparatively insignificant item if the power requirement is large, owing to the high price of fuel. This is not only true of the mining industries, but of all uses to which the steam engine can be put. In the New England States the cost of steam coal is about $3.50 per ton. Many of the installations are large, and here we find the best types of large economical steam engines. In many instances, the location is very remote from any source of supply or repair, which fact conduces to the selection of more simple machines, subject, however, to other conditions.

63. Cost of Transportation.—The cost of transportation of large engines working at a high ratio of expansion may influence the selection in favor of the smaller high-speed engine. A peculiar condition of transportation, due to location, is frequently met in the Western mountainous districts, where it is required to install an engine, usually for mining and metallurgical purposes, subject to the condition that no piece shall weigh more than 500 pounds, the reason for this being that the parts must be transported on mule back through dangerous and difficult mountain passes. The simplest types of engines working without expansion and ingeniously divided into a number of parts determined by the size of the machine are usually chosen for these locations. Even here the item of fuel cost enters, which, with perhaps even the water supply, must also be transported in a similar manner as the engine parts; hence, high-grade engines working expansively have sometimes been chosen. It is almost needless to say that the first cost of such an engine is quite high, and that it taxes the skill of the designer and builder to the utmost to produce it.

64. Existing Conditions.—There are other conditions that follow as a natural result of location and which are not dependent on the cost of fuel or transportation, but which, to some extent, go far in determining the type of engine. Existing conditions at any given location may fix the type of machine, as steam pressure available, speed desired, and feasible method of coupling, and the availability of condensing feedwater. The demands of the particular service may require a special type of engine and frequently a machine of special and peculiar construction to meet the demands of peculiar local conditions, such as size and height of buildings.

INFLUENCE OF FIRST COST.

65. The item of first cost is probably the first to enter the deliberations of the constructing engineer and purchaser and throughout the determination stands out as a condition against which all other elements of the determination are weighed. Whether it be given first place or made second to the cost of fuel depends in a great measure on the amount of power demanded in proportion to the quantity of finished product. When the outlay for fuel is small compared with other running expenses, due to a small demand for power in the particular industry, low first cost may prevail; but, on the other hand, if the power requirements are large, even though the price of fuel is moderate, the saving in fuel will soon overrun the interest and extra depreciation charges against the engine high in first cost, but economical in the use of steam. Where a large amount of power is required, even though the cost of fuel is low, if no other conditions enter the problem, such as remote location from any source of supply or facilities for repair or if the labor obtainable is untrustworthy or incompetent to work an efficient installation economically, it will generally be a wise investment to install the high-grade slow-running engine, taking advantage of water for condensation if possible and compounding if available steam pressures are not too low for good results.

INFLUENCE OF FUEL COST.

Introduction.—As was pointed out in Art. **65**, the selection of an engine is to a great extent determined either by the first cost or the cost of fuel. It must be understood that for special services and extraordinary locations and conditions, special engines must be designed to meet the condition, and to a great extent regardless of first cost or the cost of fuel. These are special cases and are almost self-solving.

67. Cheap Fuel.—When fuel is cheap and, as is sometimes the case, must be burned to dispose of it, an engine low in first cost will be the natural choice. This feature should be combined with simplicity, and the engine should be of such design as to require as little attention as possible, since where cost of fuel is of little or no consequence, the whole steam plant is liable to be neglected or left to care for itself, particularly the engine. In this case, one of the single-acting vertical enclosed-crank type of engine or some similar simple engine would be the natural selection.

68. Dear Fuel.—When the cost of fuel is high, recourse must usually be taken to every known means for saving fuel.

The extent of elaboration in that direction depends on the price of fuel. The various means and devices used at the engine to secure small consumption of fuel may be briefly mentioned, as high steam pressure and multiple expansion, steam containing 70° superheat, and steam-jacketing. It may be mentioned here that superheated steam and steam-jacketing are agents working in the same direction—namely, the reduction of initial cylinder condensation, and where one is used the other is superfluous; thus, to steam-jacket a high-pressure cylinder receiving steam of, say, 50° F., superheat would be a non-paying investment. A means of securing high economy in compound-engine performance is to provide an efficient reheating receiver.

Advocates of reheating receivers claim as high as 10 per cent. gain by their use. Low-pressure cylinders are frequently steam-jacketed, the pressure in the jackets being reduced to about one-half the boiler pressure. Many builders will not use jackets on low-pressure cylinders; the best practice, however, favors their use. Serrating or corrugating the outside of cylinder liners and jacket spaces of the heads to make them more active is sometimes practiced, as is also the polishing bright of surfaces exposed to the incoming steam. A thorough system of circulation and drainage of all jackets, as well as means for freeing them of accumulated air, are essential to high economy. Small units should be avoided, if possible, combining them into as few large units as practicable, since large engines are generally more economical than small engines. All cylinders, pipes, reheaters, etc. should be covered with a non-conducting covering.

69. If condensing water is available, it should be used and a vacuum of not less than 3 inches below the indication of the barometer should be obtained. If condensing water is not available, a cooling tower may be used, remembering that it is not possible to obtain a vacuum with cooling towers much better than 6 inches below the indication of the barometer. A primary heater may be used between the low-pressure cylinder and the condenser through which the feedwater is pumped, thus extracting as much heat as possible from the steam. A steam separator should be used at the throttle valves, returning entrained water to the boiler. All other drains should be trapped to an automatic receiver pump to be returned to the boilers. Superheated steam, if heated to an effective stage, say 70° F. superheat, requires poppet valves on the high-pressure cylinder on account of the difficulty experienced in oiling and keeping any kind of slide valves tight. This type of valve is somewhat expensive compared with other types of valves. The superheat is so reduced by the time the steam reaches the low-pressure cylinder of a compound engine that Corliss or gridiron valves may be used on this cylinder. The degree of refinement to which it will be expedient for the engineer to go must be determined by the cost of fuel and the amount that the steam engine demands as against the interest and cost of repairs due to this extra outlay to secure the smaller outlay for fuel.

70. The growing practice in steam engineering is along the lines of greater economy of fuel, and the compound engine is fast finding its way in the coal-mining districts, even where the fuel is culm, which is delivered at a cost not much exceeding 50 cents per ton. Assuming that engineers in coal-mining districts have found it a paying investment to introduce high-grade engines for their purposes, there should be little question as to the advisability of elaboration in districts where the cost of fuel is not uncommonly eight times higher, provided no other militating influences present themselves.

INFLUENCE OF STEAM PRESSURE.

71. The steam pressure available may be a controlling factor where an engine is to be selected to replace an old or overloaded one. The steam pressure available is generally the element that determines whether an engine shall be a single-cylinder or multiple-expansion. Generally the steam pressure should be 100 pounds gauge pressure for effective compounding with a condenser, while 135 pounds should be available for compound non-condensing engines and 160 pounds for triple-expansion condensing engines. It is a matter of fact that the compound condensing engine has such an economical range over wide variations of steam pressure that it can be proportioned to secure results approaching so nearly the triple-expansion engine that the additional outlay for a triple-expansion engine is questionable, except for special and otherwise favorable service, such as high-duty municipal pumping engines, where three cylinders, each actuating a separate plunger, conduce not only to extreme economy of steam, but to a steady flow of delivery water, thus avoiding shocks on both pumping machinery and delivery mains.

INFLUENCE OF DURATION OF SERVICE.

72. By duration of service is often meant the life of the engine for useful work. It frequently happens that machinery is installed for the purpose of developing a doubtful industry or on speculation, when the measure of the doubt will be the controlling influence in determining the degree of efficiency and first cost of the engine. Engines and machinery are sometimes sent to distant localities or those difficult of access to perform a service, and the expenses of transportation are such as not to warrant their return, for it must be borne in mind that when an engine has been used sufficiently to be called second hand, its selling price is reduced below its former value. In such cases, engines of low first cost, but strong, simple, and well built, are usually selected. It is manifestly important in such cases that accident and costly delay by breakage be provided against.

INFLUENCE OF FACILITIES FOR REPAIRS.

73. While this influence is usually not a strong or active one, it nevertheless must be considered and provided for either in the selection of the engine or provisions for keeping it in good working order. This latter may be accomplished by providing the necessary skilled labor, tools, and supplies or by providing spare parts to replace such pieces as have been found by experience most likely to break.

Specifications for engines for foreign shipment frequently include the following parts: 1 pair of connecting-rod brasses for each end of rod; 1 crosshead shoe; 1 piston and rod complete; 1 complete set main and outboard bearing boxes; 1 eccentric strap; 1 complete releasing gear with dashpot, if of the releasing-gear type of engine; 1 steam valve and 1 exhaust valve for each engine or each side of a compound engine. If a condensing engine, the following additional parts are usually specified: 1 air-pump bucket and rod; 1 air-pump delivery deck with valve and guards complete; 1 set of India-rubber valves; duplicate sets of metallic packing to be furnished and all stuffingboxes designed for the use of fibrous packing and suitable glands to be provided.

74. While the influence of facilities for repairs may in many instances of small and even moderate-size plants dictate the simplest and strongest types of steam engines, it does not, in fact, obtain in the case of large installations, where the cost of fuel is high or even moderate. The difficulty can be met by providing either spare parts, facilities for repairs, or relay engines, and in many of the South African plants all three expedients have been found desirable. Even with the simplest and strongest types of engines, if the facilities for repair are not at hand, carrying spare parts is advisable, as accidents and defects are liable with the most carefully constructed engines.

INFLUENCE OF KIND OF LABOR OBTAINABLE.

75. This influence is one that to a great extent is controllable, but the possibility of being compelled to trust an expensive steam plant in unskilled hands even for short periods, and possibly for long ones, may have some weight in determining the type of engine. High-grade economical engines generally require superior intelligence to maintain them in that condition where the item of extra first cost may be assured as a constant and continuing profitable investment. The condition of the kind of labor obtainable may, when other conditions operate against the selection of high-grade engines, carry the choice to the simplest and strongest types of machine, but except in extreme cases it is not of sufficient weight itself to determine a selection.

INFLUENCE OF EXISTING CONDITIONS AND PROBABLE EXTENSIONS.

76. This influence has to a considerable extent been dealt with in former articles, but it will not be out of place to retrace the subject, as it is important, and a little forethought in this direction may save a considerable future outlay with less satisfactory results. Additional engine power may be obtained in many ways. One of the simplest and least expensive practices is the addition of a condenser where condensing water is available, the resulting increase of power ranging from 18 to 25 per cent., depending on the type of engine and degree of vacuum obtained. A good condenser should add from 11½ to 12 pounds to the mean effective pressure. A jet condenser at average temperatures will require about 28 times as much injection or condensing water as there is steam to be condensed; that is, 28 pounds of water at 70° F. will be required to condense 1 pound of steam at 19 pounds absolute pressure. The duplex engine is an excellent and efficient means of increasing power, as is also compounding by adding a low-pressure cylinder if condensing is practicable, even at the expense of a cooling tower. Another means of increasing the power and securing economy where provisions for the duplex engine have not been made is by installing the low-pressure side of a compound by means of an entirely separate engine and running them disconnected. Such engines act with sufficient precision for all practical purposes.

CONDENSING OR NON-CONDENSING ENGINES.

77. The question of whether to run condensing usually depends for its answer on the natural supply of cooling water available, and frequently the supply of cooling water determines the location of the steam plant. This is particularly true of large installations, which, if in large cities, are invariably on the water front. There is a decided gain by the use of a condenser, not only in fuel, but in first cost, as a condensing engine may be made, on the average, 20 per cent. smaller and almost 20 per cent. cheaper than one not provided with a condenser. The cost of air pump and condenser, if directly connected, the air pump being driven by the main engine, should be about 10 per cent. of the cost of the engine for average sizes, and should require about 2 per cent. of the power of the engine to drive it. If an independent condenser is used, which for many reasons is the most desirable arrangement, it should cost about 15 per cent. of the cost of the engine, considering here slow-running high-grade engines. High-speed and inferior engines require condensers larger in proportion, owing to the larger amount of steam used. For large plants, where abundant natural water is not to be had, the cooling tower may be used to extract the heat from the injection water and to use it repeatedly. Sometimes a pond or large pans or tanks on the roof top are devised for the purpose of cooling injection water. Both of these plans of cooling water are slow and very large areas are required and, on account of atmospheric changes, are very uncertain.

78. A number of quite effective water-cooling devices are now being regularly manufactured in units as large as 10,000 horsepower. The general principles of all are the same; they consist of a round or rectangular tower so devised that the delivery water, which the air pump delivers to the top of the tower, in its descent is divided into the greatest possible number of sprays or films; an artificial current of air traverses the surface of the water, extracting the heat from it and rendering it sufficiently cool for service as injection water. These towers, as now constructed, do not require much floor space, a 300-horsepower tower occupying a space of 8 feet × 12 feet. Self-cooling condensers have also been used to avoid steam plants becoming a nuisance in thickly populated city districts, by virtue of the suppression of all exhaust noises, although the law allows an industry to make as much noise as is reasonable and unavoidable in the pursuance of its processes. The cost of fuel usually dictates the policy in regard to cooling towers. If a natural water supply is available, it should be taken advantage of for economical reasons. The jet condenser is the favorite for all land purposes on account of its low first cost as compared with surface condensers.

ONE LARGE ENGINE OR SMALL UNITS.

79. This is a problem not only demanding large experience and most careful study of the details, but it also demands a broad and comprehensive investigation regarding future conditions. The conclusion to put in one large unit is too frequently jumped at. Generally the fewer units there are, the more economical will be the plant; but the condition of service goes far to determine the selection in this respect. In many industries departments require the use of power only a short time or at intermittent periods during the day, and frequently demand widely varying speeds for best effect; they also frequently require to be operated overtime or all night. Long-distance transmission is also involved. In such cases it will often be found that subdivided power will give the best results and, as a matter of fact, some large industries in the New England States formerly driven by one large single unit have adopted the scheme of subdivided power, dividing a single 1,400 horsepower unit into 40 smaller units of varying power. The high efficiency of electricity as a power-transmitting medium has done much to solve the problem of transmission to remote and difficult points and has also contributed to the existence of large single units; but these units should not be so large that they will be run underloaded, for an underloaded engine is about as poor an investment as can well be imagined.

ENGINE FOUNDATIONS.

80. Purpose of Engine Foundations.—One of the most important items in the installation of engines is to provide a suitable foundation, not only in order to rigidly support the machine, but also to absorb the jars and shocks due to its reciprocating motion, because if these are not absorbed, it will result in injury to the engine in question and also to adjacent property, such as other foundations, walls, and structures of any kind resting on the adjacent soil.

81. Supporting Power of Soils.—The foundation, besides having sufficient mass to absorb vibrations, should be spread out over sufficient area to prevent settling. Accepted figures for the supporting power of various soils range from 1 ton per square foot for soft clay to 5 tons per square foot for compact sand bottom, while 200 tons per square foot is given as the supporting power of hard rock in thick strata.

82. Depth of Foundations.—The depth of a foundation will vary with conditions; it should go out far enough below the surface to be free from the effects of frost or the influence of loads borne by adjacent grounds. It is rarely less than 4½ feet for small engines and rarely exceeds 22 feet for the largest engines. A horizontal slow-speed engine foundation for a 40-horsepower engine should be 7 feet deep; a 200-horsepower engine foundation should be 9 feet 6 inches deep; a 600-horsepower engine foundation should be 11 feet 6 inches deep.

Different types of engines require somewhat different designs for their foundations; experience has been and is the only teacher in this subject.

83. Absorption of Vibrations.—High-speed engines stop and start the reciprocating parts many times a minute, and hence set up severe vibrations, which must be absorbed by the foundations. In many engines of this type very careful counterbalancing is used to balance the reciprocating parts in the horizontal direction; this, however, leaves the counterbalance unbalanced in the vertical direction. This balancing in horizontal engines tends to prevent the engine sliding lengthwise upon its foundation, while in the vertical engine the revolving counterbalance tends to slide the engine upon its foundation in a horizontal plane, and the foundation in either case must be of sufficient mass to absorb the vertical and horizontal forces, since engine framings and subbases rarely have sufficient mass to absorb vibrations. When engines are on the upper floors of a building, the scheme of suspending a very heavy mass underneath the floor, but rigidly bolted through to the engine base, and thus making virtually a foundation suspended in air, has proven effective in preventing all vibrations. Care must be exercised in placing engines upon solid rock that some elastic medium, as layers of wood and hair felt, is used between the machine and the rock to prevent vibration of the engine being transmitted to adjacent property.

84. Foundation for Vertical Cross-Compound Engine.—In a vertical two-crank high-speed engine having the cranks placed opposite each other, the vertical forces act to vibrate the machine in a vertical plane parallel to the crank-shaft. Such foundations should be designed with footings, as *a* and *b*, Fig. 1, relieving the center of the foundation and thus preventing any tendency to rock on a central bearing. Fig. 1 gives the general foundation dimensions for a 700-horsepower vertical cross-compound engine running at 100 revolutions per minute.

85. Comparison of Foundations for Vertical and Horizontal Engines.—Horizontal engines usually occupy so much space in a horizontal plane that the supporting power of the soil will be very much above the load if the foundation is made as small as possible—that is, if the bolt holes through the capstones are 6 inches from the center of hole to the edge of the stone for a 1-inch bolt and 10 inches for a 3-inch bolt. There should be 4 to 8 inches of masonry outside the capstone, and if the sides are carried down straight, sufficient bearing area will be covered, except in cases of alluvial soil. Vertical engines, owing to the small

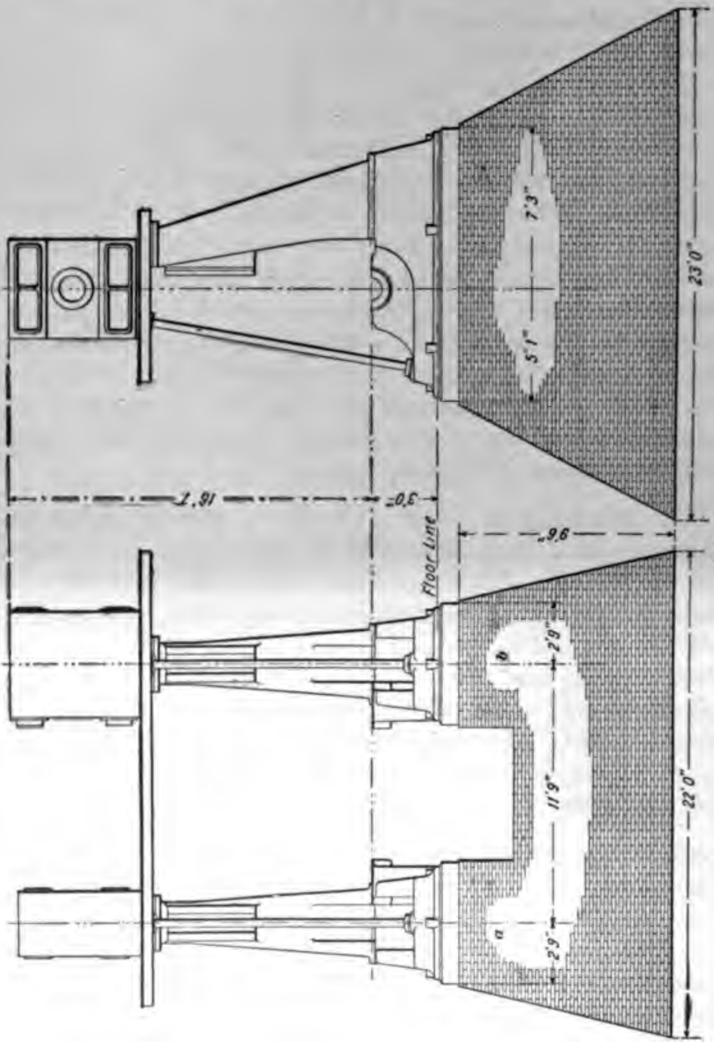


FIG. 1.

horizontal space required, should have deeper foundations than horizontal engines, and to secure sufficient bearing area, the sides may be battered to any desired extent. Bearing surface for vertical engines should be carefully calculated with reference to the nature of the supporting soil, including, of course, the weight of the foundation itself as well as the engine that it supports.

86. Foundation Material.—The material of which a foundation is made depends very much on the location and the kind of material available. Brick is the most common material; dressed stone laid in cement mortar is sometimes used; concrete is growing in favor for engine foundations. When brick is used, it should be first quality hard brick laid in Portland cement mortar; lime mortars are not suitable for engine foundations on account of their tendency to disintegrate under vibration. Stone foundations should also be laid in cement mortar.

Foundations of concrete are coming more into use; they are constructed by first providing a level and suitable footing upon which a casing of timber, embracing the outlines of the foundation, is built. This is open at the top and bottom. The foundation bolts are suspended in pipes, old boiler tubes, or wood launders, leaving a space of at least 1 inch all around the bolt; successive layers of cement concrete are thrown in and well rammed until the desired height is reached.

87. Foundation Footings.—Foundation footings are in some cases required, most frequently on the water front, where it is necessary to go many feet deep to find a sufficiently hard bottom to support the load. This is accomplished most commonly by piling, which consists of driving long sticks or timbers, as *a*, *a*, Fig. 2, down to hard bottom, placing them 2½ to 4 feet apart from center to center. A timber grating *b* is fastened to the tops of the piles and a layer of concrete *d* is deposited. Planking, as *c*, *c*, is sometimes put on the framing, which distributes the pressure, but it is considered objectionable, as it prevents any connection between the superstructure and the concrete and increases the liability of sliding. The space between the piles is frequently filled with rubble, clay, or concrete, and upon this footing the foundation proper is built. A properly driven pile, well supported against lateral flexure, may bear from one-eighth to one-tenth the crushing load, which

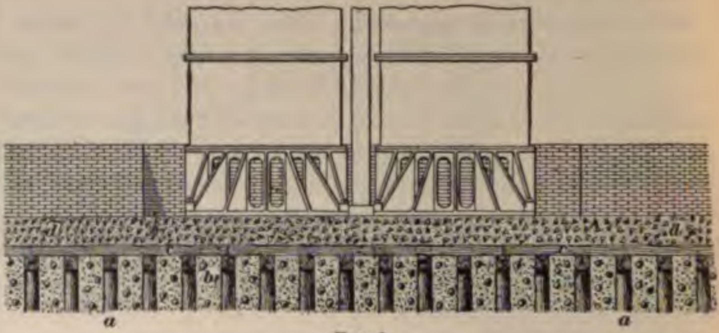


FIG. 2.

varies between 5,700 and 8,500 pounds per square inch. A pile 7 inches in diameter will bear about 12 tons. A pile can support a load of 25 tons when it refuses to move more than $\frac{3}{8}$ inch under thirty blows of a monkey weighing 1,200 pounds and falling 4 feet.

88. When rock is struck at a high level, special footings to prevent vibration must be constructed. An underlying stratum of timber or rubble or of both has been tried with questionable success; a layer of 2 or 3 feet of sand constrained laterally by a casing to prevent displacement has proved quite effective. The sand is also filled in around the sides of the foundation block. A heavy layer of asphalt is also effective in breaking engine vibrations before they reach the transmitting rock upon which the foundation is built.

89. Capstones.—Brick and dressed stone foundations usually require capstones to make a good job. These are usually granite and vary in thickness from 8 to 24 inches. Concrete foundations usually require no capstones. Instead of capstones, cast-iron sole plates are sometimes used. They are usually thin, about $\frac{7}{8}$ or 1 inch thick, with an upturned ledge around the top to keep oil from the foundation and sufficient ribs below to give stiffness to the plate, and are provided with raised planed facings to match the engine parts. They are not more expensive than good capstones and are a superior job. Every precaution should be taken to keep oil from reaching the foundations, as it will dissolve the cement.

90. Many erecting engineers make a practice of setting the capstone for the outboard bearing from $\frac{1}{4}$ to $\frac{1}{2}$ inch lower than the actual figures called for and shim up the sole plate with wrought-iron strips or plates, the contention being that if the stone is set low the bearing can be shimmed up, but if a little too high it is a difficult matter to do anything with it but to chip off the top or take it up and reset it.

91. Foundation Bolts and Washers.—The bolts are always made of wrought iron, and should be of good quality, as Burden's best-best, Catasauqua, or some equivalent brand. Foundation bolts are usually made in length nearly the full depth of the foundation, and in important work they are made upset, that is, the threaded portion is made enough larger in diameter that the bottom of the thread is still a little larger than the body of the bolt. By this means the stretch due to the pull on the bolt is distributed over the long body and not only over the threads, as would be the case if the thread were cut on a rod of uniform diameter. Foundation bolts vary in diameter from $\frac{7}{8}$ inch in small engines to 4 inches in the largest types of land engines.

92. Foundation washers are commonly made of cast iron, but for small engines wrought-iron plates from $\frac{3}{8}$ to $\frac{3}{4}$ inch thick are used. In many locations it is not possible to provide pockets for access to the foundation washers; in such cases it is the best practice to provide cast-iron washers *a*, Fig. 3. With this style of foundation washer it is possible to adjust the bolts to any desired height, or even to remove them and replace them at will. Some builders make a practice of tapping the top of the foundation bolt for an eyebolt to facilitate lifting or lowering the bolt in place.

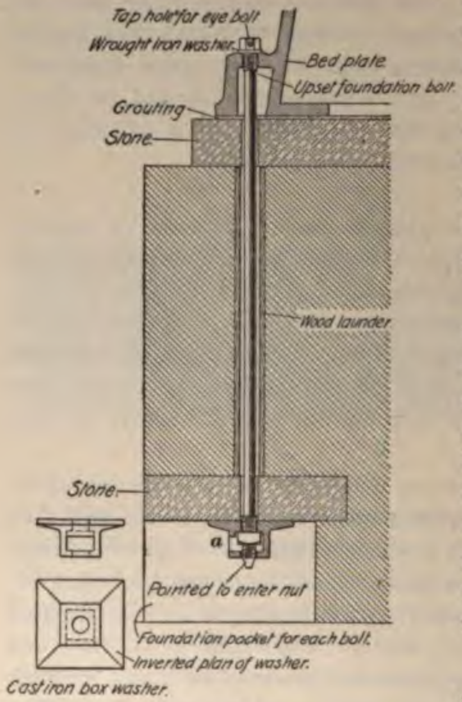


FIG. 3.

93. In large work it is quite important to have the foundation bolts removable, for if they extend through high framing or high bosses, it is necessary to lift the castings over the top of the bolts, which adds much to the cost of erection.

94. With many of the mining companies it is the custom to build foundations with pockets according to the drawing, but omitting the holes for the bolts entirely. When the capstones are leveled and grouted, they lay off the holes from the castings and drill holes for the bolts with a diamond drill. Sometimes the engine is erected, lined up, and grouted, and then the holes for the foundation bolts are drilled in place. With this arrangement it is essential in the design of the machine to see that holes for foundation bolts are not covered by any part or projection of the machine, or if this cannot be done, to provide a hole to pass the foundation bolt through.

95. Foundation Templets.—Foundation templets are used for the purpose of accurately locating the bolts, bringing them to the proper height, and holding them in position while the masonry is being built. They are constructed of wood with blocks of varying height to suit the height of the engine bosses. The center line of the engine is carefully marked on the templet with correct relation to the bolts, and at right angles to it is marked the center line of the crank-shaft. Suitable marks and dowels to facilitate putting it together, if of such dimensions that it is necessary to ship it in sections, are also provided. The foundation bolts should not be allowed to hang on the templet, but rest on stone or bricks. The bolts, if they have no adjustment, should be set originally from 1 to 2 inches higher than required, as the gradually increasing weight of the foundation will sink the soil upon which it was started, and hence the bolts may not project through the engine casting unless this precaution be taken. Bolts when used in pockets or box washers should be pointed to facilitate entering the nut. Templets for out-board bearings or compound or triple-expansion engines are usually not connected, the relative locating being done from the foundation drawing. This work is usually done by the engine contractor, who sends only skilled men for this duty, but if not done by the engine contractor, it should be checked and approved by him at a sufficient time before the work of erection commences to have all defects, real or alleged, made good satisfactory to both parties.

96. Supporting the Templet.—The supporting of the templet must be left to the ingenuity of the erecting engineer; generally it should be supported outside of the foundation, but there is no real objection to building the supporting posts into the mass and sawing them off when the foundation has reached about 18 inches from the top.

97. Setting the Templet.—Setting the templet is a simple matter, but it must be carefully and exactly done, especially if the engine is to drive a shaft by belt or gearing. The first and most important thing to do is to have the templet exact; particularly the crank-shaft center line must be square with the center line of the engine. This can be tested by measuring off from the intersection of the two lines 6 feet on the shaft center line and 8 feet on the engine center line and adjusting the lines with their intersection as an axis until the hypotenuse of the triangle measures exactly 10 feet. Then having given the templet the correct relative position and the correct levels, the only remaining thing to do is to set the center line of the crank-shaft parallel to the line shaft or its established line. This can be done by plumbing down from or to the line shaft and measuring at both extremities of the crank-shaft center line from that line to the plumb-lines. Some mechanics establish the center lines of the engine and the crank-shaft by stretching the lines considerably above the templet height and set the templet from these lines by plumbing down. It is not a very easy matter to stretch two lines in the air at exactly right angles, and hence it is a better plan to have the lines on the templet and exactly right, and then to work to the crank-shaft line as the most important one. Having properly set the templet, the bolts are passed down through the holes and the washers and nuts put into place. Each bolt must rest on a large stone slab. Old pipe or wooden boxes should be put around the bolts to allow considerable lateral adjustment of the bolts.

98. Placing the Engine.—When the foundation is built up to within 2 feet of the top, the templet is removed and the top of the foundation built and carefully leveled by means of sensitive levels and straightedges. If the engine is large and the wheel is in halves, one-half the wheel should be placed in the wheel pit first; then the framing, outboard bearing, and shaft may be placed and finally the cylinders and valve gears. The engine is leveled in a plane parallel with the center line of the shaft and cylinder by means of sensitive spirit levels and all parts resting on the foundation are wedged and shimmed up. The bolts are tightened down moderately and the space between the bed-plate and foundation, which will vary from $\frac{1}{4}$ to $\frac{5}{8}$ inch, is filled with grouting.

99. Grouting.—The grouting may be made of iron borings mixed with cement, sal ammoniac, sulphur, and water in about the following proportions: 2 parts of sal ammoniac, 1 part of sulphur, 5 parts of cement, and 40 parts of iron borings mixed with enough water to make a heavy paste. Sometimes melted sulphur alone is used, but one of the very best groutings and the most easily applied is pure Portland cement. The rust joint must be driven in, while the sulphur and cement will flow in, suitable dams being constructed to constrain it to its proper place. Bolt holes should be filled with liquid grouting. Some builders who use box bedplates fill the entire bedplate with concrete to give it solidity and to reduce the tendency to magnify knocks into pounds.

100. Setting the Outboard Bearing.—The setting and alining of the outboard bearing should be carried along with the progress of the other work, but its final adjustment is important and should be done last, by the aid of lines representing the center lines of the cylinder and crankshaft. All outboard bearings should be provided with a sole plate and means for lateral adjustment either by screws or wedges, while some are provided with means for vertical adjustment.

ELEVATORS.

(PART 1.)

GENERAL DESCRIPTION OF ELEVATORS.

INTRODUCTION.

1. Definition.—The term **elevator** is applied to that class of hoisting machinery in which a cage, cab, car, or platform is raised and lowered between fixed stops or landings.

2. Principal Parts.—In all complete elevators the following principal parts are easily distinguished:

1. The motor.
2. The car (cage, cab, or platform) and its principal guides.
3. The devices transmitting power from the motor to the car.
4. The counterbalance weights and their guides.
5. The controlling devices.
6. The safety devices.
7. Accessories.

MOTORS AND CLASSIFICATION.

3. Various kinds of motive power and, consequently, motors are used to run elevators. In practice, the classification of elevators is made according to the motive power used. The most generally accepted one, which is also the one that we shall adopt, is as follows: *Hand-power elevators, belt elevators, steam elevators, electric elevators,* and *hydraulic elevators.*

CARS AND GUIDES.

4. It is evident that elevator cars must be different for various purposes. All of them, however, have a **platform** upon which the load rests, and with few exceptions, as in sidewalk elevators, two upright **posts** connected by a **crosshead** to which the ropes are attached. Each upright carries two **guide shoes,** one on top and one on the bottom, which fit over the **guides.** The latter consist either of hardwood strips of square cross-section or bars of T iron carried up inside the hoistway and attached to suitable supports. According to the location of the elevator shaft in the building and the accessibility of the guides, they are placed either in the center of two opposite sides of the shaft or in two diametrically opposite corners, necessitating the upright posts of the cars to be placed in like manner with reference to the platforms. In the first case they are called **side-post elevators;** in the other case, **corner-post elevators.** The guide shoes are usually of cast iron, and in the case of iron guides are lined with Babbitt metal.

For freight service the cars are of the simplest kind; they are generally made of wood with iron fixtures and bracings. For passenger service a complete cage is built upon the platform, preventing any possible contact of the passenger with the hoistway. Passenger cars are now mostly built wholly of metal, though many wooden ones are in operation. Various styles of cars are shown in subsequent illustrations.

TRANSMITTING DEVICES.

5. Various transmitting devices are used with different kinds of motive power. Hydraulic elevators and a certain electric elevator have peculiar transmitting devices of their

n, which will be described in connection with these ele-
ors. All belt and steam elevators and the majority of
id and electric elevators are of the **drum type,** that is,
ı type in which the transmitting devices include a drum
rope. All these elevators, therefore, have certain pecu-
ities in common, which are pointed out beforehand to
id repetition.

. Side Travel of Ropes in Drum Elevators.—An
rent feature of the rope-and-drum drive is the deflection
he rope as it winds upon the drum. Let *D*, Fig. 1, be
winding drum and *S* the nearest sheave from which the

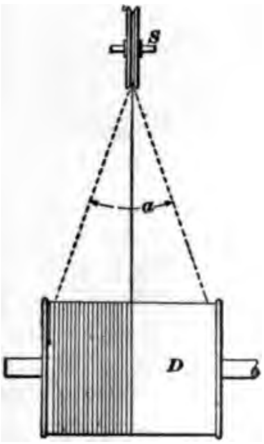


FIG. 1.

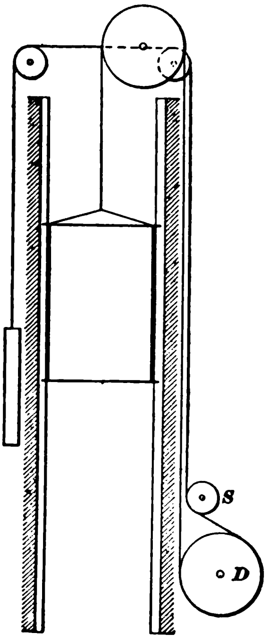


FIG. 2.

passes on to the car. It is plain that only at a certain
tion of the car the rope runs over the sheave exactly
ight; in all other positions it must be guided into the
ive. If the distance between the drum and the nearest

sheave is great, as, for instance, when the rope passes straight up from a drum located at the foot of the hoistway to an overhead sheave, the deflection measured by the angle a, Fig. 1, is but small and readily taken care of by the depth of the grooves in either sheave or drum. But if the distance is small, danger exists that the rope will jump the grooves of the drum and "ride" on itself, which may evidently cause accident. Such small distances between the drum and the nearest sheave are frequently unavoidable. Thus, in the case shown by Fig. 2, where it is required that the ropes of both the car and the counterweight shall run within the hoistway, the hoist rope must be led over an **idler** S very near the drum D, and the counterweight rope, in the case shown, will surely "ride" if no provision be made against it. These idlers are, therefore, so mounted on their shafts that they can follow the ropes as they wind upon and unwind from the drums. Such a traveling idler is sometimes spoken of as a **vibrator.** In most cases it is found sufficient to mount the idler loosely on a smooth shaft and to rely on the pressure of the rope against the sides of the groove in the idler to shift the idler along. That careful lubrication is essential to the proper working of this arrangement is evident.

7. The constant chafing of the rope against the sides of the idler groove, which is unavoidable in the arrangement just mentioned, is an objection, and if considered of sufficient influence on the life of the rope, is avoided by giving the idler a positive motion in the direction of its shaft. Figs. 3 and 4 show two ways of accomplishing this.

In Fig. 3, the idler shaft a is connected to the drum shaft by a chain and sprocket wheels, the hub of the sprocket wheel b on the idler shaft being a nut fitting over a square thread cut on the shaft and being held from moving sidewise. This causes the shaft to move in the direction of its axis, a feather preventing it from rotating. The idler c turns loosely on the shaft, but moves back and forth with it, due to collars on the shaft.

8. In the arrangement shown in Fig. 4, the chain connection is dispensed with. The idler shaft *a* is threaded but is held stationary, and the idler hub is a nut, so that while the idler revolves by the friction of the rope it travels back and forth. Since there is no positive connection between the drum shaft and the idler shaft, any slippage of the rope on the idler will bring the arrangement out of adjustment. For this reason the following plan for automatic readjustment was adopted by the Otis Elevator Company.

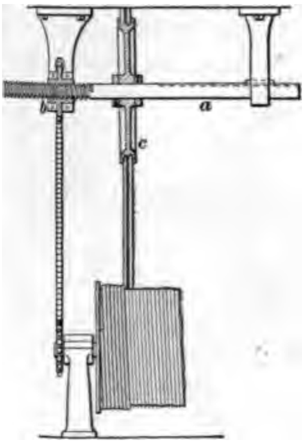


FIG. 3.

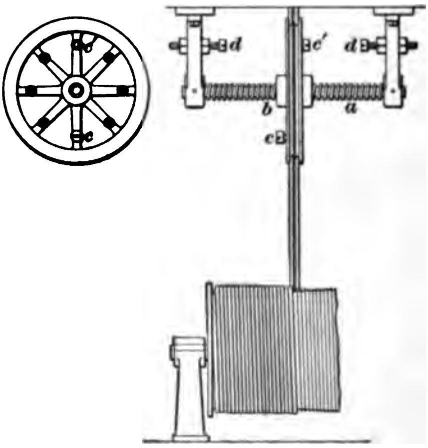


FIG. 4.

The idler is provided with stop screws *c* and *c'* that engage at the end of the travel with fixed stops *d*, *d* on the shaft supports. If for any reason slippage has occurred and the idler lags behind, it will be ahead of the rope on the following return trip and engage the stop on the shaft support before the drum comes to rest; the idler being thus prevented from turning, the rope will slip until the drum stops; on the following trip the idler will leave the stop at once and, thus readjusted, will follow the rope correctly. To allow of a fine initial adjustment, the idler has eight spokes, each drilled and tapped to receive the screw stops *c*, *c'*.

9. Absorption of Vibration Due to Gearing.—An inherent feature of all drum elevators is a certain amount of unpleasant vibration transmitted from the gearing through the drum and hoisting rope to the car. This vibration is especially noticeable in spur-geared machines; but it also exists in worm-geared ones, owing to the fact that a certain amount of backlash, be it ever so little, always exists. To reduce these vibrations to a minimum, elastic buffers, generally of rubber, are sometimes interposed between the drum and the next adjacent gear. Fig. 5 shows a way in

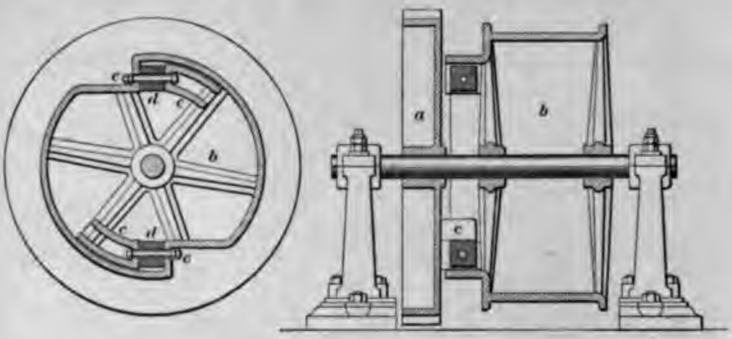


FIG. 5.

which this may be done. The gear-wheel *a* is keyed to the shaft, while the drum *b* is loose. The gear-wheel drives the drum by means of drivers *c*, *c*, which are cast in one with the gear-wheel. These drivers, instead of butting directly against metallic surfaces of the drum, butt against rubber blocks, or buffers, *d*, *d*. These buffers must be given a certain amount of initial tension, which is accomplished by the tie-bolts *e*, *e* that tie the drum and the gear-wheel together. The end view shown in Fig. 5 is taken between the gear-wheel *a* and the drum *b*, which accounts for the fact that while the drivers *c*, *c* are seen, the gear-wheel is absent. The tie-bolts must have jam nuts or some other efficient nut-locking device, which should be examined occasionally to see if the bolts have become slack.

COUNTERBALANCING.

10. In any elevator, the weight of the car and its fixtures is constant, and hence is easily **counterbalanced** to any extent desired. The simplest way to do this is to attach another rope, besides the hoisting rope, to the car, leading this second rope over one or more overhead sheaves and suspending from it the counterbalance weight, or the **counterweight,** as it is generally called, as shown in the diagrammatic illustration given in Fig. 6. In order that the car may descend when empty, the counterweight must, when so attached, always be less than the weight of the empty car with its fixtures. Evidently, with such an arrangement, no power is needed for the down trip of the car, while on the up trip, the motor must develop enough power to raise the maximum load, plus the unbalanced weight of the car. In all types of elevators in which the motor furnishes power only on the upward trip of the car, as in hydraulic elevators, for instance, the arrangement shown in Fig. 6 is the only method of counterbalancing available.

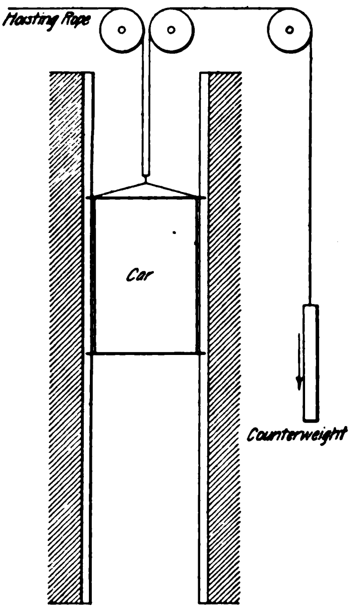


FIG. 6.

11. If in an elevator the power can be applied during the down trip as well as during the up trip, then not only the full weight of the car can be counterbalanced, but also a part of the load. An elevator thus counterbalanced is said to be **overbalanced.** This possibility exists in all drum elevators, as the motor and drum are reversible. They are, therefore, overbalanced, except when other considerations make it undesirable, to the extent of the average load by attaching the counterweight to the drum and winding the rope in an opposite direction to that of the hoisting rope, as shown in Fig. 7.

The advantage of overbalancing is easily apparent. If the load on the car is equal to the average load, no power is

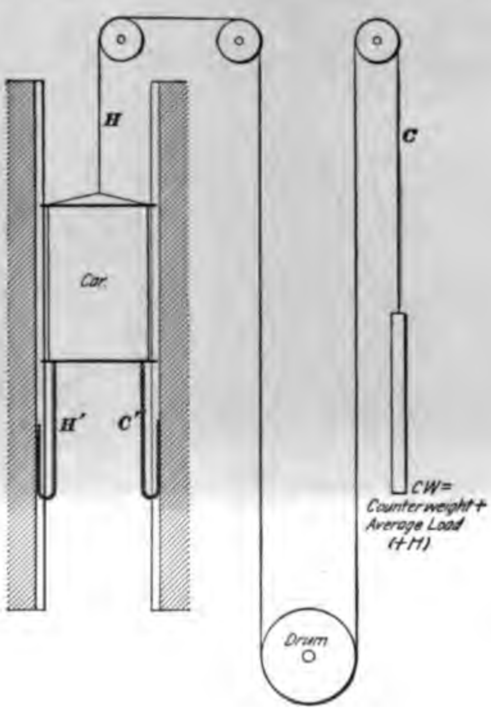


FIG. 7.

needed besides that necessary to start the machinery and keep it moving against frictional resistances. If the load is equal to the maximum load and the car is going up, the motor must furnish power enough to raise the difference between the maximum and the average load; or if the latter is one-half the maximum load, to raise one-half of the maximum load. If the car is going down empty, which is the other extreme possibility, the motor must raise the counterweight, that is, the weight of the average load. Thus a motor can be used of greatly smaller capacity, which means smaller size, less weight, and smaller cost. In connection with electric drum elevators, overbalancing also tends to equalize the current consumption.

12. By an arrangement somewhat different from that shown in Fig. 7, the stress in all the ropes and the pressure on the drum-shaft bearings may be diminished. In Fig. 8

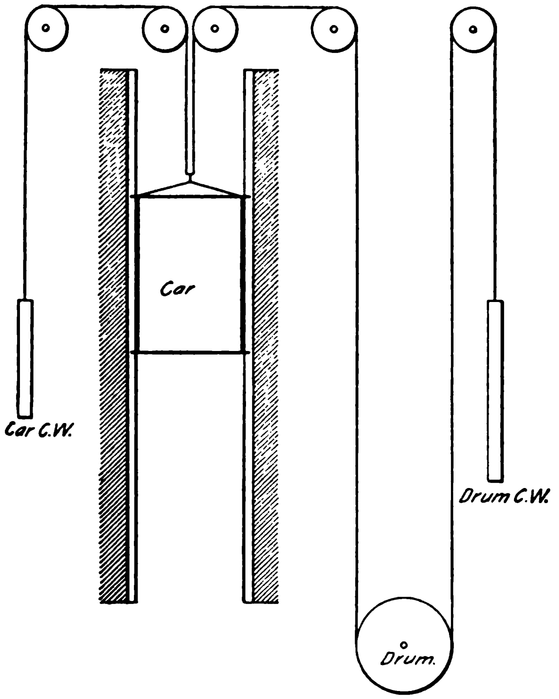


FIG. 8.

there are shown two counterweights, one attached directly to the car and the other to the drum. The car counterweight must evidently be less than the weight of the car in order to allow the car to descend when empty; the other counterweight is equal to the remaining portion of the car weight plus the average load. If the car counterweight is, for instance, one-half of the car weight, then the stress in the drum-counterweight rope and the hoisting rope is less than in the arrangement shown in Fig. 7 by one-half of the weight of the car, and the pressure on the drum-shaft bearings is less by the whole weight of the car.

13. For high lifts, the weight of the ropes themselves is a considerable item, making the counterbalancing change for different positions of the car. To avoid this, balancing chains having the same weight as the ropes to be balanced are used and are hung from the bottom of the car, either reaching all the way down to the bottom of the hoistway, in which case the chain must have the same weight per unit of length as the ropes, or reaching down only to the middle of the shaft and fastened there, in which case the chain must have a weight per unit of length double that of the ropes to be balanced. This method is indicated in Fig. 7.

The ropes to be balanced here are the hoisting rope from the car to the overhead sheave and the counterweight rope from the counterweight to the overhead sheave, denoted, in Fig. 7, by H and C, respectively. The former can be balanced by a chain H' of equal weight and an increase of the counterweight by the same amount, while the rope C can be balanced simply by a chain C'. Of course, two chains would be actually used, each weighing $\frac{H + C}{2}$.

14. All counterbalancing means an addition to the moving masses of the elevator, which, again, means an increase in the power required to set these masses in motion, as well as greater braking power to stop them. These considerations make it desirable in certain elevator types to forego the advantages of overbalancing.

COUNTERBALANCE WEIGHTS AND GUIDES.

15. The **counterweights** consist generally of cast-iron blocks carried in a frame or on a rod or rods and guided by suitable guideways. The blocks are made long and wide, but thin, in order that they may take up but little room. In hydraulic elevators the counterweights are sometimes attached to the piston rods, either inside or outside of the hydraulic cylinder.

The counterweight guides are made of angle or T iron, seldom of wood.

CONTROLLING DEVICES.

16. Kinds of Controlling Devices.—The controlling devices of all elevators consist of a **power control,** that is, means for shutting off, turning on, and regulating the power at will to start and stop the car, and some kind of a **brake,** the function of which is to effect a prompt but gradual, and therefore safe, stoppage of the car.

The power control and brake are essential parts of the motor in each case and are, therefore, located near the same; they are naturally different for different kinds of motive power, and will be described at length in connection with the various types of elevators. There is, however, with respect to the controlling devices a certain feature common to all.

Since most elevators are operated from the movable car, some flexible connection must exist between the same and the controlling devices on the motor. The means for making this connection, which we will call **operating devices,** are either mechanical or electrical. The latter is used to any extent only with a certain kind of electrical elevators, while the mechanical connection is employed on all types in the shape of a **shipper rope** running all the way from the top to the bottom of the hoistway and either simply passes through the car or is connected with some apparatus inside the car.

17. Different Operating Devices. — The simplest arrangement is a plain endless rope hung over one or more idlers and attached to the **shipper sheave** or a lever, so that a pull either up or down on the shipper rope moves the sheave or lever, which is located on or near the motor and is mechanically connected to the controlling devices of the same. This simple arrangement, which is shown diagrammatically in Fig. 9, is open to several objections, one of which makes its use undesirable in connection with all motors requiring a delicate adjustment of the controlling device, such as hydraulic motors controlled by a pilot valve and electric motors, inasmuch as the operator has no means of telling the exact position of the controlling device. Another objection is that there is necessarily a great deal of sliding of the rope through the hand of the operator, which is not only inconvenient, but may prove dangerous from broken strands. The operator should provide himself with a leather glove or use a piece of rubber hose split lengthwise.

FIG. 9.

18. In order to overcome the objections to the simple shipper rope, various devices have been invented and put into use. They all have the object of changing the up-and-down pull of the rope into the motion of a lever or crank. In Figs. 10 to 16 are shown a number of these devices as actually used, particularly in connection with hydraulic elevators. In the arrangement shown in Fig. 10, there is a three-armed lever A in the car, the long arm of which is to be swung to the right or left by the operator. To each of the short arms is connected a rope R running down over an idler carried by another three-armed lever *B* pivoted at the bottom of the hoistway. From the idlers on lever *B* the ropes *R*, *R* pass up again and over idlers fixed at the top of the hoistway, as shown. On the ends of the ropes are counterweights which are equal, and each is somewhat heavier than the equivalent force necessary to move the controlling device of the motor, which is mechanically connected to the third arm of the lever *B*. As will be easily understood, the whole arrangement is in equilibrium in any position of the lever *A*, but the equilibrium is disturbed as soon as a pull is exerted on it in either direction, in consequence of which the second lever *B* will follow the motions of *A* and stay in a position corresponding to that of *A*. The top idlers are shown in the diagram on separate shafts or studs; in reality they are placed side by side and the downward rope passed through the counterweight nearest to it, as shown in Fig. 10 (*a*). The weight is thus guided on the rope. To prevent abrasion of the rope, a rubber ring is inserted in the hole through the weight.

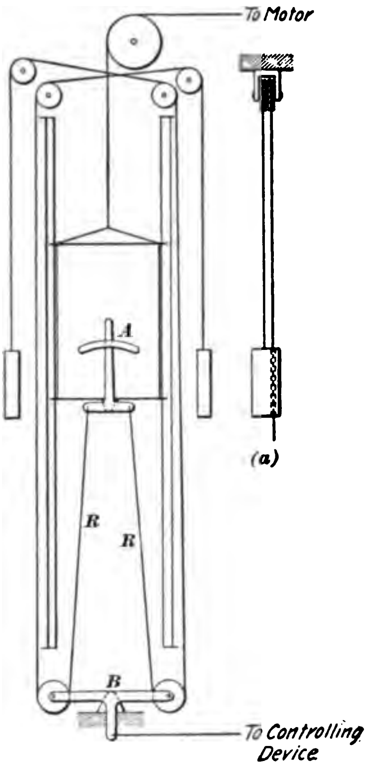


FIG. 10.

19. Another arrangement is shown in Fig. 11. The shipper rope is led from a fixed point *a* at the top of the hoistway over two idlers *b* and *c* mounted on a lever *L* pivoted to the car and handy to the operator. From the idlers *b* and *c* the rope is carried farther down, around the shipper sheave S, and thence back upwards over two more idlers c' and b', also attached to the lever L, and is finally fastened at the top at d. While the car moves up and down, the shipper sheave S is stationary unless the lever L is moved. By moving L upwards, the part $b\ c\ S$ of the rope is doubled up more, while the part $b'\ c'\ S$ is straightened out an equal amount, causing the sheave S to take a position depending on that of the lever L, in which position it remains until the lever is moved again. In an actual machine the idlers are mounted on studs, b and c' on one stud and b' and c on another stud, which are carried on a lever outside the car; the pivot of the lever is carried through into the interior of the car, where it carries the handle L' [see Fig. 11 (a)]. A hand wheel may be substituted for the handle L'.

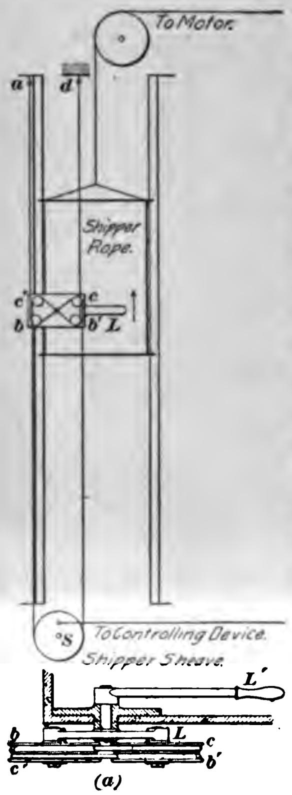


FIG. 11.

20. The same idea that underlies the arrangement of Fig. 11 is embodied in Fig. 12. Here the two branches of the rope are deflected so as to pass over two idlers i, i' on the same stud and are attached to a lever pivoted to the car; the other idlers a, b, c, and d are fixed to the car.

21. The devices shown in Figs. 11 and 12 necessitate idlers to be carried on the car, where they must, owing to the limited space available, be necessarily small. This is detrimental to the rope, especially since it is bent in opposite directions in quick succession. These objections lo not prevail in the arrangement shown in Fig. 10, where he idlers may be ample in diameter and the ropes are ent in one direction only.

22. An improvement on Fig. 12 is the operating device hown in Fig. 13, in which the ends of the ropes are ttached to the car instead of to moving weights; a single

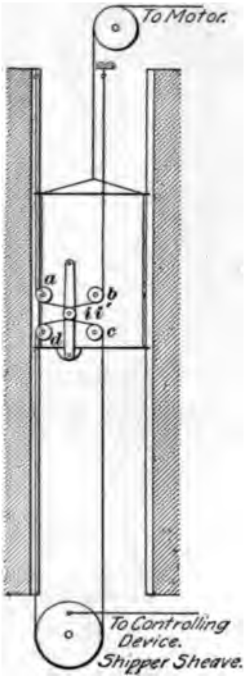


FIG. 12.

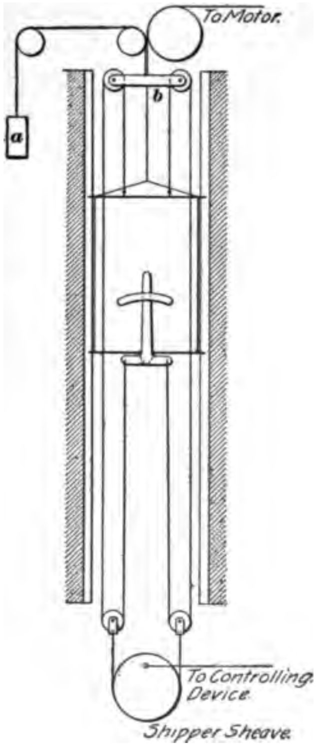


FIG. 13.

ationary weight *a* attached by a rope to a cross-bar, or ame, *b* carrying the upper idler is substituted for the oving weights.

23. This arrangement has been still further improved on in the manner shown in Fig. 14. The ends of the ropes that were attached to the car in Fig. 13 are here also attached to the lever arms, and the two ropes leading from the lever of the two idlers are crossed. It can easily be seen that by this means the motion of the lever gives twice as much motion to the sheave as in the arrangements shown in Figs. 10 and 13.

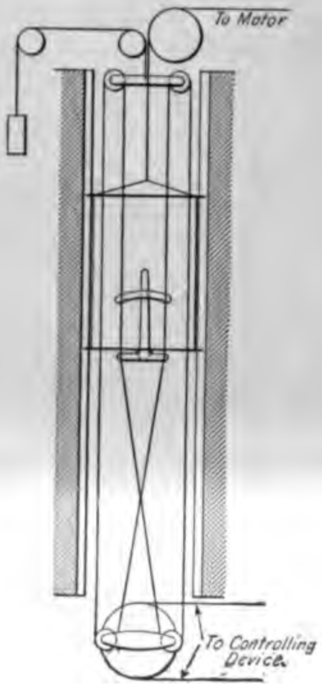


FIG. 14.

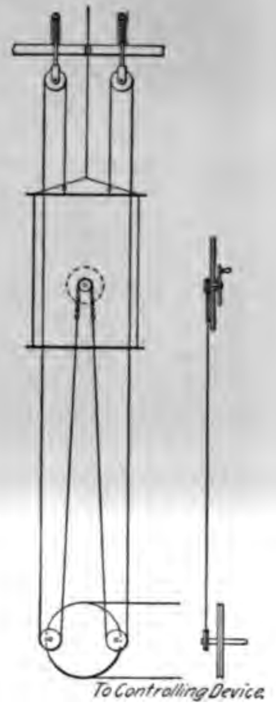


FIG. 15.

24. The operating device last described is known as the **Otis lever, or operating device,** and is now used almost exclusively on the hydraulic elevators of the Otis Elevator Company. By substituting a hand wheel or crank for the lever in Fig. 13 and attaching the lower idlers to the shipper sheave, a modification shown in Fig. 15 is obtained that is found on a good many elevators of the Otis make and is called a **hand-wheel operating device.** By elevator

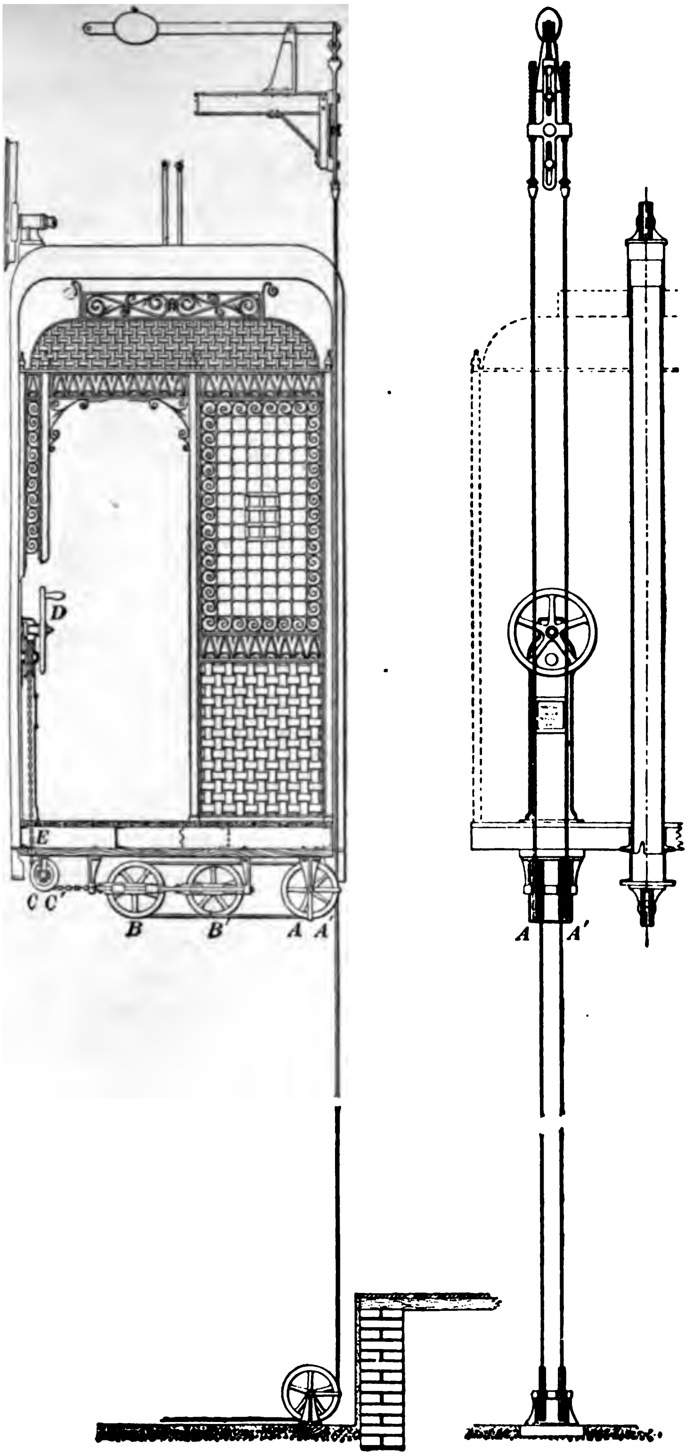


FIG. 16.

runners the operating devices are often called **controllers** and are spoken of as **lever controllers, hand-wheel controllers,** etc. Since the term *controller* is also given to the combination of switches and resistances constituting the controlling device proper in electric elevators, the practice just mentioned is not followed here in order to prevent confusion.

25. A common arrangement of a hand-wheel operating device is shown in Fig. 16. The sheaves *A*, *A'* are stationary, but the sheaves *B*, *B'*, being loosely mounted on guide rods, can be shifted by means of the hand wheel *D* and chain *E*. The chain runs over the wheels *C*, *C'*. After the explanations of the operation of the different operating devices that have been previously given, the operation of this one will be easily understood.

SAFETY DEVICES.

26. We can divide the **safety devices** used on elevators into two distinct classes: those that control the power supply, which we shall call **motor safeties,** and those that control the car independently of the power supply, which we shall call **car safeties.** The former must necessarily be treated in connection with the various styles of motors used; the latter allow of, and their importance warrants, a treatment by themselves, which will be given in its proper place.

ACCESSORIES.

27. Among **accessories** we shall class all those various appliances designed to prevent accidents from causes other than failure of the elevator or any of its parts, and to increase the comfort of the traveling public and the efficiency of the elevator service. Such appliances are automatic gates, doors, and hatchways, signals, indicators, etc.

HAND-POWER ELEVATORS.

CONSTRUCTION.

28. When an elevator is to be used but little, and especially if speed of the car is not essential, it does not pay to use steam or other motive power; **hand-power elevators** are then useful. With few exceptions they are installed for freight service only.

Figs. 17, 18, 19, and 20 show several types of hand-power elevators. Those shown in Figs. 18 and 19 are made by Morse, Williams & Co., of Philadelphia, Pennsylvania, and those shown in Figs. 17 and 20 by the A. B. See Manufacturing Company, of Brooklyn, New York.

29. Motor.—The *motor* of a hand-power elevator is represented either by a shaft actuated through a rope sheave and endless rope, the latter being pulled in either direction by hand, and examples of which are shown in Figs. 17, 18, and 19, or it is represented by a crank driving a windlass, as shown in Fig. 20.

30. Transmitting Devices.—The *transmitting devices* consist of spur gearing in connection with either a drum for a rope or chain, as in Figs. 17, 19, and 20, or a friction sheave, as in Fig. 18.

31. Operating Devices. — The *operating device* is a manila rope, preferably a four-strand and "stevedore" rope; the hoisting and counterweight ropes are generally wire ropes. In the sidewalk elevator shown in Fig. 20, chains take the place of the rope.

32. Cars.—The cars in Figs. 19 and 20 are different from the ordinary cars, inasmuch as they are supported on all four corners.

Small hand-power elevators are used largely for carrying small loads in dwellings, restaurants, libraries, etc., and are called **dumbwaiters.** The cars of these then take the shape of a box with or without shelves.

33. Guides and Counterweight.—In the elevat shown in Fig. 19 *guides* are provided on one side onl The *counterweight* in Fig. 17 is hung from a separate drur

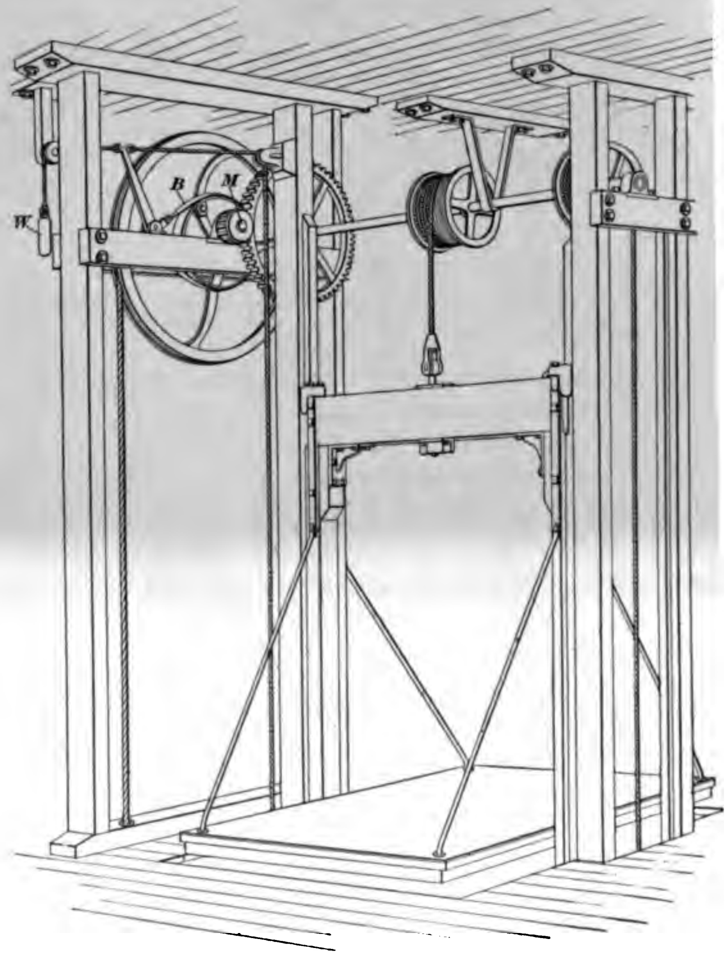


FIG. 17.

in Fig. 19 it is hung from one of the hoisting drums; i Fig. 18 it is attached to the other end of the hoisting cable and in Fig. 20 the counterweight is dispensed with entirely

When hand-power elevators are balanced, they are generally overbalanced.

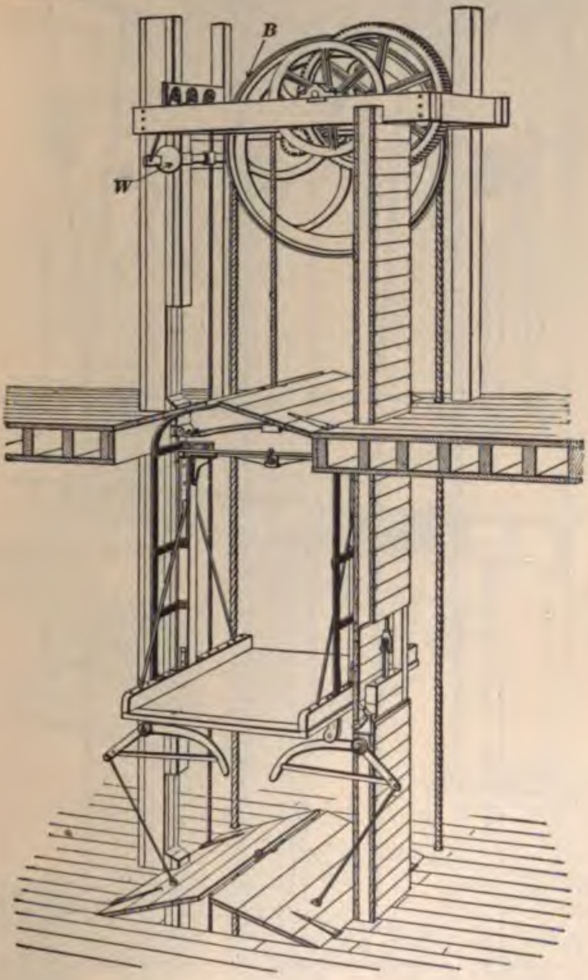


FIG. 18.

34. Controlling Devices.—The *controlling device* in Figs. 17, 18, and 19 consists only of a brake B, which is applied by a weight W and is loosened by the operator by means of a hand rope. In the windlass, or winch, type of elevator, shown in Fig. 20, the brake is actuated, both in applying and loosening it, by operating the lever L by hand. Since

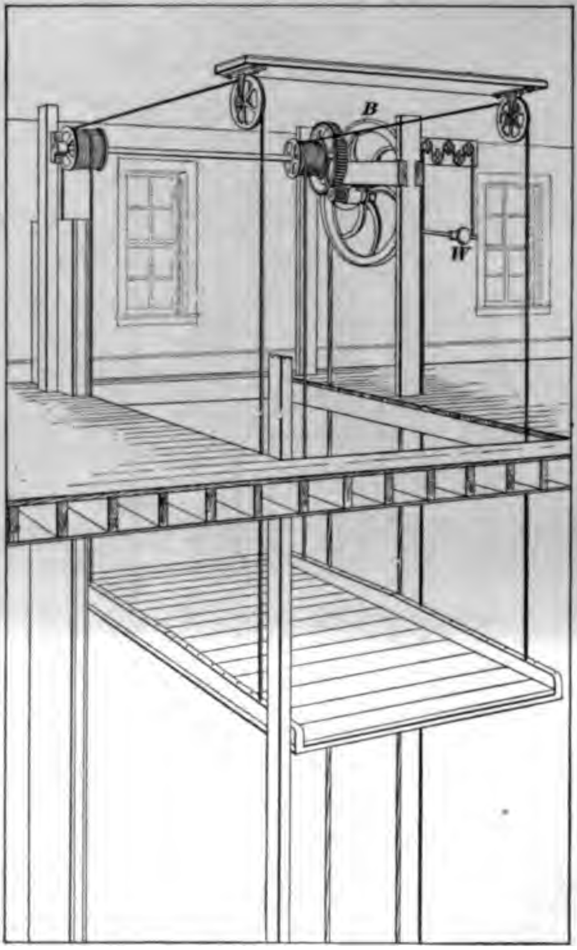


FIG. 19.

the brake is not applied automatically, a pawl P is thrown in mesh with the gearing when the elevator is at rest.

35. Motor Safeties. — Hand-power elevators have no *motor safeties.*

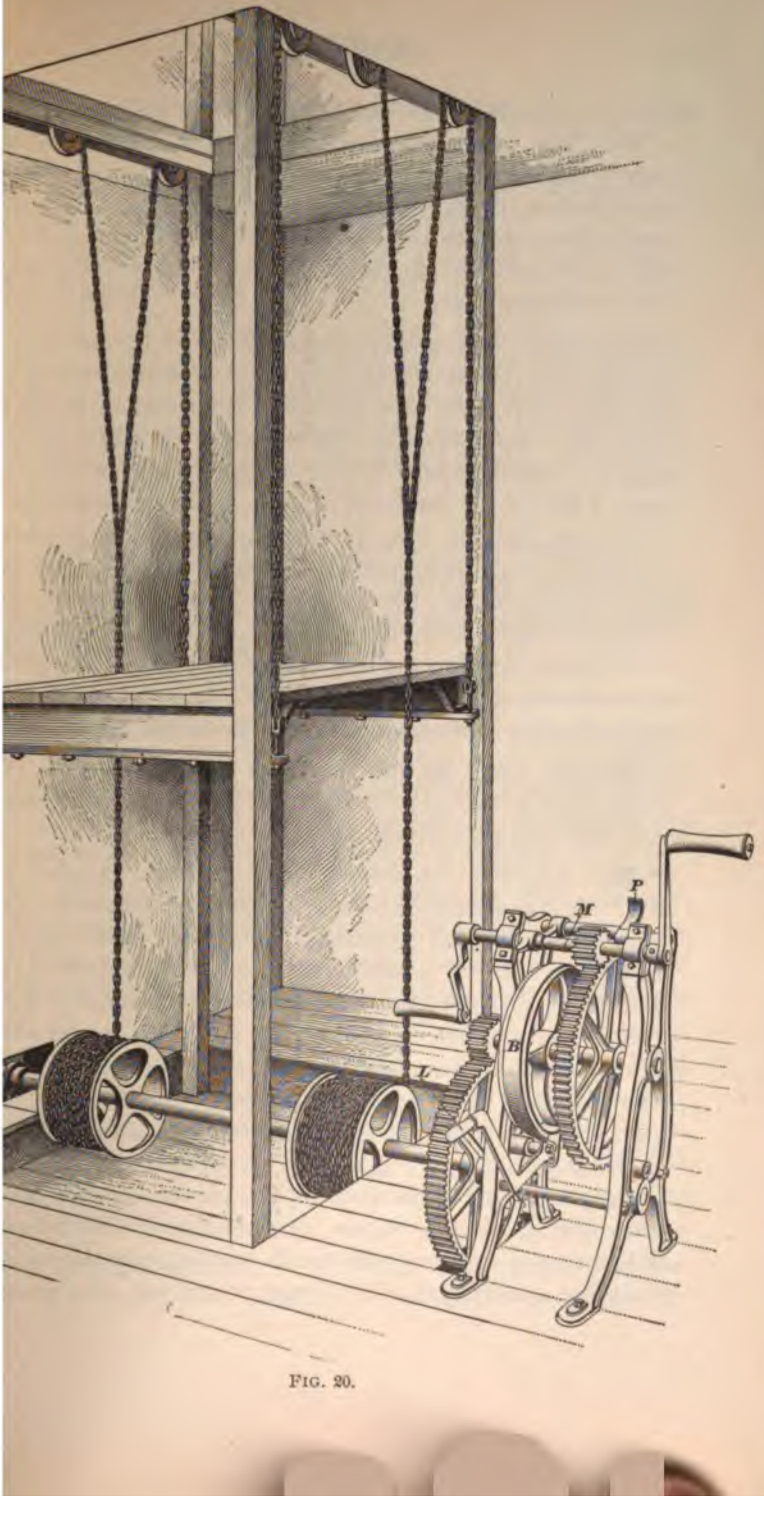


FIG. 20.

OPERATION AND MAINTENANCE.

36. The mechanisms of hand-power elevators are so simple that any one can operate them without difficulty; nevertheless, no less care should be exercised in handling them than power-driven elevators. Carelessness and neglect will prove just as dangerous with hand-power elevators as with any other type.

37. Hand-power elevators cannot be operated from the car, but are operated from any floor; a person riding on the same has no control over its movements and takes considerable risk. In operating, the operator lifts the brake and pulls the hand rope. In the design shown in Fig. 17, he must hold on to the brake rope, or **check-line,** as it is called, as long as the car is to move. This necessity is avoided by the arrangement shown in Figs. 18 and 19, the check-line passing over a number of small friction pulleys that give enough friction to the rope to hold the brake on or off after the operator has moved it by either an upward or downward movement of the hand. This device is a peculiarity of the hand-power elevators built by Morse, Williams & Co.

38. As the first and foremost rule it must be remembered that an elevator is built for a certain maximum load and that this load should never be exceeded.

39. All elevators should be started and stopped gradually. It takes more power to run an elevator up to a required speed than to maintain that speed thereafter; this additional starting power is the greater the shorter the time within which the necessary speed is attained, and the greater is also the stress in all parts of the machinery. Likewise, it takes considerable power to stop a moving elevator, which power is supplied by the braking device, and which is required to be the greater the quicker the elevator is stopped. Thus, if an elevator is stopped too quickly, enough stress may be put on the braking device to destroy it, causing accident. In hand-power elevators there is hardly any danger from quick starting, but there is from a sudden application of the brake, especially if the elevator is underbalanced and, as is often done, allowed to attain considerable speed in descending.

40. The drums, sheaves, and gears should be frequently inspected as to their fastenings to the shaft.

41. The brake needs particular attention, as the safety of the elevator depends on it.

42. If any car safety is provided, as there should be, the same should be examined frequently and carefully until the operator is satisfied that it is in good working condition; its parts should be kept well oiled and *should be kept clean*, to avoid their sticking and refusing to act in case of an emergency.

CARE OF WIRE ROPES, CABLES, AND GUIDES.

43. All that is said in Arts. **44** to **51** applies to all elevators, inasmuch as in all of them wire rope is used more or less and all have guides.

44. The wire ropes used in elevator work are made with hemp centers, to make them more pliable and thus more durable, on account of the short bends over comparatively small pulleys, or sheaves. Galvanized rope should not be used; the thin coating of zinc soon wears off, leaving the wires exposed to rapid deterioration by corrosion.

45. The wire ropes should be examined often and carefully; hoisting cables should be condemned as dangerous when the *wires* (not the strands) commence cracking. Wire ropes used for hoisting should under no circumstances be spliced.

46. Wire rope must be handled much more carefully than hemp rope, inasmuch as it is liable to kink and twist, which must be avoided on account of the harmful effect. Wire rope is best mounted on a reel that can be placed on a spindle to pay out the rope. If received from the supply house without a reel, the rope should be paid out by rolling the coil over the ground like a wheel. Wire rope should be lubricated like other moving machinery parts to preserve it.

To prevent rusting, raw linseed oil should be used and applied with a piece of sheepskin. The J. A. Roebling Sons Co. recommend to mix the linseed oil with the equal parts of Spanish brown or lampblack. The Otis Elevator Company recommend a mixture of 7 parts of linseed oil and 3 parts of tar oil. Another good preserving lubricant is made by heating and mixing well cylinder oil, graphite, tallow, and vegetable tar. When the **ropes, or cables,** as they are called frequently in elevator work, have once become well soaked, they need oiling only about every third or fourth month. They should not be allowed to become dirty and gummy.

47. In replacing worn ropes, particular attention must be given to the fastenings. In all cases where the ropes are replaced for the first time, it is best to carefully reproduce the joint as it was originally made by the makers or installators. An engineer taking charge of an elevator plant will, however, sometimes find rope fastenings of an inferior kind made by his predecessor. It may, therefore, be in order to call attention to the principal methods used by manufacturers.

48. The **shackle** used by the Otis Elevator Company is shown in Fig. 21. It consists of a split rod, the two legs A, A of which are bulged out and provided with noses at the ends. A collar B straddles the legs and eventually abuts against the noses. The rope is brought through the collar, bent over a **thimble** C, and passed back again through the collar, after which the free end is fastened by wrapping with wire. The wrapped end of the rope should be at least 8 inches long. The inside surfaces of the legs A and the outside surface of the thimble are concave to conform with the rope. Instead of the wire wrapping, clamps are sometimes used; the wrapping is to be preferred, however.

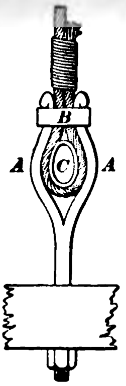
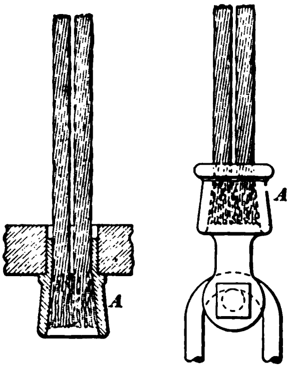


FIG. 21.

49. Another fastening is shown in Fig. 22. The rope is passed through a socket A forming part of the shackle; then it is untwisted for a short distance and the individual wires bent double. The socket is then filled with molten lead, or, better, with Babbitt metal, which should be of the best quality. The sockets should be warmed before the metal is poured to prevent chilling.



(a) (b)
FIG. 22.

50. In fastening the rope to the drum, it must be observed that at the lowest position of the car the rope must still encircle the drum several times to reduce the stress at the point of fastening.

51. The guides should not be allowed to become gummy and should, therefore, be cleaned from time to time—about twice a month—and freshly lubricated. Gummy guides cause the car to alternately stick and free itself, making its motion jerky; and a bad case of sticking may cause the car to drop a distance great enough to break the cable and thus cause serious accident. In cleaning guides, a judicious occasional use of kerosene oil is recommended. For a lubricant on steel guides good cylinder oil is used; some use a composition that is seven-eighths cylinder oil and one-eighth plumbago, well mixed. Wooden guides are greased with No. 3 Albany grease or lard oil; a mixture of tar oil and wax is also recommended by some.

BELT ELEVATORS.

DEFINITION.

52. The term **belt elevator** is applied to that class of elevators that are driven directly by belts from line shafting, which shafting, in turn, may be driven by any prime mover and may be used for driving other machinery at the same time. Belt elevators are used for freight service principally, seldom for passenger service.

INSTALLATION.

53. The shaft from which the power is taken revolves

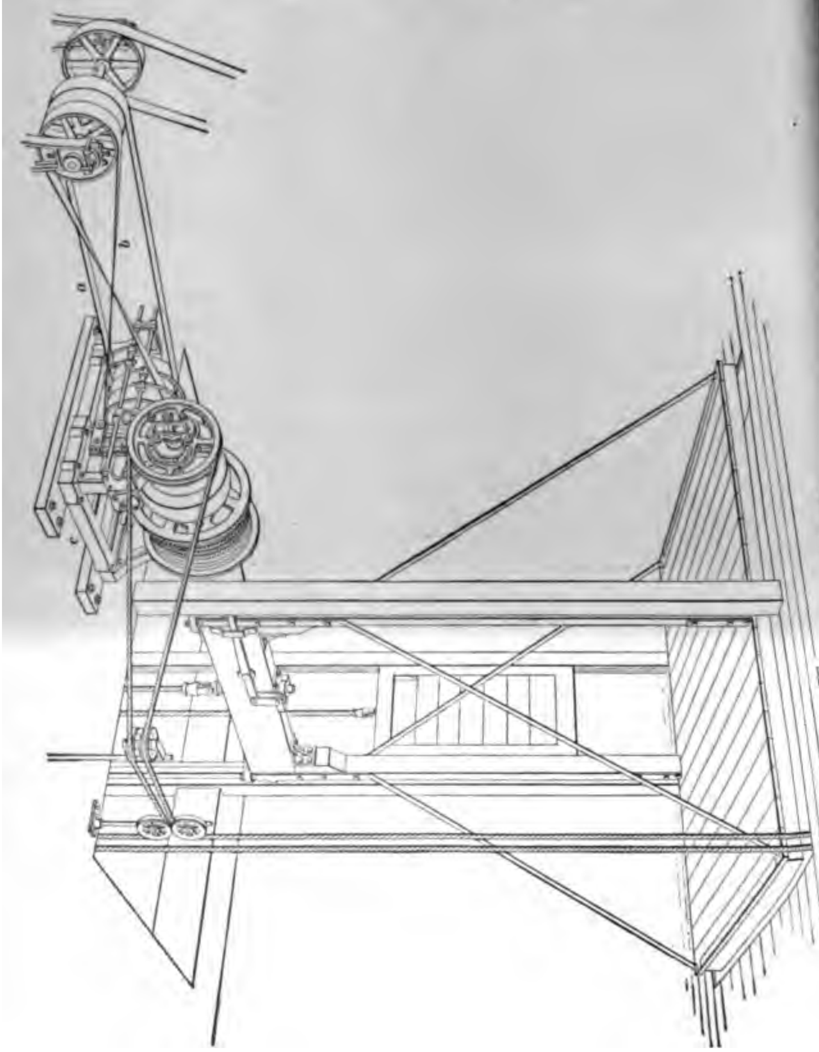


Fig. 23.

continually in the same direction independently of the motion of the elevator — that is, uncontrolled by the operator. The power is transmitted from this shaft to the elevator machine either direct, if the shaft is conveniently located, or by a countershaft. In either case, the shaft or countershaft carries a wide pulley that drives two belts, an open one *a*, Fig. 23, and a crossed one *b*. The elevator machine *c* is preferably placed on the ceiling, as shown, to save floor space, but it may be put on the floor as well. In many cases it will be possible to place it directly over the hoistway and thus save the expense of overhead sheaves.

GENERAL DESCRIPTION OF PARTS.

54. Following up the various parts again in the order named in Art. **2,** they present themselves as follows:

The *motor* of a belt elevator is simply a shaft carrying two loose pulleys and one tight pulley; it is designated by *M* in the various illustrations following hereafter.

The *transmitting devices* consist of either worm-gearing or spur gearing connecting the shaft *M* with the drum, worm-gearing being by far the more common arrangement.

When *counterbalancing*, worm-geared belt elevators are generally overbalanced; spur-geared ones are not. The reason for this is that worm-gearing works much smoother than spur gearing; it starts and stops gradually, offering much more resistance during the period of getting up speed, and acts as a kind of brake by itself in bringing the elevator to rest. The addition to the moving masses due to overbalancing, therefore, greatly increases the jerkiness of motion in spur-geared machines, while it has little influence that way in worm-geared ones.

The *controlling devices* consist of a pair of belt shifters, constituting the power control, and a brake, both being operated simultaneously by a shipper rope.

CONSTRUCTION OF CONTROLLING DEVICES.

55. In a belt elevator the controlling devices must be constructed in such a manner that the following requirements are fulfilled:

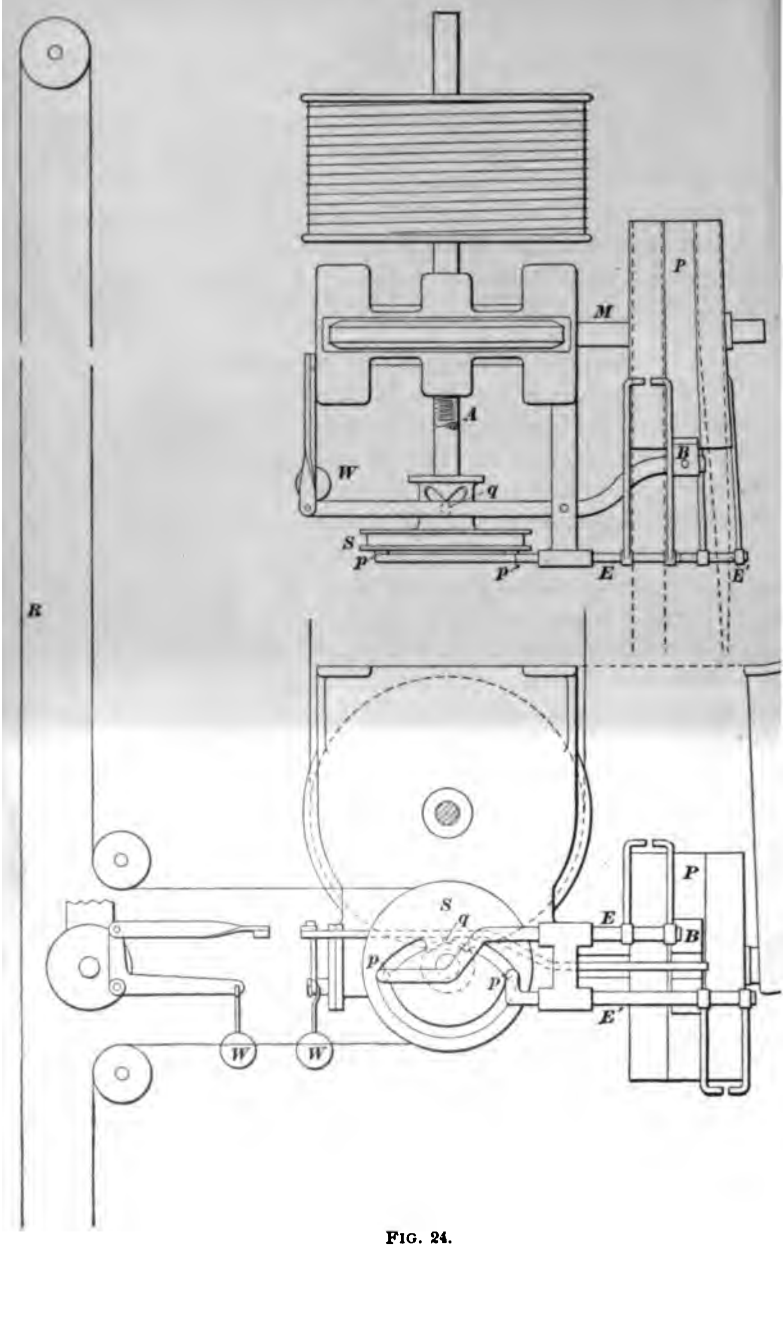


FIG. 24.

(*a*) When the shipper sheave is in the central position, both belts must be on their respective loose pulleys and the brake must be *on*.

(*b*) When the shipper sheave occupies the extreme right- or left-hand position, one belt must be on the tight pulley, while the other must be on its loose pulley, and the brake must be *off*.

(*c*) In their respective driving or non-driving positions, the belt shifters must be locked in place, so that they cannot be accidentally shifted.

(*d*) It must not be possible to throw the shipper sheave over too far.

(*e*) The central position of the shipper sheave must be distinctly defined, so as to give the operator warning against overthrowing the sheave from the one extreme position to the other. It is evident that this danger always exists.

56. The requirements stated in Art. **55** are met in various ways in practice. Fig. 24 shows in a diagrammatical way a typical arrangement for this purpose. The figure does not represent an actual machine, but was prepared to show the various elements of a belt elevator, separately explaining their functions. The shipper sheave S has two cam grooves into which enter corresponding pins p, p' on the ends of the belt shifters E, E'.

The cam groove, it will be noticed, has one concentric middle portion and two eccentric side portions.

When the sheave is in a central position, as shown, both belts are on their respective loose pulleys. A pull downwards on the shipper rope R swings the shipper sheave around to the left; the pin p of the left-hand belt shifter enters the left-hand eccentric portion of the cam groove and is thus forced to the right, while the other pin p' travels in the concentric portion of the groove and thus remains stationary. The open belt is thus shifted on to the tight pulley P while the crossed belt remains on its loose pulley; the elevator car then ascends. On pulling upwards on the rope R, the reverse takes place and the elevator car descends.

On the hub of the shipper sheave a V-shaped cam groove is formed, into which enters a pin *g* in the middle of a lever that carries on the one end a brake shoe *B* and on the other a weight *W*, which latter is so connected to it by means of a system of links that it tends to keep the brake on the tight pulley; a swing of the sheave either to the right or left lifts off the brake. Thus requirements (*a*) and (*b*), Art. **55**, are fulfilled. Requirement (*c*) is met by the shape of the cam groove, and not only are the shifters locked to the sheave in whatever position the same may be in, but also while one shifter is being moved the other is held positively and immovably in place by virtue of the concentric position of the cam groove.

Requirement (*d*) is met by the proper length of the groove, and requirement (*e*) by the sharp corner of the V groove, which will make itself distinctly felt to the touch of the operator.

57. As already said, the simple shipper rope is used for an *operating device*, special devices, such as described in Arts. **18** to **25**, being uncalled for, owing to the comparatively slow speed of belt elevators and to the fact that the car begins to move only after the belt has been shipped a considerable distance, so that it requires but little skill to complete the shift during the accelerating period of the car.

MOTOR SAFETIES.

58. Limit Stops on Shipper Rope.—In all elevators that are run by motive power the danger exists, if no provision be made against it, that, through the operator failing to arrest the car on time, the car or counterweight may be hoisted against the overhead work, causing damage and accident. Such danger does not exist in hand-power elevators with their slow speed, the resistance immediately being felt by the hand when the car strikes an obstruction. It is, therefore, one of the provisos in every power-elevator design that the power be shut off and the car be automatically arrested at the limit of its travel up or down.

The means adopted for this purpose are called **limit stops**, and are of various designs. In all cases where a shipper rope passes straight through the car, knobs or buttons are clamped on the same, against which the car strikes when nearing its upper or lower limit of travel, thus operating the shipper sheave automatically. This means, of course, operates only as long as the shipper rope is intact. As it may easily occur that the shipper-rope connection is broken or the rope is otherwise ineffective, limit stops are also always provided on the motor itself.

59. Limit Stops on Motors.—For drum elevators the most common arrangement is that shown in Fig. 25. Let A

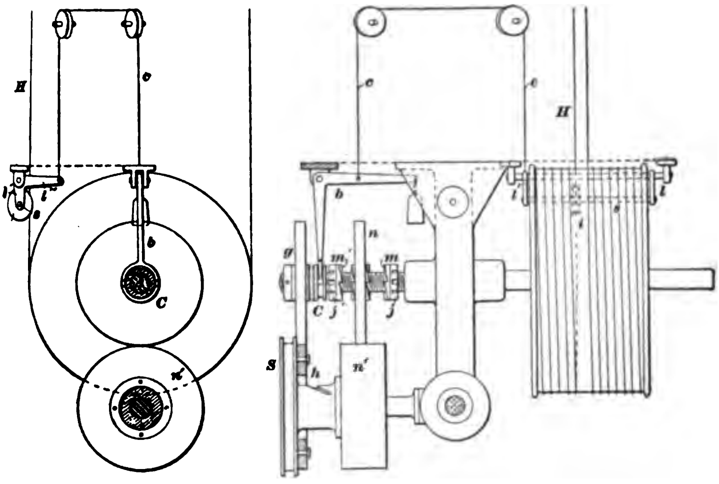


Fig. 25.

be a continuation of the drum shaft shown broken off in Fig. 24. A screw thread is cut on this shaft and a gear-wheel n, the hub of which forms a nut, is placed on the threaded portion of the shaft; this gear-wheel meshes with another similar wheel n' bolted to the shipper sheave. It is evident that when the sheave is stationary and the drum rotates, the gear-wheel n will be prevented from revolving with the drum shaft, but will travel on the same in an axial direction either towards or from the drum, according to the sense of rotation of the latter. The hub of the wheel *n* has claws on either side, as shown, corresponding to similar claws formed on two other nuts *m* and *m'* that are clamped by jam nuts *j* and *j'*, or in some other manner, securely to the drum shaft *A*. Now, it will be easily understood that when the wheel *n* travels either way, it will eventually be engaged by either one of the revolving nuts *m* or *m'* and be swung around, carrying with it the shipper sheave, with the effect of cutting off the power and applying the brake. The nuts *m* and *m'* can easily be adjusted to any position on the threaded portion of the drum shaft, and can thus be made to act when the car reaches the upper or lower limit in the hoistway.

SLACK-CABLE SAFETY.

60. Should the elevator car be obstructed in its descent by gummy guides or for any other reason and the motor continue to pay out the cable, the car would, if released suddenly, drop and most likely break the cable, causing damage; or should the car not drop, but be resting, for instance, on the bottom of the hoistway, the slack cable might still cause damage by getting into revolving parts of the machine. In any case, if the hoisting cable becomes slack, it will quickly be riding over itself on the drum or otherwise get entangled and must be straightened out again, which entails much labor and annoyance. A frequent occurrence is a slack cable produced by a careless handling of the shipper rope. It can often be noticed that when an operator in going up has missed his landing, he hastily reverses the machine to make his error good; the result is that the hoisting cable becomes slack. Now, most car safeties are so arranged that they will bind the car to the guides on the cables becoming slack. In his perplexity at the sudden stoppage of the car, the operator is likely to forget to shift the shipper rope so as to stop the machine, and the latter goes on paying out rope. Provision is made against such an emergency by contrivances called **slack-cable safeties.**

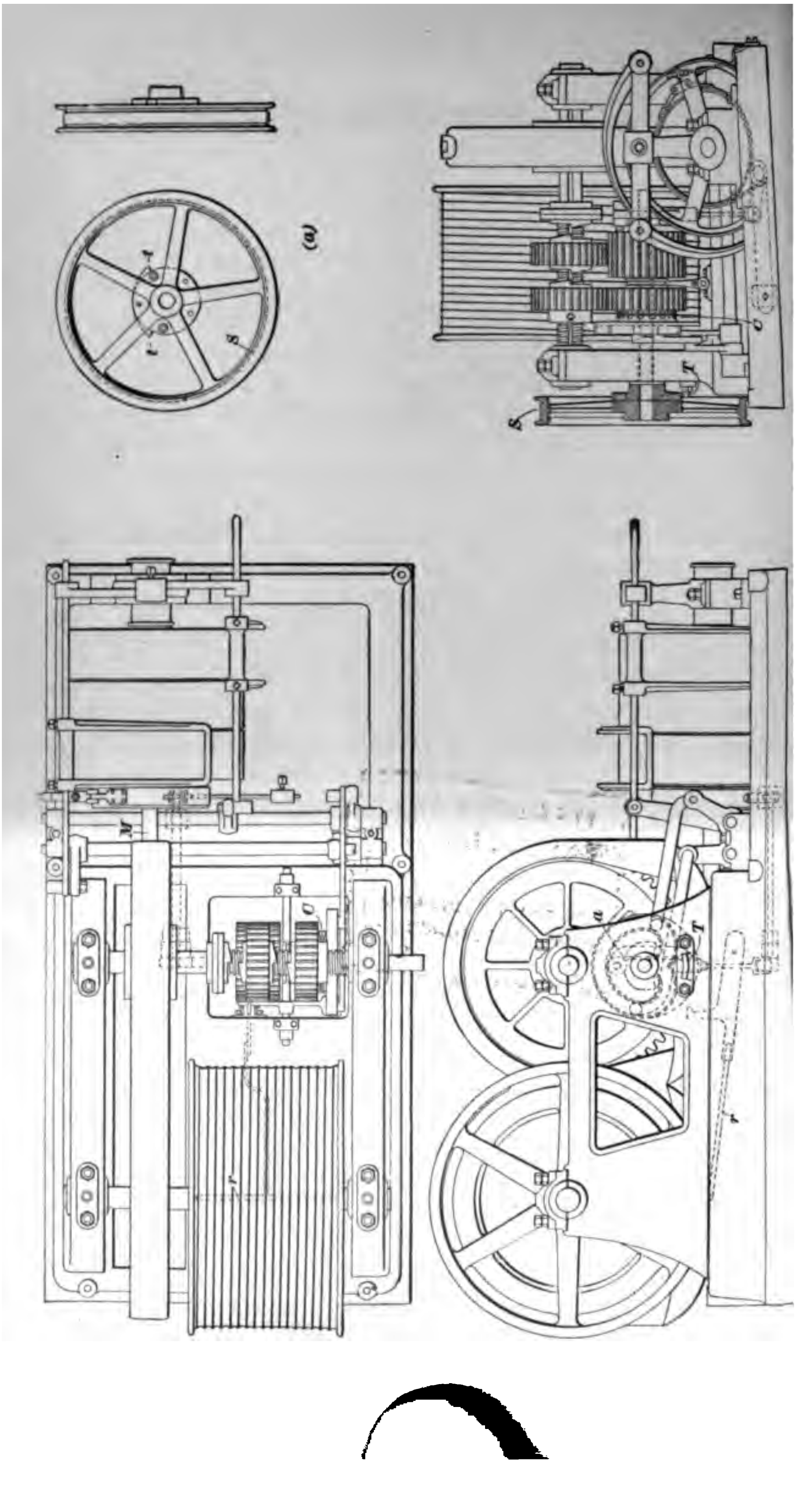
Fig. 25 shows the principle underlying such an arrangement: An idler *i* travels axially on its shaft *s* with the hoisting rope along the drum. The shaft *s* is supported on levers *l*, *l'* pivoted in a convenient manner. A cord *c* leads from the arm *l''* of the lever *l'* over sheaves to a bell-crank *b*, one arm of which is weighted, while the other engages a clutch *C*. As long as the hoisting rope *H* is taut, the idler *i* is pushed outwards against the weight on the bell-crank *b*; but should the hoisting rope become slack, the weight on the bell-crank *b* will cause the clutch *C* to engage with a gear-wheel *g* mounted loosely on the drum shaft, and will cause the same to revolve with the drum shaft. The gear-wheel *g* meshes with another gear-wheel *h* fastened to the shipper sheave, so that the latter will be swung around when the hoisting cable becomes slack.

61. The principles illustrated by Figs. 24 and 25 are found embodied in all belt elevators in various ways.

EXAMPLES OF BELT ELEVATORS.

62. Fig. 26 is a plan, elevation, and side view of a worm-geared belt-elevator machine built by The Whittier Machine Company, Boston, Massachusetts, and designed to be placed on the floor. While there is no particular difficulty about understanding the operation of the machine, a few explanations will nevertheless be in order. The machine has two worms, one left-handed and one right-handed, actuating two worm-wheels that mesh together. This combination prevents the end thrust, which is unavoidable in single-worm machines and saves the power necessary to overcome the frictional resistance due to it. There is, consequently, also no wear to the end of the shaft, and no step bearing is required, as in single-worm machines.

With regard to the controlling devices, it will be noticed that the belt-shifter cam groove is continuous. Special provision is, therefore, made against throwing the sheave over too far by fastening a stop-plate *T* to the frame, and



by stops t, t formed on the hub of the shipper sheave, as shown in detail in Fig. 26 (a).

The central non-driving position of the controlling device is made perceptible to the hand of the operator by nicking the brake cam, as shown in dotted lines at a in Fig. 26.

The limit stops are arranged practically in the same manner as in the typical drawing given in Fig. 25. The slack-cable safety is, however, radically different, inasmuch as the tension of the hoisting rope is not made use of; but, instead, the weight of one or more turns of rope hanging from the drum underneath in case the cable should become slack. For this purpose a rod r is placed across and underneath the drum, which rod is attached to the end of a weight-actuated lever that is tripped and closes the clutch C when there is any weight resting on the rod r. Both kinds of slack-cable safeties are extensively used.

63. Fig. 27 shows a worm-geared elevator built by Morse, Williams & Co., Philadelphia, Pennsylvania. In this machine the various requirements are fulfilled by slightly different means than have so far been shown, although the principles are the same. The difference lies in the manner in which the belt shifters are moved. Instead of the shipper sheave carrying the cam, the belt-shifter bars a and b have slotted cam pieces a' and b' attached to them, and the sheave carries two buttons, or projections, p and p'. It is easily seen that the effect is the same as when the shipper sheave carries the cam, with one advantage. As will be shown presently, it saves a good deal of complication in the way of gearing to put the shipper sheave loosely on the drum shaft instead of placing it on a separate stud, or shaft, in line with the shifter bars, as was done in the machines shown in Figs. 24 and 26.

As this, however, throws the center of the shipper sheave out of line with the shifter bars, the distance between must be bridged over. When using a cam on the sheave, this is done ordinarily by interposing double-arm levers l and l', as shown in Fig. 28, so that one advantage is gained at the sacrifice of another in that case. By arranging the parts as in Fig. 27, the necessity of the double-armed levers is avoided, making the machine so much simpler. Both types are in extensive use, however.

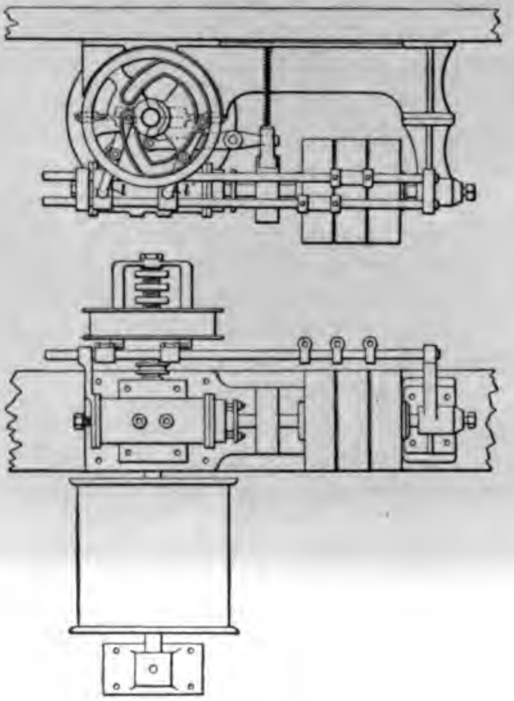


FIG. 28.

Reverting to Fig. 27, it will be seen that in turning the sheave to the right, for instance, the right-hand belt will be shifted on to the tight pulley *A* by virtue of the button *p'* entering the cam groove on the corresponding shifter-bar cam piece *a'*. The left-hand button *p*, however, will leave its cam *b'* entirely, and if no provision were made against it, the left-hand shifter would be unprotected, that is, liable to be shifted accidentally. To avoid this, that is, to lock the stationary shifter bar in place while the other is being moved (see requirement (*c*), Art. **55**), a circular groove *g* is formed in the shipper sheave *S*, which fits over pins *h* and *h'* inserted in the shifter cam pieces *a'* and *b'*.

The central, or non-driving, position, as well as the extreme right and left positions, are strongly defined to the operator by the shape of the brake cam, which has a wide, flat surface for the neutral position and two smaller flat surfaces for the end positions, with the effect that when either of the corners *c* or *c'* of this cam passes through a position vertically below the center of rotation, the sheave will come to a quick and sudden stop.

64. It was mentioned in Art. **63** that by placing the shipper sheave on the drum shaft a simpler arrangement for the limit stops can be had. The usual arrangement is clearly shown in Figs. 27 and 28. The shipper sheave is provided with a yoke *y*, which takes the place of the hub. The yoke has formed on it a feather or rib *f*, upon which slides the traveling nut *n* that eventually engages with the fixed nuts *m* and *m'* in the manner already described. It is thus seen that the gearing shown in Figs. 24 and 26 is dispensed with. The yoke arrangement is in most extensive use and is found on almost every drum-elevator machine.

OPERATION AND MAINTENANCE OF BELT ELEVATORS.

65. The operation of belt elevators requires but little skill, the speed being comparatively slow; nevertheless, a certain amount of practice is required to "make the landings" exactly.

66. The operator *should never rely on the limit stops* to make a top or bottom landing, but should always operate the rope as at any intermediate floor. The limit stops are provided for an emergency and not for general use. They should be tried, however, once or twice every day, to see if they are operative and correctly set. The operator is to be cautioned against sudden reversals of the controlling device.

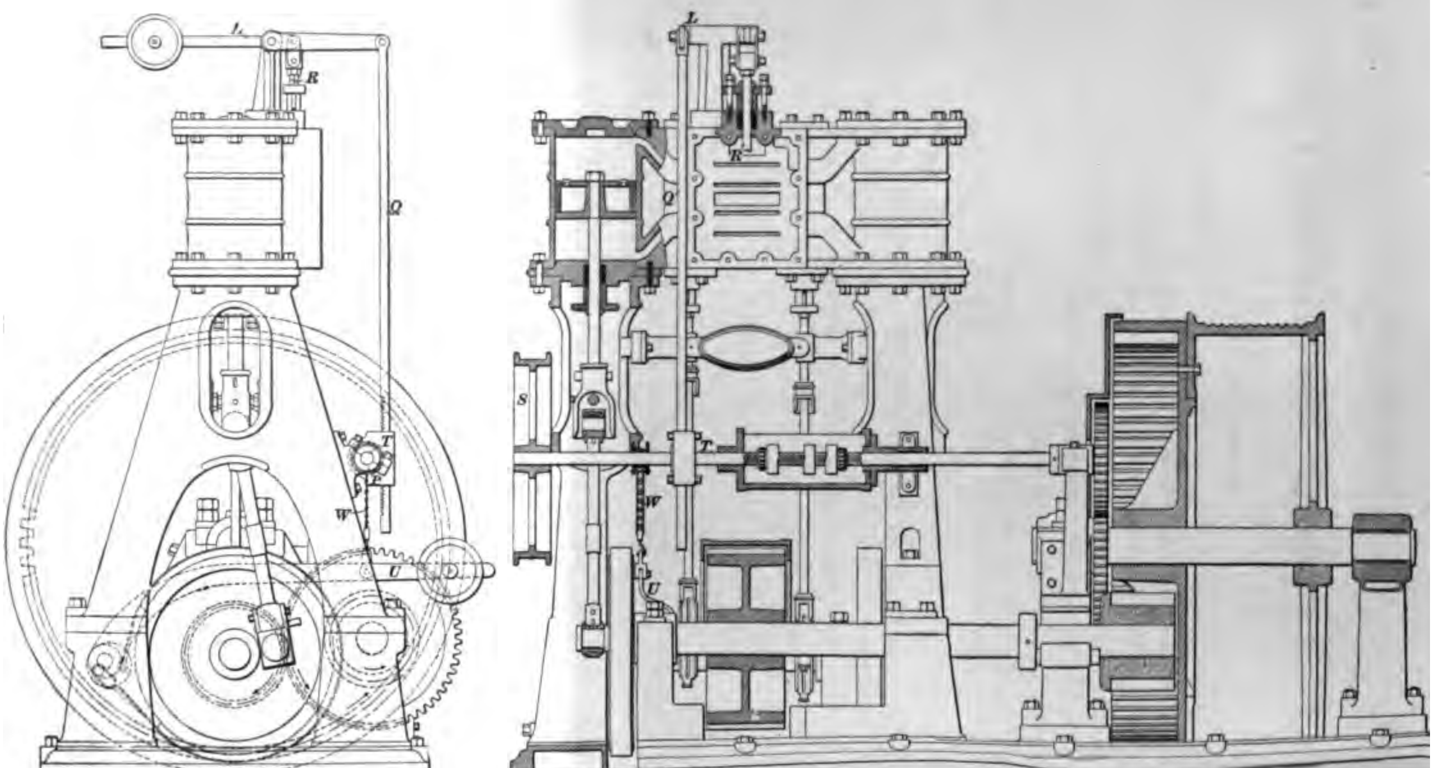


Fig 25

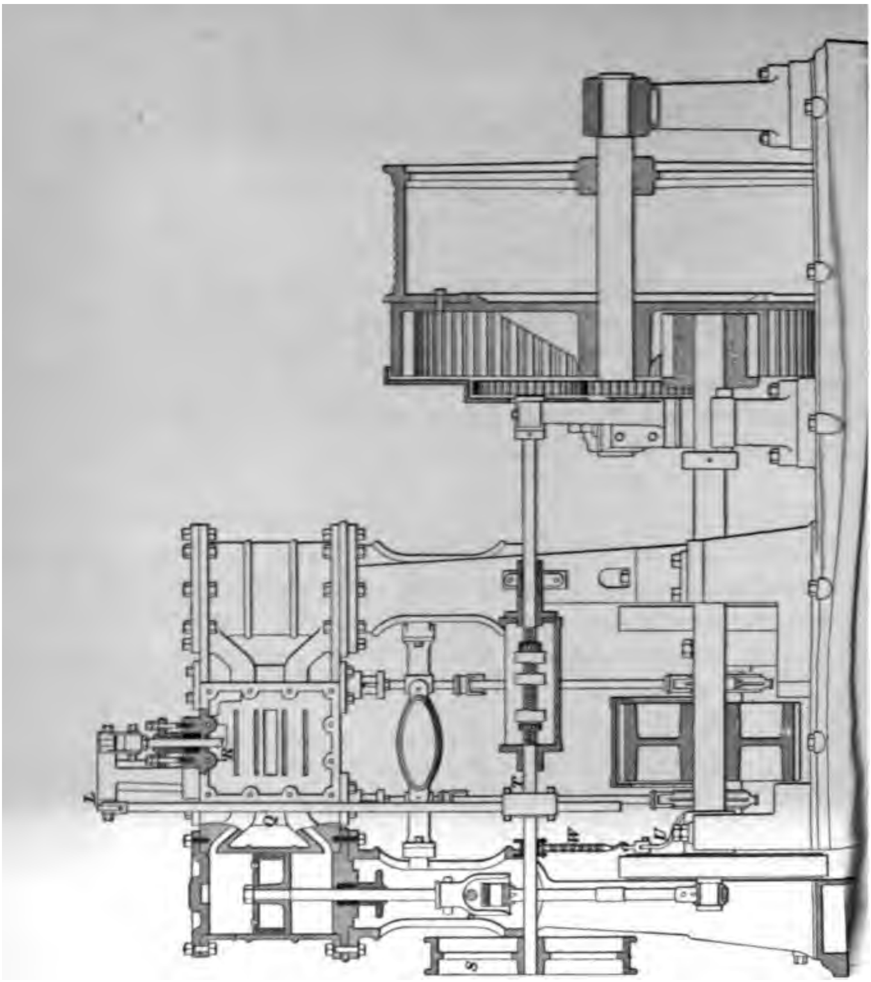
71. Worm-gearing when new should, if possible, be less heavily loaded than when run in. A judicious observance of this rule is sure to prolong the life of the gearing considerably. Although conscientious manufacturers run in their worm-gearing before shipment, they can naturally do so only to a limited extent. It is said that the best oil to use on the worm bath is castor oil. The fact, however, that castor oil thickens when it becomes heated and that more or less heat is developed on worm-gearing, makes it desirable to use a mixture of 2 parts of castor oil and 1 part of the very best cylinder oil. Upon getting warm the cylinder oil runs freely, thus compensating for the property of castor oil mentioned.

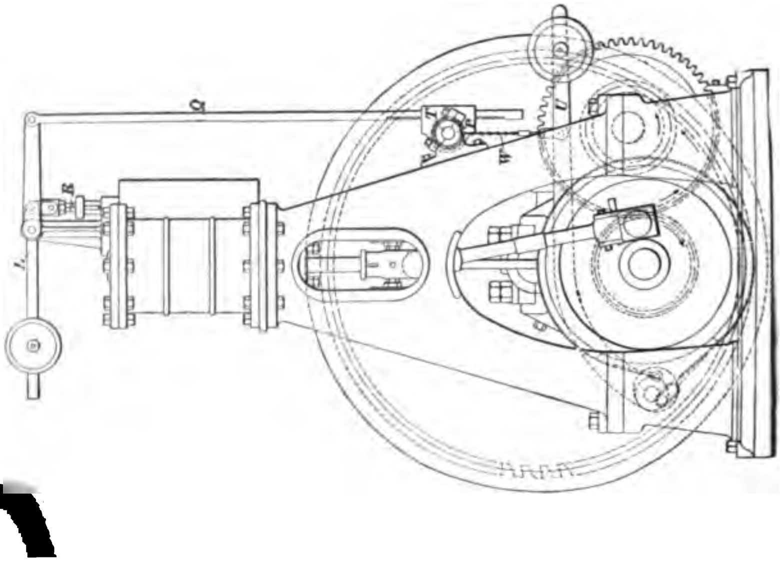
72. Particular attention is to be paid to the lubrication of the thrust, or step, bearings of the worm, which should be renewed as soon as they show signs of cutting, since they will rapidly go from bad to worse. The step is generally made adjustable. The adjustment should be such that there is a little end play for the worm-shaft, say a scant thirty-second of an inch. This end play gives the oil a chance to enter between the bearing surfaces at every reversal of the worm.

If the steps are screwed up too tight, they will run hot at once and soon seize. The same as the worm and wheel, the step bearing requires to be run until a full uniform bearing surface is attained by a mutual adjustment through wear of the journal and its step. This mutual adjustment can be greatly facilitated by placing a leather washer behind the step.

73. Overhead sheave boxes must not be neglected. They should be kept lubricated with heavy grease in summer, with an addition of cylinder oil in winter.

74. Belt elevators should ordinarily not be run at a greater car speed than 60 feet per minute. The pulley speed should not exceed 400 revolutions per minute.





82. When the car is at rest, the reversing valve occupies the position shown in Fig. 30 (*c*), where all ports are covered. A motion of the reversing valve in an upward direction will start the engine and hoist the car; a downward motion of the reversing valve will let the car descend.

83. The stem *R* of the reversing valve (see Fig. 29) is attached to a lever *L* that is actuated by a rack *Q* and

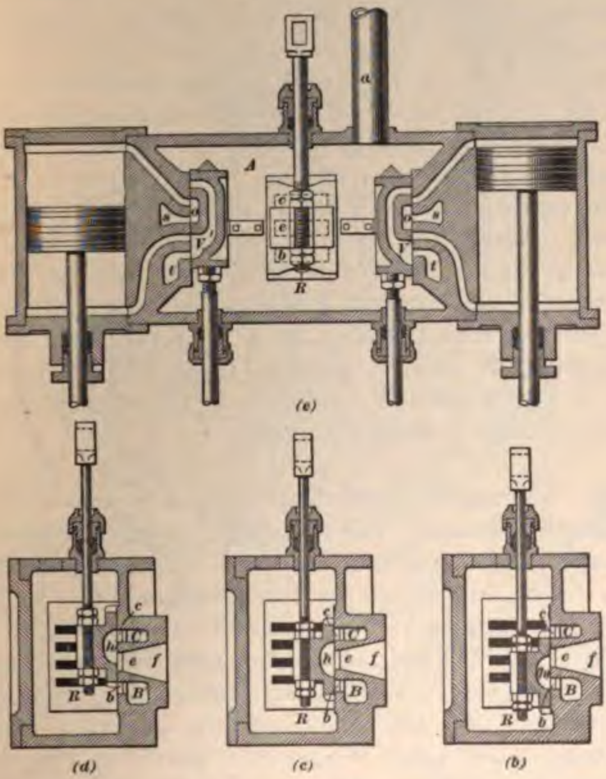


FIG. 31 (*b*), (*c*), (*d*), (*e*).

pinion *P* from the shipper-sheave shaft *T*, which latter is also connected to the brake lever *U* by a chain *W*. The operation of the brake is plain from the drawing.

84. The reversing valve in the machine shown Fig. 31 (*a*), which is a **Crane worm-geared steam e vator,** is somewhat different from the Otis valve ju described. Its action will be understood from Fig. 31 (*b*), ((*d*), and (*e*). Fig. 31 (*e*) is a vertical section through t two engine cylinders and the valve chest showing the stean distributing slide valves *V*, *V'* in section and a front view the reversing valve *R*, while Fig. 31 (*b*), (*c*), and (*d*) are transverse sections showing the three positions of the reversing valve. The action is as follows: Steam enters through the pipe *a*, Fig. 31 (*e*), the steam chest *A*. If the reversing valve *R* is moved to the position shown in Fig. 31 (*b*), the port *c* is opened, thus allowing steam to flow through the passage *C* into the cylinders, while the exhaust passes through passage *B*, port *b*, cavity *h* of the reversing valve, exhaust port *e*, and duct *f* into the atmosphere. To stop, the reversing valve is moved to the position shown in Fig. 31 (*c*), when it closes the passages *B* and *C*. To reverse the machine, the reversing valve is moved to the position shown in Fig. 31 (*d*). Steam then enters *A* through *a*, as before, but goes through port *b* and passage *B* to the distributing valves *V*, *V'*, while the exhaust passes through passage *C* and port *c*.

85. The manner in which the engine is reversed may be explained as follows: The port *c* and passage *C* connect with the steam passage *s*, and the port *b* and passage *B* connect with the steam passage *t* in the valve seat of each engine. With the reversing valve in the position shown in Fig. 31 (*b*), the live steam is admitted through *s* into the central cavity *o* of the steam valves, while exhaust takes place through *t*. It is thus seen that the valves are now indirect. When the reversing valve takes the position shown in Fig. 31 (*d*), the live steam passes through *b* and *B* and through *t* into the steam valves, while now the exhaust steam passes through *s* into the cavity *h* of the reversing valve, and thence into *e* and *f*, and finally into the atmosphere. In this position the steam valves act as direct valves and give a motion to the engines contrary to that obtained when the valves act as indirect valves.

86. There is no brake shown on this machine. The worm having a sufficiently low pitch to make it self-locking, a brake is often dispensed with. When used, however, it consists of a wooden brake shoe, which is pressed against the wheel by means of springs and released by live steam.

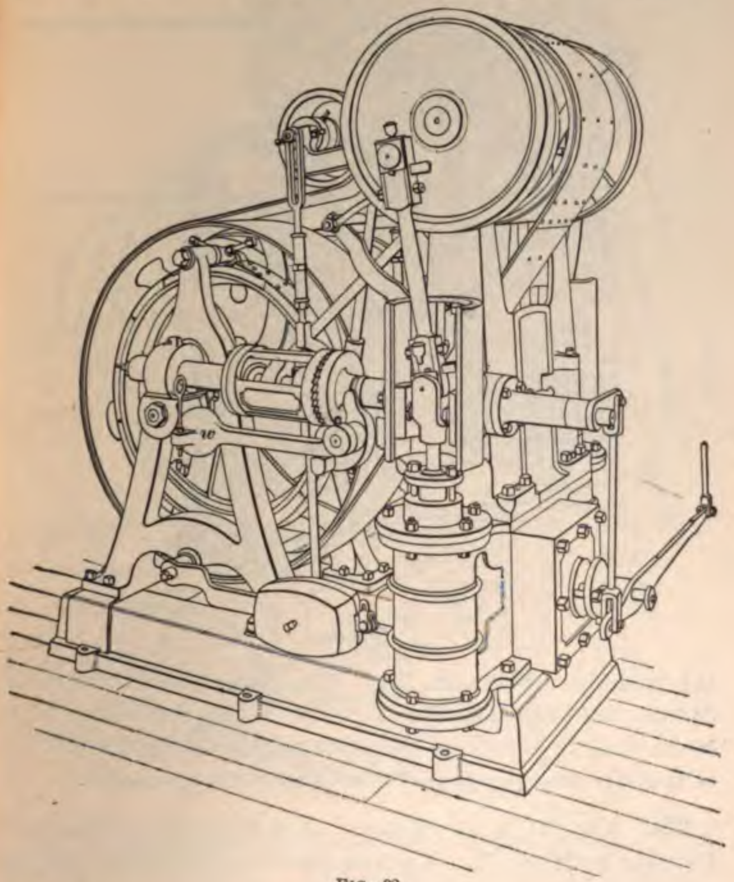


FIG. 32.

87. In the machine shown in Fig. 32, which is a belt and spur-gear steam elevator built by the Otis Elevator Company, a rotary reversing valve is used. Its action is much the same as that of the Otis reversing slide valve previously described.

88. Fig. 33 is an illustration of a belt and spur-gear steam elevator. The machine is built by the Otis Elevator Company of Chicago, formerly the Crane Elevator Company. The same kind of reversing valve is used in it as in

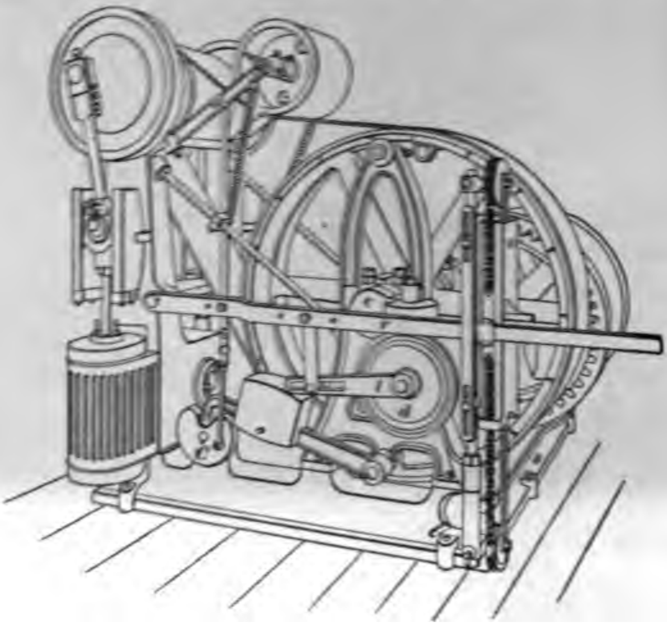


FIG. 33.

the machine shown in Fig. 31. Among the controlling devices in this machine is to be noted the heart-shaped brake cam C, the deep notch of which marks the central, or non-driving, position of the controlling mechanism.

89. Fig. 34 is an example of a belt and worm-geared elevator built by The Whittier Machine Company, now consolidated with the Otis Elevator Company.

90. Motor Safeties.—Motor safeties are provided in all cases, either in the shape of limit stops of the ordinary yoke type, as shown in Figs. 29 and 32, or of special design with the same underlying principle as shown in Fig. 31 (*a*).

91. The device used in the machine shown in Fig. 31 has some additional features, and is, therefore, shown in detail in Fig. 35.

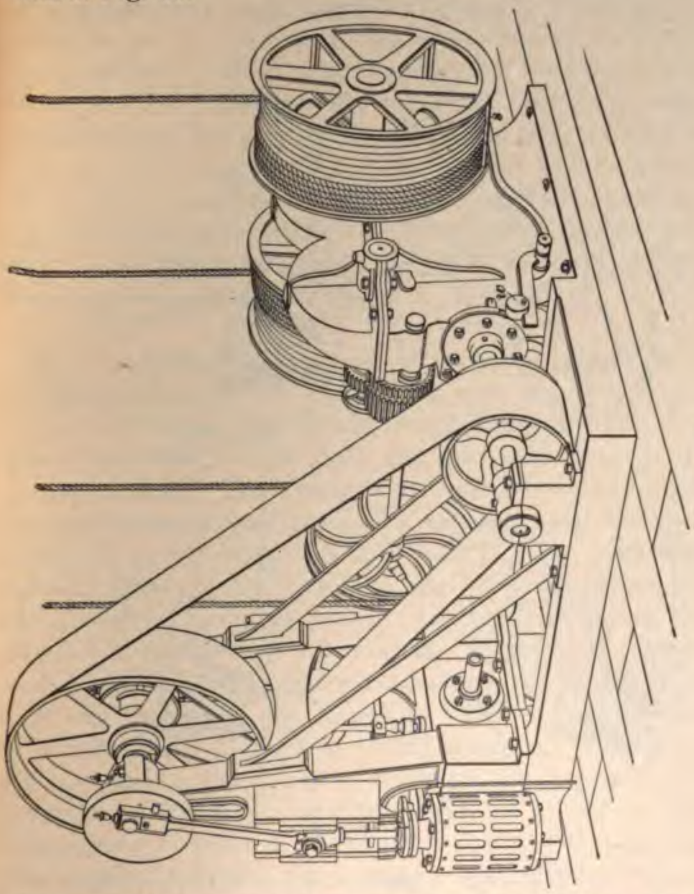


FIG. 34.

The winding-drum shaft carries a pinion *p*, Fig. 35 (*a*), meshing with a gear *g*. This latter gear has a second pinion *p'*, which is solid with it and meshes with another gear *g'* mounted loosely on the winding-drum shaft. This gearing, which is similar to the back gearing of a lathe, is such that the gear g' will make less than one revolution for the whole number of revolutions of the drum shaft necessary to lift the car to the top. To the gear g' is attached a drum having long slots [see Fig. 31 (*a*)] into which are fitted adjustably the cams c, c', shown in Fig. 31 (*a*) and Fig. 35 (*b*), (*c*), and (*d*). These cams are located on the

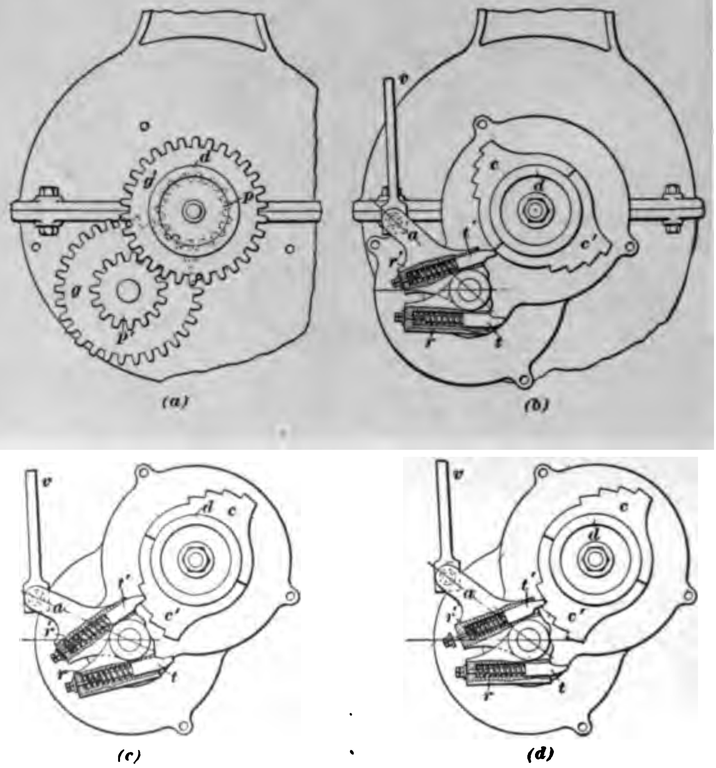


FIG. 35.

drum d in different planes and have two or more ste
as shown, and engage eventually each one of two sprin
actuated triggers t, t' mounted in rockers r, r' on a stud a
in planes corresponding to those of the cams. One of t
rockers, which are rigidly connected to form one piece, h
an arm a, to which is connected the rod v leading to t

valve lever m, as shown in Fig. 31 (a). When the car reaches the top or bottom, respectively, of the hoistway, either cam c or c', as the case may be, engages its particular trigger t or t' and pulls or pushes the valve rod v.

Fig. 35 (b) shows the position of the various parts when the car is about midway in the elevator shaft. Suppose now the car to go up; then, as it nears the limit of its upward travel, the cam c' will engage the trigger t' with the first one of the steps and thus move the valve rod v until the trigger passes over the first step, as shown in Fig. 35 (c). This will slow down the car; on a farther motion of the car the second step of the cam will come into contact with the trigger, pulling the valve rod still farther, thus slowing down the car still more, and so forth, until the steam valve is entirely closed and the car stops.

92. The gradual choking off of the steam supply by the successive steps has the effect that with a heavy load on the car the latter will finally reach the top very much slower than with a light load, and may even stop short of the last landing. Conversely, if the apparatus is so adjusted that with a heavy load the car will finally stop at the lowest landing, it may do so with a light load, but only very slowly and even may not reach the landing. The apparatus illustrated in Figs. 31 and 35 provides for these conditions, inasmuch as it permits the operator on the car to operate the controlling device to some extent even after the automatic stop has performed its function. This is accomplished by making the triggers t, t' spring actuated. The springs will not yield to the action of the cams, the pressure being a transverse one, but they will yield if a pull or push, respectively, is exerted on the valve rod v by the operator. Thus, if it is found, for instance, that after the cam c' [see Fig. 35 (c)] has acted upon the trigger t' so as to slow down the car, the latter moves too slowly, the operator may pull on the rod v and bring it into the position shown in Fig. 35 (d), the spring of said trigger permitting this, and thus partly reopen the steam port.

the gear g' will make less than one ... has a different whole number of revolutions of the ... of the disk d is a to lift the car to the top. To the ... a worm and gear drum having long slots [see Fig. ... c). This plate has a fitted adjustably the cams c, c', connected to the disk d in Fig. 35 (b), (c), and (d). These ... on adjustable stops and ... turn with it, thus centering

(a)

(b)

...Slack-cable safeties are generally ... elevators, although shown only ... consist in all cases of a rod ... of the winding drum and so ... by the weight of any loose ... when so depressed, it releases a ... turn, acts upon the controlling ... The aforesaid rod is seen in ... weight at w. In Fig. 32, the ... throws in a clutch that causes the ... valve rock shaft.

... AND MAINTENANCE.

... steam elevator is exceedingly ... to every one able to run a steam ... cannot be bestowed on the hoist- ... safety appliances; the weights ... being usually great, the risk to ... by neglect on the part of the ... great.

... elevators, attention must be paid par- ... A breakage of this belt, it will ... and all there is upon it on the ... results if the latter should prove ... belt performs a duty much more ... it runs over a large and a ... to give it as great as possi- ... small pulley; it must also run in

that there is always con-
efore, be of the best
well cared for. The
tanned stock; the pieces
such a way that the hide
lt. The pieces should be well
ps. They should not be more
ending laps, and should be joined
making a perfect joint. A *straight*
under any circumstances. Besides
the cement used must be very pliable
short turn of the belt under the idler and
pulley. The belt should be riveted as a pre-
st a lap becoming loose, so that the rivets may
defective lap together until it is discovered and
red.

cing belts must not be resorted to, as the laces soon
, due to running over the small pulley.

is recommended by elevator men to give the belt an
ional dressing with castor oil to keep it pliable.

ELEVATORS.

(PART 2.)

ELECTRIC ELEVATORS.

INTRODUCTION.

1. Treating elevators in the order of their development, the hydraulic elevator would follow after the steam elevator, because the electric elevator is the latest competitor in the field. Nevertheless, as most electric elevators are of the drum type, and therefore similar in many ways to hand, belt, and steam elevators, they will be considered before the older type.

INDIRECT-CONNECTED ELECTRIC ELEVATORS.

BELT-CONNECTED, BELT-SHIFTING ELEVATORS.

2. The first mode of application of the electric motor to elevator machinery was simply a substitution of an electric motor for whatever kind of power was previously used for driving the line shafting of an ordinary belt elevator. The motor was started by an ordinary main switch and starting box and ran continuously in one direction, the elevator being controlled in the same manner as other belt elevators. If such an elevator is not in constant use, the electric motor must be stopped and started frequently, which, with an ordinary switch and hand starting box, compels the operator to go to the starting box every time the elevator is used. To avoid this, the switch and starting box are operated by a hand rope running through the car in the same manner as the shipper rope, or to avoid the handling of the two ropes, the shipper rope may serve both for shifting the belts and for cperating the switch and rheostat.

BELT-CONNECTED, BELT-SHIFTING, REVERSIBLE-MOTOR ELECTRIC ELEVATOR.

3. General Description.—By introducing a reversing switch instead of the single switch, the motor can be reversed by reversing the current in the armature. The necessity for two belts, an open and a crossed one, is then obviated, and one belt between the countershaft and elevator machine is sufficient, this belt being shifted from a loose pulley to a tight one to start the car in either direction, that is, up or down.

4. Belt-shifting electric elevators being nothing but combinations of belt elevators with an electric motor, we can confine our remarks with respect to the various parts of these elevators to motors and controlling devices, all the other parts being the same as in ordinary belt elevators.

5. Motors.—For belt-shifting elevators, continuous-current, constant-potential, shunt-wound, single-speed motors are generally used, and since the motor starts without load, no rush of current that might injure the armature takes place at starting. Any kind of alternating-current motor may be used for belt-shifting elevators when the motor runs continuously. When, however, the motor is to be stopped and started frequently, polyphase synchronous motors or induction motors must be used, because these motors will start by themselves, while single-phase motors will not.

6. Controlling Devices.—Aside from the belt shifters in belt-shifting elevators, the power control consists of a switch and a rheostat. For combinations in which the motor runs continuously in one direction and is started and stopped only occasionally, the ordinary switch and starting box operated by hand are sufficient. If, however, the switch and rheostat are to be operated by a hand rope or other operating device from the car, special mechanisms become necessary, since the simple pull on the hand rope cannot give the necessary motions. To prevent a possible damaging rush of current in starting such an electric motor as is used in elevator work, the main switch is closed with all the starting resistance in the armature circuit, which resistance is then gradually cut out as the speed of the motor increases, until the motor is finally (when running at its normal speed) connected directly to the mains. After stopping, this resistance should all be in again, so as to make the apparatus ready for the next start; and since starting may follow quickly upon stopping, this restitution of the apparatus to its starting conditions after stopping must be effected quickly. When the switch and starting box are manipulated by hand, the above requirements can be easily fulfilled, but not when they are operated together from a hand rope. To obtain the required motions, various contrivances have been devised and are largely used. A few examples are given.

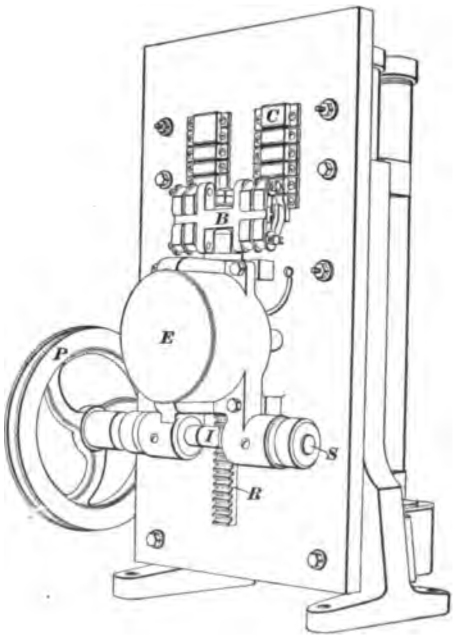


FIG. 1.

7. Mechanically Operated Rheostats. — The most natural way to gradually cut out the starting resistance as the speed of the motor increases is to mechanically connect the starting box to the motor shaft. Fig. 1 shows an apparatus made by the Automatic Switch Company and designed to be used with motors running always in one direction, that is, in our case, with an indirect-connected or belt-shifting, non-reversible elevator machine. The pulley *P* is belted to a smaller pulley on the motor shaft or countershaft and drives a shaft *S* having formed on it a two-toothed pinion *I*. When the motor is running, a rack *R* is drawn into mesh with the pinion *I* by means of an electromagnet *E* energized by a coil in shunt with the motor circuit. As soon as the circuit is closed and the motor commences to revolve, the rack ascends and with it the contact bar *B* that is carried on its upper end. The contact bar passes successively over the contacts *C*, gradually cutting out resistance. As soon as the current is broken, the magnet is deenergized and the contact arm drops back, the rack *R* springing out of gear with the pinion.

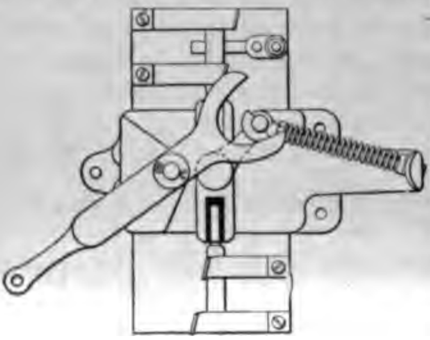


FIG. 2.

8. In connection with the starter shown in Fig. 1, a simple snap switch is used, such as is shown in Fig. 2; the action of this will be readily understood. It is operated either by hand or by a separate hand rope or cord running parallel to the shipper rope in the hoistway.

9. Fig. 3 is a diagram of an installation using the starting box shown in Fig. 1. Fig. 4 is a diagram of the connections; this will prove useful to engineers wishing to drive existing belt elevators by an electric motor.

10. In case a belt-shifting elevator is to be run with a single belt, the motor must be reversible. A **reversing**

switch is then used instead of the single snap switch shown in Fig. 2. Such a reversing switch, made by the Automatic Switch Company, is shown in Fig. 5, which also gives a diagram of the connections. The reversing switch has

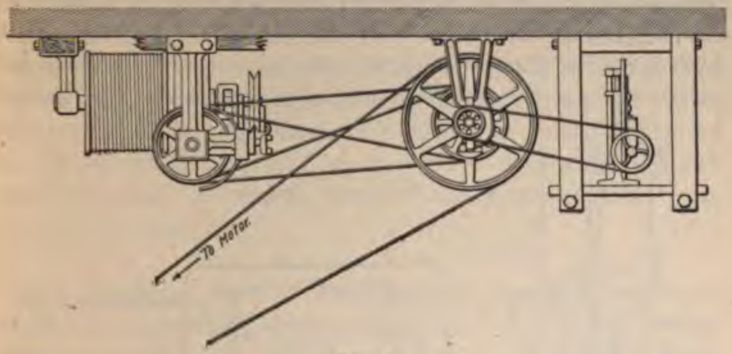


FIG. 3.

four sets of contacts a, b, a', b', each consisting of three clips, and two blades B and B', which are insulated from each other. The clips are connected with the terminals of the various parts (motor armature, field, resistance,

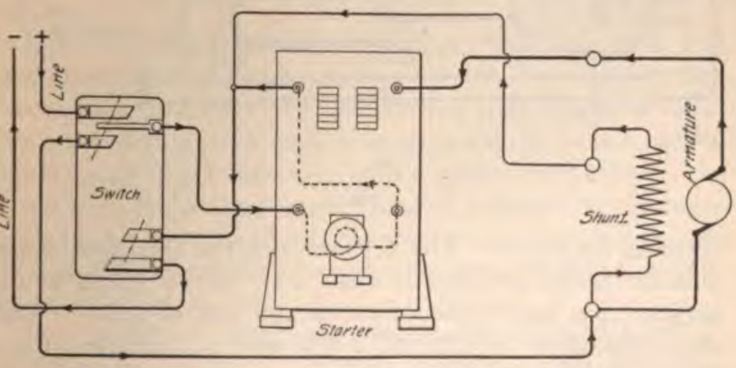


FIG. 4.

and starter magnet), as shown. When the switch is pulled up, blade B connects the three clips at a and blade B' connects the three clips at b. This allows the current to flow through the armature, the shunt field, and the resistance, and the elevator ascends. When the reversing switch is pulled down, B connects the three clips at a' together and B' connects the three clips at b'. This reverses the flow of the current through the armature, because the wires on the switch that connect the upper and lower horizontal clips are crossed; the current in the shunt field flows in the same direction, no matter whether the switch is up or down; hence, pulling down the switch

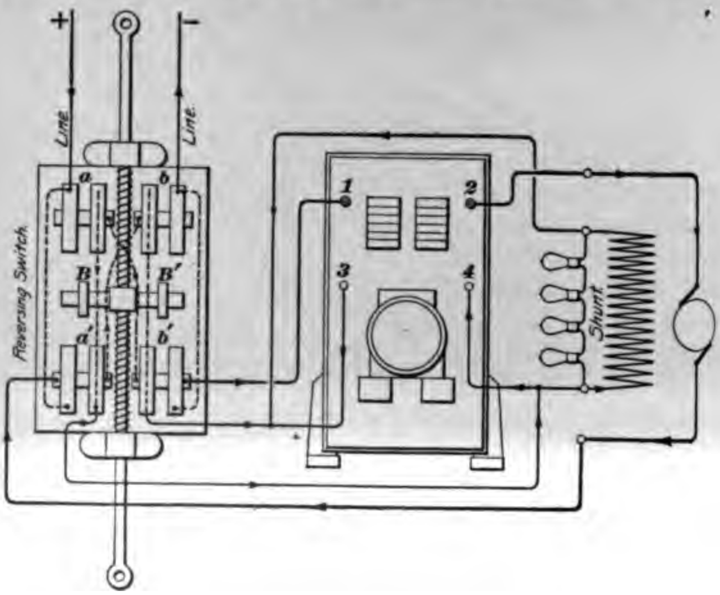


FIG. 5.

reverses the motor. The terminals of the armature resistance are shown at *1* and *2*; *3* and *4* are the terminals of the magnet that throws the rack into and out of gear. With this explanation the student will be able to trace the path of the current without difficulty.

11. It is often observed on opening the circuit that there is considerable sparking at the clips connected to the shunt field. This is due to the self-induction of the field. To reduce this sparking, it is a good plan to connect across the shunt a series of incandescent lamps having a combined voltage of from 6 to 8 times that of the line current; that is, in case a 110-volt lighting current is used, a series of, say, four 220-volt lamps is inserted, through which the induction current of the field is discharged. Since the starter is belted to the machine or countershaft, it will be reversed with it; it must, therefore, be so arranged that it will lift the cross-bar *B*, Fig. 1, no matter in which direction the motor runs. This is done in this kind of starter by substituting for the two-toothed pinion *I* an eccentric operating a pawl. Otherwise the "reversible starter" is the same as the "non-reversible" one.

12. Another kind of mechanically operated starter is shown in plan and elevation in Fig. 6. It is made and patented by the Otis Elevator Company. Its action is different from the apparatus described in the foregoing article in so far that it is not connected mechanically to the motor or countershaft but to the main-switch spindle, and the gradual cutting out of resistance is obtained by a dashpot. The following description is taken from the patent specifications:

A box *A* contains in its rear part *A'* resistance coils, and in its front part the operating mechanism, the essential features of which consist of a snap switch *1*, an arm *2* for operating the snap switch, and a brush-carrying arm *3*, which brush operates in connection with a resistance device *10*; the brush arm *3* is, in the present instance, provided with a counterbalance *9* and controlled by a dashpot *4*; arm *3* is mounted on a shaft *5*, by means of which it is operated in the manner described later.

The switch *1* comprises essentially a knife blade *7*, mounted on a pivot *6*, adapted to engage and disengage the contacts *8*, *8*, and connected to this knife is a cam *16* having a notch *17*, into which projects the end of the arm *2* for moving the cam; the cam is further provided with recesses and projections *18*, with which a spring catch *15* cooperates, under the stress of a spring *15'* for holding the switch

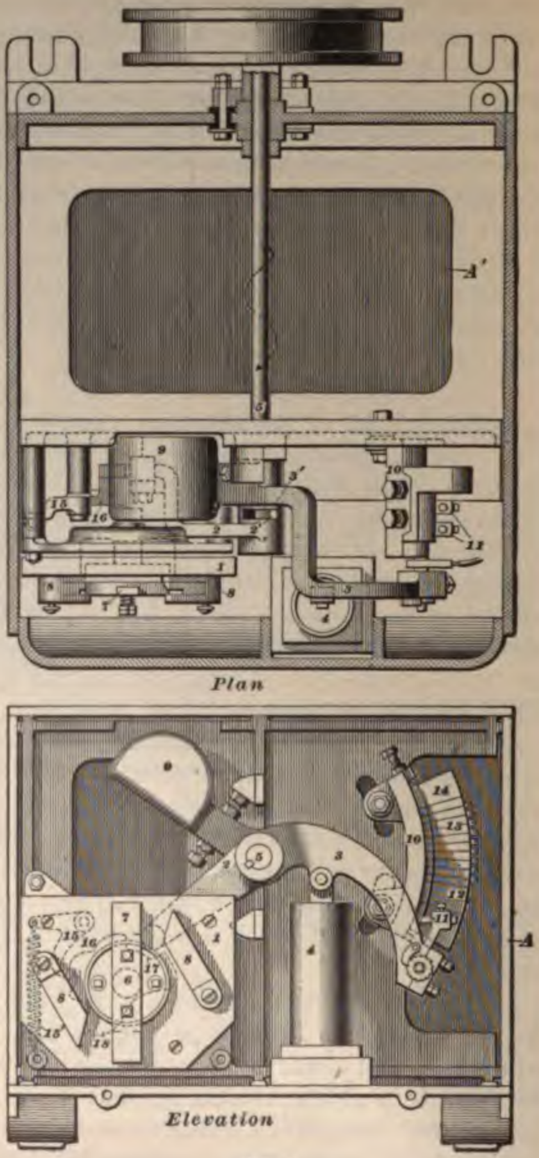

A'
9
3'
10
15
16
2
2'
11
1
8
7
3
4
5
Plan
14
13
10
6
2
3
5
7
1
15
16
8
17
4
12
11
A
8
15'
18
Elevation

FIG. 6.

in different positions and for making it complete its movements after it has been started, so as to produce the sudden engagement and disengagement in the manner well known in connection with snap switches. The arm *2* is rigidly connected to the shaft *5* so as to move therewith, while the brush-carrying arm *3* is loosely mounted on the shaft *5*; interposed between the two arms is a catch, or stop, so arranged that the arm *2* may move independently of the arm *3* when the parts are in one position, but when it is moved in the opposite direction and the arm *3* is in another position, they will move together. This catch consists of a projection *2'* on the hub of the arm *2* working in a slot *3'* in the hub of the brush-carrying arm *3*.

The brush-carrying arm *3* carries a brush *11* adapted to bear on the resistance-contact device *10*, and the contacts are arranged so that the contact *12* will permit a considerable movement of the brush before any of the resistance is cut out. While the contacts *13* are connected by the resistances in box compartment *A'* in the usual way, the contact *14* is connected directly with the line; so that while the brush is on the contact *12* all the resistance is included in the circuit, and as it sweeps over contacts *13* more or less of the resistance is cut out until it bears on the contact *14*, when all the resistance is out of the circuit. This resistance device *10* is made on the arc of the circle and is adjusted in the box by means of lugs and bolts engaging slots in the frame of the box.

In the figure, the circuit is shown open and all the resistance is included in the circuit, the catch *2'* bearing on one side of the slot *3'* of the brush-carrying arm *3*, holding the parts in the position shown. If, now, the shaft *5* is turned in the direction of the arrow, that is, to start the motor, the arm *2* operating through the cam *16* will move the switch blade 7 so as to engage the contacts *8*, the spring catch *15* riding over the projection *18* and tending to complete the throw of the switch arm as it enters the adjacent depression on the other side of the projection *18*, making a snap switch. The catch *2'* moves through the slot *3'* and leaves the brush-carrying arm *3* free to move, which, under the influence of the counterbalance *9*, it commences to do at once, but its movement is retarded more or less by the dashpot *4*. The parts are so arranged that before the brush *11* moves off the resistance contact *12*, the switch *1* has closed the circuit through the contacts *8* and the brush-carrying arm moves gradually over the resistance contacts, cutting them out, until the brush *11* bears on the contact *14*, by which time the motor has come up to speed. When the shaft *5* is turned in the direction opposite the arrow, that is, to stop the motor, the projection *2'* bears on the side of the slot *3'* so that as the arm *2* is turned to open the switch *1*, the brush *11* is moved over the resistance contacts, insuring the inclusion of the resistance in the circuit. It will be noted that the slot *17* in the cam *16* is of such dimensions as to permit the inclusion of a greater part of the resistance contacts before the knife blade *7* is actually moved from the contacts *8*.

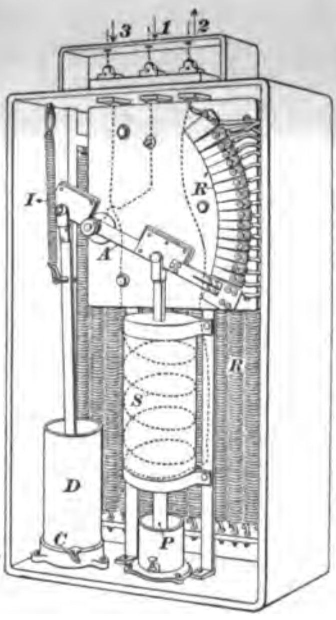


FIG. 7.

13. Solenoid Rheostats.—Instead of the weight *9*, Fig. 6, a solenoid is used in many starting devices. This permits the rheostat to be mounted separate from the switch, no mechanical connection between the two being required. The switch alone is mechanically operated by the hand rope or other operating device. Fig. 7 shows one form of solenoid rheostat, as manufactured by the Elektron Manufacturing Company. The armature current enters at the binding post *1*, whence it goes to the contact arm *A*, through the series of resistances *R*, and out at the binding post *2*. The solenoid current, taken from the main switch, enters at binding post *3*, goes through the windings of the solenoid *S*, and leaves at binding post *2*. As soon as the main switch is closed, the solenoid is energized and draws in the iron plunger *P*, raising the arm *A*, and thus making the contact piece at the end slide over the sectors *R'* of the rheostat and cutting out resistance from the armature circuit. In order that this may be effected gradually, the other end of the arm *A* is connected by a rod with a piston fitting in a dashpot *D*. In moving downwards, this piston must displace the air in the dashpot, and the speed with which this may be done is regulated by the stop-cock *C*. To bring the apparatus back to its original position at the breaking of the circuit, the piston end of the arm is provided with a spring *I* that is put in tension while the resistance is cut out. On opening the circuit, the spring pulls up the arm and dashpot piston, and in order that this may be effected quickly the dashpot has a relief valve that will open while the piston is going up.

14. The apparatus described in the foregoing articles as applicable to belt-shifting elevators are used for a number of other purposes, among which their connection with electrically driven pumps for hydraulic elevators is of special interest.

DIRECT-CONNECTED ELECTRIC ELEVATORS.

DIRECT-CONNECTED, BELTED ELECTRIC ELEVATOR.

15. The second step taken in the development of the electric elevator was the elimination of the countershaft and the tight and loose pulley, and the substitution therefor of a belt connecting the motor directly with the elevator machine. The mechanisms used in belt elevators for shifting the belt then became superfluous. Although the elimination of the countershaft seems a small and natural step to take, it makes a great change in the working conditions of the elevator, ice in the belt-shifting types the motor starts without oad, which is applied only after the motor has attained its normal speed; while in the direct-connected type, the motor nust start under load. There is nothing gained by having the motor and the elevator separate and belted together, and therefore direct-connected belted elevators are never used; they are described here only to help us to arrive gradually at the form of elevator now commonly used.

DIRECT-[illegible] ELEVATORS.

16. Connection [illegible] nd Machines.—The work-g conditions of the [illegible] nected belted elevator are not changed when the [illegible] oupled directly to the shaft of the elevator machi[illegible] he modern type of electric vator this is always [illegible] notor being mounted on the ne base with the mac[illegible]ne.

17. Motors.—Since in [illegible] ct-connected electric elevators the motor must start [illegible] r load and must, therefore, have a strong torque, it [illegible] also get up speed rapidly though gradually. Of these two conditions the last-named one is fulfilled by peculiar controlling devices that are described below, while the first-named one is fulfilled by the construction of the motor itself, which is generally of the compound-wound type—a series-field coil serving to give the necessary torque at starting and the shunt coil steadying the field. The series coils are generally cut out when the motor has attained normal speed, after which the motor runs as a simple shunt-wound motor.

Of alternating-current motors, only the two-phase or three-phase induction motors prove satisfactory for direct-connected electric elevators, since they will start under load with sufficient torque. These motors behave, as far as their action in the elevator combination goes, just like shunt-wound continuous-current motors.

18. Transmitting Devices.—The transmitting devices between the motor and car consist, with few exceptions, of worm-gearing, drum, and rope. The worm-shaft is almost invariably coupled to the motor shaft by a flange coupling, serving at the same time as a brake pulley. Both single worm- and double worm-gearing are used, as will be seen from the illustrations given farther on, the double worm being used mostly on heavy machines, to avoid the end thrust of the worm-shaft. Such heavy machines are also frequently provided with back gearing. Ordinarily, however, single worm-gearing is used, great care and ingenuity being displayed in the design of the step bearings for the worm.

19. Counterbalancing.—Direct-connected electric elevators of the drum type are always overbalanced.

20. Controlling Devices.—The power control of direct-connected electric elevators is entirely electrical, there being no belts to shift or similar mechanical operations to perform; but, besides breaking the current, the motor must be reversed. Hence, besides the simple snap switch and rheostat already mentioned in connection with belt-shifting electric elevators, a **reversing switch** or **pole changer** is needed.

In elevator practice, the complete apparatus necessary to control the electric motor—the **power control,** as we have called it—is called a **controller,** especially if the various parts of it are built together in such a way as to make a separate, self-contained piece of machinery. A number of different forms of such controllers are used by the various manufacturers of electric elevators, and they will be described with the various designs shown.

21. Brakes.—The braking arrangements used are either entirely mechanical, that is, such as are used in connection with belt and steam elevators, or electrical mechanical, or wholly electrical.

22. Operating Devices.—In the majority of electric elevators the operating devices are mechanical, such as hand ropes, hand wheels, and levers. Electrical operating devices

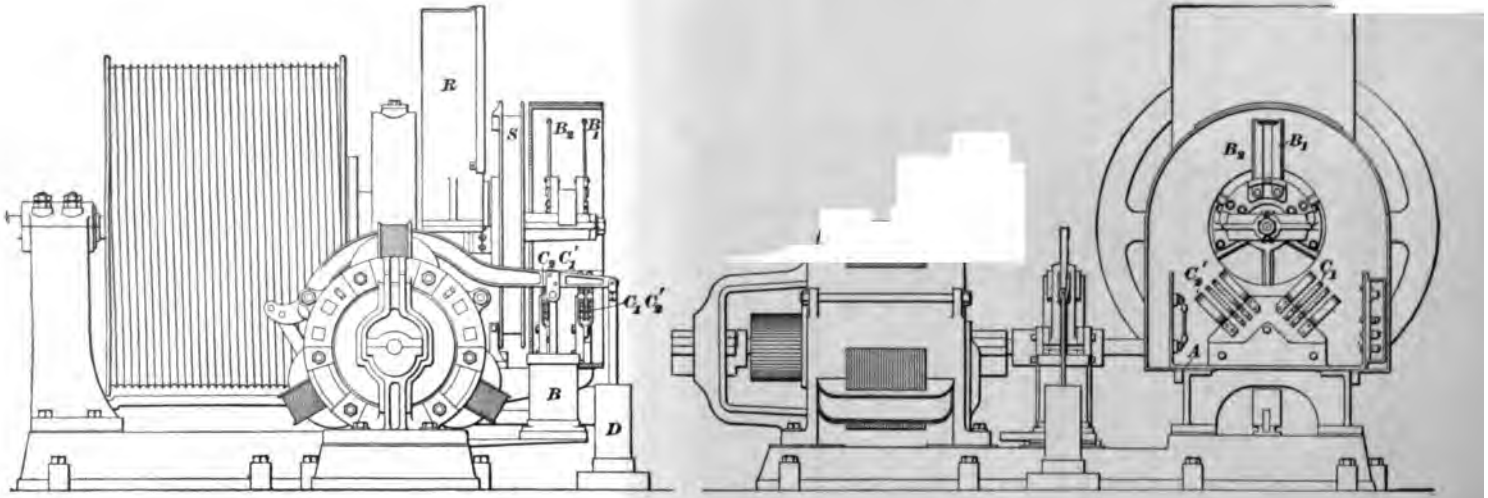


FIG. 8.

are being introduced, however, with success in connection with the magnet system of control, which is described later.

23. Motor Safeties.—Motor safeties are used in various forms; they are either mechanical or electrical or both.

EXAMPLES OF ELECTRIC ELEVATORS.

INTRODUCTION.

24. The examples of electric elevators here given do not represent all the various designs in the market, nor does the order in which they are described indicate any superiority of design of one make over another. A careful study of these will give a person enough insight into the construction and operation of this class of machinery to enable him to handle other makes of machines.

ELEKTRON ELEVATORS.

25. Motors.—Fig. 8 is an end and side elevation of an electric elevator made by the Elektron Manufacturing Company. The motor is the well-known Perret multipolar machine, shunt-wound.

26. Transmitting Devices.—The transmitting devices are single worm-gearing, drum, and rope. The arrangement of the step bearing of the worm is shown in Fig. 9. Alternate phosphor-bronze and steel disks are used to distribute the wear. The worm-shaft is attached to the motor shaft by means of a flange coupling *F*, which serves at the same time as a brake pulley.

27. Simple Controller.—The Elektron Manufacturing Company uses various kinds of controllers for various kinds of elevators. The simplest arrangement used is a double-throw switch attached to the hub of the shipper sheave *S*, Fig. 8, and a solenoid rheostat placed anywhere conveniently near the machine; such a rheostat is shown in Fig. 7 and another form in Fig. 10.

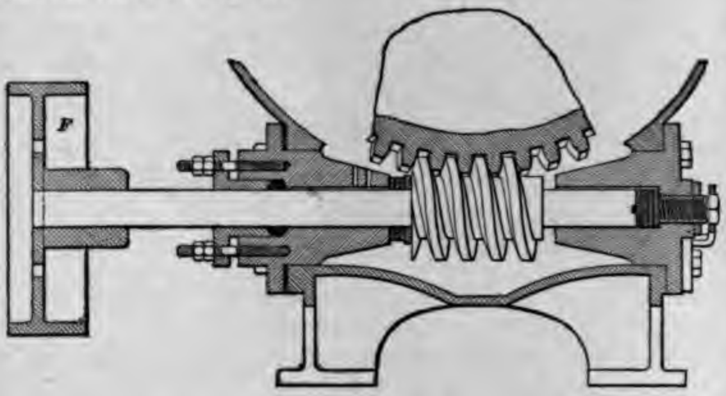


Fig. 9.

The switch consists of a casting A, Fig. 8, supported on the frame of the machine and carrying four sets of clips C_1, C_2, and C_1', C_2', to which the necessary line, field, armature, solenoid, and electric-brake connections are made as shown below. The switch blades B_1, B_2 attached to the shipper sheave engage the clips C_1, C_2, or C_1' C_2' for the up trip and the down trip, respectively. In Fig. 8 the blades are shown in their neutral position; that is, when the elevator is at rest. It will be seen that to start the elevator up or down, the sheave with the blades must be turned through an arc of 135°, the clips being set at right angles. This long travel is given for the purpose of giving the rheostat arm time to fall back into its starting position before the current in the armature can possibly be reversed; it also helps to reduce sparking and flashing at the clips.

28. Ordinary Brake.—The brake used in these machines is, for ordinary service, a simple mechanical one, which is released by a cam on the shipper sheave through a system of levers and applied by a weight, as with belt elevators. For passenger service, an electrical-mechanical brake is used, which is released by an electromagnet and applied by gravity. This arrangement is shown in Fig. 8, in which the brake magnet is marked B; the rapidity of action of the same is regulated by a dashpot D.

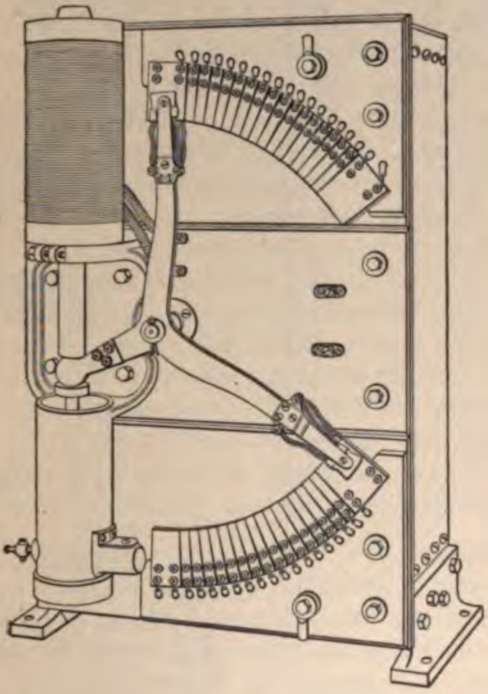
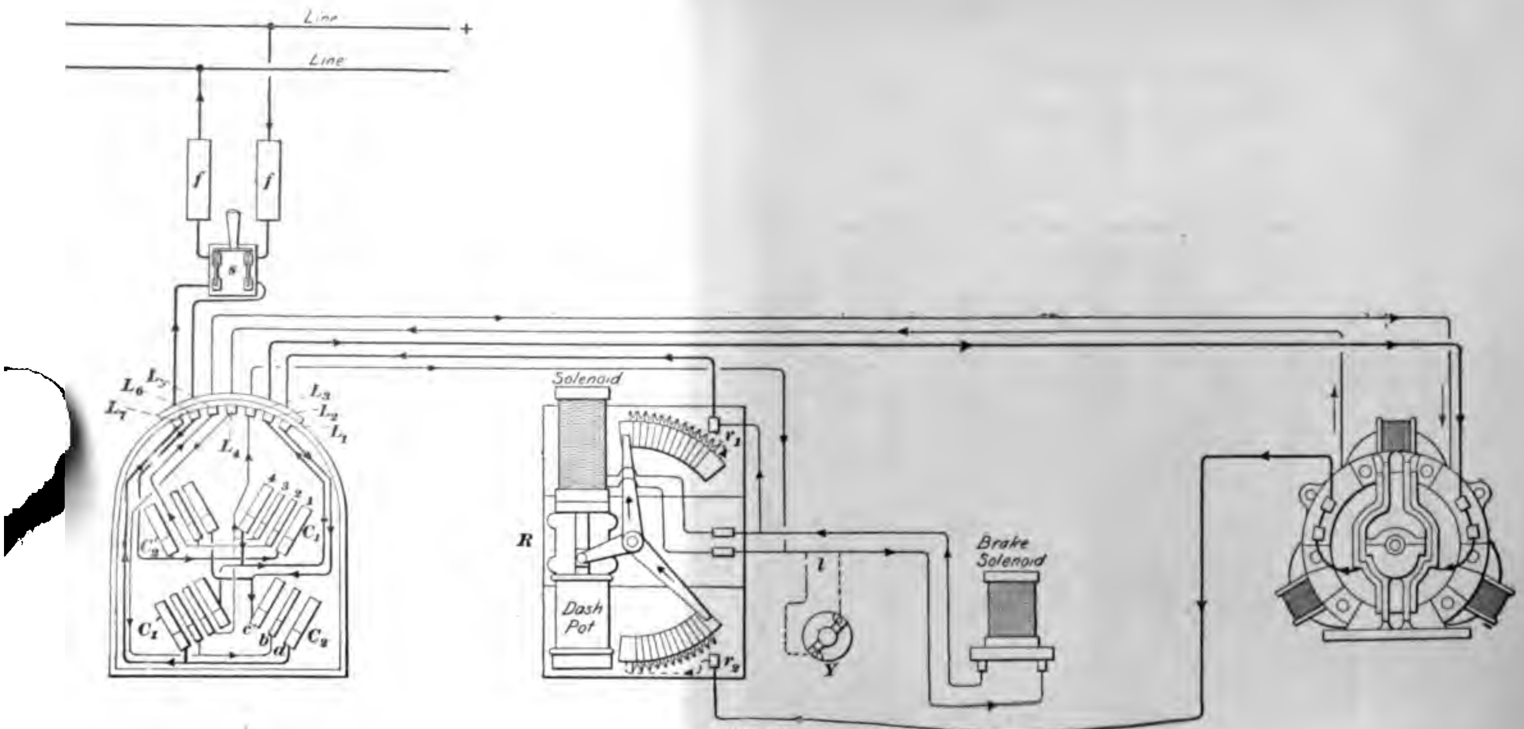


FIG. 10.

29. Fig. 11 is a diagram of the electrical connections between the switch, rheostat, brake, and motor. It will be useful to follow out these connections. The lines are connected through the fuses f, f and the double-pole switch s to the elevator switch at the binding posts L_8 and L_7. Supposing the blades of the switch to be thrown to the right, that is, across the clips C_1 and C_2, and the current to enter at the binding post L_6, then it passes first to clip *1* of the set C_1, whence it divides by means of the switch blade among the clips *2*, *3*, and *4*. From *2* it passes to binding post L_5, thence through the field windings of the motor, back to the binding post L_4, thence to the clip b of set C_2,



Line
+
Line
f
f
s
L_6
L_5
L_7
L_3
L_2
L_1
L_4
4 3 2 1
C_1
C_2
C_1
C_2
c
b
d
Solenoid
R
r_1
r_2
Dash Pot
l
Y
Brake Solenoid

FIG. 11.

over the blade crossing this set of clips to clip a, thence to binding post L_1 and to the line, thus completing the shunt circuit for the field. From clip *3* the current goes to the binding post L_2, through the solenoid windings of the rheostat R to the binding post r_1 of the rheostat to the binding post L_1 of the switch, to clip c of set C_2 over the blade to the clip a, to the binding post L_1 to the line, thus completing the circuit through the solenoid. From clip *4* the current goes to binding post L_2, thence through the armature of the motor to the binding post r_2 of the rheostat, through the lower half of the resistance, through the rheostat arm and the upper half of the resistance to binding post r_1, to L_1, c, a, L_1, and line, thus completing the armature circuit. Throwing the blades to the left, we will find, in following out the three circuits again, that the current traverses the field circuit in the same direction as before, but that the current in the armature is reversed, thus reversing the motor. The electromagnet windings of the brake are in shunt with the solenoid circuit, as is easily seen from the diagram.

30. The operation of this elevator is as follows: When the shipper sheave is thrown over to the right or left, the brake magnet is energized and tends to slowly release the brake, since the dashpot prevents too sudden a release; at the same time the solenoid is energized. This tends to slowly cut out the resistance from the armature circuit; the dashpot prevents too quick an action, and it is so adjusted that all the resistance will be cut out by the time the motor reaches its normal speed. Upon breaking the circuits, the brake is at once applied and the resistance arm drops back into its original position, ready for another start.

31. Dynamic Brake.—On high-speed elevators, in order to get a particularly smooth stop, the Elektron Manufacturing Company uses, in addition to the electrical-mechanical brake, a so-called dynamic brake, which, indicated in Fig. 8 at R, is usually placed on a bracket between the shipper sheave and worm-gear case. It is shown in detail in Fig. 12 and consists of a switch lever L, actuated by a cam on the operating sheave, and a variable resistance.

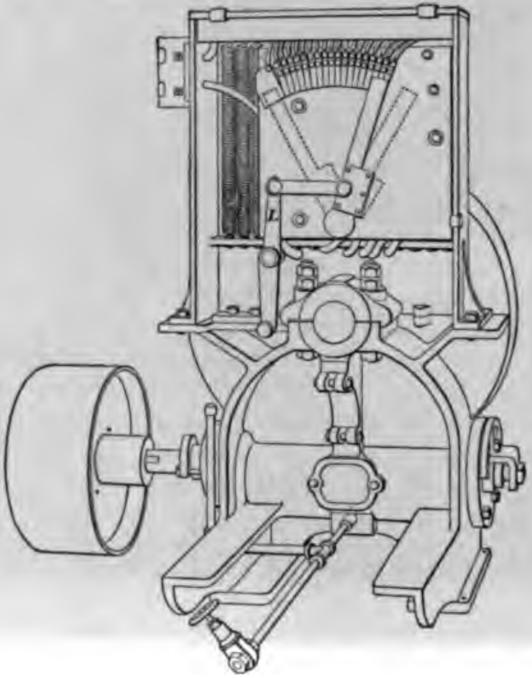


FIG. 12.

This resistance is so connected to the system that the armature is short-circuited through it immediately after the circuit from the line is broken to stop the elevator, thus acting as a stopping resistance, the motor acting as a dynamo and sending a current through the resistance. This has the effect of slowing the motor down quickly but smoothly, like a brake, and more smoothly than an ordinary frictional brake. The smoothness of the stop is made still more marked by the resistance being gradually cut out of the armature short circuit as the motor slows down, the cam operating the lever L being so constructed as to first cut in all the resistance at the instant the main circuit is broken; on being turned farther by the operator, the switch lever is caused to brush over the resistance contacts, thus gradually cutting the resistance down to zero. Of course this short circuit is opened before the elevator is started again. As has been said, the dynamie brake is used only in addition to the ordinary brake, the latter being necessary to hold the car stationary after it has been stopped.

32. Fig. 13 shows diagrammatically the connections when the dynamic brake is used. The field must necessarily remain excited after the armature circuit is broken and the armature short-circuited, in order to make the motor act as a dynamo. The field is, for the sake of simplicity, kept excited all the time, but in order to cut down the current thus constantly wasted while the elevator is standing still, a resistance is inserted in the fields. When the elevator is started, this resistance is short-circuited, thus giving the fields the full current due to its windings and, consequently, the full torque available. When the elevator is stopped, the resistance is cut in, choking the field current, but leaving it strong enough to give sufficient magnetism to get a dynamic-brake effect.

33. Speed Regulating Controller.—Another type of controller used by the Elektron Manufacturing Company is shown in Figs. 14 and 15, while the diagram of connections is given in Fig. 16. It is evident that the combinations described in the previous article do not allow of any regulation of speed, the motor being simply shunt-wound with an unchangeable field. The purpose of the arrangement now to be described is to give speed regulation, which is accomplished by a changeable resistance in the field. The controller is mechanically operated.

As seen in Fig. 14, there are two cams *I* and *II* operating the armature and field-resistance arms A_a and A_f, respectively. Both arms are provided with dashpots D_a and D_f. Two more cams *III* and *IV*, shown in Fig. 15, operate the reversing switch, or pole changer, P; the one cam is intended to throw the switch for going up and the other for going down. While not visible in the illustrations, there

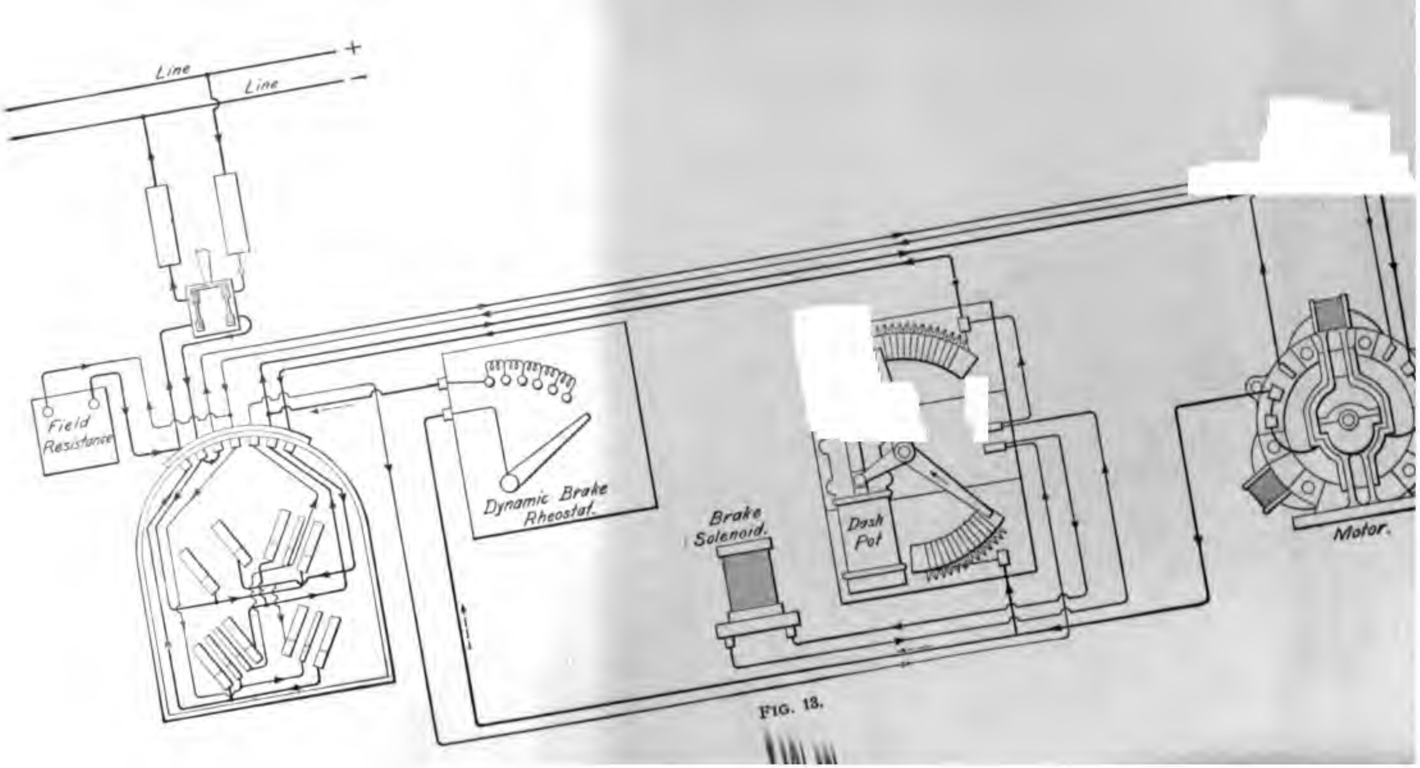

Line
Line
+
−
Field Resistance
Dynamic Brake Rheostat.
Brake Solenoid.
Dash Pot
Motor.

FIG. 13.

are other cams that operate various knife switches. All these cams are mounted on the shipper-sheave shaft S. The brake is the same as in the previous design.

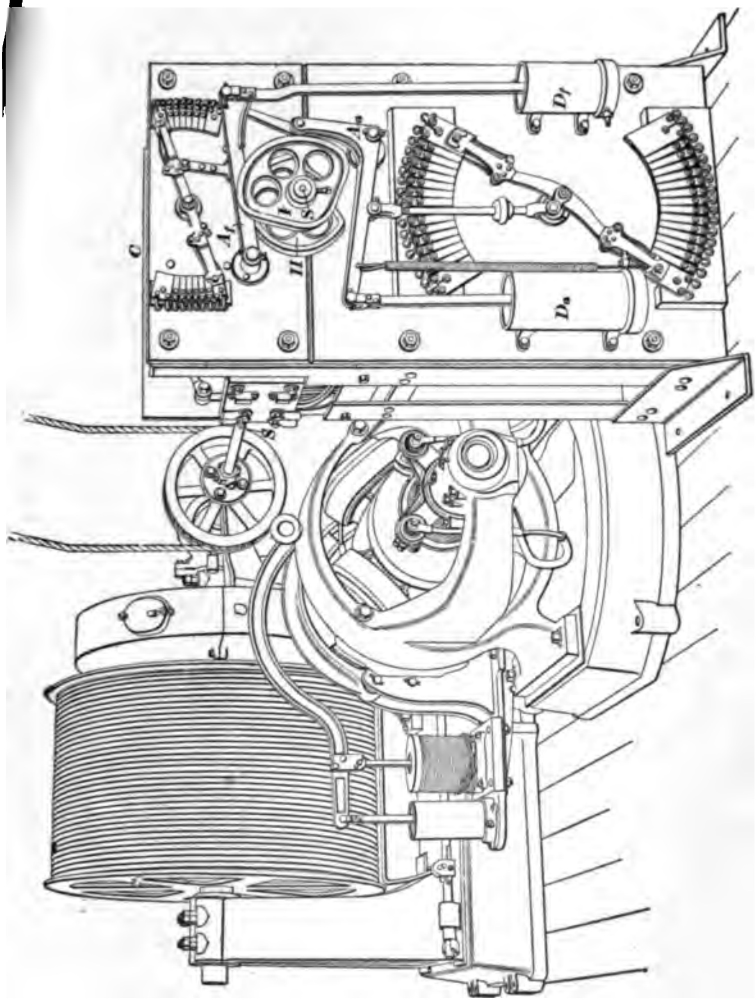


FIG. 14.

34. Fig. 16 is a diagram of the connections for this controller. (a) shows the external connections between motor, brake, and connection board B; (b) gives the internal

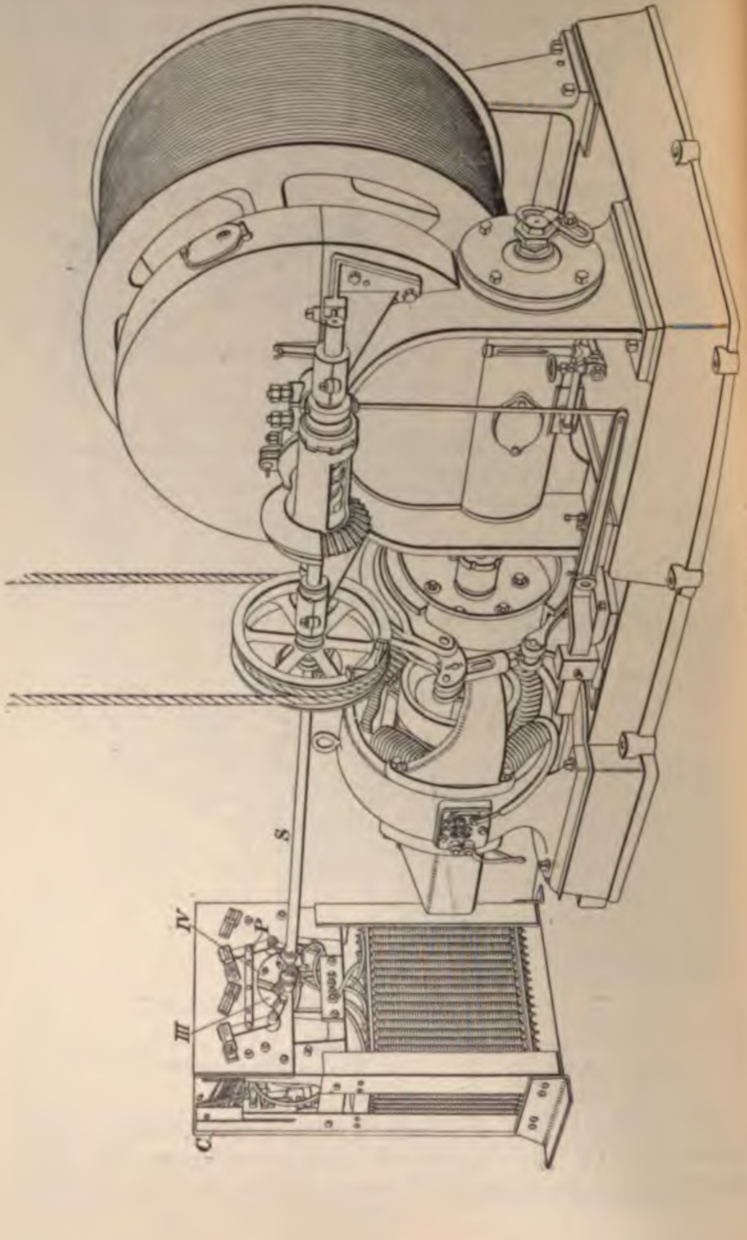


FIG. 15.

connections between the connection board B and the various clips and resistance blocks inside the controller. By swing-

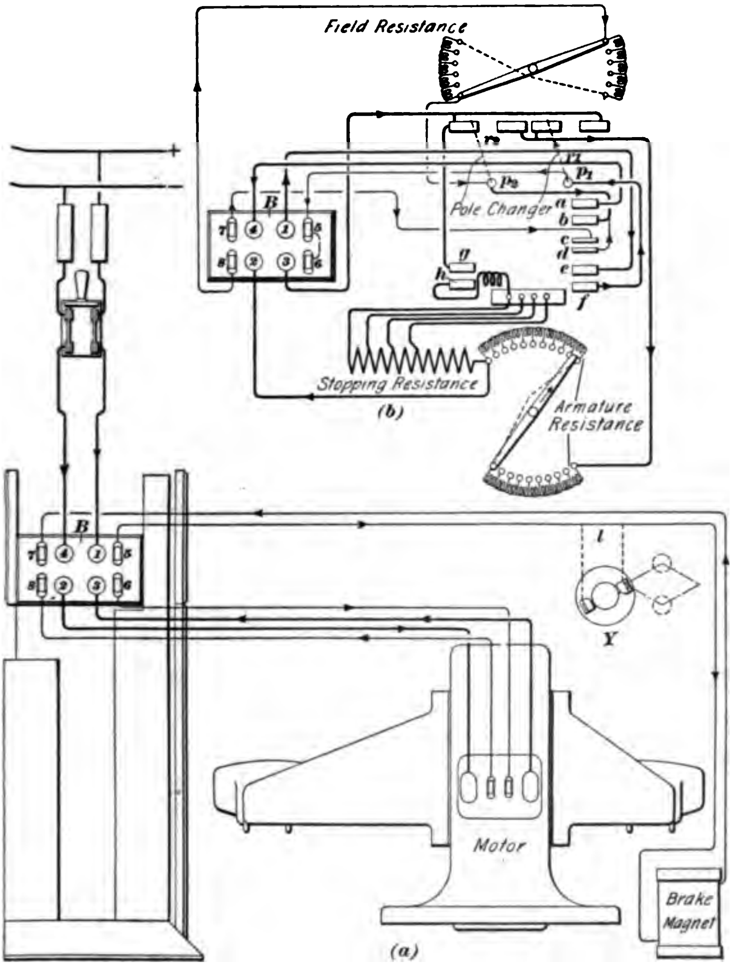


FIG. 16.

ing the shipper sheave to the right or left, switch blades connect the clips a and b, c and d, and e and f, completing the circuits. Thus, supposing the current to enter the system from the line at the binding post *1*, it goes to the clip *e*, over a blade or knife to the clip *f*, thence to the pivot p_1 of the pole changer, where it divides. One branch goes through the pole-changer arm r_1 and the armature resistance to binding post *2*, thence through the armature back to the binding post *3*, thence through the other pole-changer arm r_2 to the pole-changer pivot p_2, to the clip *b*, over the knife to the clip *a*, thence to the binding post *4*, and back to the line, thus completing the armature circuit. The other branch of the circuit goes from p_1 to the binding posts *5* and *6*, which, in turn, are connected, respectively, to the brake-magnet circuit and the shunt-field magnet circuit. The other terminal of the brake-magnet circuit is connected to the binding post *7*, whence the current flows over clips *c* and *d*, and *b* and *a* to the binding post *4*, and back to the line. The other terminal of the field circuit is connected to the binding post *8*, whence the current flows through the field resistance to p_2, *b*, *a*, post *4*, and back to the line.

35. The cam *I*, Fig. 14, on the shipper-sheave shaft is so arranged that after the circuits are closed the armature-resistance arm A_a is free to move, which it does slowly under the retarding influence of the dashpot D_a, gradually cutting out resistance until at the normal speed of the motor all resistance is cut out. After turning the shipper sheave a little farther, the cam *H* controlling the field-resistance arm A_f is released, but is retarded by the dashpot D_f. Thus the field resistance is slowly *cut in*, weakening the field and speeding up the motor.

36. Another pair of clips *g* and *h*, Fig. 16, is so connected that when a switch blade is thrown across them, the armature is short-circuited through the stopping resistance. This switch *g h* is closed and the armature short-circuited when the other circuits are opened.

37. Motor Safeties.—The usual motor safeties, viz., limit stops and slack-cable safety, such as we have met in connection with belt and steam elevators, are used in the Elektron elevators. Their arrangement is shown in Fig. 15.

38. Another motor safety used is in the shape of a switch controlled by a centrifugal governor running in unison with the car, and which opens a switch in the brake circuit when the car attains undue speed. This safety is indicated at *Y* in Figs. 11 and 16, and is connected in series with the brake solenoid by opening the solenoid circuit at point *l* and inserting switch *Y*, as indicated by the dotted lines.

SEE ELECTRIC ELEVATORS.

39. Motors.—Fig. 17 shows one of the standard machines built by the A. B. See Manufacturing Company. A bipolar, drum-armature, compound-wound motor is used.

40. Transmitting Devices.—Among the transmitting devices, the step bearing shown in Fig. 18 is of peculiar construction. Both steps, that for the up trip and that for the down trip, are located at the free end of the worm-shaft and are easily accessible. The one is adjustable by means of the plug *P* in the cap *C*, while the other is made so by means of the nut *N* on the threaded free end of the shaft. The other end of the worm-shaft passes through a stuffingbox *S*, as in other machines. The worm and lower part of the worm-wheel are constantly running in oil.

41. Controller.—The controller, as shown in Fig. 17, is placed on top of the motor and consists of a box with three compartments, one of which is accessible from doors *O*, and another one from similar doors on the opposite side. The first, shown open in Fig. 19, contains the main reversing switch *M* and three snap switches *N*, *N'*, and *U*, the blades, or knives, of the latter being mounted on the same lever, but insulated from one another. The switches are operated by a bar *B*, which, in turn, is linked to the rack *R*, Fig. 17, and operated by a pinion *P* fastened to the shipper sheave *S*. The opposite compartment contains a solenoid dashpot, a resistance lever, and resistance contacts very much the same as those shown in Figs. 7 and 10. The third compartment is located between the first-named two and contains resistance coils of German-silver wire. The walls of the compartments

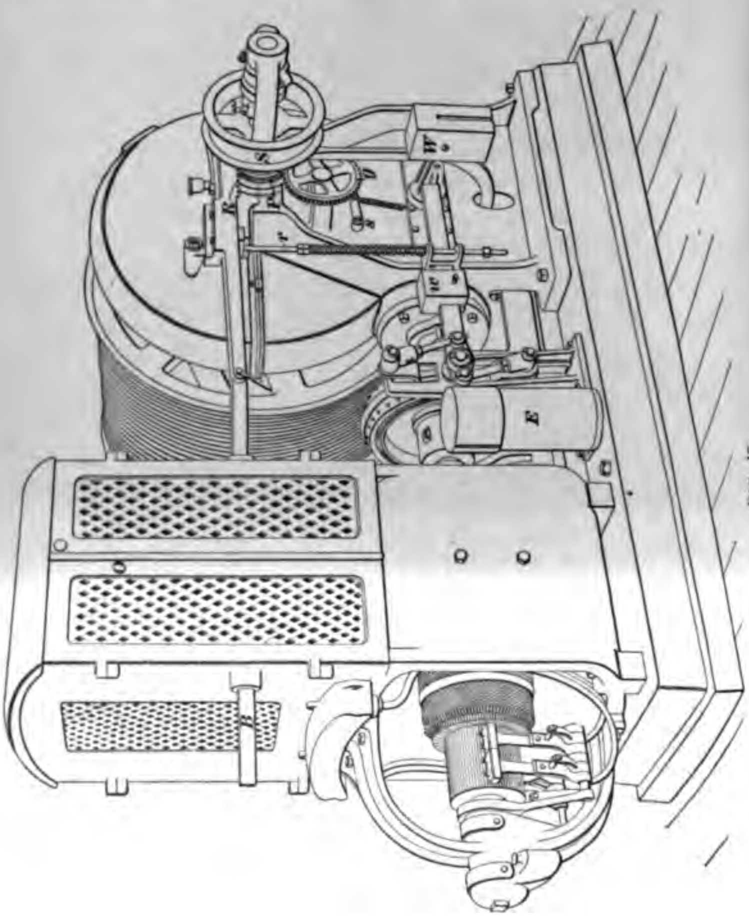


FIG. 17.

are cut away wherever they are not needed for the support of contacts or mechanisms, so as to give ventilation to the resistance coils; the doors and sides of the controller are perforated, as shown in Fig. 17, for the same purpose.

Brake.—The brake used in this machine is con-
l mechanically and electrically. A spring-cushioned
rod *r*, Fig. 17, is operated by an arm fastened to the
er sheave and forces the brake lever down to apply the

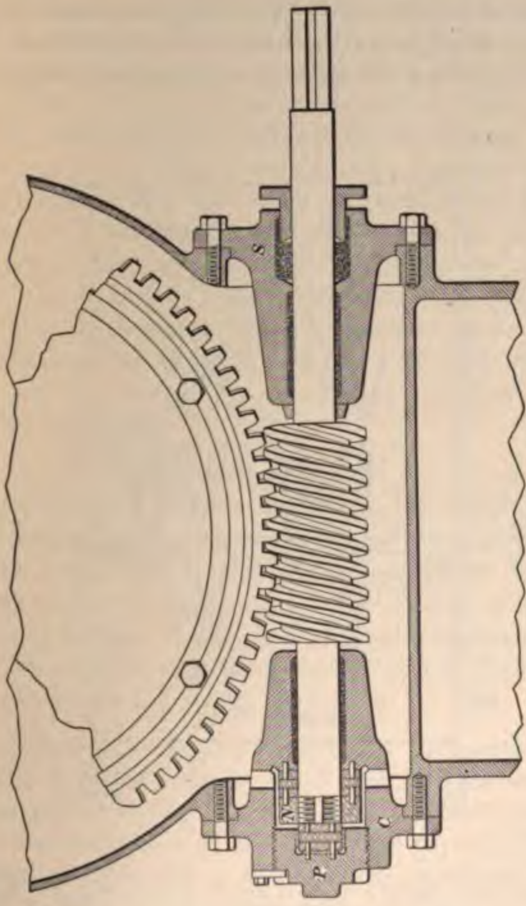


FIG. 18.

A solenoid *E* holds off the brake as long as there is
nt in the armature with which the solenoid is connected
ies. A weight *W* applies the brake, when the current
oken. There is also a dynamic-braking effect, the

armature being short-circuited through resistanc current is shut off from the machine.

43. Motor Safeties.—This machine is particula provided with motor safeties. Not only the usual tra nut, limit-stop, and clutch-operating slack-cable saf provided, but an extra limit switch is also provided breaks the current through the armature and brake s

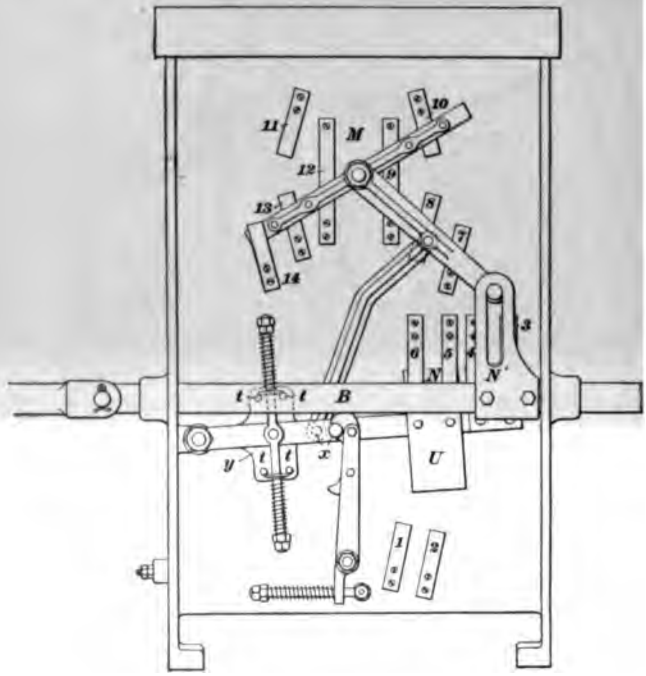


FIG. 19.

at the limits of car travel. This switch *s*, Fig. 17, lo the worm-gear casing below the drum shaft, is spring a and tripped by a stop on a gear *g*, which is one of a train of gears driven from the drum shaft. The w throws in the clutch that connects the drum shaft shipper sheave when it is tripped by slack cable.

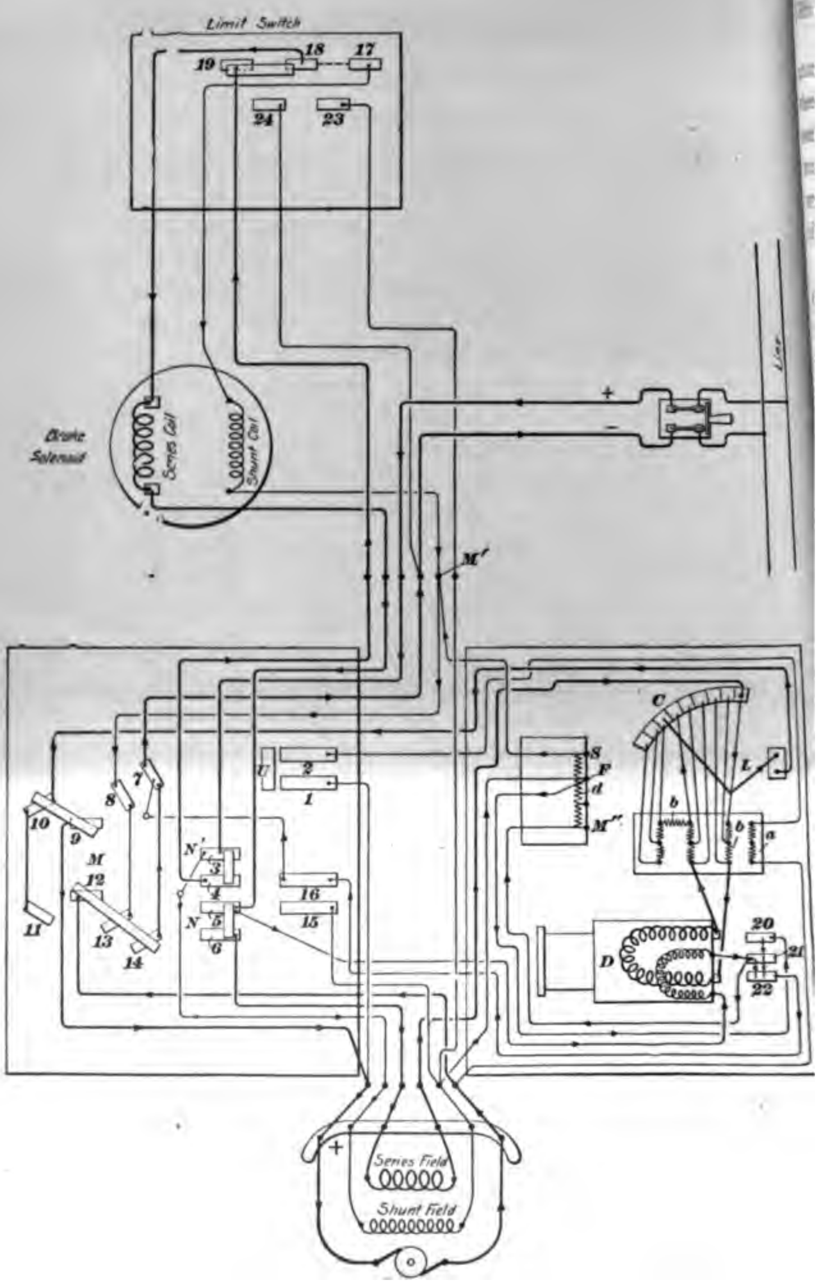
44. Electrical Connections.—Fig. 20 is a diagram of the electrical connections. The contact pieces are marked in the diagram the same as in Fig. 19. The circuits for the position of the controller shown in this figure are as follows.

45. *Armature Circuit.*—In the armature circuit the current passes through the + line to clip *3*; from clip *3* to clip *4* over the blade of the switch; from clip *4* to clip *19*; and from clip *19* to clip *18* over the switch blade, which is open only when the car overtravels the normal limits of travel; from clip *18* the current passes through the series coil of the electric brake to clip *5* and then to clip *6* over the switch blade; from clip *6* it passes through the series field of the motor, through the armature resistance and series coils on the armature resistance solenoid to clip *10* and then to clip *9* through the switch blade; from clip *9* it passes through the armature to clip *12*; from clip *12* to clips *13* and *14*; from clip *14* to clip *7*; and from clip *7* to the – side of the line.

46. In the armature circuit, when the pole changer is reversed from the position shown in Fig. 19, the current passes from the + line to clip *3* and then to clip *4*; from clip *4* to clip *19* and then to clip *18*; from clip *18* to clip *5* and to clip *6*; from clip *6* to clip *10* and on to clip *11*; from clip *11* to clip *12* and then through the armature, in a reverse direction, to clip *9* and then to clip *8*; from clip *8* to clip *7* and then to the – side of the line.

47. *Dynamic-Brake Circuit.*—When the controller is in its neutral position, that is, when the current is shut off from the machine, clips *1* and *2* are bridged by the switch blade *U* and the motor is short-circuited through the resistance, passing from clip *1* through the armature and then through the short-circuit resistance *a* to clip *2*.

48. *Electric Brake.*—The shunt coil of the electric brake obtains its current from clip *17*, clips *17*, *18*, and *19* being bridged by one switch blade, which is operated by the stop motion mentioned in Art. **43** and which stop motion



Armature

FIG. 20.

automatically breaks connection between clips *17*, *18*, and *19* when the car overtravels its normal limits. This switch is essentially an automatic safety switch, for it not only breaks the line current before it passes through the armature, but also breaks the current flowing through the shunt coils of the brake solenoid.

The + side of the shunt coil is connected to the separate clip *17* instead of to the clip *18* in order that upon breaking the circuit the armature circuit may be disconnected from the electric-brake circuit, thus allowing the brake to act at once. Otherwise, the motor still running would send enough current through the shunt coil of the brake solenoid to keep it energized and thus prevent its action. The electric-brake circuit is, therefore, from clip *17* through the shunt coil to the terminal M', and from M' to clip *8* or *13*; from clip *13* to clip *14*, or from clip *8* to clip *7*, and thence to the — side of the line.

49. *Path of Current in Starting Box.*—The shunt coil of the solenoid D, Fig. 20, gets its current from clip *5*; and after the current passes through the coil it enters clip *21*. The switch blade, or knife, that bridges clips *20*, *21*, *22* is drawn out of contact with the clips when the plunger of the solenoid reaches the end of its travel, when all the resistance in the armature circuit is thus cut out. Before the switch blade is removed, the current crosses on it to clip *22*; from clip *22* it passes to clip *16*, and thence to clip *7*, whence it goes to the — side of the line.

When the contact is broken, the current is forced to pass from clip *21* to and through the resistance d from the terminal M'' to the terminal S; from the terminal S it passes to the terminal M', then to clip *8*, and so on to the negative side of the line. The resistance d is introduced in this circuit for the purpose of reducing the heating in the shunt coil and to reduce the current consumption after the solenoid has done its maximum work.

50. *Field.*—The field circuit of the motor is as follows: The current passes from the + line to clip *3* and thence through the field to the terminal *F* of the resistance *d*. From *F* it passes to clip *20* and thence to clip *22*, to clip *16*, to clip *7*, to the negative side of the line. When the armature resistance is all cut out, contact between clips *20*, *21*, and *22* is broken, and the current is forced to pass through the portion of the resistance between the terminals *F* and *S* to the negative side of the line, provided the parallel connections from clip *15* to clip *16*, or at the limit switch from clip *23* to clip *24*, are broken. This resistance weakens the field on the motor and causes it to run at a higher speed. The contact between clips *23* and *24* is automatically made and broken when the car gets within about a floor from the top or bottom of its travel, and by the same stop motion that operates the limit switch. When the switch blade connects clips *23* and *24*, the resistance in the field is short-circuited; the field strengthens and the motor slows down. The switch blade bridging clips *15* and *16*, although situated at the machine, is withdrawn directly by the operator in the car during the last few inches of travel of his controlling lever, and he is thus enabled to weaken the field on the motor and run at a higher speed, but only after the car passes the first floor from the top or bottom and after all the resistance is cut out of the armature circuit.

51. In some of the See machines, the field is broken every time the motor stops. Fig. 20 is a diagram of a machine where the field is *on* all the time. Whether the field is to be left on or off is determined by the duty of the elevator. When the high-speed attachment is left off, a change in connections from those shown in Fig. 20 is made, Fig. 20 being a diagram of connections for a high-speed elevator running 250 feet per minute and over.

OTIS ELECTRIC ELEVATORS.

52. **Motor.**—The Otis Elevator Company makes a number of styles of electric elevators. They are all of the drum type, but have various kinds of controlling devices. Figs. 21

and 22 illustrate what may be termed the standard type of Otis elevators.

The motor used is the Eickemeyer bipolar, drum-armature, compound-wound type, the series coils of the field being

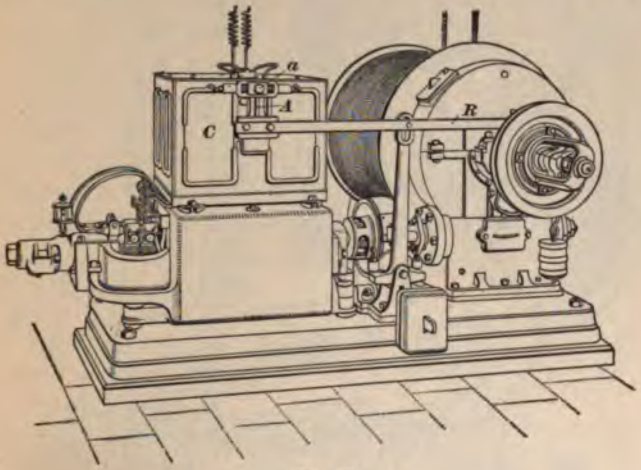


FIG. 21.

cut out after the starting resistance has all been cut out, that is, when the motor has acquired normal speed. This is done both on the up and down trip of the car.

53. Transmitting Devices.—With regard to the transmitting devices, it may be mentioned that either single or double worm-gearing is used, the latter for the larger sizes generally. In connection with the single worm a peculiar kind of step bearing is used. The purpose of this arrangement, shown in Fig. 23, is to increase the bearing surface, without enlarging the diameter of the step, by dividing the pressure between two surfaces, viz., the end surface *s* of the shaft and the ring-shaped surface *s'* of the bushing *B*. Now, it is well known to any mechanic that it is next to impossible to make the wear equal on two such separate surfaces unless special provision is made for it. This provision consists in this case of a couple of small levers *l*, *l* having three

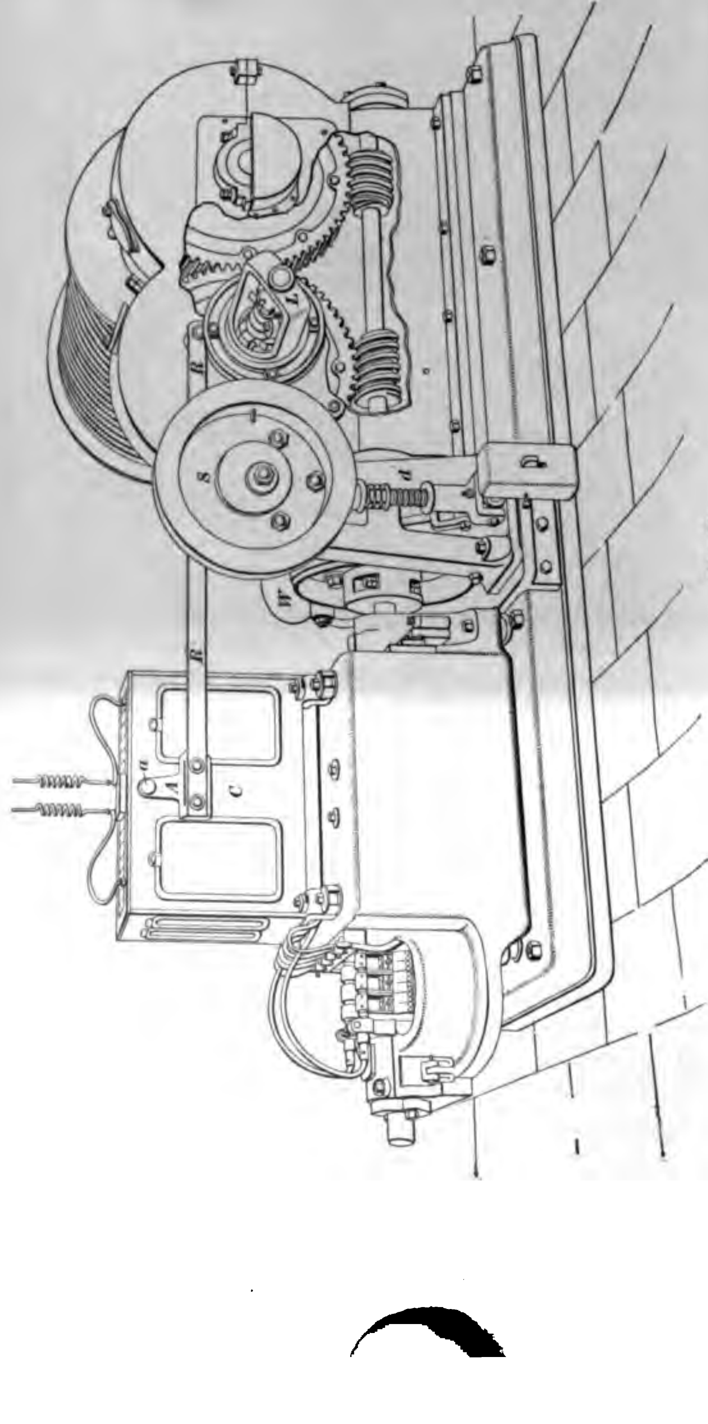


FIG. 22.

points each. One of these points, in the middle of one side of the lever, rests against an adjusting screw *S*, which is provided for the purpose with a circular groove. Of the other two points on the ends of the other side of the levers, one rests on the step plate *P* and the other on the bushing *B*. If the bushing wears faster at *s'* than the step plate wears at *s*, the shaft will move to the right, which will cause the levers to press on the bushing, and vice versa. Thus, the pressure is equally distributed over both surfaces *s* and *s'*. The screw *S* serves to take up the wear. The little equalizing levers *l*, *l* are held in place by being placed in slots in the sleeve or bushing *B*, and by a pin *p* that fits into semicylindrical grooves in the end of the levers. Buffers between the worm-gear and drum are used on all Otis electric elevators to absorb vibration.

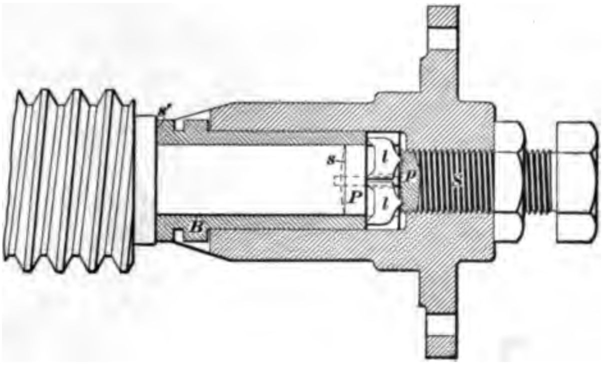


FIG. 23.

54. Controlling Devices.—The controller of the Otis elevators is box-shaped and is usually mounted on top of the motor, as shown at *C* in Figs. 21 and 22. It is operated by a rod *R* attached to the shipper sheave, which rod has an arm *A* on the other end, which engages by means of a part *a* with another arm or crank, hidden underneath the arm *A*; this crank is fastened to a shaft that reaches inside the controller box. In Fig. 24, which is a drawing showing the interior mechanism of the controller, this shaft is marked *s*. For clearness, the two parts (*b*) and (*c*) of the mechanism are

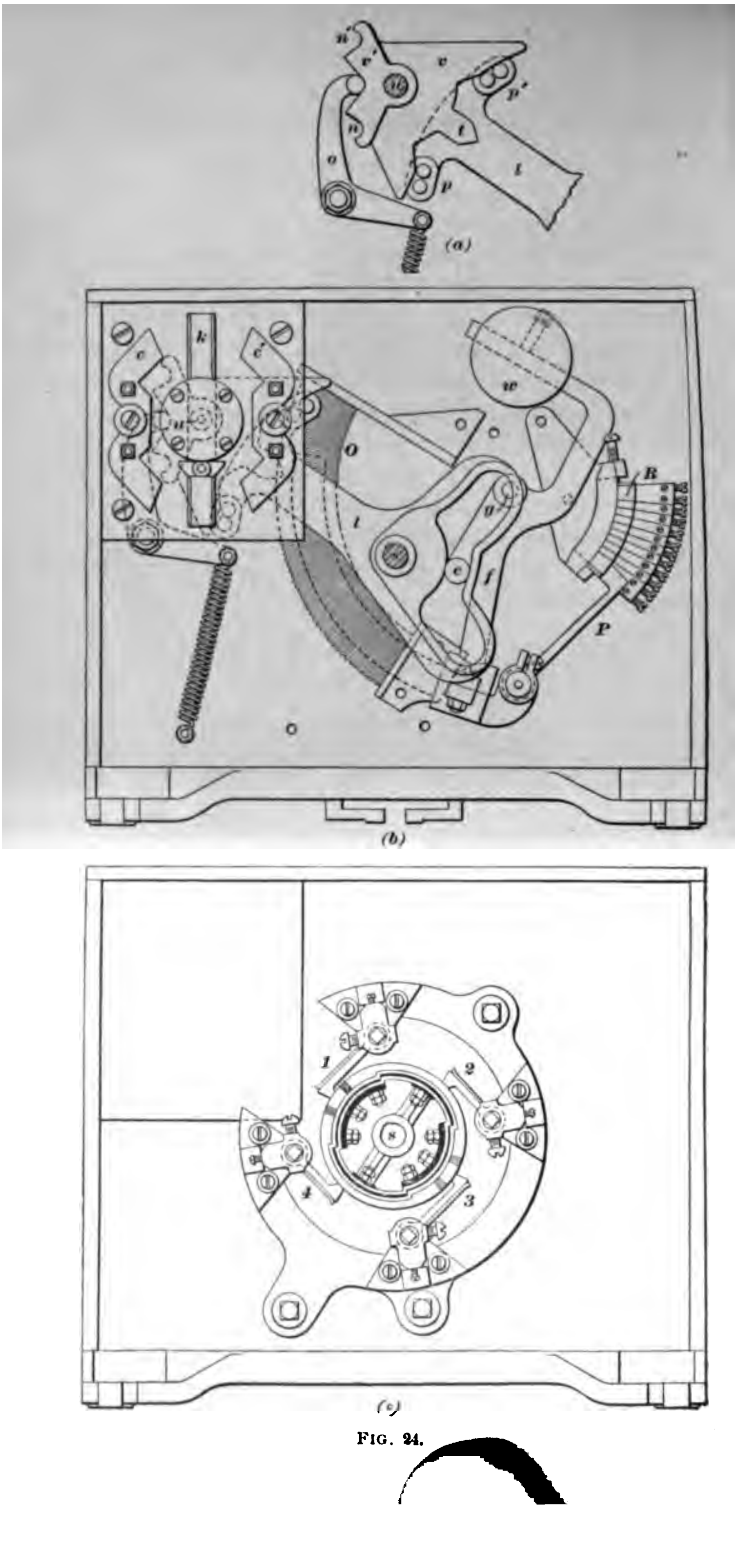


FIG. 24.

shown apart, while in reality part (c) is in front of part (b). Fig. 24 (a) is a detail view of some of the parts not very clearly shown in (b), where they are shown in dotted lines. The following is a description of the mechanism: the portion (c) contains the **reversing drum** mounted on and keyed to the shaft s; it has four contact plates insulated from one another. On these contact plates, of which two are long and two short, there rest four brushes *1*, *2*, *3*, and *4*, 90° apart. By turning the drum to the left, brushes *1*, *2* and *3*, *4* are made to rest on the same long contacts; while by turning the drum to the right, brushes *1*, *4* and *2*, *3* are brought into connection. The brushes are so connected to the armature and line that by turning the drum as aforesaid, the current in the armature is reversed. This will be plain from the diagram of connections given in Fig. 25. Behind the drum there is also fastened to the shaft s a lever l, Fig. 24 (a) and (b), carrying pins p and p', which, when the shaft s is turned, engage a tooth t formed on a plate v pivoted at u. The plate v carries another plate v' having notches into which falls the end of a spring-actuated bell-crank lever o. By turning the shaft s, the plates v and v' are first turned around u until the end of the lever o rides on one of the sharp corners of the plate v', whereby the spring of lever o is stretched. Turning the shaft s a little farther makes the end of the lever engage the inclined planes n or n', which are so located that the spring causes the plate v' to make an additional quick rotary motion.

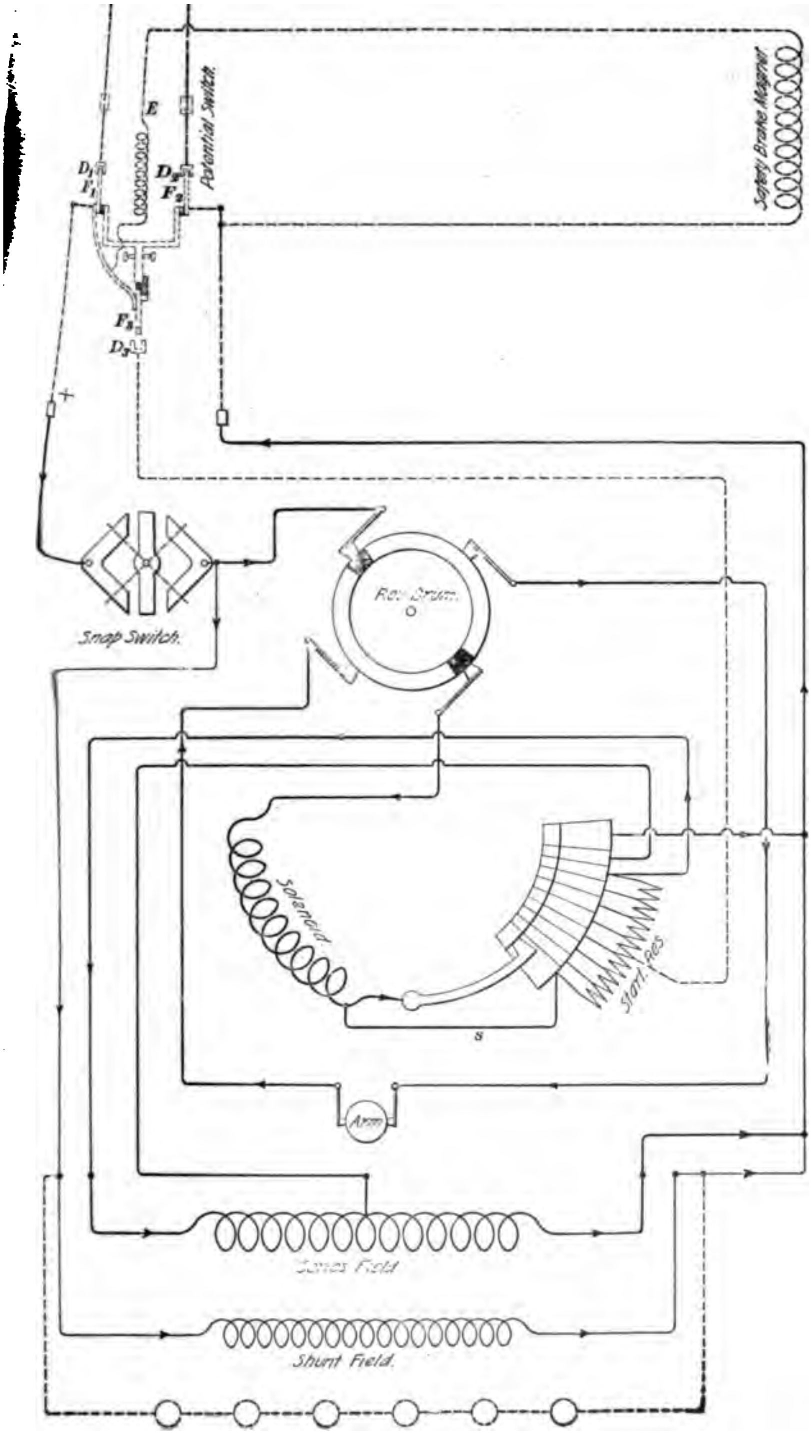
On the pivot u is fastened the blade k of the knife switch shown in the upper left-hand corner of Fig. 24 (b), and the quick rotary motion of the plate v' causes this blade k to snap between the clips c and c' of the switch. It is evident that on returning the mechanism to its middle position, the same snap action is caused by the two middle inclined planes of the plate v', so that the switch blade k is quickly withdrawn from the clips c, c', thus avoiding the formation of arcs; this is really the main object of the snap switch.

The other end of the lever l is formed into a cam of peculiar shape, which engages a pin e of a double-armed lever f

pivoted at g. This lever f has fastened to its lower end a curved magnet core entering a solenoid O, as well as a contact arm P arranged to slide over resistance contact blocks R. The greater part of the weight of the magnet core and arm P is counterbalanced by a weight w on the other arm of the lever f, so that when free to move, the magnet core, while having the tendency of swinging out of the solenoid, will be pulled back into the same as soon as the current will produce enough magnetism to overcome the unbalanced weight of the core and the arm P. The lever f becomes free to move, however, only after the shaft s has been turned enough to make the circuit at the snap switch, the cam on the lever l holding all parts in position until then.

55. Supposing that the solenoid and the resistance R are in series with the armature, it will be seen that the operation of this apparatus is as follows: First the circuits are closed with all the resistance in the armature circuit and the motor starts up. By the time the motor has gained some speed the lever f is set free, and if the speed of the motor is such that the counter-electromotive force is enough to cut down the armature current to the desired amount, the solenoid will not hold the core, the latter will swing out, and the arm P sliding over the contact blocks R will gradually cut out the starting resistance. Should for any reason the armature current increase above the normal, the solenoid will pull back the core, throwing resistance into the armature circuit. It is thus seen that the solenoid performs two functions: first, that of cutting out the starting resistance; and second, that of a safety device. In stopping, the lever f is brought back into the original position by means of the cam on the lever l, making the arrangement ready for starting again.

56. The diagram of connections given in Fig. 25 will be readily understood. It is to be noticed that the series windings of the field are cut out after all starting resistance is cut out. A safety wire s connects the end of the solenoid with the first resistance contact. This wire will keep the



Safety Brake Magnet
Potential Switch
E
D_1
F_1
D_2
F_2
F_3
D_3
+
Snap Switch.
Rev. Drum.
Solenoid
Start. Res
s
Arm
Shunt Field.

FIG. 25.

circuit closed even if, for some reason, the contact brush of the solenoid lever should fail to provide sufficient contact and thus stop the motor. The resistance coils are placed in a compartment of the controller box back of the mechanism shown in Fig. 24.

57. Brakes.—The brakes on the Otis elevators are of the band type. In the simpler forms, a steel band faced with leather encircles the pulley and is so connected to a weighted lever that the weight applies the brake. The lever is linked to the controller rod in such a manner that when the shipper sheave is turned either to the right or to the left the brake is released.

58. For high-speed service elevators, such as are shown in Fig. 22, a different kind of brake is used, for the reason that in such elevators the car must be stopped almost instantly without any possible slipping when the limits of travel are reached; while at any floor stop, midways of the travel, such instant stoppage is not so essential. The brake is, therefore, so arranged that it will be set in action by the limit stop much quicker and more effectively than by the ordinary device. The arrangement is shown in detail in Fig. 26.

On a stand A is a bearing a in which a short shaft s can revolve. To this shaft is keyed a crank-arm C, which in turn is connected by a rod R to the yoke of the limit-stop device L, Fig. 22. On the shaft s there is also keyed an eccentric E carrying another eccentric E'; the strap D encircling this outer eccentric is connected by a spring-cushioned rod d to the brake lever, and to it is also fastened the shipper sheave S, so that the latter, with the outer eccentric, turns upon the inner eccentric as a pivot. The outer eccentric has an arm C', Fig. 26, connected to the controller crank by a rod R', Fig. 22. To stop the car at intermediate landings, the brake is applied by turning the shipper sheave into the position shown in the figure, the outer eccentric pressing down on the brake lever. When, however, the limit stop is set in action, the inner eccentric is turned, which, having a greater throw than the outer one, gives more pressure to the brake.

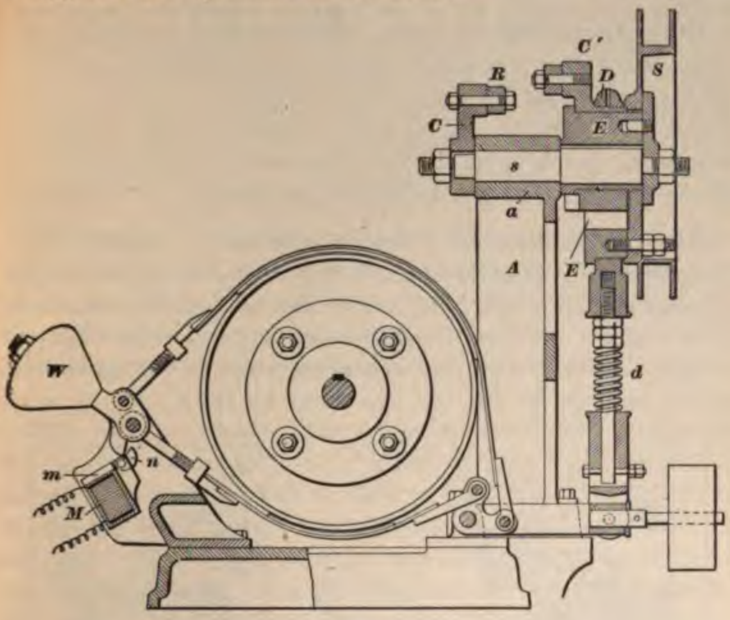


FIG. 26.

59. Another feature of the brake shown in Fig. 26 is the **safety magnet** *M*. This magnet serves to automatically apply the brake if the current should for any reason be interrupted in the system, and is placed in shunt with the motor, together with a so-called potential switch (of which we shall speak later), as shown in Fig. 25 in dotted lines. The armature *m* of this magnet has a projection, or nose, *n* which normally, that is, when a current of sufficient magnitude circulates through the magnet winding, holds in suspense a weight *W* connected to the free end of the brake band, as shown in Fig. 26. As soon as the current falls below the normal, the weight *W* trips the armature *m* and tightens the brake band. After the trouble causing this safety arrangement to act has been remedied, the weight is replaced into the position shown in Fig. 26

by operating the brake in the regular way. Dynamic braking is also resorted to.

60. Operating Devices.—For standard passenger and freight elevators, the simple hand rope is generally used; for high-speed elevators, hand-wheel devices or levers are preferred. To prevent accidental reversal of the motor in stopping, the tripping device (the lever l and the plate v) shown in Fig. 24 has considerable lost motion, or backlash.

61. Motor Safeties.—Besides the safety magnet brake above described, the usual limit-stop arrangement, consisting of yoke and traveling nut, and a clutch operating the slack-cable safety, the Otis Company generally installs with the magnet brake a so-called **potential switch.** This switch, shown in detail in Fig. 27, has three blades F_1, F_2, F_3, with

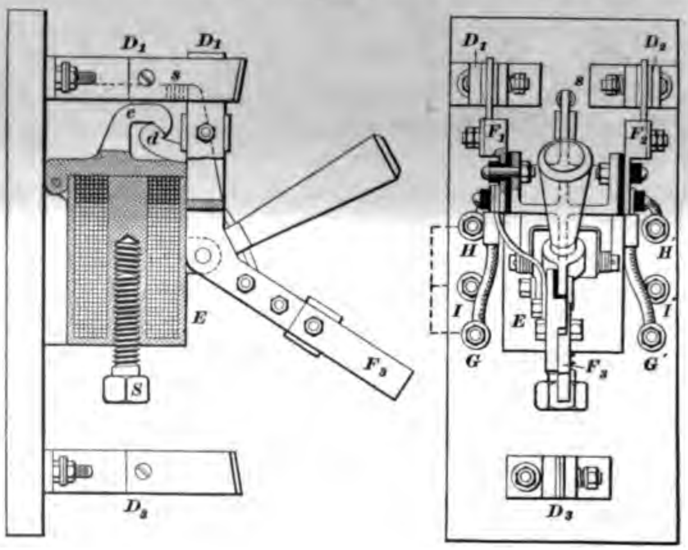


FIG. 27.

three corresponding double clips D_1, D_2, D_3, of which the first two are connected, as shown in Fig. 25, to the line wires, and the third D_3 to a wire leading to about the middle of the starting resistance. Blades F_1 and F_2 are connected to the motor circuit as shown, and F_3 to F_1. An electromagnet E placed in the shunt across the line in series with the safety-brake magnet holds the blades F_1 and F_2 in contact with the clips D_1, D_2 by means of a catch c on the armature of the magnet engaging a projection d on the fulcrumed lever carrying the blades. A spring s counteracts the magnet and causes the blades F_1, F_2 to leave clips D_1, D_2 and the blade F_3 to engage the clip D_3 when the current in the magnet windings falls below the normal. This has the effect of breaking the main circuit, releasing the safety brake, and thereby short-circuiting the armature through more or less of the starting resistance, according to the position of the resistance arm at the time. This short-circuiting acts as a brake on the motor, as is well known.

62. The usefulness of the potential switch extends beyond the use just explained. In Fig. 28, a method of connecting up the potential switch is shown, by which the potential switch not only performs its function in case of a fall of electric potential, but also in case of an undue increase of current in the line. For this purpose the switch magnet E, Fig. 27, has two windings with opposite magnetizing effect. One winding (the one next to the armature of the magnet) terminates in the binding posts H, H', while the other terminates in binding posts I, I'. These posts are, respectively, connected so as to throw the magnet winding $H\,H'$ in series with the armature of the motor, and the coil $I\,I'$ in series with the safety magnet brake, as in the previous case.

The coils of the electromagnet are so proportioned that under normal conditions the shunt coil $I\,I'$ gives a stronger magnetic field than the series coil $H\,H'$, and since they are wound in opposition to each other, the shunt coil will thus normally hold the switch closed. But if the potential in the line falls below the normal, the switch will be opened, the magnet not holding against the spring. Again, if the current in the armature circuit rises above the nórmal, the series coil of the magnet will produce a stronger field than normally, with the effect of weakening the field produced by the shunt coil, so that eventually the magnet will be demagnetized enough to let go of the switch lever. It is thus seen that the switch operates not only under a fall of potential

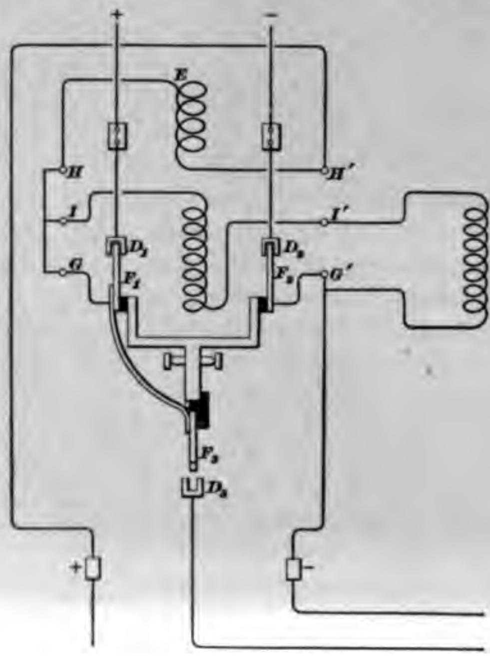


FIG. 28.

but also under an excess of current. The screw S shown in Fig. 27 serves to regulate the shunt field by screwing it in or out, decreasing or increasing, respectively, the resistance of the magnetic circuit of E.

ELEVATORS OPERATED BY ALTERNATING CURRENT.

63. While direct current is preferable for the operation of electric elevators, in many cases alternating current is the only source of power that is available. Two-phase or three-phase alternating current is generally used for elevator operation. Prior to the introduction of the two-phase and three-phase systems, alternating current was very little used for motive purposes because the single-phase alternating current motor would not start of its own accord under load; on the other hand, two-phase and three-phase motors give a good starting torque and will run up to speed in much the same way as a direct-current motor. An alternating-current induction motor consists of two main parts: the primary, or stator, which is the stationary part, and the secondary, or rotor, which is the revolving part.

The primary consists of a laminated body provided around its inner circumference with slots in which the primary coils are placed. These coils are connected together, and the terminals connect to the line when the motor is in operation. The secondary, or rotor, is also a laminated body provided with slots around its circumference in much the same way as a direct-current armature. In many induction motors, each of these slots contains a heavy copper bar, which is connected to a copper ring at each end of the armature, thus forming what is known as a squirrel-cage winding. In other types of machines, especially those that must give a good starting effort and are started and stopped frequently, the armature is provided with a three-phase winding and the three terminals brought out to collector rings mounted on the armature shaft. This is done so that resistance may be inserted in series with the armature windings when the motor is being started, and thus allow a good starting effort to be obtained without an excessive rush of current. In some cases resistance is inserted in series with the field, or stator, at starting instead of in series with the armature. This avoids the use of collector rings, but it does not give as good a starting effort for a given current as when the resistance is used in series with the armature. The student should note particularly that in the alternating-current induction motor no current is led into the armature from the line; in fact, there is no connection between the armature and the line. The armature currents are set up by the inductive action of the constantly shifting magnetic field that is set up by the two-phase or three-phase currents in the stationary field winding. This point should be borne in mind, as it will aid in understanding the connections to be described later.

OTIS ELECTRIC ELEVATOR WITH ALTERNATING-CURRENT MOTOR.

64. General Description.—Fig. 29 shows an Otis elevator operated by a three-phase induction motor *A*. This motor is of the type manufactured by the General Electric Company and is arranged so that a resistance is inserted in series with the armature windings at starting. In order to allow the insertion of this resistance, the armature is provided with three collector rings, shown at *b*, contact being made with the rings by means of carbon brushes. The motor operates the drum by means of a worm-gear, as already described in connection with other elevators. The starting, stopping, and reversing are controlled by a shipper sheave *S* operated from the car. When the shipper sheave is moved in either direction, a cam moves the rod *r* back and forth. In the figure the shipper sheave is in the neutral position. *R* is the reversing switch that connects the motor to the line and controls the direction of rotation of the motor. This switch is operated by the cam *c*. The shaft *s* of the switch carries a number of arms, which engage with suitable contacts when the switch is moved to either the up or down position. The cam *c* has three prongs that engage with prongs on a segmental gear *G*, and when the car reaches the limit of its travel in either direction, the traveling nut on the drum shaft causes *G* to open the circuit and stop the motion of the car. In the position shown in the figure, switch *R* is open; when thrown to the right, it makes connections for the car to go up, and when thrown to the left, it reverses the motor. Enough backlash is given between the prongs of the cam *c* and the lugs on the wheel *G* to insure safety against overthrowing the switch. When switch *R* is operated, the motor starts up with all the resistance in the armature circuit and means must be provided for cutting out this resistance as the motor comes up to speed. This is accomplished by the controller shown at O.

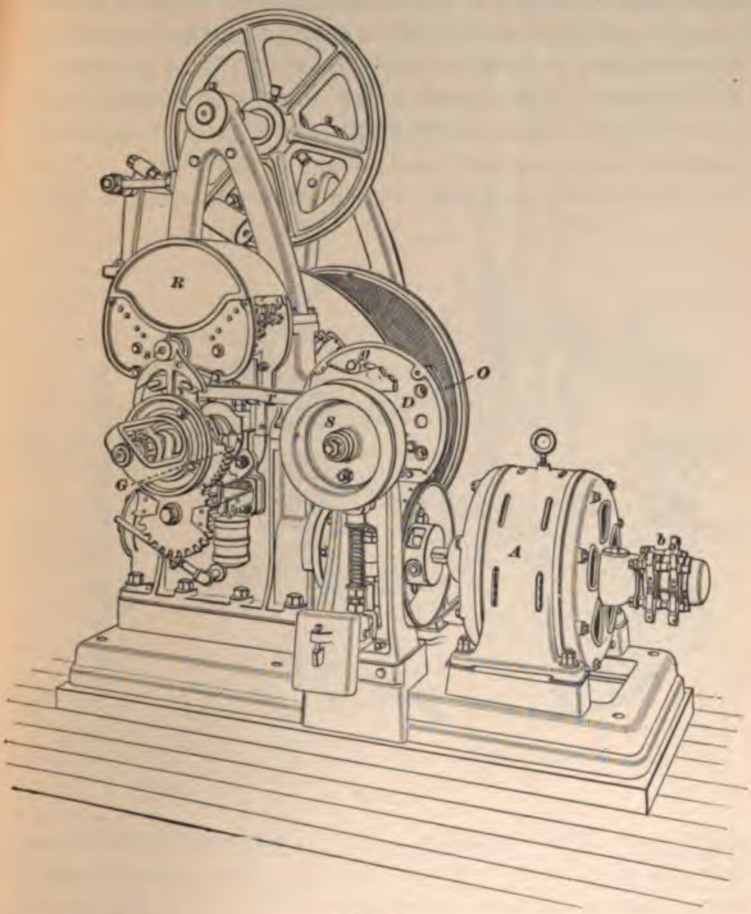


FIG. 29.

65. The Controller.—The controller is operated by means of the rod r, which raises the roller g whenever r is moved. The roller g is mounted on the end of a lever, as indicated in Fig. 30 (b). Fig. 30 (a) shows a rear view of the controller. D is the supporting cast-iron plate that carries the slate pieces S, on which are mounted a number of contacts l_1, l_1, l_2, l_2, etc. The hinged fingers f_1, f_1, f_2, f_2, etc. also carry contact pieces, and in the position shown in the figure, the fingers are in connection with their respective contacts mounted on S. When they are in this position, all the resistance is cut out and the motor runs at full speed, as will be shown later. As soon as r, Fig. 29, is moved, roller g is raised and casting E, Fig. 30 (a), is forced down, thus compressing the spring F and raising all the fingers. At the same time, the reversing switch is closed and the motor

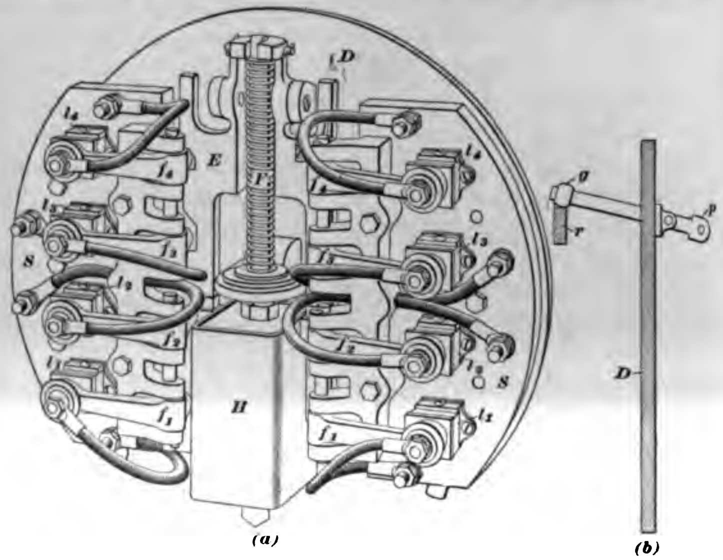


FIG. 30.

starts up with the resistance in. When roller g rides over the cam on r, the spring F forces up E, the upward motion being gradual because of the dashpot H. Casting E is provided with a number of cams, or notches, so placed that as E rises, the fingers f are closed down in pairs; i. e., the two lowest fingers first make connection with their contacts, then the next pair, and so on until all the contacts are closed, as shown in the figure. The closing of each of the pairs cuts out a section of resistance in each of two of the motor windings.

66. Connections and Operation.—The operation of the reversing switch and controller will be understood by referring to Fig. 31, which gives the electrical connections. *R* is the reversing switch and *M* the main switch, which is operated by hand and is only used when the motor is to be cut off entirely from the line. Switch *R* is provided with six clips *4*, *5*, *6*, *4'*, *5'*, *6'*, which engage with the blades or

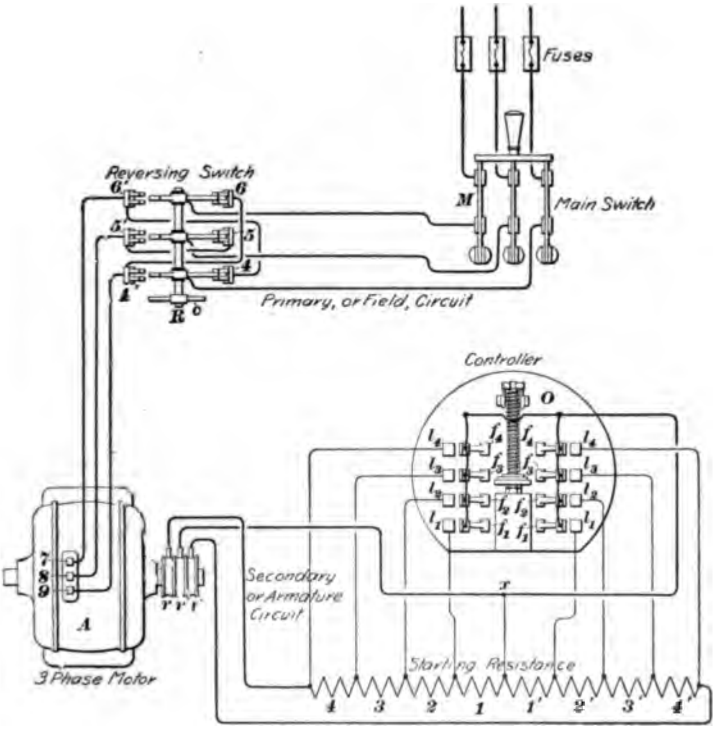


FIG. 31.

contact arms mounted on the shaft of the switch when the shaft is rocked by means of the cam *c*. In the position shown, the switch arms engage the right-hand clips and connection with the left-hand row is broken. The field terminals of the motor are *7*, *8*, *9*; and it is easily seen that when *R* is thrown over, the connections of *7* and *9* to the line are interchanged, thus reversing the motor. There is no resistance in this primary circuit, and the secondary or armature circuit in which the controller O is placed is entirely separate from the primary. In the position shown, all the fingers f_1, f_2, etc. are raised off the contacts l_1, l_2, this being the position they occupy at the moment of starting. The induced armature current in flowing from ring r' to r must take the path r'-x-*1*-*2*-*3*-*4*-r, thus passing through four sections of resistance. Also, in flowing from r' to r'' it must pass through the four resistance sections *1'*, *2'*, *3'*, *4'*. The insertion of this resistance in the armature windings keeps down the rush of current through the primary and results in a good starting effort. As the casting E, Fig. 30 (a), rises, fingers f_1 and contacts l_1 make connection, thus short-circuiting sections *1* and *1'* of the resistance. As E rises still farther, sections *2* and *2'* are cut out by f_2 and l_2 making contact, and so on, until all the fingers are down and all the resistance cut out. In passing from ring r' to r, the current now takes the path r'-f_4-l_4-r and there is no resistance in circuit. When, therefore, the fingers are all down, rings r, r', and r'' are connected together and the induced armature currents are provided with a closed circuit in which there is no resistance other than that of the copper armature conductors and the connecting wires.

67. The number of steps of resistance depends on the service to which the elevator is to be put. For example, some controllers are provided with only three sets of contact fingers, as it is found that three sections of resistance are sufficient to give a smooth start. The connections for a two-phase motor are practically the same as those shown, so that it is not necessary to describe them in detail.

ELECTRIC ELEVATORS WITH MAGNET CONTROL.

68. General Features of Magnet Control.—In most of the controlling devices so far described for electric elevators, the cutting out of the starting resistance is accomplished by means of an arm carrying a contact that slides over a series of plates, or contacts, connected to the sections of the resistance. This method works very well if the contact brush and contact plates are kept in good condition, but if either of them become rough or burned, the starting rheostat rapidly gets into very bad shape on account of the poor contact and consequent burning action. This is especially the case if the motor requires a large current for its operation, because the larger the current, the more perfect must be the connections made by the rheostat contacts, and a contact that is at all defective will very soon give rise to burning and cutting.

69. In order to avoid the use of a sliding contact with its accompanying contact plates, the so-called magnet system of control has been devised, in which the resistance is cut out by a series of electromagnetic switches, each one of which operates independently and which is so designed that it will handle a large current with very little burning or arcing. As these switches are simply of the make-and-break variety and have no sliding contacts, any small amount of burning that may take place does not interfere with the operation of the controlling outfit. There are many ways in which the system of magnet control may be applied. The electromagnetic switches may be arranged to operate automatically as the motor increases in speed; they may be controlled entirely by a controlling switch on the car, or part of them may be controlled automatically and part from the car. These resistance-controlling switches, together with the other electromagnetic switches necessary for closing the main circuit and reversing the armature connections, are mounted on a switchboard, which is usually separate from the elevator motor and hoisting mechanism.

70. Elementary System of Magnet Control.—Before taking up an elevator with magnet control, we shall consider the elementary arrangement shown in Fig. 32. This diagram is intended merely to illustrate the principle and does not represent any special controller. It shows an ordinary shunt motor M with its starting resistance R controlled by the two magnets S, S'. The starting and reversing switch is shown at A, and in this case it is supposed to be operated by hand. Of course, if the motor were used in connection with an elevator, switch A could be operated from the shipper sheave. When the switch is in the position shown, the motor runs in one direction, and when it is thrown over so that the blades occupy the position shown by the dotted lines, the motor is reversed. The starting resistance is divided into two sections a, b, which are successively

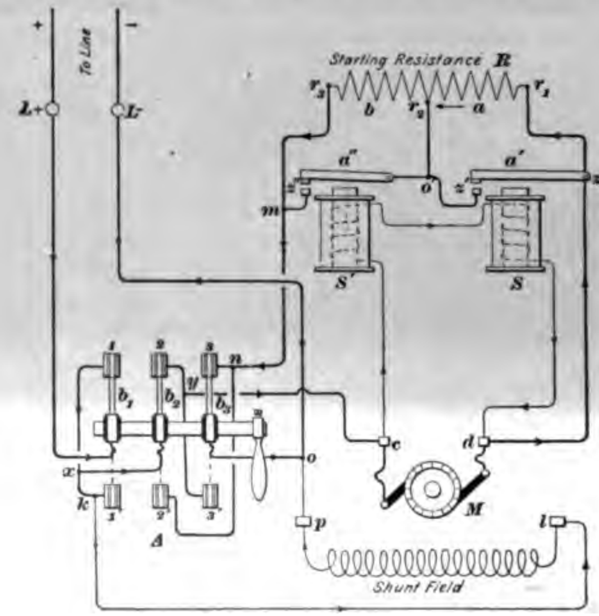


FIG. 32.

short-circuited by the electromagnetic switches S, S' when the motor comes up to speed. The windings of S, S' are connected in series across the armature terminals, forming a shunt circuit to the armature. When the main switch is closed, all the resistance is in series and the pressure across the armature terminals and coils S, S' is very small; consequently, very little current flows through S, S'. However, as the motor speeds up, its E. M. F. increases and the pressure across the brushes increases, and this increases the current through S, S'. The armatures of these switches are so adjusted that S will operate with a smaller current than S'; consequently, as M comes up to speed, S closes and cuts out section a of the resistance by short-circuiting it. As the speed increases still further, the current through S and S' becomes strong enough to operate S', and section b is short-circuited, thus connecting the motor armature directly to the line. Suppose that the motor is to be started and that switches A, S, and S' are in the positions shown. The path of the current through the shunt field is as follows: $L+-b_1-1-k-l-$ through shunt field$-p-o-L$. The path of the main current through the armature and starting resistance is $L+-b_1-1-x-b_2-2-y-c-$ through armature of motor$-d-z-r_1-r_3-3-b_3-o-L$. When the current through the shunt-magnet circuit $C-S'-S-d$ has become strong enough to pull down armature a', contact is made at z' and the main current on reaching z takes the path $z-a'-z'-o'-r_2-r_3-3-b_3-o-L$, thus flowing past section a of the resistance that is short-circuited. When S' operates, the current takes the path $z-a'-z'-o'-a''-z''$, and so on, the whole of the resistance being thus short-circuited. Any arcing, or burning, that may occur will take place at contacts z' and z'', and this can easily be taken care of by providing suitable contacts. Moreover, it will be noticed that the closing of an armature short-circuits the resistance, and that when an armature opens, the circuit is not broken, because the current still has the alternative path through the resistance. The result is that when the armature leaves its contacts there is but little sparking.

71. When the motor is to be run in the reverse direction, switch A is thrown over to the position indicated by the dotted lines. This does not change the direction of the current through the shunt field, but it reverses the current through the armature, the path being as follows: $L+-b_1-1'-x-b_2-2'-n-r_3-r_1-d-c-y-3'-b_3-o-L$. Since the current through the armature is reversed while that in the field remains the same, the direction of motion is reversed.

The scheme of using electromagnetic switches to control the starting resistance has been embodied in the controllers of a number of different manufacturers. It has been found that it is not necessary to provide a great many resistance sections and resistance-controlling switches in order to give a smooth start. The actual number needed depends, of course, on the conditions under which the motor is operated. With an ordinary sliding-contact rheostat, it is necessary to provide quite a large number of resistance sections, in order to keep the voltage between adjacent contact plates down to the small amount necessary to avoid sparking when the arm slides from plate to plate. With electromagnetic switches the number of sections can be much smaller, because this precaution is not necessary. Moreover, when the cutting out of resistance is controlled by switches that are in turn controlled by the counter E. M. F. of the motor, the resistance is never cut out until the armature has come up to such a speed that it is able to take care of the increased current. The resistance is, therefore, cut out just when the armature is ready for it and not before; such being the case, fewer resistance sections are necessary than if the cutting out were controlled by hand.

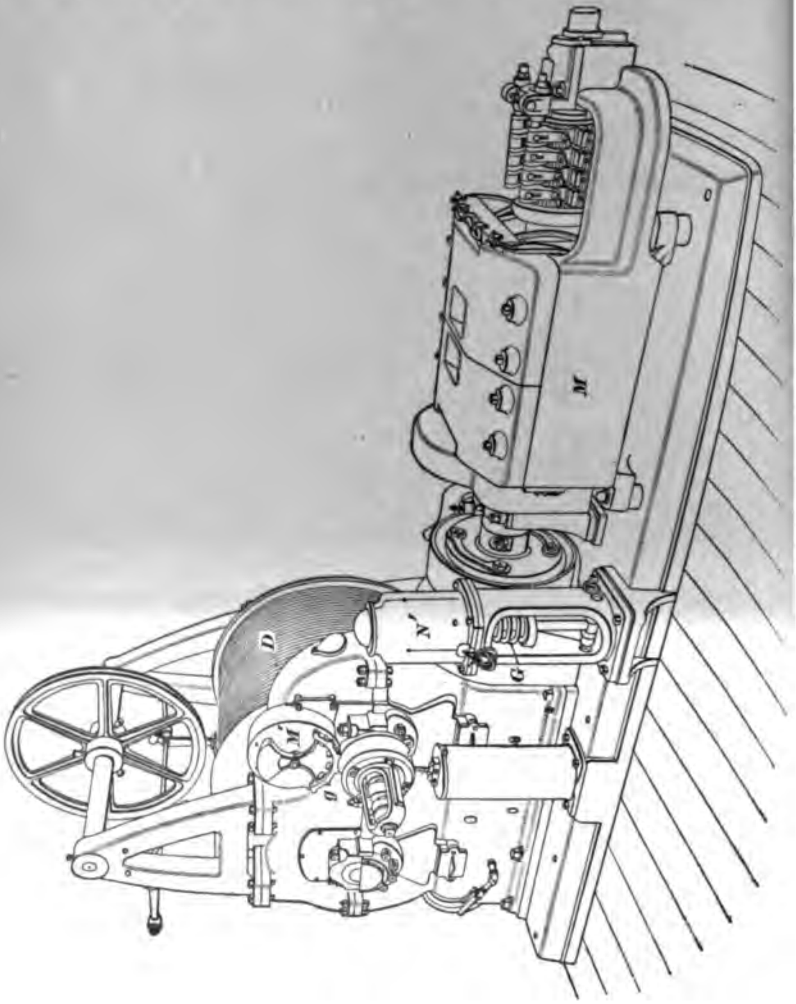
72. With most high-speed passenger elevators using this method, the switches that perform the same duties as *A*, Fig. 32, are operated by electromagnets or solenoids, thus doing away with the shipper sheave with its cable, cams, and other switch-operating devices and replacing them by an electric cable connecting the car-operating switch to the switchboard.

73. The car-operating switch replaces the ordinary operating wheel or lever used for operating by means of a cable. The cable running from the operating switch to the switchboard carries the wires that connect to the electromagnetic switches, and as these switches require only about ¾ ampere for their operation, the wires in the controlling cable do not need to be large. This method of control is being used quite largely for various kinds of service, and, as pointed out above, it has advantages over the older sliding-arm method of controlling resistance. In order to illustrate its application in practice, we will describe two controllers made by the Otis Elevator Company and covered by patents owned by them.

OTIS ELEVATOR WITH G. S. MAGNET CONTROLLER.

74. General Description of Elevator Machine.—Fig. 33 shows a direct-connected Otis electric elevator for use with magnet control. The motor *M* operates the drum *D* by means of double worm-gears. This particular machine is provided with back gearing between the motor shaft and worm-shaft, so that unusually heavy weights, such as safes, may be lifted. It will be noticed that there is no electric controller connected to the machine other than the brake magnet *N'* and the stop-motion switch *M'*. The brake magnet is a powerful solenoid that operates against the spring *G*, so that when the magnet is energized the band brake is released, and when current ceases to flow through the magnet, the brake at once goes on. The stop-motion switch *M'* will be described more in detail when the electrical connections are taken up. Its function is to cut off the current and stop the motor whenever the car approaches the limit of its travel in either direction. Under ordinary running conditions, the intermittent gear *g* remains in the central position shown in the figure. When the car approaches the limit of its travel, the safety nut on the shaft of the worm-gear causes a pin to engage with *g*, thus making it swing over. This operates a switch arm inside the casing *M'*, which breaks electrical connections and slows down the motor. When the safety nut makes another revolution, *g* is swung over another notch and the motor is stopped completely. The mechanical features of the hoisting machine are similar to those that have already been described and do not call for special attention.

75. General Description of Otis G. S. Magnet Controller.—Fig. 34 is a general view of the Otis G. S. magnet controller. The controlling devices are mounted on a heavy



slate panel *A*, which is in this case supported on an iron framework *B* that also serves to house the resistance coils. With many controllers, the resistance is placed in a case arranged behind the switchboard. The various electromagnetic switches necessary for controlling the direction of motion of the car and the cutting out of the starting resistance are mounted on A.

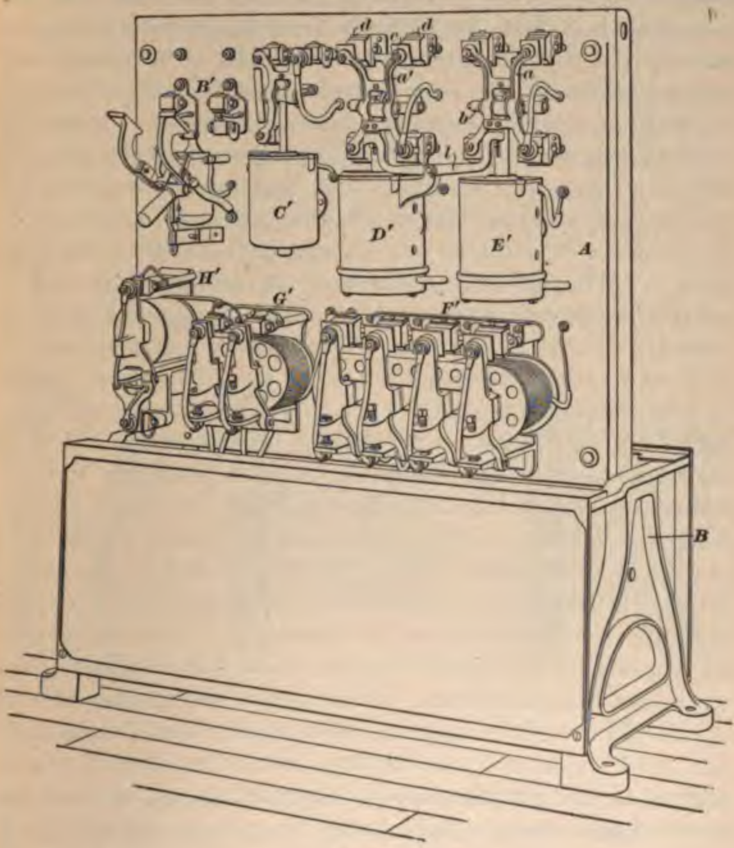


FIG. 34.

In Fig. 34, B' is the potential switch, the use of which has already been explained. It is a protective device and is not concerned with the regular starting and stopping of the elevator. When the elevator is in operation it remains closed. Switches C', D', and E' control the main current.

Switch H' controls the brake and the two groups of switches G' and F' control the resistance. The group of four switches F' controls the starting resistance and the pair of switches G' controls the stopping resistance. With these controllers the motor is stopped by allowing it to act as a generator, thus providing a dynamic-braking action in addition to that of the band brake. In order to allow a smooth braking action, the current generated by the motor is passed through a resistance, and this resistance is cut out or in by magnets G'. The main operating magnets C', D', and E' are of the solenoid type, and when they are not excited the plungers are down and the upper switch contacts, as c, for example, are separated from the fixed contacts d. The movable contacts c are mounted on rocker-arms a, a' pivoted as shown at b. The plungers of the two switches D' and E' are connected by a lever l, as shown, so that when one contact lever a is up, i. e., the upper terminals in contact, the other lever a' is down, and it is impossible for both levers to occupy the up or down position at the same time. The operation of these switches will be understood more clearly by referring to Fig. 35 (a). Switches F' and G' are arranged as shown in (b) and switch H' is as shown in (c). These sketches are intended merely to indicate the operation of the switches, so that the diagram of connections to be given later may be readily understood; hence, particular attention has not been paid to the mechanical details. In (a), when the magnet draws up the plunger, lever a is moved so that c and d make contact, and contacts c' and d' are, of course, opened. Contacts d, d' are graphite blocks mounted on spring holders, the object of the graphite being to prevent damage from burning or sparking, and especially to obviate the danger of the contacts fusing, or sticking, together as might possibly occur if both were of copper. Fig. 35 (b) shows the construction of the resistance-controlling switches; f is the magnetizing coil, which is made to serve for the whole group of switches by embracing the series of cores h, as indicated at F', Fig. 34. G is the magnet casting that carries the series of cores h, opposite each of which is hinged the armature a carrying an insulated contact c, which makes contact with d when the armature is drawn down. When current flows around coil f, all the cores are magnetized to about the same degree, but the armatures are not all attracted because they are adjusted to different distances from the pole pieces s by means of adjusting screws p that rest against lugs r. The armature with the shortest air gap between a and s is first attracted, then the next, and so on, the armatures closing in succession as the magnet increases in strength on account of the motor speeding up. The resistance is thus automatically cut out by steps, as explained in connection with Fig. 32.

Fig. 35 (c) shows the switch indicated by H' in Fig. 34. It is practically the same as (b), except that it is provided

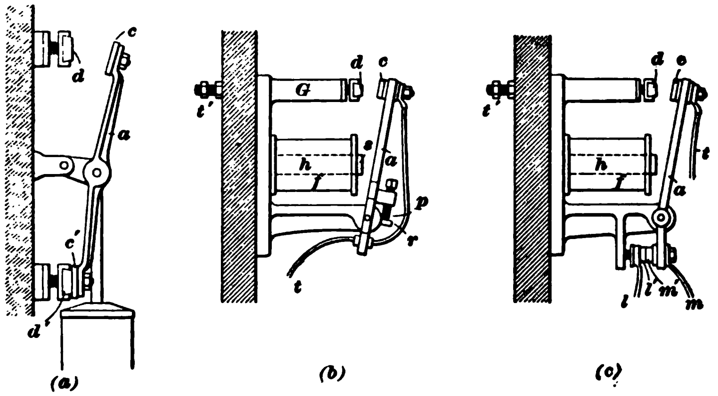


FIG. 35.

with two insulated back contacts l', m' to which the leads l m are connected. When armature a is unattracted, l' and m' are in contact; when a is attracted, contact between l' and m' is broken and contact between c and d is closed. The switch contacts that are most liable to arcing are provided with magnetic blow-out coils. These are coils provided with an iron core so placed that a magnetic field is set up between the contacts, and as soon as the arc forms, it is forced across the field and broken almost instantaneously.

76. Car-Operating Switch.—Fig. 36 shows the style of car-operating switch used with the magnet controller. When the motor is stopped, the handle occupies the vertical position and is thrown to the left or right, according as the car is to go up or down. When the cover is closed and the switch in use, sliding contacts *c*, *c* bear against the arcs *a*, *a*, *b*, *b*; when the switch is off, they bear against the insulating pieces *d*, *d*. The contacts on the back of the operating lever press against segments *e*, thus making the required connection. By adopting the construction shown, no current flows through the hinge *f*. The exact arrangement of the contact segments varies with different controllers, as the starting and running requirements are not always the same for different installations. The operating switch for which the connections are shown in Fig. 38 is somewhat simpler than that shown in Fig. 36, and requires fewer wires and contact arcs, but its general construction is the same.

FIG. 36.

77. Stop-Motion Switch. — Fig. 37 shows two views of the stop-motion switch shown at *M'* Fig. 33. The use of this switch has already been explained, but Fig. 37 shows the construction and will aid in understanding the electrical connections. The arm *a*, which is operated by the intermittent segmental gear *g*, normally occupies the horizontal position, but is swung around whenever the car reaches the limit of its travel. Contact brushes are mounted on the arm, and these rub on the contact arcs *b*, *b*. When the arm is swung

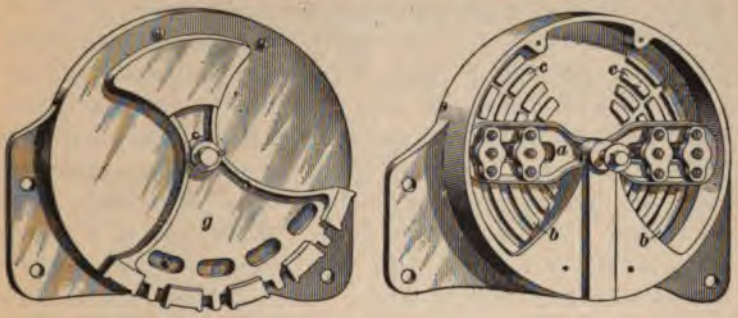


FIG. 37.

around in either direction, one set of brushes leaves the long contacts *b*, *b* and passes on to the short pieces *c*, *c*. Arcs *c*, *c* are, with the controller to be described, not connected to anything, but serve as bearing pieces; as soon, therefore, as the contact arm slides on to them, electrical connections are broken, which causes the motor to stop.

78. Connections for G. S. Controller.—Fig. 38 shows the general scheme of connections for the G. S. controller. In order to simplify the diagram, the relative positions of a few of the parts have been changed; for example, the starting resistance and the extra-field resistance $r\,r'$ are shown connected directly to their switch contacts. The relation of the various switches is the same as shown in Fig. 34, except that their order is reversed, because all the connections are made on the back of the board; corresponding switches in Figs. 34 and 38 are lettered alike. In order to facilitate the tracing out of connections, all *fixed* contact pieces on the switches have been shaded, while all *movable* pieces have been left open. For example, on switch E' the shaded contact pieces $5'$, $6'$, $7'$, and $8'$ are mounted on the slate panel and the open contacts *5*, *6*, *7*, and *8* are mounted on the tilting arm shown in Fig. 35 (*a*). Also, contacts that touch each other are marked with similar figures, i. e., when switch *E'* is pushed up, *5* makes contact with *5'* and *6* with *6'*. A few details, such as blow-out coils, have been omitted, as they are not necessary to illustrate the operation of the controller. The car-operating switch controls switches *E'*, *D'*, *C*, and *H'*; switches *M'*, *F'*, and *G'* operate automatically. The main switches *E'* and *D'* are each provided with two coils. One of these coils is of fine wire, and the current in it is controlled by the car-operating switch. The lower coils are of coarse wire, and carry the main motor current; these coils are arranged below the fine-wire coils and, when energized, hold the switch down. When switch *C'* operates, *13* makes contact with *13'*, *14* with *14'*, and contact between *15* and *15'* is broken. When *H'* operates, contact is made between *16* and *16'* and broken between *17* and *17'*. When switches *G'* and *F'* operate, contact is made between *1* and *1'*, *2* and *2'*, etc. The movable contact pieces of the stop-motion switch *M'* occupy the horizontal position shown until the car reaches the limit of its travel in either direction. The full-black segments on this switch are not connected to anything, being in this case bearing surfaces only. Switch *P'* is provided with a pair of contacts on each side of the "off" position. Two contacts *u'* and *d'* are longer than the others *f u* and *f d*, so that the lever makes contact with the former before the latter. To avoid confusion, a wire is shown connected to the lever instead of carrying the current to it through sliding contacts, as is done on the switch shown in Fig. 36. An additional safety switch *S* is sometimes provided to stop the elevator in emergencies, but it is not in use under normal conditions.

79. Type of Motor Used With G. S. Controller. Before taking up the action of the controller, it will be well to consider briefly the type of motor used with this system of control. In order to get the elevator under way quickly, it is necessary that the motor should give a strong starting

torque. This is provided for by the series field. The shunt field furnishes the excitation after the motor has attained its speed. In addition to these two windings, a third, or extra-field, winding is provided. This winding aids in providing a field when the motor is being brought to a stop, by allowing it to act as a dynamo; it also aids to some extent in providing a strong field at starting. It should be remembered that a shunt-wound motor will run as a generator if it is disconnected from the mains when up to speed and a path provided between the brushes for it to send a current through; it is not necessary to reverse either field or armature connections in order to make it generate, as is the case with a series motor.

80. Operation of Controller on First Point.—Suppose that the car is to be run up and that the lever of *P'* is moved to the left until the arm comes in contact with the long arc *u'*, but does not touch the contact *f u*. The operating current then flows as follows, starting from point *18* on the + side of the potential switch: *18*, through coil of switch *H'*, through coil of the "up" magnet *D'*, through wire *u u* to stop-motion switch *M'*, to contact strip *u'*, by way of the horizontal strip, through flexible car-operating cable to contact *u'* on car-operating switch, through lever to *w*, thence through safety switch and wire *y y y* to the negative side of the potential switch. This current operates switches *D'* and *H'*. Switch *D'* is drawn up by the fine-wire coil and contact is made between *9*, *9'* and *10*, *10'*, as indicated by the dotted lines. This allows current to flow through the shunt field by way of the path +–*18*–*9'*–*9*–*D*–through shunt field–*11*–*19*–*20*. The operation of *H'* releases the brake by connecting points *16* and *16'*, because point *D* connects through *D'* to the + side of the circuit, and when *16* and *16'* are in contact, the other terminal of the brake magnet connects to the negative side of the circuit. This releases the brake and allows the motor to start as soon as current flows through the armature. When switch *H'* operates, points *17*, *17'* are separated so that no current can flow through coil *k*, and, hence, switches *G'* are open so long as the motor is working. As soon as switch *D'* is forced up, switch *E'* makes contact between *7*, *7'* and *8*, *8'*, because of the connecting lever *1*, Fig. 6. The main current then takes the path indicated by the arrowheads as follows: *18–9'*, *10'–9*, *10–8–8'–I*–through armature of motor to *E*–through series coil of switch *E'–7'–7–14–H'–G'–3'–2'–4'*–through whole of starting resistance to *F*–through whole of series field to *H–19–20* to negative side of the circuit. The current through the series coil of *E'* holds the switch arm down firmly. The motor, therefore, starts up with the two sections of the series field and all the starting resistance in series with the armature. The extra field is in series with the resistance *r r'*, and the two together are in shunt with the armature. This is easily seen by tracing the path through the series field, beginning at point *D*, as follows: *D*–through extra field–*K'–4–r'–r–15'–15–14*–etc.

Fig. 39 represents, diagrammatically, the connections that are made on the first position of the switch *P'*. The

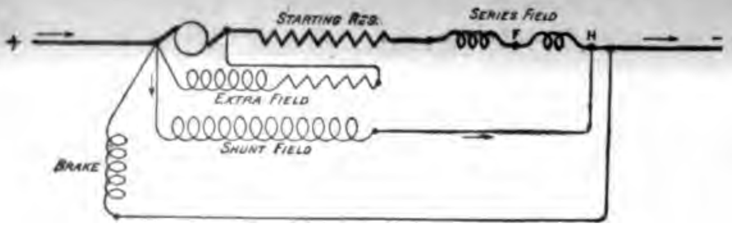


FIG. 39.

extra field does not supply nearly as much magnetizing power at starting as when the motor is stopping, because, at starting, the pressure across the armature terminals is small. On this point of the controller, therefore, the motor starts up, but would run the elevator at a slow speed because of the resistance in circuit. It should be noted that as long as the operating handle rests on *u'* or *d'*, the resistance switches *F'* are inoperative, because one terminal of coil *h* connects to contact *13* on switch *C'*, which is open.

81. Operation on "Fast Up" Position.—When the handle of P' is moved over farther, so as to make contact with the $f\,u$ (fast up) contact, current flows through the solenoid of switch C', thus forcing the switch lever up and making contact between *13*, *13'* and *14*, *14'*. At the same time contact is broken between *15* and *15'*, thus opening the circuit through the extra field, which is now no longer needed, as the motor is by this time well under way. The operation of this switch allows current to flow through the magnet coil h and resistance x', as indicated by the dotted arrows, beginning at point *21*. This operates the group of switches F'. First *2* and *2'* are connected, thus cutting out the first section of resistance, then *3*, *3'*, cutting out the second section, then G, G', cutting out what is left of the resistance and also one half of the series field; finally, H and H' are connected, thus cutting out the other half of the series field. The motor has now attained its maximum speed, and the path of the main current is +–*18–9'*, *10'–9*, *10–8–8'–I*–through armature–*E–7'–7–14–H'–H–19* to negative side of circuit. The motor now operates as a plain shunt machine with no resistance in circuit.

82. If the operating switch were moved in the reverse direction, switch E' would be moved up and switch D' would be down, as can be readily seen by tracing the connections. The main current then takes the path *18–6'*, *5'–6*, *5–12–12'*–through series coil of *D'–E*–through armature–*I–11'–11–14*, and so on as before. The current in the armature flows in the reverse direction to what it did before, while the current in the fields remains unchanged; the car, therefore, moves down.

83. Action of Controller on Slowing Down and Stopping.—Suppose that the elevator is running on the "fast up" point and that the handle is moved back until it leaves contact $f\,u$, but still rests on contact u'. Switch C' will be opened, and this will cause switches F to open, thus cutting the resistance and series field back into the circuit: the extra field will also be connected, because *15* and *15'* will make contact, the resistance r r' being in series with the extra field. Quite a large current will now flow through the extra field because the potential across the armature is high. When, therefore, the handle is moved back from the fast position, the field of the motor is greatly strengthened and all the resistance is cut back into circuit, thus rapidly lowering the speed of the motor. On account of the decrease in speed and the cutting in of the resistance, the pressure across the brushes is considerably decreased when the handle is moved from the fast position. When the operating handle is moved to the off position, switches D' and H' are opened. D' breaks connection with the line, and H' sets the brake by separating points *16*, *16'* and opening the circuit through the brake magnet. At the same time, points *17*, *17'* are brought into contact, thus connecting coil k across the armature terminals. The pressure across the armature terminals is large enough to cause switches *4*, *4'* and *1*, *1'* to close, thus cutting out r' r and connecting the extra field across the armature. The armature thus generates current, which takes the path *1*–*8'*–*8*–D'–extra field–K'–*4*–*4'*–*14*–*7*–*7'*–E. The current through the extra field remains in the same direction as it did when the motor was run from the line and hence assists in keeping the field magnetized and bringing the motor to a stop quicker than if the shunt field only were used. As the motor slows down, the magnetization supplied from the shunt-field coil diminishes; hence, the provision of the extra field supplied with the current that the motor furnishes when running as a generator greatly increases the braking action. The generating action soon slows the motor down, and as the pressure across the armature terminals decreases, switches *4*, *4'* and *1*, *1'* open in succession, because K is no longer able to hold them. This cuts resistances r', r back into circuit with the series field, thus making a smooth stop and leaving the resistances r, r' in series with the extra field ready for the next start. Of course, while this action is taking place, the band brake is also on because the dynamo-braking action decreases as the speed decreases, and hence would not answer, in itself, for bringing the motor to a full stop. All these actions take place in a very short space of time, but the effect is to stop the motor smoothly and quickly, and the car is at all times easily controlled by switch P', the cutting out of the resistance and the connections necessary to produce the dynamo-braking action being made automatically.

84. The stop-motion switch M' merely brings about automatically the same connections that P' should, in case the operator failed to move P' when the car reaches the limit of its travel. When the contact arm is swung around by the action of the traveling nut, as already explained, contact is first broken, if the car is ascending, between the $f\,u$ and f contact arcs, thus slowing down the motor; and as the car travels still farther, contact is broken between arcs u and u', thus applying the brake and stopping the car.

OTIS NO. 6 MAGNET CONTROLLER.

85. General Description of No. 6 Controller.—Fig. 40 shows the general arrangement of the Otis No. 6 magnet controller. This is a later type than the G. S. controller previously described, and although its mechanical details are quite different, its principle of operation is almost identical. The resistance, which is usually in the form of cast-iron grids for controllers of large capacity, is arranged behind the board and does not appear in the figure. The various switches are marked A', B', C', *1*, *2*, *3*, *4*, *5*. Switches A', B', C', and *1* are operated by the car-controlling switch; the other switches operate automatically. Whenever a switch, for example C', operates, its plunger c is drawn up, thus bringing the copper disks d, d' up against the contact fingers f, f'. When a switch is deenergized, its plunger drops and the disks make contact with the lower fingers where any are provided. When a disk is drawn up, it first makes contact with the auxiliary carbon contacts x, and as it is pulled up still farther, it bears against a copper contact on a finger hinged at the same point as the finger that carries the carbon contact. When a disk drops, it first breaks contact at the copper surfaces, and the final break

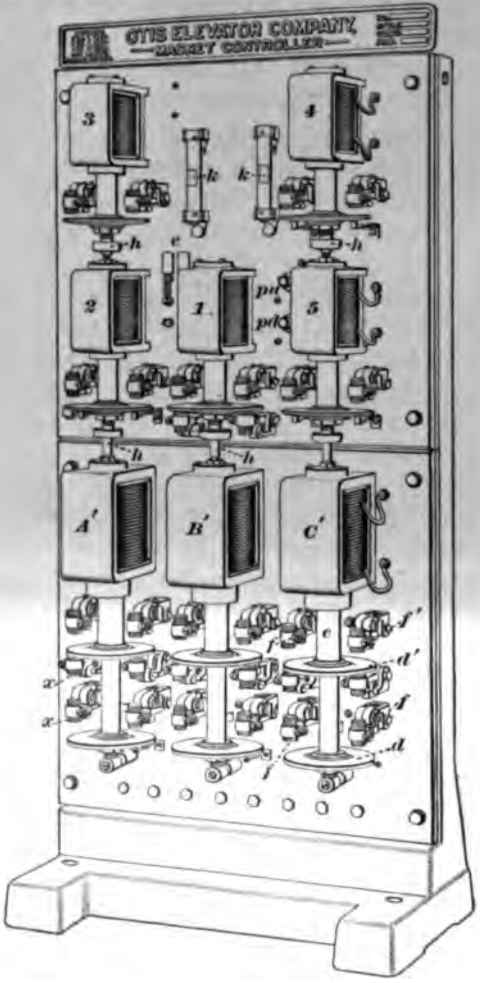


FIG. 40.

takes place between the copper and the carbon terminal that there is no danger of sticking. The carbon pieces threaded, so as to prevent their working loose and slid through the holders. The plungers with their contact plates are free to revolve, and the motion of the switch gradually works them around so that whatever burning takes place is spread around the whole disk instead of in one place only. Switches *2*, *3*, *4*, and *5* operate automatically, one after the other, and the voltage at which they operate is adjusted partly by regulating the initial position of the plunger by means of the adjustable stops *h*, and partly by inserting a resistance in series with each solenoid. The main fuses are shown at *k*, *k*; they are of the enclosed type. The small knife switches shown at *c* are used for cutting off the car-controlling switch, so that the motor cannot be started from the car. Push buttons *p u* and *p d* are used to allow the motor to be operated from the board. These devices are very useful when tests are being made to locate trouble, but under ordinary working conditions they are not in use. Switches *A′* and *B′* control the direction of motion of the car. When *A′* operates, the car descends, and when *B′* operates, it ascends. Switch *C′* closes and opens the main circuit.

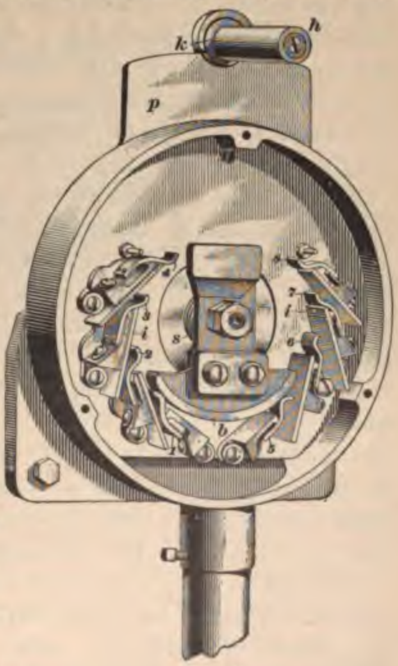
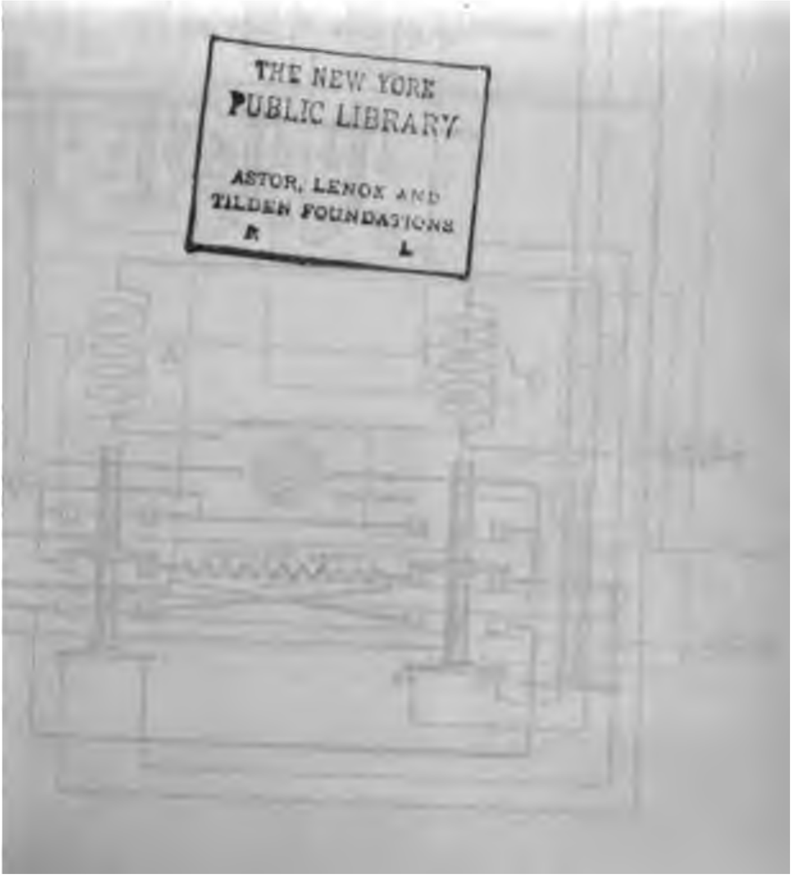


FIG. 41.

86. No. 6 Car-Controlling Switch.—Fig. 41 shows the car-controlling switch used with the No. 6 controller, the cover being removed in order to show the working parts. The operating handle is shown at *h*, and it normally occupies the central, or off, position. When moved to the left, the car ascends, and when moved to the right, it descends. The arm carries a contact arc *b* that makes contact with the fingers *1*, *2*, *3*, *4*, etc. when the handle is moved from its central position. The arc *b* is of such a length that when the handle is moved to its extreme position in either direction, it makes contact with all four fingers on the side to which it is moved. The handle *h* is held at the central position by means of a spiral spring *s*, so that if the operator releases the handle, it at once returns to the off position. When the handle is at the off position, the projecting rim *k* rests in a notch in the plate *p*, and in order to move the handle, it must first be pulled out against the action of a spring. Insulating pieces *i* are inserted between the fingers, as shown, in order to avoid short-circuiting.

87. Connections for No. 6 Magnet Controller.—Fig. 42 shows the connections of the No. 6 controller. In this diagram the positions of the switches, resistances, and motor armature and field windings have been arranged so as to make the diagram simpler and easier to follow than if the various parts were located in the same positions that they occupy on the controller. The connections are, however, the same as used on the controller shown in Fig. 40, and corresponding switches are lettered alike. The operation is on the whole very similar to that of the G. S. controller. Terminal *x* of the operating circuit is connected to the + line and terminal *y* to the − line. By throwing the small switch *c* down into the dotted position *c''*, the car cannot be operated from *P'*. Switch *m* is normally open, but it can be thrown so as to connect *t* and *d* or *t* and *u*. If *c* is thrown down to the position *c''* and *m* is thrown down so as to connect *t* and *d*, magnet *A'* is energized, and if push button *pd* is then pressed, current will flow through the coil of *C'* and the elevator will move down. If *m* be thrown up so as to connect *m* and *u* and button *pu* pressed, switches *B'* and *C'* will be operated and the car will move up. In other words, the small switches and push buttons allow the machine to be operated from the controller while the switch *P'* is cut off.



88. Operation of No. 6 Magnet Controller on Starting Position.—Assume that the operating handle is moved to the left so as to bring the arc z in contact with u. Starting from x, the path of the operating current will be as indicated by the arrows through the coil of B' to the negative terminal y. Note that this current passes through the small contacts *1* and *2* of switch A', so that unless switch A' is down, B' cannot be drawn up, and it is impossible, therefore, for both switches to be drawn up together. When z is moved still farther so as to bring it in contact with the finger $p\,u$, a current is set up through the operating circuit, which includes the solenoid of switch C'. This current may be traced as follows: x–c–c'–$p\,u$–$p\,u$ to contact $p\,u$ on stop-motion switch–p–P–through solenoid of C'–*6*–*5*–*4*–*3* to line. This current operates C', which closes the main circuit, releases the brake, and connects the shunt field and extra field across the armature. The various paths of the current are indicated by the arrows, bearing in mind that B' and C' are now up. A powerful magnetic field is provided by the series coils, and as the motor comes up to speed, switches *2*, *3*, *4*, and *5* operate, thus short-circuiting the resistance and the series field. For example, when switch *2* closes, the main current passes from terminal R_0 to R_2, thus short-circuiting the first two resistance sections. When the handle of P is advanced so that z makes contact with finger $f\,u$, switch No. *1* is operated. This breaks connection between R_0 and M, thus cutting out the extra field. The switches are now all up except A' and the motor runs with a shunt field only; the resistance is all cut out and the motor runs at its maximum speed. When C' is up and when switch *2* operates, contact is broken between the small terminals *3* and *4*, so that the current through C' has to take the path through the resistance r_p to the negative side of the circuit. This resistance is inserted to prevent undue heating of C' and also to save current. Also, when switch *5* is operated, the current through the coils of *2* and *3* is cut off, thus preventing these coils from heating and cutting off the current necessary to energize them. When the core of *2* drops, contact is established again between points *3* and *4*, but in the meantime it has been broken between *5* and *6* so that the current through C' still flows through the resistance r_p. The coils of *1*, *4*, and *5* have considerable resistance in series with them and do not overheat. It is, of course, necessary that these three should remain up while the motor is running, otherwise the extra field resistance, and series fields would not be cut out, while with switches *2* and *3* it is not necessary that they should remain up after sections *2* and *3* of the resistance have been cut out. The voltage at which switches *2*, *3*, *4*, and *5* operate is adjusted by means of the resistances r_3, r_4, r_5, switch *2* having no resistance and, therefore, operating at the lowest voltage. When P' is moved to the right, switch A' is energized and the elevator descends because the direction of the current through the armature is reversed, while that in the fields remains the same as before. The action of the stop-motion switch is the same as in connection with the G. S. controller and needs no special description.

89. Operation of No. 6 Controller on Stopping.—When P' is moved back, contact is first broken with the *fu* finger. This drops switch *1* and cuts one section of resistance into circuit as well as connecting points R_0 and M, thus cutting in the extra field. When contact is broken with finger *pu*, all the resistance is cut in and the main circuit is opened because switch C' is dropped. In fact, the switches *2*, *3*, *4*, and *5* will likely operate before contact is actually broken between *z* and *pu*, if the operating switch is not moved too quickly, because the cutting in of the first section of the resistance and the extra field will lower the speed and thus cut down the E. M. F. applied to coils *2*, *3*, *4*, and *5*. When C' drops, the brake is applied because the circuit through the brake magnet is opened. When the main circuit is opened by C', the armature is still able to send a current around the local circuit E–S_1–through stopping resistance –D–through extra field and extra-field resistance–M–R_0–R_0–R_0–I, because switch B' has not yet dropped.

This generating action through the series field, together with the brake, will soon stop the motor, even if P' is not moved to the vertical position and connection broken with finger u. The car can, therefore, be stopped without the necessity of operating the direction-controlling switch. For example, if the elevator were making an up trip, stopping at each floor, the handle would be moved far enough to break contact with $p\,u$ only, and B' would remain up during the whole trip. When P' is moved to the off position, switch B' drops and the stopping resistance is connected directly across the armature terminals.

AUTOMATIC ELECTRIC ELEVATORS.

90. General Description.—An automatic elevator is one that does not require a regular operator, but is so arranged that it can be controlled by the passenger. These elevators are largely used in private dwellings where the elevator is not used very frequently, and where it would not be desirable or convenient to have an elevator boy.

91. Automatic electric elevators are, with the exception of the controlling devices, similar to other direct-connected electric elevators. There are a number of different styles of them, but the general method of operation is about as follows: A push button is provided at each landing, and in the car there are as many push buttons as there are floors. A passenger at the third floor wishes, say, to go to the first floor. He presses the button at the third floor and the elevator comes up or down, depending on what location it may be in at the time, and when it reaches the third floor it stops automatically, at the same time unlocking the door. The passenger then gets in the car, closes the door of the elevator shaft, and presses the first-floor push button in the car. The car then descends until it reaches the first floor and stops there of its own accord. In the automatic elevator made by the Otis Company, the various devices are so arranged that when the elevator is once started by the passenger it cannot be interfered with by any other person. Also, it is not necessary that the push buttons should remain closed while the elevator is in motion. All that is necessary is to press the button for an instant and then release it. Various safety devices are also introduced; for example, it is impossible to operate the elevator if any of the doors of the shaft are open, and no person on any of the floors can possibly start the elevator if anybody at any of the other floors is getting on or off. We will describe two types of Otis elevator provided with automatic control, and these will serve to illustrate the principle of automatic control in general. The two types are practically the same with the exception of the automatic floor controller, which is geared to the elevator drum.

OTIS AUTOMATIC ELECTRIC ELEVATOR.

92. General Description.—Fig. 43 shows an Otis automatic electric hoisting machine provided with their older type of floor controller. In general appearance it will be noted that the machine is much the same as those previously described in connection with magnet control. About the only difference is the addition of the floor controller shown at *C*. A spiral contact band *a* is mounted on an insulating drum, which is moved sidewise by a coarse screw as it revolves, so that the contacts *b* always press against the band. The various contacts *b* connect to the push buttons at the different floors. The controller *C* is used to determine the direction of motion of the car when any given button is pressed, and also to stop the car when it reaches its destination; its action will be understood when the electrical connections are described. The strip *b* is arranged in spiral form on the drum simply to avoid the use of a drum of large diameter.

93. Magnet Controller for Automatic Elevator.—Fig. 44 shows the magnet controller used with the automatic elevator. *A'* is the main switch that controls the direction of rotation of the motor. A swinging armature *x* is hung as shown, between the poles of the two ironclad magnets z, z'. When it hangs in the central position, the circuit is open. When a button is pressed, one or the other of the magnets is excited, depending on the direction in which the car is to move. The armature is drawn over and contact established between the swinging terminals y attached to the arma-

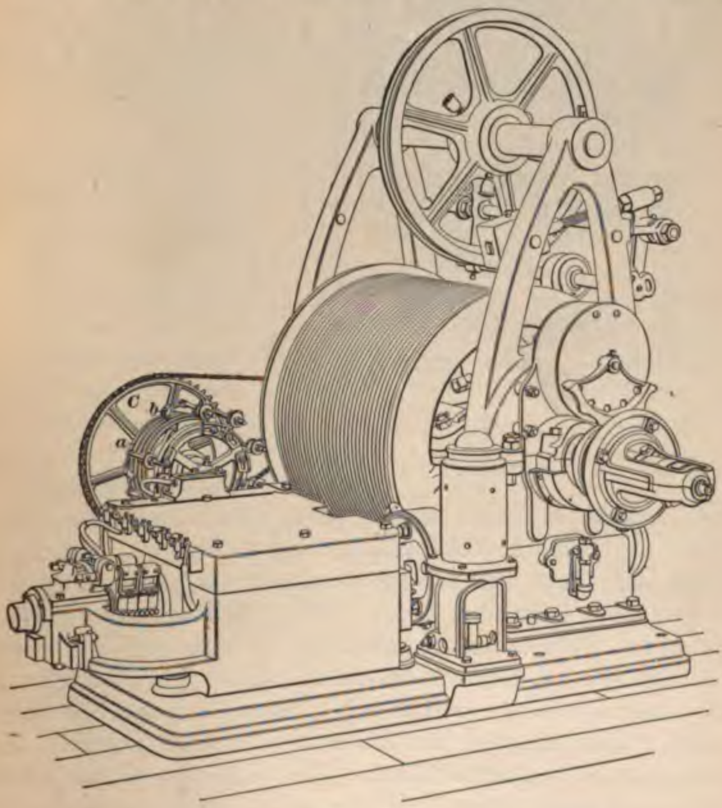


FIG. 43.

ture and the fixed terminals w. The pair of magnets C' outs out the starting resistance, and magnet B' closes the main circuit and releases the brake. The devices shown at d, e, f, g, h, and k are known as **floor magnets**, and their function is to hold the push-button circuit closed after it has been once pressed, even though the operator releases the push button itself; their action will be explained later. Magnets l, m are provided to prevent interference with the elevator from other floors until the party who is already operating it is through.

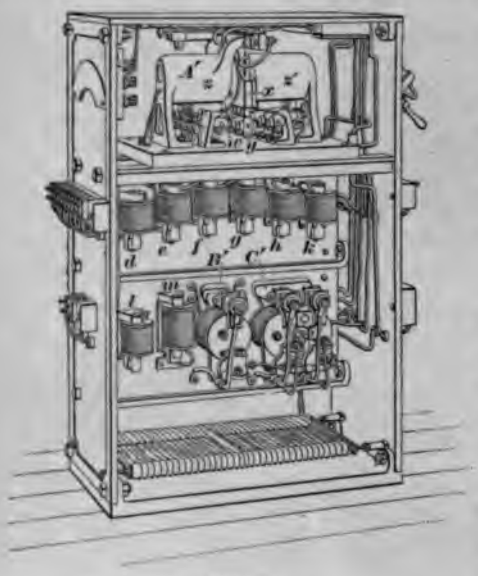
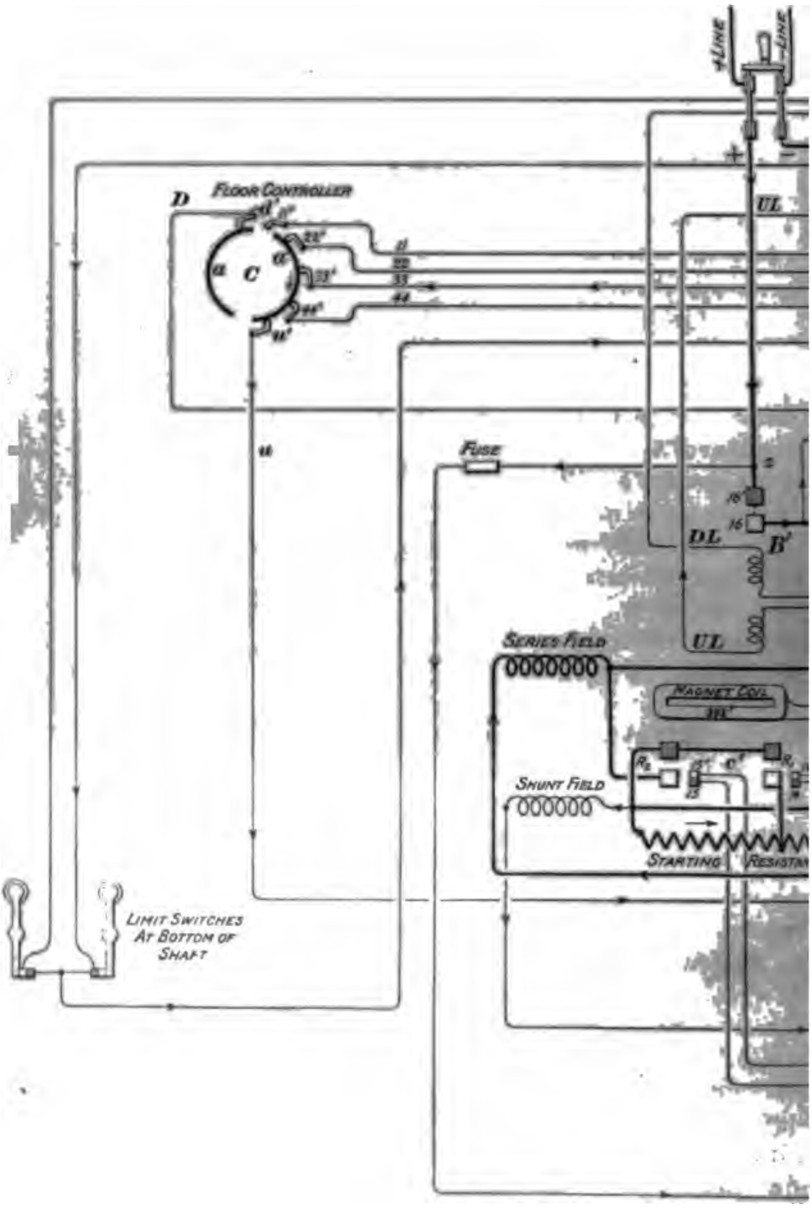
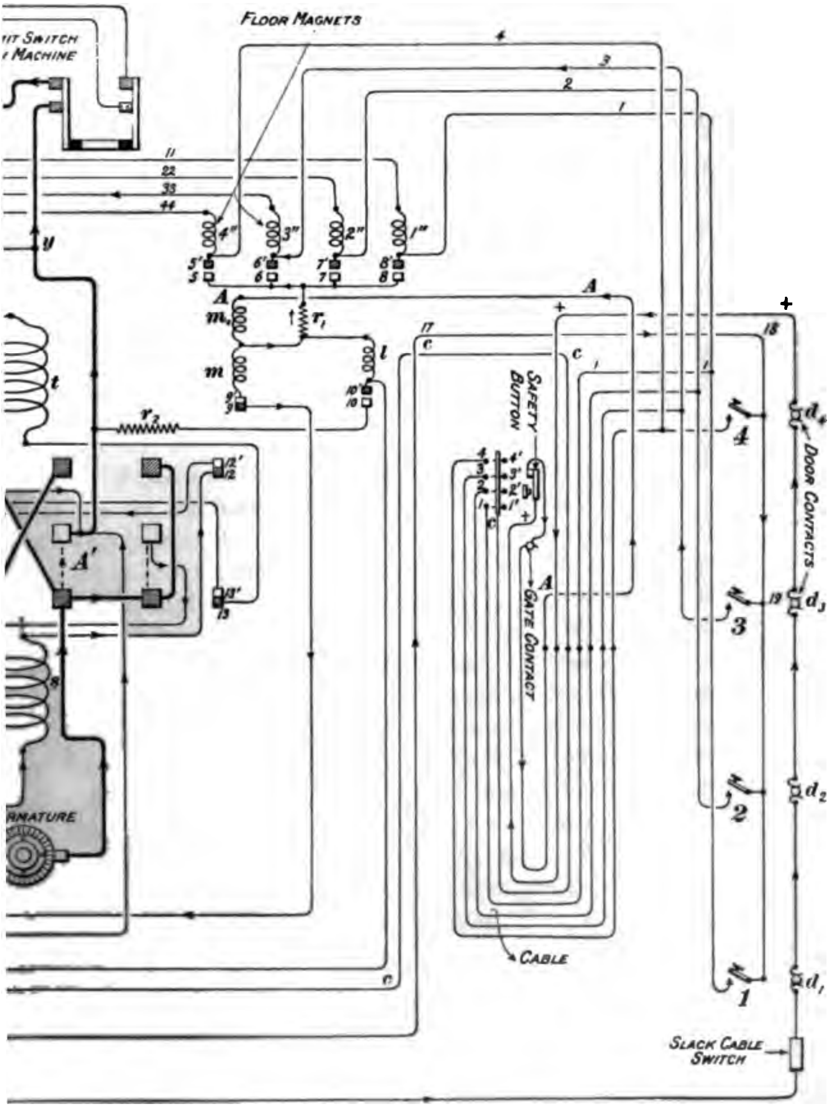


FIG. 44.

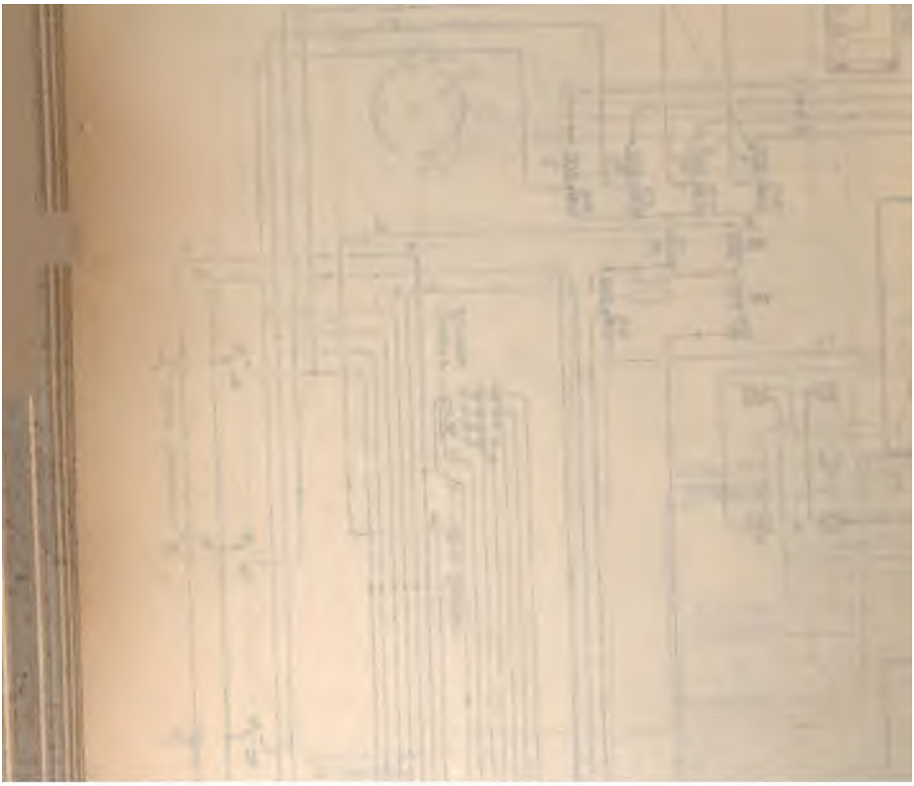
94. Connections for Automatic Elevator.—Fig. 45 shows a diagram of connections for an Otis automatic elevator with the style of floor controller shown in Fig. 43. Like the preceding diagrams, the location of some of the parts has been changed in order to make the connections easy to follow. The diagram shows the connections necessary for the control of the elevator from four floors. A' is the reversing switch; the open contacts are mounted on the swinging armature and the shaded contacts are fixed. When coil s is excited, the armature is drawn towards s, and the open contacts make connection with the lower row of fixed contacts, as indicated by the dotted lines. We will assume that with the connections shown, the car moves up when coil s is energized and down when t is energized. Switch B' closes the main circuit; $1''$, $2''$, $3''$, $4''$ are the small floor magnets shown at d, e, f, etc., Fig. 44; 1, 2, 3, 4 are the push buttons on each floor; $1'$, $2'$, $3'$, $4'$ are the push buttons on the elevator; d_1, d_2, d_3, d_4 are door contacts. These contacts are in series and are connected together as shown only when



+LINE
-LINE
FLOOR CONTROLLER
D
C
UL
DL
B'
FUSE
SERIES FIELD
MAGNET COIL
SHUNT FIELD
STARTING
RESISTAN
LIMIT SWITCHES
AT BOTTOM OF
SHAFT



FLOOR MAGNETS
SAFETY BUTTON
GATE CONTACT
CABLE
DOOR CONTACTS
SLACK CABLE SWITCH
ARMATURE



all the doors of the shaft are closed. If any one of these contacts is open, i. e., if any door is not closed, it is impossible to start the elevator. The two switches for cutting out the resistance and the series-field coils are shown at c', and they are operated by magnet m'. C is the floor controller driven from the winding drum. The two segments a, a' represent the strips a shown in Fig. 43. On the drum itself they are put on spirally in order to allow a small diameter of drum, but here they are shown as two arcs, so that the connections may be more easily followed. With the position of the controller shown, the car is at the bottom of the shaft, i. e., at the first floor. As the car ascends, the controller turns as indicated by the full-line arrow, and a' slides from under *22'*, *33'*, *44'*, and u' in succession. At the same time, segment a comes in contact with *11'*, *22'*, *33'*, and *44'* in succession.

95. When switch A' is pulled by coil t, contact is broken between the auxiliary contacts *12*, *12'*, thus opening the circuit through coil s. Also, when A' is operated by coil s, contact is broken between contacts *13*, *13'*, thus opening the circuit through coil t. It is, therefore, impossible for both s and t to be excited at the same time. When switch c' makes contact at R_1, contact is broken between *14* and *14'*; when contact is made at R_2, contact is broken between *15* and *15'*. When the armature of magnet $m\ m_1$ is drawn up, contact is broken between points *9*, *9'*; when the armature of l is attracted, contact is made between points *10*, *10'*. When magnets *1''*, *2''*, *3''*, etc. operate, contact is made between *5* and *5'*, *6* and *6'*, etc. When buttons *1'*, *2'*, *3'*, or *4'* are pressed, contact is made between wire c and the wire running to the floor magnet corresponding to the button that has been pressed.

96. Operation of Automatic Elevator.—As already stated, the controller C indicates that the elevator is at the first floor. Finger *11'* is open-circuited, and, pressing either button *1* or *1'* would not start the elevator, because it is already at the first floor. Suppose a passenger wishes to get on at the third floor and then go down to the first floor. He presses the third-floor button *3*, and the elevator moves up to the third floor and stops there, at the same time automatically unlocking the door of the shaft. After the passenger has entered the elevator and closed the door, he presses button *1'* in the car and the elevator descends to the first floor and stops. The way in which the control is brought about will be understood by following the circuits in Fig. 45.

97. When button *3* is pressed, the operating current, starting from point z on the main + wire flows through the slack cable switch, through door contacts d_1–d_2–d_3–d_4, through cable to car and through safety button on car to *A–A–A–*m_1*–m–9'–9–14'–14–17–18–19*, through push button *3–3*, through floor magnet *3''–33–33*–finger *33'*–strip a'–finger u'–u–u–magnet *s–12–12'*, through *UL* magnet of B', through limit switch on machine, through limit switches at bottom of shaft, to negative side of the circuit at y. This operating current accomplishes several results. In the first place, it closes magnet *3''* so that contacts *6* and *6'* are brought together. This provides a path for the operating current that is independent of the path through push button *3*. The current can now flow along *A A* through coil m_1–r_1–*6–6'* through coil *3''*, and so on as before. Consequently, after button *3* has been pressed the car will start up, even though *3* be released again, and it is not necessary for the passenger to keep the button pressed until the car reaches its destination. All that is necessary is to press *3* long enough to allow *3''* to attract its armature. When the operating current takes the path through *6–6'*, a resistance r_1 is in circuit, as only a small operating current is needed to hold the armatures after they have been attracted. The operating current flows through s and one coil of switch B'. Hence, the reversing switch is pulled to the up position and the main circuit is closed, thus allowing the main current to flow as indicated by the arrows and starting up the motor. The operating of these switches also allows current to flow through the shunt-field coil and brake solenoid, thus releasing the brake. When coils m, m_1 are energized, points *9*, *9'* are separated, thus breaking all connection between wire *A A* and wire *17*, *18*, *19*, which leads to one side of all the floor buttons; consequently, as soon as one button, in this case *3*, has been pressed, the buttons on all the other floors are cut out of service, and it is impossible for any other parties to operate the elevator. As the motor speeds up, switch R_1 operates because coil m' is connected across the armature terminals, and this cuts out the greater part of the resistance, at the same time separating points *14*, *14'*. When sufficient speed is attained, switch R_2 operates and cuts out the remainder of the resistance and the series field, at the same time separating points *15*, *15'*.

98. All the time that the car is going up from the first floor to the third, controller *C* is turning as indicated by the arrow, until, when the third floor is reached, *a'* slides from under finger *33'*, thus interrupting the operating current and stopping the motor. The motor is stopped by the band brake and no provision is made for a dynamic braking action, as these elevators are not intended for high-speed service.

99. After the car has stopped at the third floor and automatically unlocked the shaft door, the passenger slides back the door, thus opening the operating circuit at contacts d_3 and making it impossible for any persons on the other floors to start up the elevator while he is getting on. After closing the door and thus reestablishing contact at d_3, he presses button *1'*. This allows the current to flow as follows, starting from *y* as before: *z*–slack cable switch–d_1–d_2–d_3–d_4, through cable to safety button on car–*A*–*A*–*A*–through m_1–*l*–*15'*–*15*–*c*–*c*–*c*–*c*–*1'*–*1*–*1*–*1*–*11''*–*11*–*11*–*11'*–*a*–*d'*–*D*–*t*–*13*–*13'*–*D L*, through limit switch on machine, through limit switches at bottom of shaft to *y*. It must not be forgotten that by the time the elevator has reached the third floor, fingers *11'* and *22'* are resting on contact strip *a*, and hence are in connection with the wire *D* that runs to *t*. Switches *B'*, *A'*, and *C'* therefore operate as before, except that *A'* allows the current to flow through the armature in the reverse direction and reverses the motor. Floor magnet $1''$ makes contact between 8 and $8'$, so that the operating current will continue to flow even after the button $1'$ is released. Contacts 9, $9'$ are also separated, so that the push buttons 1, 2, 3, 4 are cut out while the elevator is in motion. Magnet l is excited and makes contact between 10 and $10'$, thus allowing current to flow to the negative line by the path A–m_1–l–$10'$–10–r_2– –, and this current holds 10 and $10'$ in contact, even though the operating current through $1''$ is interrupted by the controller C when the elevator reaches the first floor. When the elevator reaches the first floor, it is automatically stopped by the controller, as already explained, but contacts 9 and $9'$ are still separated and contacts 10 and $10'$ closed, because current still flows through the path 2–d_1–d_2–d_3–d_4–A–A–A–m_1–l–$10'$–10–r_2– –. The result is that no one can interfere with the elevator because pushes 1, 2, 3, 4 are cut out. This current through m_1 and l remains to flow until the door is opened, thus breaking the circuit at d_1 and allowing the armature of m and l to drop. After the passenger has gotten out and after the door has been closed again, thus bridging the break at d_1, the elevator may be operated from the other floors, but not before; thus avoiding the possibility of accident while the passenger is getting out.

By tracing out the connections and bearing in mind the action of the controller C, the student will see that the car is under complete control at all times, and that it is practically impossible for any person to interfere with the operation while another person is using it. The unlocking as well as the opening of the doors on these elevators is usually automatic.

OTIS AUTOMATIC ELECTRIC ELEVATOR WITH NO. 2 FLOOR CONTROLLER.

100. General Description.—The style of floor controller shown in Fig. 43 and indicated at C, Fig. 45, has been superseded by a later type shown at C, Fig. 46. Both styles are, however, in use, so that it has been thought advisable to illustrate both of them. The floor controller in Fig. 46 is considerably different in construction, but it accomplishes the same results as the older type; it is mounted on top of the motor and driven by a chain *a* running over a sprocket wheel on the end of the drum shaft.

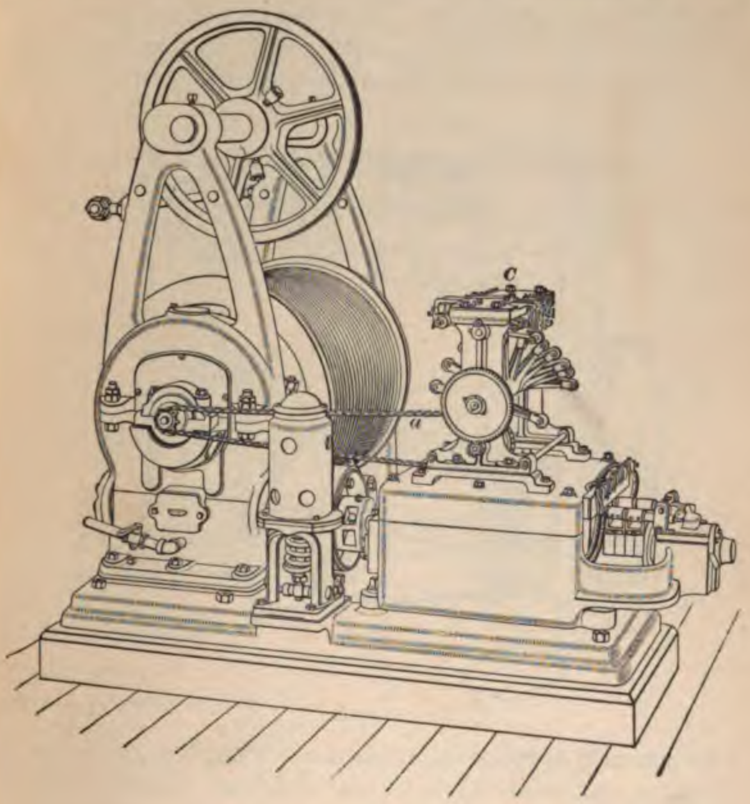


FIG. 46.

The controller *C* serves also as a limit switch, so that it is not necessary to provide the hoisting machine with the usual traveling nut operating a limit switch. Fig. 47 is a larger view showing one side of the controller. The sprocket wheel *s* is revolved by means of the chain, and by means of reduction gearing turns a shaft *b* through an arc proportional to the total rise of the elevator. On this shaft a number of arms *c* are mounted, each arm carrying a small roller *d*. On shaft *e* a number of cams *f* are loosely pivoted, each cam carrying at its end a cross-contact piece *g*; the shape of these contact pieces is more clearly indicated by the large one shown at *g'*. Each contact has a groove in which the rollers *d* press as shaft *b* revolves, thus forcing up the contact and making connection between the two clips

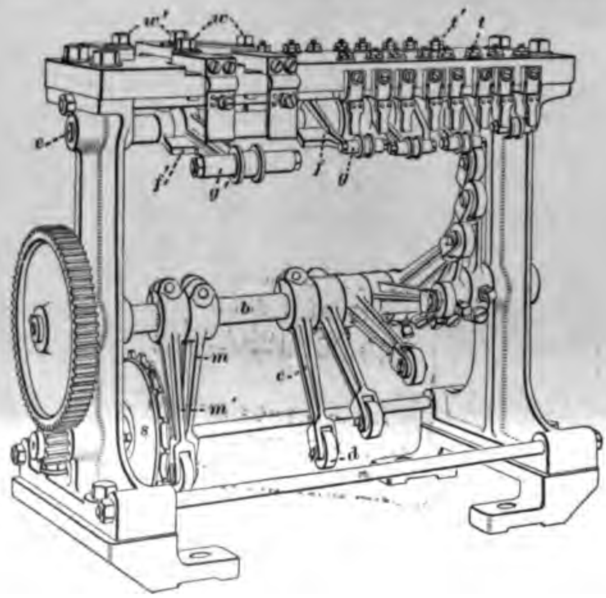


FIG. 47.

with which it is brought in contact. There are two rows of contacts, one on each side of the controller, together with a corresponding number of arms, cams, and cross-contact pieces. The two pairs of large terminals shown at *w*, *w'* connect to the main circuit, contact being made between them by means of the large cross-contact piece *g'* and a similar one on the other side of the controller. These two main switches are operated by the arms *m*, *m'*. This controller accomplishes the same result as the one shown at *C* in

Fig. 43, but uses a series of mechanically operated switches in place of a series of brushes with a sliding contact.

101. Connections for Otis Automatic Elevator With No. 2 Floor Controller.—Fig. 48 is a diagram of connections for this type of controller. It will be noted that it is very similar to Fig. 45, the limit switch on the machine and switch *B'* being omitted. The up-and-down magnets on switch *A'* are reversed in position from that in the first diagram, but this is immaterial, as the direction of rotation of the motor may be kept the same in both cases by reversing the armature terminals at the motor. The two large cross-contact pieces on the controller are shown at *g'*, *g''*, and the small contacts are indicated at *11'*, *33'*, *22''*, etc., there being but three small movable contacts on each side in the diagram, because the elevator is controlled from four floors only. With the diagram as shown, the car is supposed to be at the first floor. All the left-hand contacts of the controller are out and all the right-hand ones are in, connecting the floor magnets to the *u* line and allowing current to flow through the up magnet of switch *A'* when any one of the push buttons *2*, *3*, or *4* is pressed. As the car moves away from the first floor, *11'* closes, and when it reaches the second floor, *22'* opens. As it moves away from the second floor, *22''* closes, and when it reaches the third floor, *33'* opens; and so on. When the car reaches its upper limit of travel, switch *g'* is opened and when it reaches its lower limit, *g''* is opened, thus cutting off the main current. When the elevator descends, the switches open and close in the reverse order. The small arrowheads show the path of the operating current when button No. *3* is pressed to bring the car up to the third floor. The large arrowheads show the path of the main current. It will not be necessary to trace these through, since outside of the part through switch *C* they are practically the same as explained in connection with Fig. 45. Ordinarily, the switch *A'* would open the main circuit and switches *g'* and *g''* are intended more as a safety device in case *A'* does not operate.

SPRAGUE-PRATT SCREW ELEVATOR.

102. General Description.—The hoisting mechanism of this elevator differs in a marked degree from those previously described; Fig. 49 shows the general construction. It has no winding drum, the cable being taken up over a number of multiplying sheaves. The hoisting rope *H*, or rather set of four ropes, passes over the fixed sheaves *S'* and movable sheaves *S* and is anchored at *A*. The motor *M* revolves a long screw *E*, which is directly coupled to the motor shaft. On this screw is a nut *N*, which is not connected in any way with any other part of the mechanism. A crosshead carrying sheaves *S* is arranged to slide on the base *B*, and when screw *E* is revolved by the motor, the nut bears against the crosshead and moves the sheaves *S* to the right, thus taking up the cable and raising the car. The construction of the nut *N* and the sheave bearings is such that there is very little friction, and the efficiency of the hoisting mechanism is so high (about 70 per cent. from car to motor) that when the car is descending, the pull against the crosshead revolves the screw and motor in the reverse direction, thus driving the motor as a generator. The sheaves are usually designed to give a multiplication of 8 to 1, so that the amount of rope that the machine takes up is 8 times the travel

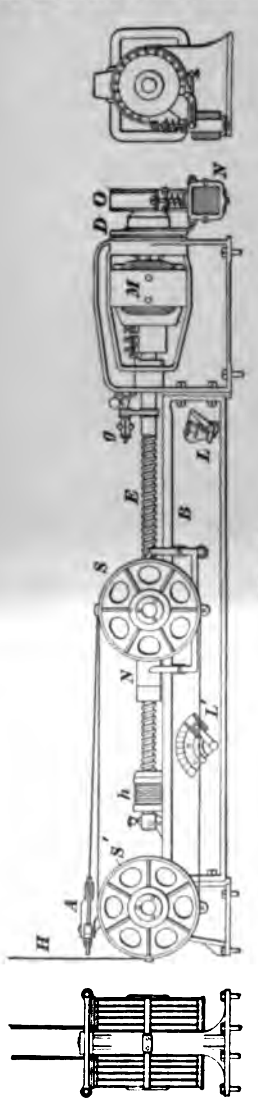


FIG. 49.

of the screw. For high rises and high speed, there is a further multiplication of 2 to 1 on the counterbalance. The ropes lead from the car over the overhead sheaves, down around a sheave on the counterbalance, up to and anchored at the top of the building. The ropes leading to the machine are attached to the bottom of the counterbalance. There are four of these ropes, as indicated in the end view, Fig. 49, two of them passing around the eight sheaves on one side of the machine, and the other two passing around the eight sheaves on the other side. The travel of the car is, therefore, 16 times that of the nut. The screw *E* is always under tension, no matter what the load on the elevator may be and no matter whether it be moved up or down. This is necessary with this type of elevator because the construction of the nut and screw is such that the pressure between them must always be in the one direction, and the tension on the rope is the only driving power that the elevator has when descending. These machines are not, therefore, overbalanced.

103. Motor.—The motor used with the Sprague-Pratt elevator is of the ordinary direct-current four-pole type with compound field winding. It is mounted at the right-hand end of the machine, as shown at *M*, Fig. 49, and needs no special description.

104. Transmitting Devices.—The transmitting devices of this elevator are of special interest. The use of the screw, traveling nut, and sheaves makes the action similar in many respects to that of a hydraulic elevator. The sheaves are mounted on roller bearings so as to run with little friction, and the traveling nut is arranged so that the thread of the screw bears against balls. Fig. 50 shows a section of the **ball nut.** Steel balls *a* are arranged as shown, and when the screw turns, these balls revolve and work their way along through the nut, passing in at one end, traveling through the nut, and returning by way of the channel *b* in one side. The rolling friction of such a

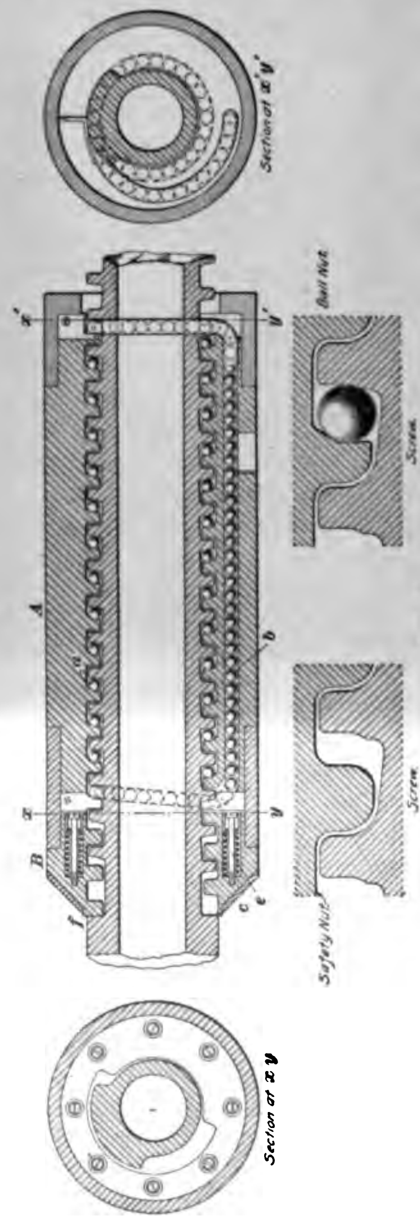


FIG. 50.

nut is very much less than the sliding friction of an ordinary nut. In addition to the ball nut A, there is provided a safety nut B, which is without balls, because under ordinary circumstances there is no pressure taken up between its threads and those of the screw. The safety nut is provided for two purposes, namely: to prevent slack cable and also to hold the crosshead in case the threads on the ball nut or screw should strip. This last contingency is something that never occurs if the elevator receives any kind of inspection, but as these elevators are intended for passenger service, it is advisable to take every possible precaution. When the car is drawn up, there is a thrust between the conical bearing c and the crosshead that carries the movable sheaves, and since the friction of this conical bearing is much greater than the friction of the screw, the nut does not revolve, but travels along the thread, thus pushing the crosshead and raising the car. When the car descends, the pull on the rope runs the screw backwards, and with it the motor, which now runs as a generator. When the pressure on the nut is released, the screw continues to revolve on account of the momentum of the armature, and the cable would be slackened if the nut did not revolve with the screw. As soon, however, as the pressure on bearing *c* is released, springs *e*, which are normally compressed, force nuts *B* and *A* apart, thus bringing the threads of the safety nut into contact with those of the screw and producing enough friction to make the screw and nut revolve together and thus hold the crosshead stationary. Again, if the threads of the ball nut should wear excessively, or strip, the pressure is taken up on the safety nut, which then revolves with the screw and indicates the defect. A buffer *h*, Fig. 49, is provided for the nut to strike against when it reaches the limit of its travel corresponding to the lowest position of the car. When it reaches the upper limit, the end of the nut comes up against the shoulder *f*, Fig. 50, and any further turning of the screw simply causes the nut to revolve with it. The nut shown in Fig. 50 is the later type using a hollow screw of large diameter with $\frac{3}{8}$-inch balls. The balls used on earlier types of the machine were $\frac{1}{2}$ inch in diameter, but this size was found to be rather small. The nut shown contains 320, $\frac{3}{8}$-inch balls. The nut used formerly had 240, $\frac{1}{2}$-inch balls.

105. Thrust Bearing.—The thrust of the screw is taken up by a special form of thrust bearing, which is located at *D*, Fig. 49, on the back end of the motor frame. The thrust is taken up on a large number of small rolls placed between two hardened steel plates. One plate is carried by the field yoke and the other revolves with the shaft. The small rolls, 180 in number, $\frac{1}{2}$ inch in diameter by $\frac{3}{16}$-inch face, are placed in openings, arranged in spiral form, in a bronze plate. A plate containing these rollers is shown in

Fig. 51; this is placed between the two hardened plates previously mentioned, and the whole thrust bearing is arranged so as to run in oil.

FIG. 51.

106. Brake.—The elevator is provided with a band brake controlled by a solenoid. This brake is shown at *O*, Fig. 49, and consists of a steel band lagged with wood. The band covers about three-fourths of the circumference of the brake wheel. The solenoid *N* operates against a spring, so that when the magnet is excited the brake is released, and when it is demagnetized the brake is at once applied by the spring.

107. Limit Switches.—Two limit switches *L* and *L'*, Fig. 49, are mounted on the base and are operated by projections on the traveling crosshead, so that if the sheaves reach the limit of their travel in either direction, the motor is stopped. Switch *L* is ordinarily closed, and when the car reaches the upper limit of its travel it is opened, thus opening the main circuit and applying the brake. When the car is descending, switch *L'* is normally open and when *L'* is operated at the lower limit it is closed, thus cutting in a resistance across the motor and gradually cutting it out with further motion of the crosshead. A centrifugal governor *g* is belted to the screw, and if the speed exceeds the allowable limit, this governor opens a circuit and effects an application of the brake.

108. Method of Control.—The method of control used with the Sprague-Pratt elevator is similar in many respects to the magnet-control method previously described. The magnet type of controller might be used with this type of elevator, but many of the Sprague-Platt machines are equipped with a controller in which resistance is cut out by means of a sliding arm moved by a small *pilot motor*. The closing of the main circuit and the reversing of the motor is accomplished by means of electromagnetic switches very similar to those shown in Fig. 40. The pilot motor is under the control of the car operator and is operated by means of a car-operating switch in a manner similar to that already described in connection with magnet control.

109. Sprague-Pratt Vertical Type Elevator.—Most of the Sprague-Pratt machines have been of the horizontal type shown in Fig. 49. In cases where two or more elevators are required, these horizontal machines are placed one on top of the other, thus economizing space, but a number of machines have been built so that they may be placed vertically in the same way as a hydraulic elevator. Fig. 52 shows the general arrangement of one of these vertical machines. The motor *M* is at the bottom of the shaft. The fixed sheaves *A* are mounted just below the lower limit of the counterbalance, and the movable sheaves *S* travel up and down in guides. The rope running to the sheaves is fastened to the under side of the counterbalance, and there is a multiplication of 2 to 1 between the counterbalance and the car, as indicated. The vertical type has some important advantages over the horizontal type. In the horizontal type, the long screw always tends to sag more or less, thus producing uneven wear. This sagging effect also produces uneven wear on the motor bearings and on the thrust bearing.

FIG. 52.

When the machine is placed in the vertical position, these effects are done away with entirely, and the additional advantage is gained that the weight of the screw, armature, and sheaves tends to counterbalance some of the thrust and thus reduces the effective pressure on the thrust bearings.

FRASER DIFFERENTIAL ELEVATOR.

110. General Description.—This elevator has not as yet been widely used, but as it is very simple in construction and easily controlled, it is probable that it will prove valuable for many kinds of service. It is manufactured by the Otis Elevator Company. The principle on which the elevator operates is an interesting one and will be understood by referring to Fig. 53; *A* is the car; *B* and *C* two pulleys that revolve in opposite directions, as shown. *W* is the counterweight and *D* an endless rope passing over the pulleys *B*, *C*, and around pulleys *E* and *Y* on the car and counterweight. Pulleys *B* and *C* are driven by independent sources of power, so that their speed with regard to each other may be changed. If the circumferential speed of *B* is exactly the same as that of *C*, it is evident that the rope *D* will simply pass around over the pulleys and the car will remain stationary. If, however, the circumferential speed of *C* is made greater than that of *B*, the rope will be passed over *C* faster than it is taken up by *B* and the car will descend. If the circumferential speed of *B* is greater than that of *C*, the rope will be taken up by *B* faster than it is paid out by *C*, and the car will ascend; the greater the difference in circumferential speed, the greater is the speed of the car. It should be noted that the action depends on the difference of *circumferential speed*, or upon the difference in speed at which the rims of sheaves *B* and *C* travel. Pulleys *B* and *C* may or may not revolve at the same speed when the car is stationary, depending on whether or not they have the same diameter.

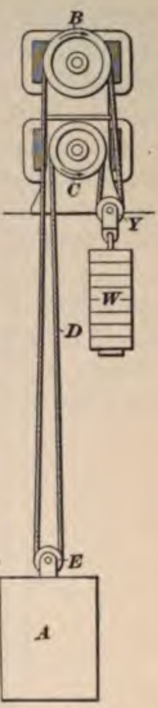


FIG. 53.

This type of elevator allows the car to be stopped, raised, and lowered without stopping or reversing the driving motors. This, of course, is a great advantage. Usually electric motors are used for driving *B* and *C*, though steam engines or a combination of engines and motors could be used. Another advantage of this elevator is that it does not require a winding drum with its accompanying gearing.

111. Fig. 54 shows the general arrangement of an elevator embodying this principle and driven by means of two electric motors. *B* and *C* are the two pulleys, the circumferential speed of which is varied by changing the speed of the motors *m*, *m'*. The endless rope *D* is in this case not attached to the car, but runs around a pulley *E* carried on the bottom of the counterweight *W* and around the pulley *Y* carried on the under end of the rope-tightening device *N*. The ropes *L* are attached to the car, and after passing over sheave *M* are attached to the counterweight. The ropes *F* also attach to the top of the counterweight, and after passing over sheave *G* and through cross-bar *3*, are fastened to cross-bar *4* of the tightening device. By drawing bar *4* down on the threaded rods, the ropes can be tightened to any desired degree. The speed of the motors is controlled from the car, and in making a trip they are not

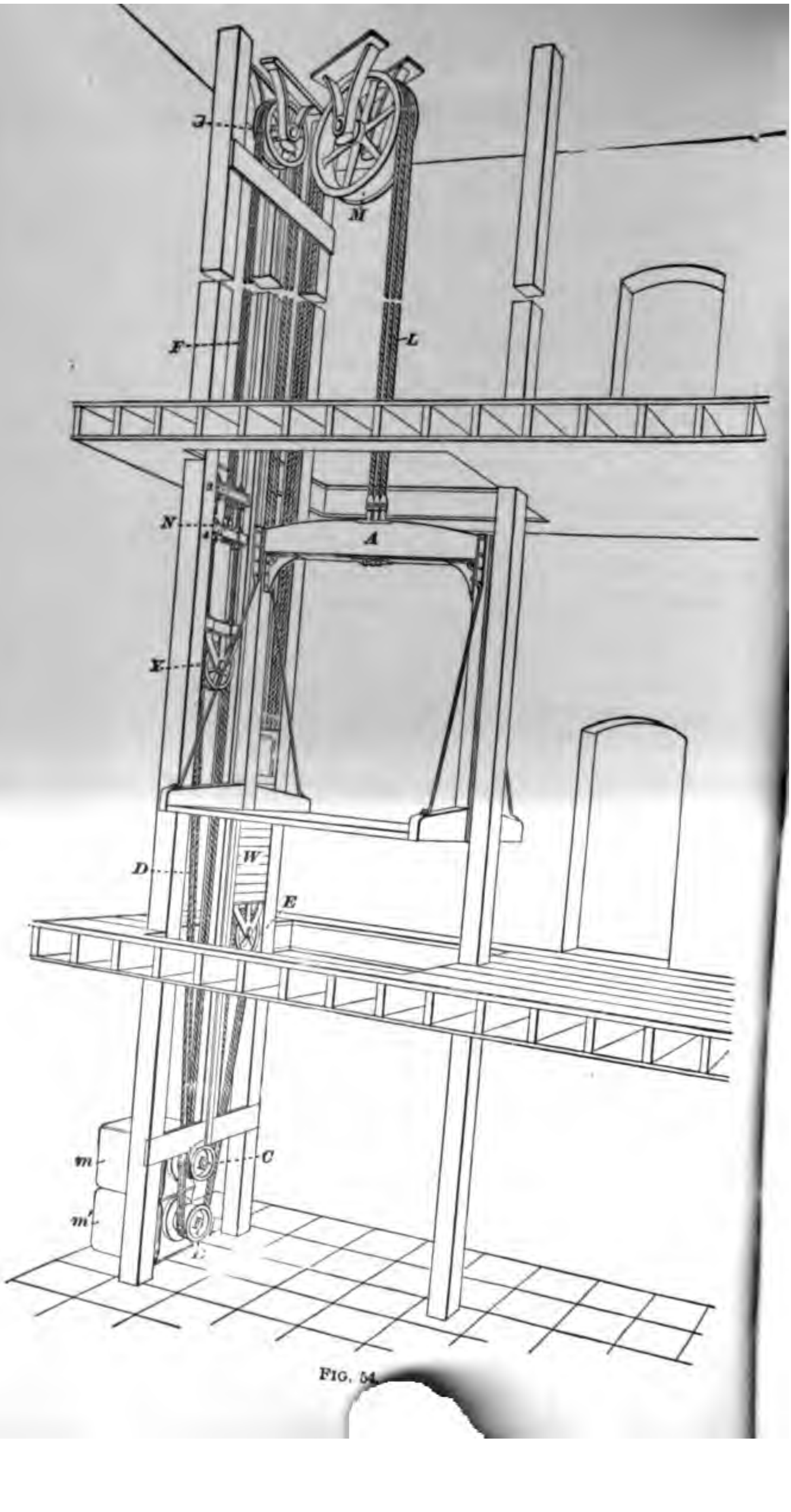


FIG. 54.

stopped when the elevator stops at the various floors; they are merely made to run at the same speed by means of the car controller. While the car is ascending, pulley *B* runs faster than *C*, and in order to make it descend, all that is necessary is to make *C* run faster than *B*. The variations in speed are readily accomplished by varying the field strength of the motors. The controller is arranged so that when the handle occupies the central position, the speed of both motors is alike and the car is stationary, and when moved to either side of the center, the speed of either one or other of the motors is changed. A small auxiliary-operating handle is also provided in connection with the main handle, so that by pulling up on it the operator can stop both motors when the elevator is not in use.

ELEVATORS.

(PART 3.)

HYDRAULIC ELEVATORS.

INTRODUCTION.

1. Hydraulic elevators are still considered by the majority of engineers as being the most suitable for large passenger-service plants with their high lifts and great speeds, although the electric elevator has since its advent become a powerful competitor. The hydraulic elevator is intrinsically safe, reliable, smooth-acting, and under perfect control. It requires comparatively less care in operation than the electric elevator, the mechanism being very simple. The cost of maintenance is small, the wearing parts being few and easily and cheaply replaced.

On the other hand, the hydraulic elevator is cumbersome, requiring much space, especially—and this is the case in most large plants—where the water pressure available is not high enough for direct use in the elevator cylinders, so that the installation of steam pumps, reservoirs, or tanks, and the necessary piping becomes necessary, not mentioning a boiler plant, which we may assume, for the sake of the comparison, as being already in existence for other than the elevator service. Thus, the first cost is great compared with that of an electric elevator plant, which, in case the right current is already available, either from a central station or from an isolated lighting plant in the building, consists of the elevator machine only, which may even be placed on top of the hoistway, and in case the current must be generated expressly for the elevator, of an additional steam-engine- or gas-engine-driven dynamo, but no cumbersome tanks or piping. The installation is thus simple and cheap, the space needed but small. There are advantages, then, in both systems, and which one to select depends on many circumstances which must be weighed against one another by the architect and owner, but not by the operating engineer, who should have no prejudice against the one or the other, but should be equally familiar with both.

PLUNGER ELEVATORS.

SERVICE.

2. The simplest kind of hydraulic elevator is the **direct-acting** or **plunger elevator.** It is also the oldest kind of hydraulic elevator and has been used for a long time, both for freight and passenger service, for short lifts. It has been until recently considered unsuitable for high lifts and high speeds, and is therefore found installed in great numbers as yet only for sidewalk lifts, slow freight elevators, and similar service. Lately, however, the possibility of using this type of elevators for greater lifts and speeds has been recognized, and it is safe to predict that they will be more frequently installed than before and for even the severest service.

CONSTRUCTION.

3. Fig. 1 shows a plunger elevator made by Morse, Williams & Co., of Philadelphia, Pennsylvania, for short lifts.

4. Motor.—The motor in this machine consists of a vertical cylinder *A* sunk into the ground below the bottom of the hoistway and a plunger *P*. The cylinder is closed at

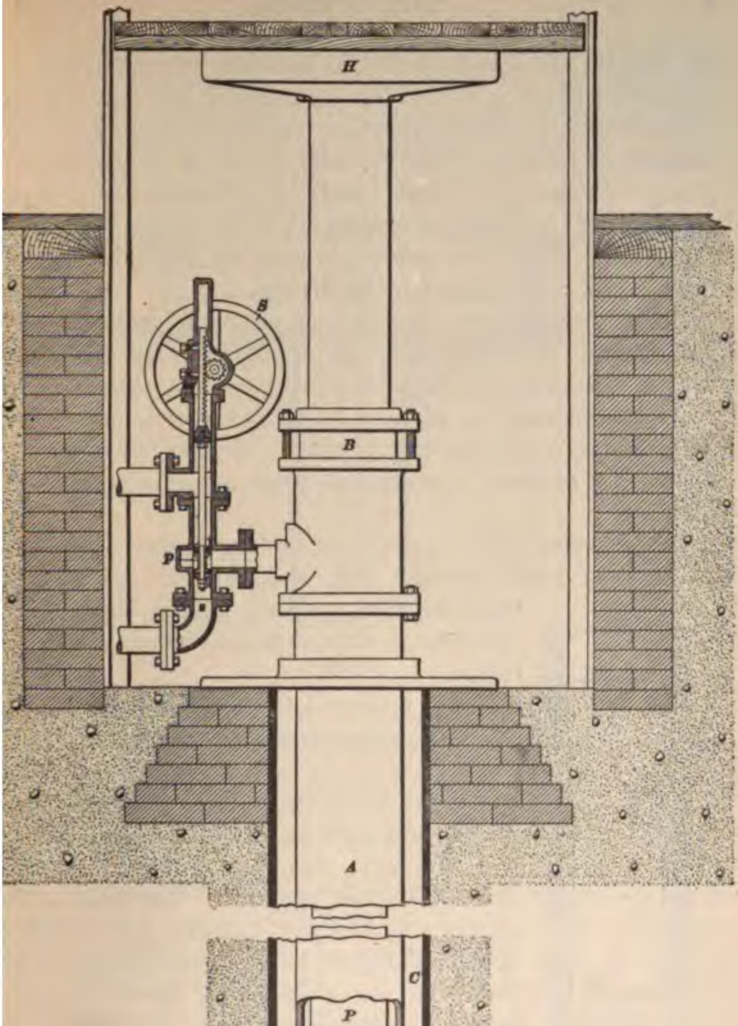


FIG. 1.

the bottom and has an enlarged head above ground containing the stuffingbox B and an opening for the pipe through which the water under pressure enters the cylinder and is discharged from it. The cylinder must, of course, be sunk plumb. If the subsoil is soft earth, it is first necessary to sink a steel pipe C, called the **casing**, through which the earth is removed. When the subsoil is rock no casing is required, the hole for the cylinder being drilled.

For high lifts the cylinder is made up of sections. The Plunger Elevator Company, of Worcester, Massachusetts, use steel tubing, which they square up and thread in the lathe, connecting the sections by means of couplings. This insures a perfectly straight cylinder. Before burying in the ground the cylinder is tested and given a coat of preservative paint.

The plunger when required to be long is also made up of sections of steel tubing. Fig. 2 shows the special joint used by the company named above. The plunger is turned to uniform size and polished.

FIG. 2.

5. Transmitting Devices.—As the car rests directly on top of the plunger, there are no transmitting devices, such as drums, ropes, and sheaves. The car is fastened to the plunger, which is provided with a head H for the purpose. The head shown in Fig. 1 is simply a cast-iron plate clamped to the plunger. This arrangement, while sufficient for unbalanced small elevators, would be dangerous for large counterbalanced ones, inasmuch as should the connection between the head and the plunger give way, the counterweights would jerk the car upwards against the overhead work. Great care is, therefore, taken in balanced elevators of this kind to make the aforesaid connection very rigid and reliable. The manner in which this is done by the Plunger Elevator Company is shown in Fig. 3.

The plunger has a flange formed on its upper end that fits into a corresponding recess of the head *H*. The latter, in turn, is securely bolted to the framework of the car platform. Besides this flange connection a second security

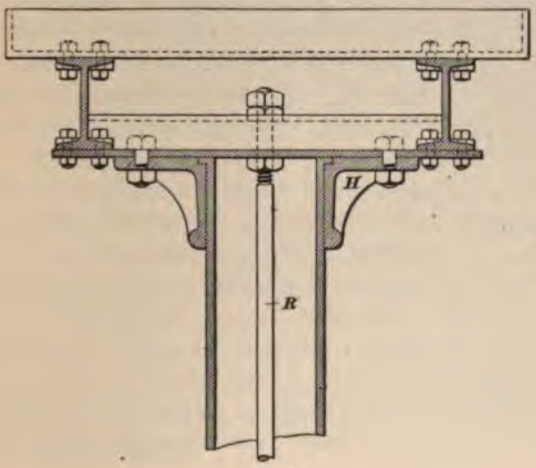


FIG. 3.

against the parting of the car and plunger is provided by a tie-rod *R* which runs all the way through the plunger, through the bottom of the same, and through the framework of the car platform. Instead of the rod *R* a loop of galvanized iron rope is often used for the same purpose.

6. Counterbalancing.—Low-lift plunger elevators are generally not counterbalanced at all. High-lift elevators are counterbalanced, but not overbalanced, since the power acts only on the up stroke of the plunger. Enough of the weight of the car and plunger is left unbalanced to secure the descent of the car at the proper speed when empty. The upward pressure of the water on the plunger gradually diminishes as the plunger goes up by an amount corresponding to the increasing height of the water that displaces the plunger. To equalize this change of pressure, the counterweights are suspended from cables of such size that the weight per each foot of their length passing over the overhead sheaves will be equal to the weight of 1 foot in height of water displacing the plunger.

7. Controlling Devices.—The controlling devices consist simply of a balanced three-way water valve operated by a simple shipper rope, or a shipper rope in connection with some more elaborate operating device. The simple shipper rope is generally used with the smaller machines, while an operating device of more elaborate form is used for the larger machines.

The valve in a hydraulic elevator constitutes the only controlling device, being power control and brake at the same time. As a power control it shuts off the power at the will of the operator; as a brake it is so designed as to shut off the water gradually by throttling. This object is most easily attained by a piston valve, which type of valve is used exclusively. Thus, while there is no brake in the common meaning of the word in hydraulic elevators, it is, nevertheless, there as in any other elevator, but in a different form.

This identity of power control and brake is one of the intrinsically valuable features of the hydraulic elevator, since by opening the water passages more or less, the speed of the car can be regulated to a nicety and in harmony with the load it carries, which feature is not easily attained, if attained at all, in any other kind of elevator. Generally the valve is proportioned by the installators so that when fully open it will give the empty car the maximum speed permissible; but by the use of stops the valve throw can be adjusted to any car speed. Such stops are generally in the shape of knobs or buttons clamped to

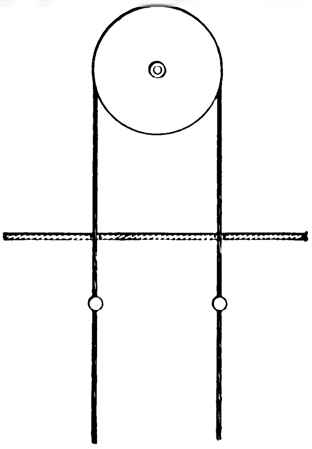


FIG. 4.

the shipper rope and striking against some fixed projection, as shown in Fig. 4. These stops are called **back-stop buttons.**

8. The valve used on the Morse, Williams & Co.'s elevator is shown in section in Fig. 1. A peculiar feature of this valve is the shape of the piston p, which is seen to be wedge-shaped, in consequence of which the water passes to and from the machine gradually and without shock. The operation of the valve will be readily understood from the drawing; on shifting it one way by means of the shipper rope passing over the sheave S, water flows from the *supply* into the *machine* and exerts a pressure on the plunger, lifting it and the car. By shifting the valve in the opposite direction, communication is established between the *machine* and the *discharge*, and the elevator descends. In the intermediate position, the valve shuts off all communication of the machine with the supply and discharge and the elevator is at rest, the plunger being supported on a column of water confined in the cylinder.

9. In larger machines the controlling valve is preferably moved by a motor piston, which is operated by a **pilot valve.** The pilot valve is in turn controlled by the shipper rope from the car. The arrangements of pilot valves and main valves differ in different installations, but are easily understood in every case by inspection. We shall encounter the pilot valve again in connection with piston elevators, when descriptions and drawings of several types will be given and their purpose explained.

10. Safety Devices.—The plunger elevator is the safest elevator built. The ordinary knobs or buttons used on the shipper rope as limit stops are the only motor safeties provided, and even should the limit stop fail to operate the valve at the top of the run, the counterweight would reach the ground and the car stop; should the limit stop fail to operate at the bottom of the run, the car would simply come to rest on the cylinder. In order to avoid damaging the cylinder head in case this should happen, buffer springs are

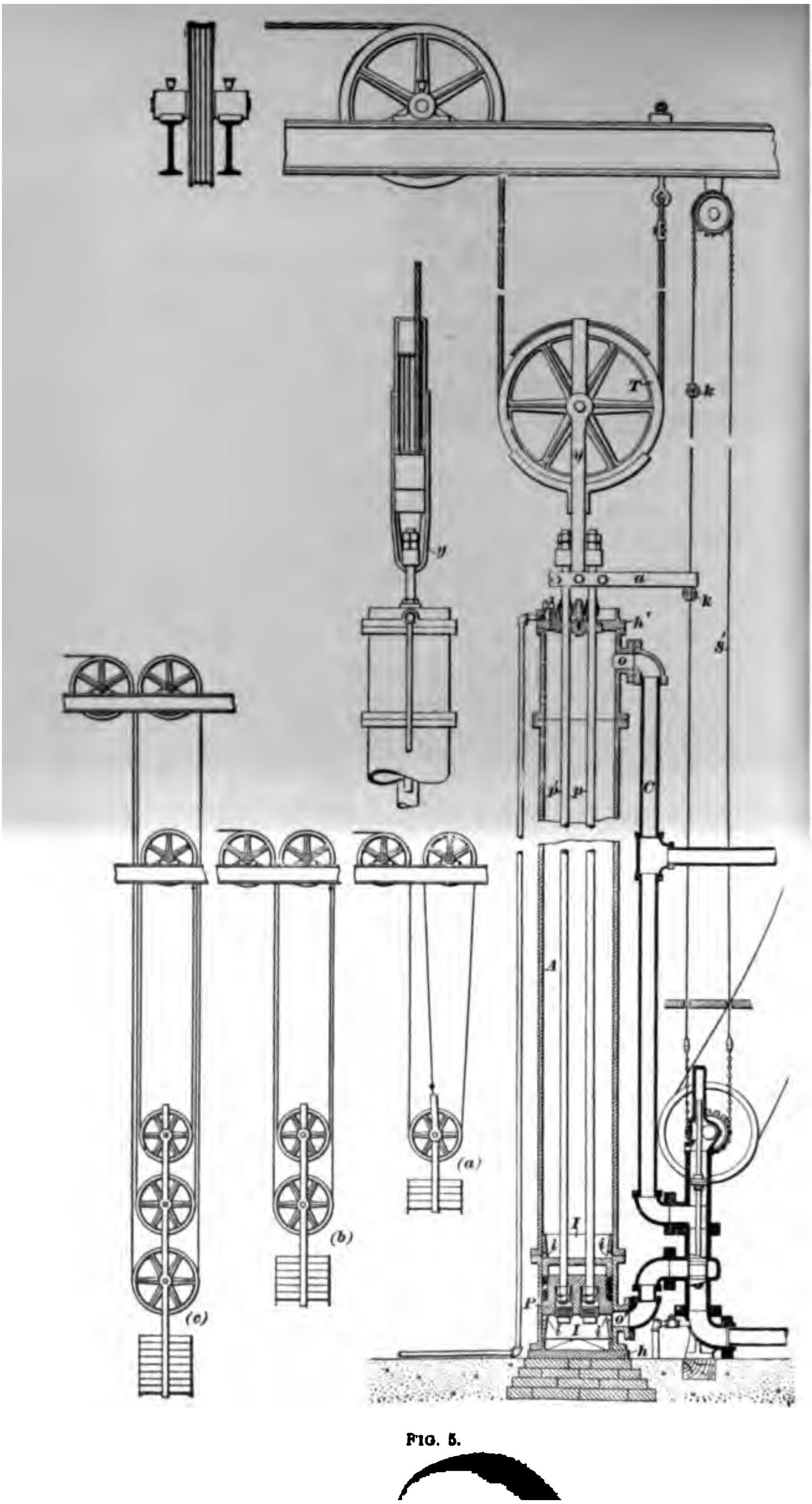


FIG. 5.

often placed on top of the cylinder head, especially when the speed of the elevator is considerable. Car safeties, which are essential on all other elevators, are not needed in plunger elevators, for the car cannot fall, since the plunger always rests on a column of water that is driven out through comparatively small openings; it may, however, in case the valve should fail to operate, attain a speed that would be undesirably great, though not dangerous. To provide against this, the simple expedient of putting in the discharge pipe a throttle valve controlled by the pressure corresponding to the velocity of the exhaust is resorted to when necessary. The car cannot be violently pulled against the overhead work by the counterweight as long as the connection between the car and the plunger is secure, which security is easy of attainment.

PISTON ELEVATORS.

ADVANTAGES.

11. While the plunger elevator treated in the previous articles is simplicity itself, it has some disadvantages. The hydraulic cylinder and plunger must have a length equal to the lift, and for each trip of the car a volume of water is used equal to the area of the plunger multiplied by the lift. In the piston elevator, by introducing multiplying sheaves the hydraulic cylinder can be made considerably shorter, and thus the volume of water used for each lift reduced accordingly. There are two types of piston elevators in general use. In one the cylinder is *vertical* and in the other *horizontal.*

VERTICAL HYDRAULIC PISTON ELEVATORS.

12. The vertical type is considered better than the horizontal type, and is always installed when circumstances will permit, chiefly for the reason that generally headroom is more available than floor space. Fig. 5 is a section through the cylinder, piston, and valve of a simple machine of this kind, as built by Morse, Williams & Co., of Philadelphia, Pennsylvania.

13. Motor.—Following up the various parts, we have as the motor a cylinder *A* and piston *P*, the former consisting of a number of cast-iron flanged sections bored and faced true and bolted together at their flanges; a bottom head *h* and a top head *h'*, which latter contains the stuffingboxes for the piston rods *p* and *p'*. The cylinder has two openings *o* and *o'*, at the top and bottom, respectively.

14. Transmitting Devices.—The transmitting devices consist of wire ropes running over sheaves, one or more of which are carried in a yoke *y* attached to the piston rods, while the others are supported in bearings on overhead beams. The main figure shows but one **traveling sheave** *T*; the car in this case moves twice as fast as the piston, and the elevator is said to be geared in the **ratio** 2 : 1. Fig. 5 (*a*), (*b*), and (*c*) shows the arrangement of sheaves for the ratios 3 : 1, 4 : 1, and 6 : 1, respectively.

15. Counterbalancing.—As we shall see presently, matters are arranged in most vertical elevators so that the cylinder is always full of water. This gives rise to an advantage in counterbalancing. The piston is always carried on a solid column of water and thus forms a counterweight that will come to rest at the moment when the power is cut off, that is, the flow of water stopped; contrary to a free counterweight, it will thus not produce a tendency to teeter the car up and down by its momentum when the power is suddenly cut off. The counterweights in these elevators are, therefore, preferably placed wholly or at least partly on the piston or piston rods, as shown in Fig. 5 (*a*), (*b*), and (*c*). As the power acts only on one side of the piston, the counterweights must be less than the car weight by an amount sufficient to make the car descend at the proper speed when empty.

16. Controlling Device.—The controlling device consists of a balanced three-way valve operated by a shipper rope in the usual manner, the rope being provided with back-stop buttons. The action of the motor under its control is as follows: The space of the cylinder above the piston is always filled with water under pressure, the supply pipe being connected with this space directly through the **circulating pipe** *C*. The other end of the circulating pipe is connected with the space of the valve chamber between the two valve pistons. If the valve pistons be moved downwards, so as to bring the upper valve chamber and thus the space of the cylinder above the piston into communication with the space below the piston, there will be the same water pressure on both sides of the piston. The car, being heavier than the piston with the counterweights, will cause the latter to ascend while it is itself descending, and will force the water from above the piston through the circulating pipe into the space under the piston. For the ascent of the car the valve pistons are raised so as to put the space of the cylinder below the piston into communication with the discharge pipe; there is then pressure only on top of the piston, and the same descends, raising up the car. In the position shown in Fig. 5, the valve closes the space below the piston against both the supply and the discharge, so that the piston is held between the water pressure from above and a confined water column from below.

The object of making the water circulate from the top to the bottom of the piston is primarily to make the effective pressure on the piston the same at all points of the stroke, which otherwise would not be the case. Imagine that the cylinder was open at the top and bottom and the piston at the top of its travel, and that water be poured on to the piston from above; then the latter would descend under the influence of the weight of the column of water above the piston, which would be nothing at first, but would gradually increase towards a weight equivalent to the total contents of the cylinder. Imagine, now, that the space below the piston is filled with water, the piston again being at the top; then the column of water underneath it will exert a suction on the piston corresponding to the height of that column, as long as the column is not higher than 34 feet, which suction will gradually decrease to nothing as the piston descends. Thus by having the space below the piston filled with water the same net force is exerted on the piston at all points; for, while the pressure of the water above it increases, the suction of the water below it decreases at the same rate. In concise technical terms, then, the object of the circulation of the water from the top to the bottom of the piston is to balance the head of the water above the piston.

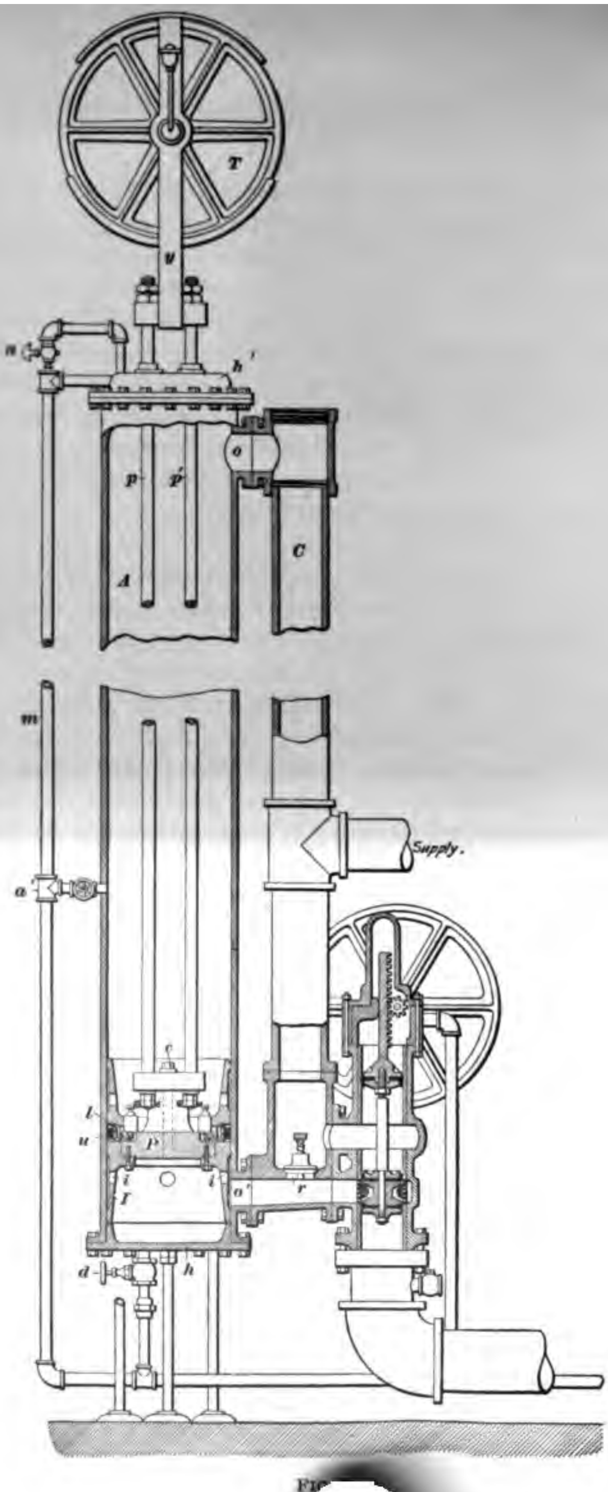
17. Safety Devices.—The safety devices consist of the usual *car safeties* used for suspended cars and *motor safeties*. Limit stops take the shape of knobs or buttons *k*, *k*, Fig. 5, on an endless rope *S'*, which are operated by a projecting arm *a* on the piston rod. The top and bottom heads would of course stop the travel of the piston either way, but it would not be safe to intrust them with that duty, as breakage may result by the piston striking them. The latter should not, therefore, ordinarily travel so far as to strike the heads. There being a possibility, however, that this might occur through a failure of the valve to operate, the piston is provided with an **apron** *l* on each side; each apron has a number of holes *i*, *i* through it and partially closes the ports *o* or *o'* and thus reduces the speed of the piston before it reaches the heads. The holes *i*, *i* allow the water to enter on the return stroke.

18. It can easily be understood that every elevator should be started and stopped gradually to avoid shocks, and that there always exists the danger of overthrowing the controlling device beyond the neutral point.

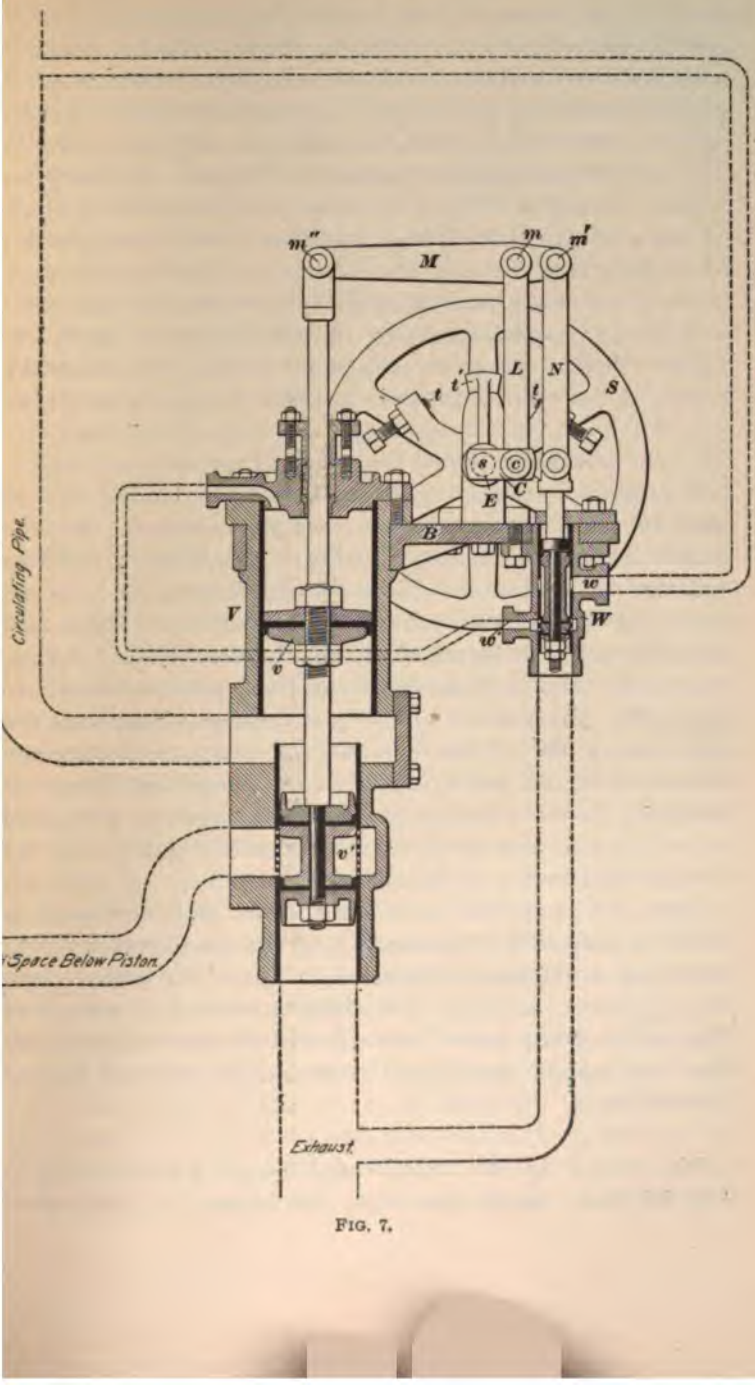
Referring to Fig. 5, it will be understood that when the piston is going down the car is ascending, and if the valve is suddenly closed, the flow of the water from the space below the piston through the discharge pipe is suddenly stopped. The momentum of the piston and car will tend, however, to continue the motion, resulting in a thud of the piston against the column of water thus confined. To avoid this **water ram,** as it is called, it is good practice to interpose in the discharge pipe between the cylinder and valve a **relief valve** r, as shown in Fig. 6, which is a drawing of an **Otis vertical elevator** of much the same design—with the exception of some details, to which we shall refer below—as that shown in Fig. 5. The danger of producing a shock by the careless handling of the operating device on the down trip of the car is not so great, inasmuch as the column of water above the piston is not confined in the cylinder on closing the valve, being always in communication with the supply pipe and through it with the pressure tank and its air cushion. A relief valve for the down trip is, therefore, deemed superfluous.

19. Pilot Valves.—For high-speed hydraulic elevators (600 feet per minute and more), the insertion of the relief valve is not sufficient to guard against shocks, it being extremely difficult to start and stop gradually by operating the main valves directly; nor is it possible to regulate the speed readily by opening the valve more or less, so that one of the most valuable features of the hydraulic elevator is curtailed. This has led to the introduction of the **auxiliary, or pilot, valve,** already referred to in Art. **9.** Such a valve as built by the Otis Elevator Company is shown in Fig. 7, of which the following is a brief description:

Contrary to the direct-operated valves shown in Figs. 5 and 6, the main valve V, Fig. 7, composed of the pistons v and v', is not balanced, but the upper piston v has a larger area than the lower double one v'; the valve is, therefore, also called a **differential valve,** there being always a pressure against the under side of the upper piston v depending on the difference between the areas of the pistons v and v'. On a bracket B fixed to the main valve casing is supported the auxiliary, or pilot, valve W, which is simply a piston valve of small dimensions; the casing of this valve has an inlet w connected with the circulating, or supply, pipe and



FIG



m''
m
m'
M
L
N
S
t
t'
t
s
c
E
C
B
V
v
w
W
w'
v''
Circulating Pipe.
Space Below Piston.
Exhaust.

FIG. 7.

an outlet w' connected by a pipe to the space above the upper piston of the main valve, as shown. In the position of the two valves shown in the illustration, the communication between them is shut off, the pilot-valve piston covering the outlet port. The upper space of the main valve is filled with water wholly confined, so that the tendency for upward motion of the piston v is checked in a position where the lower piston cuts off the circulation of the water, when, as we know, the elevator is at rest. By lowering the pilot valve, communication is established between the supply, or circulating, pipe and the space of the main valve above the piston v, which presents its whole area to the incoming water; as the upward pressure below it is less, owing to the difference between its area and the area of the lower piston v', it will descend with the effect of allowing a circulation of water from the top to the bottom of the cylinder, so that the car descends. If, now, the pilot valve be brought back into the position shown in Fig. 7 (the main valve being in its lowest position for the down trip of the car), it would check any farther downward motion of the main valve and the same would thus remain set for the down trip. Again, if the pilot valve were raised beyond the position shown in Fig. 7 (the main valve still being in its lowest position), the space above the piston v would be connected with the exhaust and the main valve would ascend and keep on ascending to the neutral position (elevator at rest) and beyond it (elevator descending), unless the pilot valve be brought back to the neutral point.

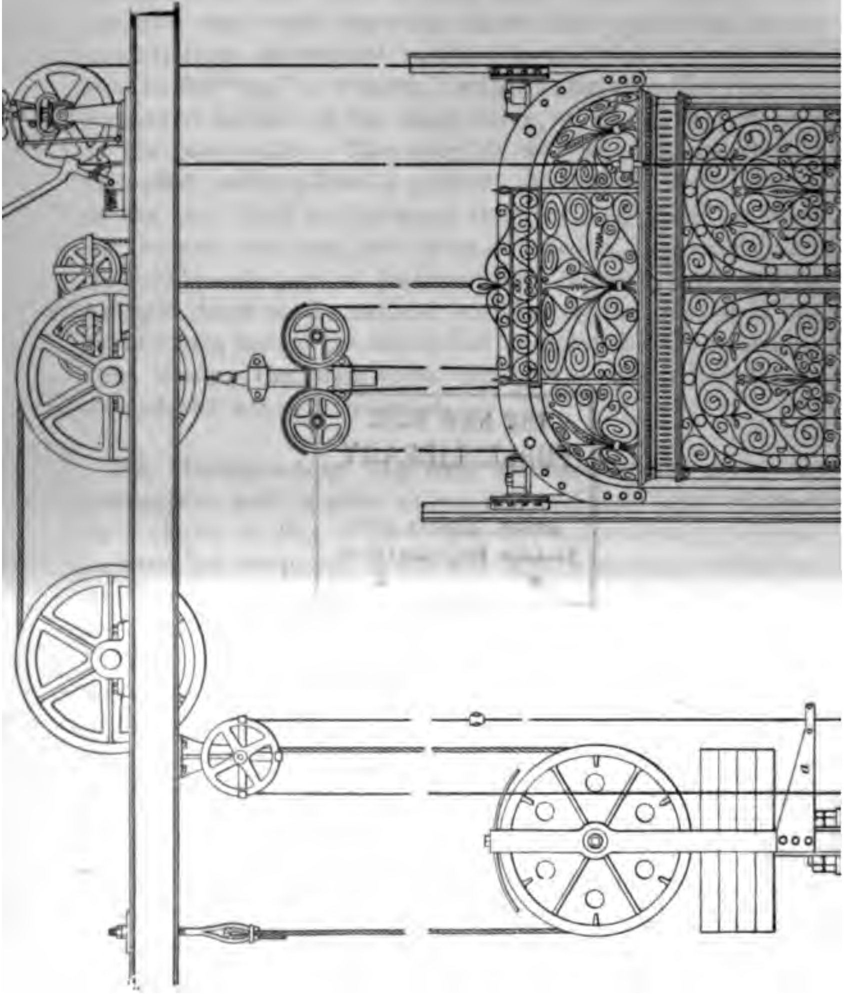
Thus, if no provision be made further than described, it would be necessary, in order to stop the car during a downward trip, to throw the pilot-valve operating device completely over, to wait until the elevator came to a stop, and then to throw the device into the central (neutral) position. The same complicated operation would be required for the upward trip.

20. To avoid the complicated operation mentioned in Art. **19,** the two valves are so connected by a system of linkwork that the pilot valve closes automatically without affecting the operating device in the car (shipper rope, lever, or hand wheel) when the main valve reaches its extreme upper or lower positions. This is brought about in the following manner: The shipper sheave *S* is mounted on the bracket *B*, its shaft *s* carrying a crank *C*, the crankpin *c* of which is connected to a double-armed lever *M* by a link *L* and a pin *m*. To the right of the pin *m* is another pin *m'* that serves as a pivot for a link *N*, which is connected at the other end to the stem of the auxiliary valve. A third pin *m''*, to the left of the pin *m*, connects the lever *M* with the main-valve stem. Stops *t*, *t*, *t'* on the shipper sheave and its stationary bearing, respectively, limit the motion of the crank *C*.

The operation is as follows: Starting, as before, from the position of the valves shown in Fig. 7, the piston *v* is held stationary between the water pressure from below and the confined water above, so that the pin *m''* forms the pivot of the lever *M* when we move the shipper sheave to the right. The crank *C* then pulls down the lever and with it the pin *m'*, link *N*, and the pilot-valve stem, thus lowering the pilot valve to the position in which it admits water into the main valve, which then moves downwards. As soon, however, as it commences to move, it raises up the pilot valve, the crankpin *c*, as well as the link *L* and the pin *m* being now stationary, which latter then serves as the pivot for the lever *M*. The leverage is so proportioned that by the time the main valve has reached its lowest position the pilot valve will be closed, that is, it will have returned to the position shown in Fig. 7, checking further motion of the main valve, the crank *C*, however, remaining in its lowest position. If it is now desired to stop the car during its down trip, the sheave, and with it the crank, is brought back to the neutral position. The pin *m''* being, now, once more the pivot for the lever *M*, the pilot valve is raised above its neutral position, the main valve rises, and by the time it has risen far enough to shut off circulation of the water it has dragged the pilot valve back to its neutral position. All parts are now again placed as shown in Fig. 7 and the cycle may be repeated.

21. Though it takes many words to describe it, the operation of this valve is very simple and reliable. The operator may with impunity throw the operating device quickly from its neutral position to the right or the left, that is, for "up" or "down," without affecting the gradual, measured motion of the main valve, which is the purpose of the pilot valve. Moreover, it will be understood that the pilot valve allows a perfect regulation of the speed of the car. For by throwing the operating wheel or lever on the car over only part of its full swing, the pilot valve will make only part of its travel and, consequently, will be brought back to its neutral position by the action of the main valve before the latter has completed its full stroke, thus leaving the main valve but partly open, whereby the flow of the water is throttled.

22. Independent Top and Bottom Stop-Valve.—In connection with a pilot valve, the ordinary kind of limit stop shown in Fig. 5 operating the valve directly cannot be used, for the piston or car will still be moving, while the quick move of the pilot valve has long been completed. It becomes necessary, then, to introduce an independent valve for stopping the car at its limits of travel. Such a valve is shown in Fig. 8 at *Q*, and its construction and operation are as follows: Into the passage leading from the space below the elevator piston to the exhaust, a cylindrical shell *q* having three passages is inserted, of which the upper passage leads to the relief valve (see Art. **18**). Either of the two passages *t* and *t'* may be closed by the rotary valve, shown to an enlarged scale in Fig. 9, which consists of a spindle *s* passing through stuffingboxes of the valve casing and carrying a valve body *v* composed of a sleeve and flanges fitting the inside of the shell *q*. The flanges of the valve body are notched out to receive the valve proper *w*, which fits with considerable play in the notches, as shown



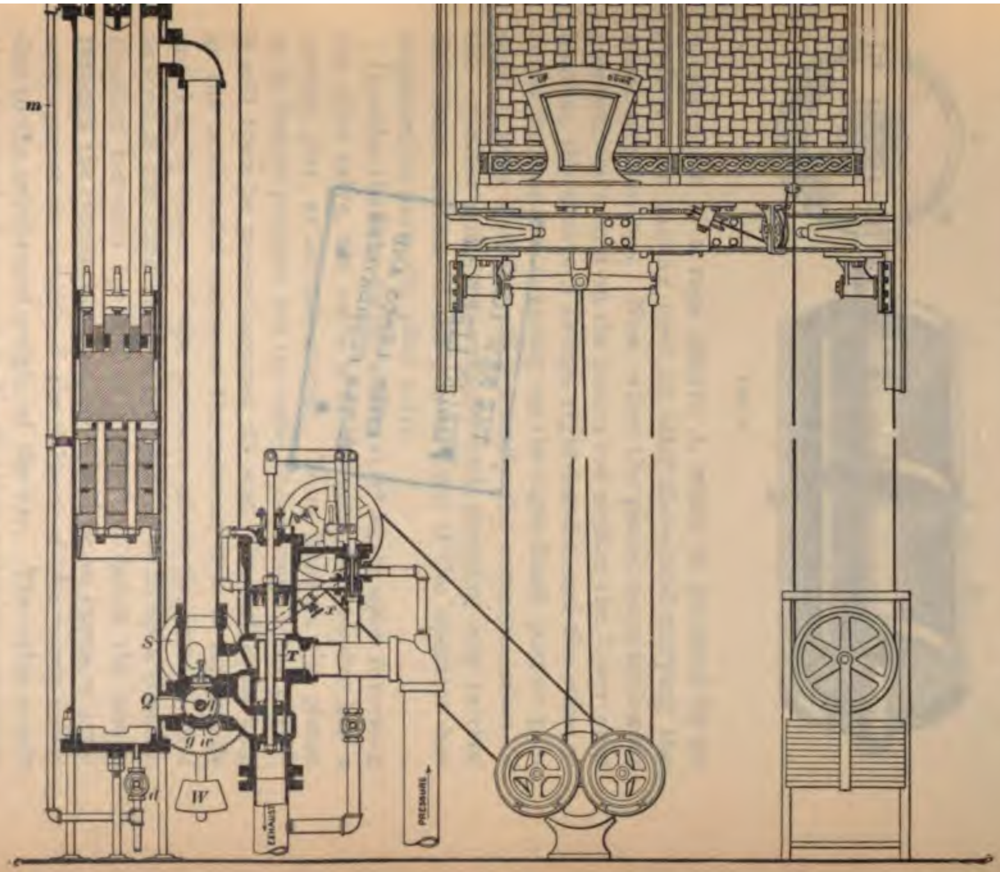


FIG. 8.

in Fig. 9. The valve spindle carries on the outside of the casing a gear-wheel *g*, Fig. 8, actuated by a weight *W* that tends to keep the valve in the neutral position shown in Fig. 8. The gear *g* meshes with a smaller gear attached

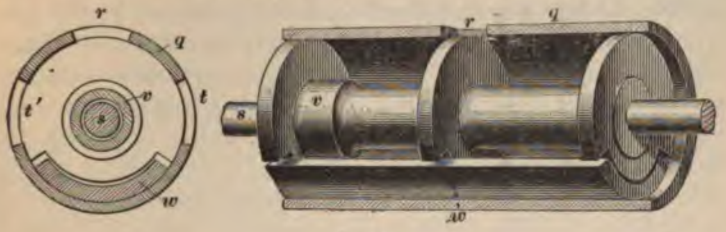


FIG. 9.

to the shaft of a rope sheave *S*, which is actuated by an endless rope passing over an idler above and carrying the usual stop buttons. Now, when the piston nears its lowest position, the arm *a* on the piston rod strikes the lower stop button; the sheave *S* swings right-handed and the valve *w* turns left-handed, covering up the right-hand port or passage *t*, thus shutting off gradually the communication of the cylinder with the exhaust and stopping any farther downward motion of the piston, even if the operator has neglected to move the pilot valve.

In order that the elevator may start again upon reversing the pilot valve, the valve *w* of the rotary stop-valve has a certain play, as already stated. Thus, imagine the piston in its lowest position and the valve *w* covering up the port *t*; it will then be pressed against its seat (the shell *q*) as long as the main valve uncovers the exhaust or is in the middle position, by reason of the pressure above the piston. But as soon as the main valve is reversed, so as to open communication between the spaces above and below the piston through the circulating pipe, there will be an excess of pressure against the outside of the valve *w* of the rotary valve, due to the unbalanced weight of the car. The valve *w* will then be lifted off its seat and will allow water to pass below the piston, which then commences to rise. Presently, the arm *a* will leave the lower stop button and the rotary valve will swing back to its neutral position by virtue of the weight *W*, Fig. 8. Similar action takes place at the extreme upper position of the piston.

23. Throttle.—A difference will be noticed in the construction of the main valve between that shown in Fig. 7 and that shown in Fig. 8, there being interposed in the latter case a metal sleeve between the upper single piston and the lower double one. This sleeve, which is called the **throttle** and is designated by *T*, Fig. 8, is fastened to the valve rod, or stem, and in its neutral position shuts off the supply from the space between the valve pistons. Otherwise, the connections and passages are the same, the supply being in constant communication—except when shut off by the throttle—with the circulating pipe by two branch passages leading from the annular chamber around the throttle to the circulating pipe. In order to show this clearly, a horizontal section through the middle of the throttle and its casing is given to an enlarged scale in Fig. 10 (*a*).

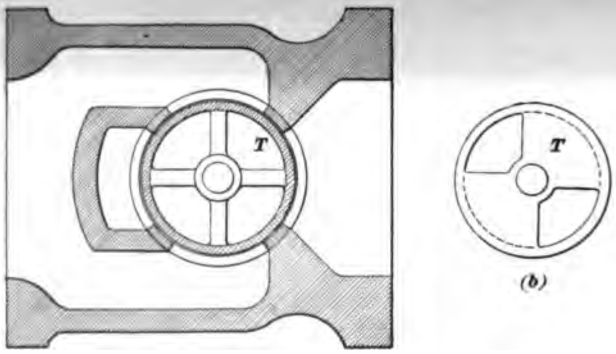


FIG. 10.

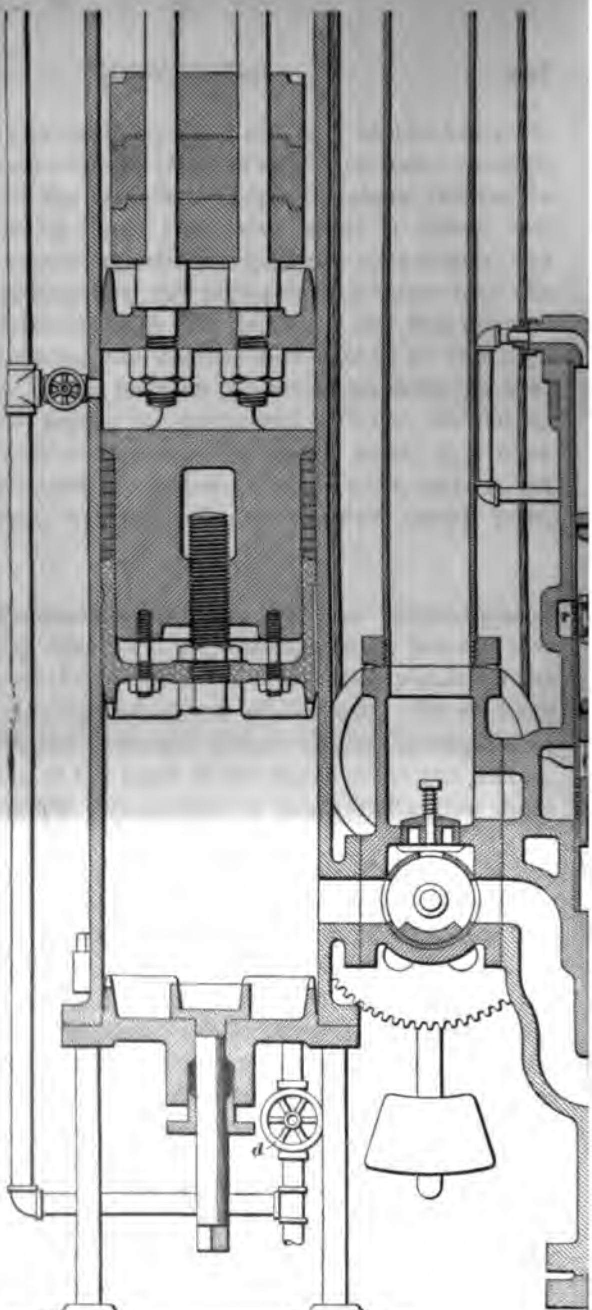
The purpose of the throttle is a threefold one. (1) It serves, if carefully adjusted, to deaden the noise occasioned by the circulating water. (2) It serves as a brake while descending, in case of an extra load on the car, preventing it from attaining undue speed. This is accomplished by the top of the throttle sleeve being partly closed, as shown by the plan view given in Fig 10 (*b*), thus allowing only a small amount of water to pass through, that is, throttling it. (3) If any pipe or connection between the supply and valve should break, the water cannot back up from the circulating pipe out through the supply port faster than it can leak around the outside of the throttle.

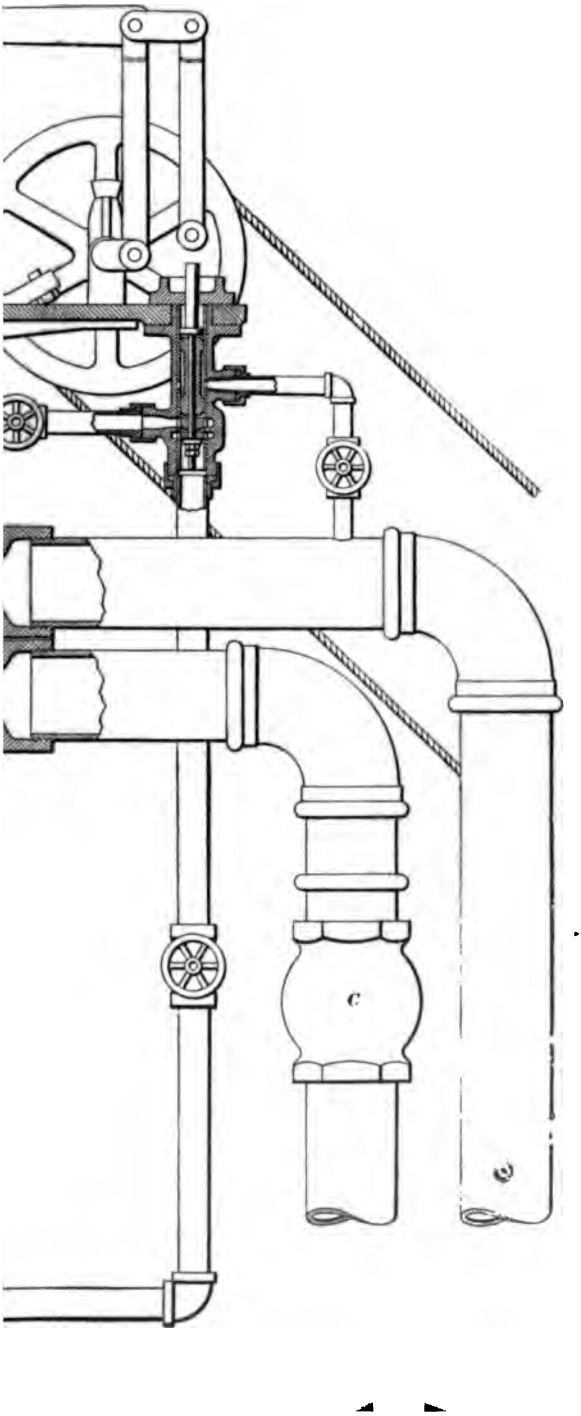
24. The throttle is but loosely fitted to its seat, or lining, so that there is always some leakage around it, otherwise the elevator could not be started from its position of rest, since there would be no outlet for the water between the large and small piston of the differential valve while descending, and to the inlet while ascending. This leakage is sometimes solely depended on to give the differential valve the initial start, but oftener a by-pass pipe x, Fig. 8, leads from the supply chamber of the pilot valve to the space under the upper piston of the differential or main valve. This by-pass pipe is provided with a globe valve, by means of which the rapidity of the initial start can be regulated.

25. Double-Power Vertical Hydraulic Elevator.—In modern office buildings safes and other heavy furniture are frequently moved about, and one of the elevators in such a building is, therefore, generally designed for a much heavier load than the others. The necessary power is obtained from an extra-high-pressure tank. In order, however, that this elevator may be used for ordinary loads as well, with no greater expense than the others, a special valve is used that permits it to be used at will either with the ordinary low pressure or with the high pressure. Such a valve, built by the Otis Elevator Company, is shown in Fig. 11. The upper valve v is, in this case, a piston valve straddling in its neutral and lowest position the high-pressure port. The throttle T has ports t, t. When the valve stem is moved down, the water circulates as in the ordinary hydraulic machine and the car descends. When the valve stem is moved up, the discharge is opened, as in the ordinary machine, and as the low-pressure inlet is connected to the top of the cylinder through the ports *t*, *t* and the circulating pipe, it causes the car to ascend. For heavy loads the valve stem is raised still farther until *T* comes opposite the high-pressure inlet *r*, and this opens the passage for the high-pressure water into the top of the cylinder through the ports *t*, *t* and the circulating pipe, thus giving the car the full benefit of the high pressure. Since in this position of the valves both the low- and high-pressure supply are connected with the circulating pipe and thus with each other, the water would flow from the high-pressure tank to the low-pressure tank, were it not for a check-valve *C* inserted in the low-pressure supply pipe, as shown.

26. Non-Circulating Systems.—In the vertical-piston elevators thus far described, the distinguishing feature was the circulation of the water from above the piston to the space below it during the ascent of the car. As we have seen in Art. **16**, the principal object of this arrangement was the balancing of the head of the water above the piston. Incidentally, certain advantages in counterbalancing were obtained by it also (see Art. **15**).

It is considered in practice that a ratio of 6 : 1 is the limit for vertical machines. In certain designs, however, the ratio has been carried much above this value for the purpose of making the cylinder very small. Now, since the head of the water becomes less and less the shorter the cylinder is made, it becomes unnecessary to balance it when the ratio is very high, say 10 : 1, for instance. In such cases the circulating pipe is dispensed with; the water then enters and leaves on one side of the piston only and one end of the cylinder is left open to the atmosphere. Fig. 12 shows an elevator of this kind made by The Whittier Machine Company, of Boston. There are quite a number of these machines in operation. The ratio of the particular machine illustrated is 10 : 1, there being a set of five fixed and five traveling sheaves on each side of the cylinder;


d


C

from each of the two sets a rope passes to an overhead sheave and thence to the car. The fixed sheaves are arranged below the traveling sheaves, the latter being attached to a crosshead carried on the piston rods and guided on rails *R*, *R*. The piston moves *up* for the ascent of the car and *down* for the descent, so that the piston rods are in compression. Moreover, the piston moving in the same direction as the car cannot, as in the case of the previously described vertical machines, be utilized as a counterweight, but must itself be counterbalanced, which is done in the manner shown in the illustration, *W*, *W* being the weights. The controlling valve is much of the same construction as that shown in Fig. 5.

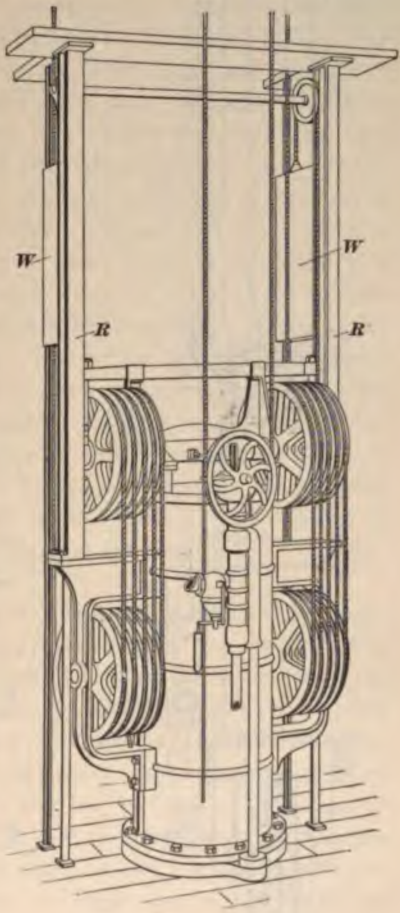


FIG. 12.

27. While in high-ratio vertical elevators of the kind shown in Fig. 12 the circulation of water is dispensed with, owing to the small head of the water, it becomes entirely dispensable when the cylinder is placed horizontally. All horizontal hydraulic piston elevators are, therefore, based on the non-circulating system.

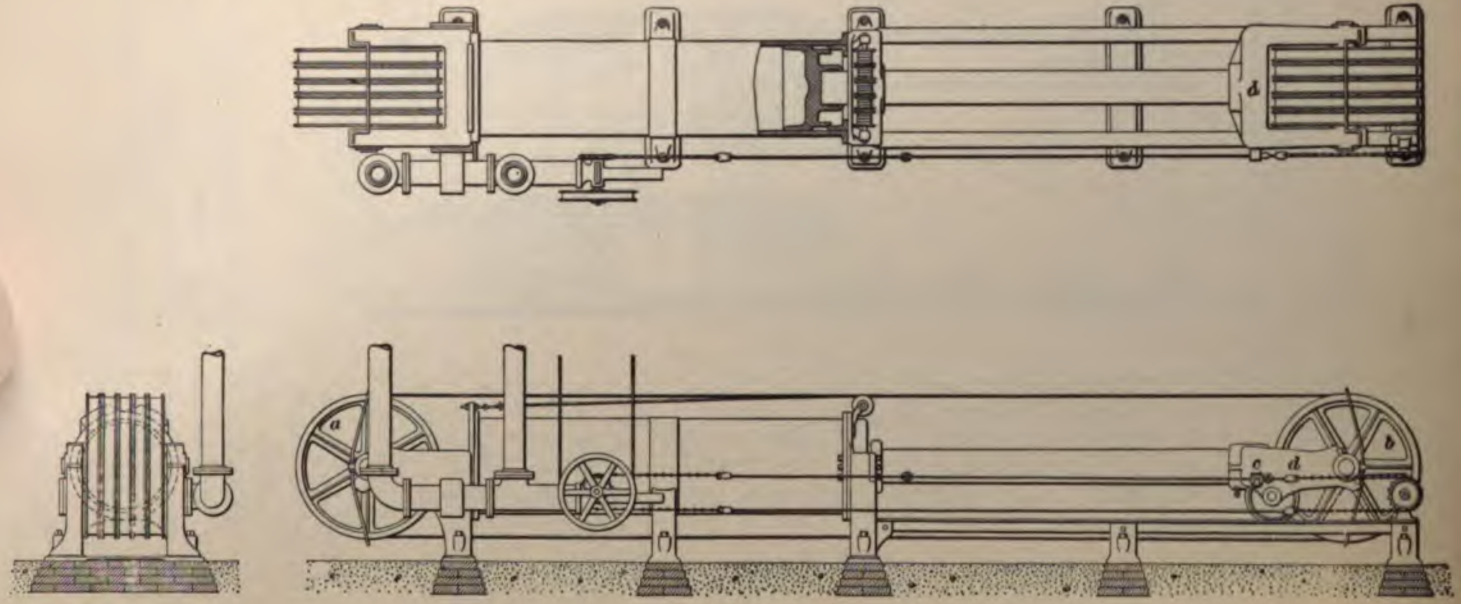


FIG. 13.

HORIZONTAL HYDRAULIC PISTON ELEVATORS.

28. Advantages.—Although the floor space occupied by a vertical elevator cylinder is comparatively small, this floor space is required on each floor of the building, and where there are a number of elevators, the aggregate necessary space amounts to more than can in many instances be conveniently spared. Moreover, it becomes necessary, in case of a battery ot elevators, to provide a separate well for the cylinders. Again, the long, upright cylinders so placed in a comparatively narrow well are inaccessible for the greater portion of their length. For these reasons preference is given to the horizontal type of elevator when there is sufficient floor space more available in the basement of the building than on the floors above. But under the most favorable conditions, floor space, even in the basement, is always limited, and it is desirable, therefore, to make the cylinders short, which necessitates a high ratio of the transmitting devices. This is generally chosen as 10 : 1. The sheaves in these machines are arranged either so as to put the piston rod in compression or so as to put them in tension.

29. Compression Type.—A simple machine of the compression type, built by Morse, Williams & Co., is shown in Fig. 13. The fixed sheaves are placed at the rear end of the cylinder and the hoisting rope is carried above and below the cylinder from the fixed sheaves *a* to the traveling sheaves *b* back and forth and is finally led off from the former to the car. The drawing calls for but little explanation. The controlling device consists of the three-way valve illustrated in Fig. 5; the motor safeties are limit-stop buttons carried on an endless chain or rope and actuated at the extreme positions of the piston by an arm or projection *c* on the crosshead *d*. The endless chain runs over a sprocket wheel fastened to the shipper-sheave shaft.

30. Tension Type.—The general arrangement of the tension type of horizontal hydraulic machines is shown in Fig. 14. Both the fixed and the traveling sheaves are

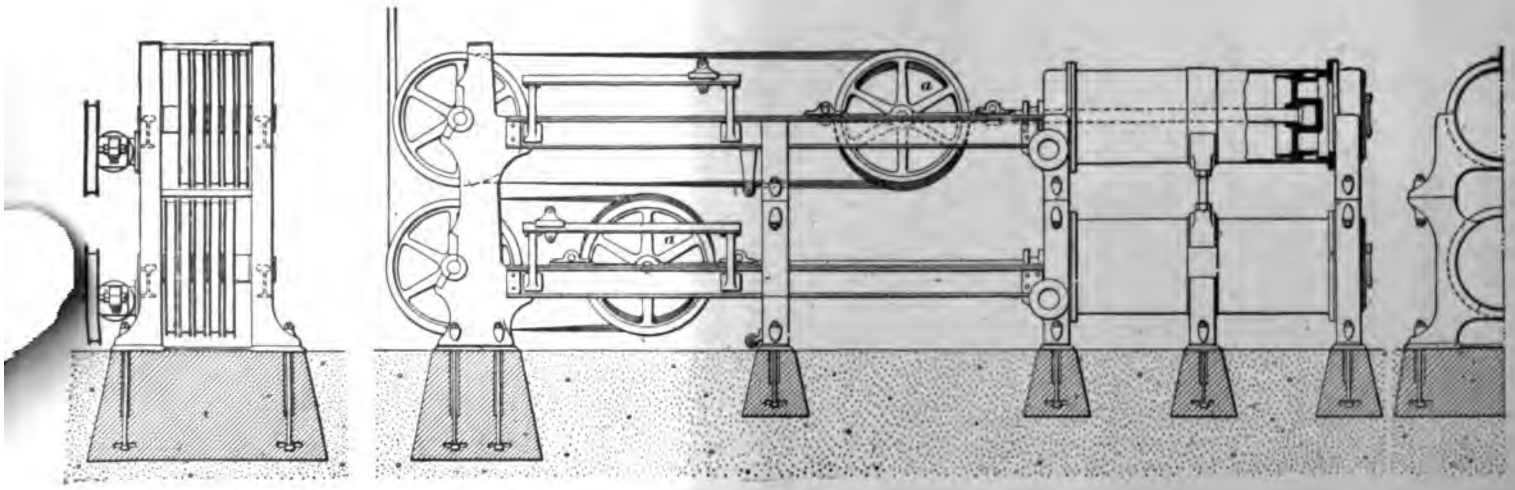


FIG. 14.

located at the front end of the cylinder. It will be noticed that the traveling sheaves *a*, *a* are mounted in the crosshead at an angle to the horizontal plane. This is necessary in order that the ropes shall not "ride" off the grooves when the two sets of sheaves come close together at the end of the stroke. This precaution is deemed unnecessary in the compression type, the sheaves being always apart a distance greater than the length of the cylinders.

There are several advantages in the tension type of machines: (1) The piston rods can be considerably smaller. (2) The distance between the fixed and traveling sheaves is smaller, being only about one-half as long as that in the compression type; this is an item of importance when the fact is taken into consideration that teetering of the car is often due to the whipping of the ropes in horizontal machines, which action increases as the distance between the sheaves becomes greater. This action, by the way, is absent in vertical elevators. The whipping of the ropes is reduced as much as possible by supporting rollers shown in Figs. 13 and 14. In the tension type these rollers are supported on a shaft that again rests on guide shoes traveling on rails.

31. The compression type of horizontal elevators has the advantage that no stuffingbox is needed for the piston rod, the water entering behind the piston only. The front end of the cylinder generally has a simple yoke through which the rod passes.

When there is more than one elevator in a building, the cylinders are preferably mounted in pairs on top of each other; such a pair is then called a **double-deck machine,** and this arrangement is shown in Fig. 14.

32. Fast-Service Compression-Type Elevator.—Fig. 15 is an illustration of an elevator machine of the compression type built by the Otis Elevator Company, of Chicago (formerly the Crane Elevator Company). This machine is intended for fast passenger service and is therefore

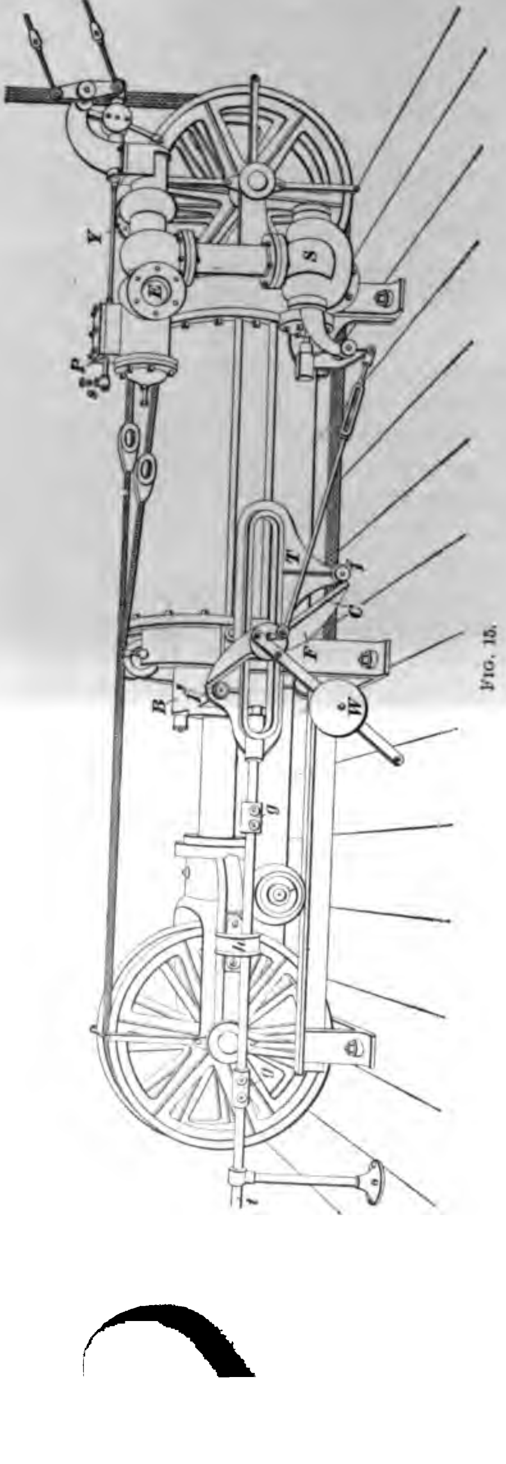


FIG. 15.

fitted with a pilot valve *P*, involving the same principles as the Otis valve described in Arts. **19** and **20**, and has an automatic stop-valve *S*.

33. The pilot valve, main valve, and stop-valve are shown in detail in Fig. 16. The pilot, or auxiliary, valve is a slide

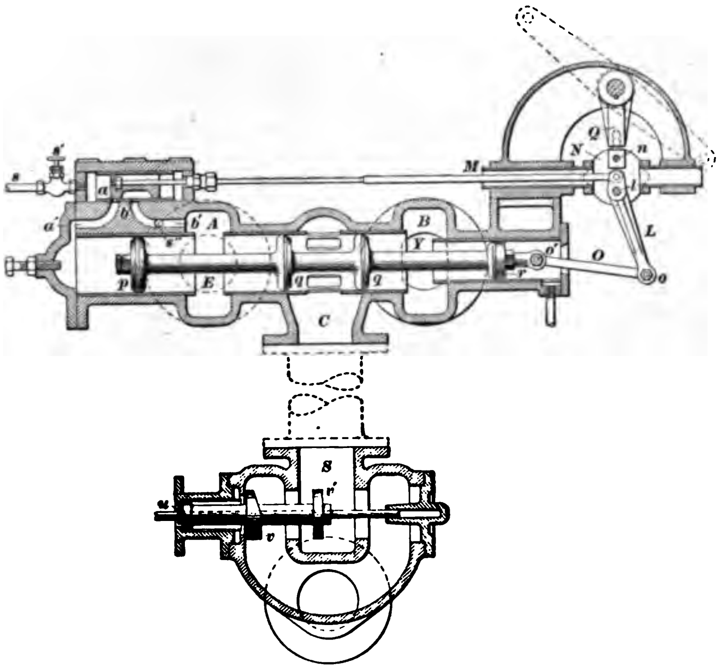


FIG. 16.

valve; its seat has two ports *a* and *b* opening into passages *a'* and *b'* of the main-valve casing. The passage *a'* leads into a space of the main valve behind a piston *p*, while the passage *b'* communicates with a chamber *A* in front of that piston, which chamber, in turn, is connected with the exhaust pipe *E*, Figs. 15 and 16. Of the two other chambers *B* and *C* of the main-valve casing, *B* is connected to the supply pipe *Y*, and *C* to the cylinder by way of the valve S. The main valve consists of two single pistons and one double piston: the piston p, already mentioned, the double piston q, and the piston r, the latter being of smaller diameter than the others. The valve chest of the pilot slide valve is connected with the supply by a pipe s.

The operation of the valves is as follows: In the position shown the valves are at rest, the port a being closed and the pressure on the piston q towards the left (due to the difference in area of q and r), thus acting against a body of water confined in the space behind the piston p. If the pilot valve is moved towards the right, it uncovers the port a and water under pressure enters the space behind the piston p; the area of this piston being greater than the difference of the areas of q and r, it moves towards the right, thus connecting the chambers A and C; that is, connecting the cylinder with the exhaust, and hence the elevator car moves down. If the pilot valve is moved from its neutral position and to the left, the passages a' and b' are connected; that is, the space behind the piston p is put into communication with the exhaust. The excess pressure due to the difference of areas of q and r then causes the pistons to move to the left, opening communication between the chambers B and C; that is, it connects the cylinder with the supply, and hence the elevator car moves up. The speed with which the main valve responds to the pilot valve is regulated by the valve s' in the supply pipe s on one hand and furthermore by a screw s'' that can be made to enter more or less into the passage b' by turning it from the outside.

For the same reason that was given in connection with the Otis pilot valve, Art. **19**, the pilot valve must return automatically to its neutral position. The mechanism that accomplishes this is similar to that used in the Otis valve. The valve stem of the pilot valve is connected to the short arm of a two-armed lever L, which is pivoted at l to the central double disk-shaped piece N of a sliding sleeve M. The long arm of the lever L is connected by means of a link O to the stem of the main valve. The central piece N is connected at n with a one-arm lever Q, the shaft of which is operated by a lever or sheave actuated by a shipper rope from the car. When the lever Q is thrown to the left, the sleeve M moves to the left, carrying the lever L and the pilot-valve stem with it, the point o' at which the link O connects with the main-valve stem being the pivotal point of the motion. As soon as the main valve commences to move to the left, that is, after the pilot valve is set by the shipper rope, the point l becomes the pivotal point, and the pilot valve is pulled back to its original position. Similar action takes place when the lever Q is moved towards the right.

34. The action of the automatic stop-valve S is as follows: The valve has three pistons v, v', and u, of which the first two serve to close the circular openings leading from the inlet to the outlet. The piston u is at all times actuated by whatever pressure there is on the cylinder, forcing it to the left and thus keeping the circular openings referred to open. The valve stem is connected by a lever and rod to a cam F, Fig. 15, pivoted to the frame of the machine. This cam is ordinarily held, as shown in the illustration, between two rollers f and f' by means of a weight W attached to it. The rollers f and f' are placed on a movable frame T guided horizontally as shown and called the **tappet.** On the guide rod t of this tappet are fastened the limit-stop buttons g, g to the right and left, respectively, of a projection, or arm, h on the crosshead of the traveling sheaves. In either of the extreme positions of the crosshead, the arm h comes in contact with one of the buttons, pushing the tappet T and thus operating the stop-valve and shutting off the communication between the main valve and cylinder.

The valves v and v', Fig. 16, are not fitted very closely, so that there is a certain small amount of leakage, which enables the valve to start back slowly as soon as the pilot valve and main valve are reversed; as soon as the arm h, Fig. 15, leaves either of the buttons g and g, the weight W causes the valve to open quickly and wide. In case the leakage around the valves v and v', Fig. 16, proves too slight, a small direct pipe connection (not shown) is made between the middle chamber C of the main valve and the top of the cylinder at the closed end. This allows a small quantity of water, which is regulated by a stop-valve, to enter or leave the cylinder independently of the automatic valve S when the pilot valve is reversed so as to give the valve S the start. This pipe connection also serves the purpose of permitting the escape of air that may have accumulated in the cylinder.

35. The hydraulic elevators described are by no means the only ones that are made or that are in operation. They are typical constructions, however, and a person will, if their principles are clearly understood, readily comprehend other designs as well.

PUMPS, TANKS, PIPES, AND FIXTURES.

GENERAL ARRANGEMENT.

36. In cases where a natural water supply or a street main with sufficient pressure is available, the elevator may be directly connected with it. Such cases are rare, however, and therefore a pumping plant is almost always included in an elevator installation. This pumping plant consists usually of one or more pumps, a pressure tank, and a discharge tank suitably connected by piping provided with the necessary valves and other fixtures.

37. A typical installation of an hydraulic elevator is shown in Fig. 17. The pump P takes the water from the discharge tank D and forces it into the pressure tank T, whence it enters the elevator cylinder C through the supply pipe S. It leaves the elevator cylinder through the discharge pipe t, which carries it back to the discharge tank.

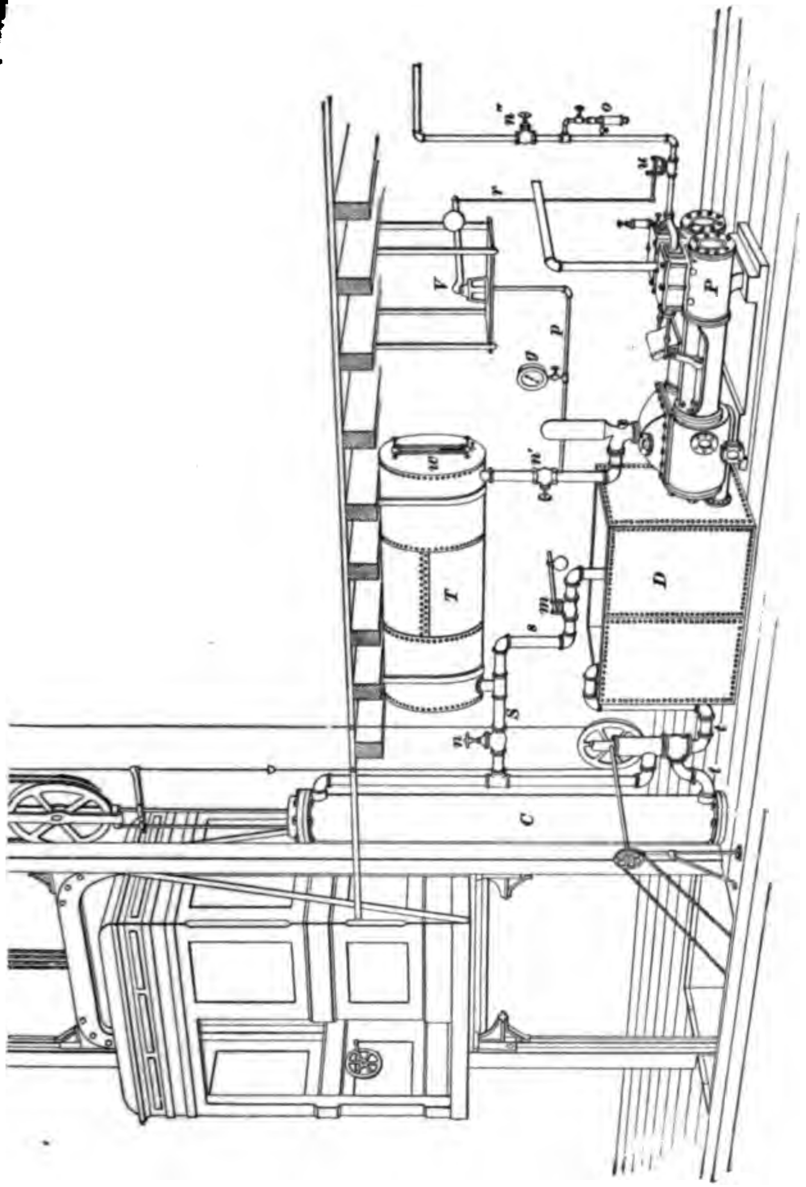


FIG. 17.

The water is thus used over and over again; this is an important item where water rates are high, as is the case in most cities and towns.

PUMPS.

38. Since, with the usual arrangement of pumps, cylinders, and tanks, the pump may work continually while the cylinder takes a quantity of water out of the pressure tank only for every other (the up) trip of the car, the pump need be only large enough to supply the average quantity of water per unit of time, supposing the cars to be running continuously up and down. Since there is more or less interruption of traffic, the pumps will generally even then supply more water than is necessary and will have to be stopped and started frequently. For such intermittent service duplex steam pumps or electric pumps are most suitable and are, hence, generally used, although geared pumps, belt-driven pumps, and gas-engine power pumps are occasionally met with.

TANKS.

39. Open tanks, formerly installed in great numbers on the roofs of buildings to furnish the necessary head, are gradually disappearing, and the closed pressure tank, as *T*, in Fig. 17, placed in the engine room, takes its place almost exclusively. Such closed pressure tanks are often placed at the top of the building also, thus utilizing both the natural head and the air pressure. In such a tank the required water pressure is obtained by having the tank partly filled with air and compressing the same by pumping in the water, so that it is really air pressure that gives to the water the necessary head. By leakage as well as by absorption in the water the quantity of air in the tank gradually grows less and must be renewed. In the smaller installations, such as is shown in Fig. 17, the necessary air supply is obtained through a vent in the suction pipe of the water pump; in large installations separate air pumps are provided for the purpose.

ACCUMULATORS.

40. The pressure used in ordinary closed-tank installations ranges generally between 90 and 120 pounds per square inch. In some cases for high buildings these pressures have to be exceeded, and then hydraulic accumulators are installed instead of the pressure tanks. These high-pressure installations require also different designs of cylinders and other parts of the plant, but since there are but comparatively few of these installations in operation we shall forego treating them in detail.

AUTOMATIC STOPPING AND STARTING DEVICES FOR PUMPS.

41. Kinds of Starting Devices.—The stopping and starting of the pumps are effected automatically by various devices. In one kind of these devices the height of the water in the tank is made use of by means of a float to operate the steam valve of a steam pump or the switch and rheostat of an electric pump; in another kind, the pressure in the tank is utilized to do the same thing by means of a pressure valve. Floats are used only with open gravity tanks.

42. Pressure-Regulated Starting Valve.—A device of the second of the above-named classes is shown in Fig. 17 at *V*. It consists of a pressure valve of much the same construction as a steam-boiler safety valve. It is connected to the pump discharge pipe or directly to the pressure tank by a small pipe *p*, into which is inserted a pressure gauge *g*. The weighted lever of the valve *V* is connected to the throttle valve *u* of the steam pump by a rod *r* in such a manner that the throttle valve shuts off steam when the weight on the lever of the valve *V* is balanced by the required water pressure in the pressure tank, and opens to admit steam when the pressure falls below the required amount. A sight-feed oil cup *o* is generally placed in the steam pipe in advance of the throttle valve *u*, in order to insure the proper working of the same and to prevent it from sticking.

43. Ford Regulating Valve.—In the device shown in Fig. 18, the two valves *V* and *u* spoken of in the previous article are combined into one. This device, which is largely used in elevator work and is manufactured by Thomas P. Ford, of New York, consists of a spring-actuated steam valve *U* and a water piston *V* moving in a little cylinder *W* under the influence of the water pressure. It is easy to see that by adjusting the spring properly the steam valve can be made to close when the water pressure on the piston *V* exceeds a certain required amount. The regulating valve should be placed in the steam supply pipe in a vertical position between the steam chest and an ordinary throttle valve. The oil cup should be placed so as to allow the oil to pass through the regulating valve. The pipe connecting the pressure tank with the pressure cylinder of the regulating valve should be provided with a globe valve and a union next to the valve, in order that the cap may easily be removed for repacking the piston *V*. A drip pipe should be connected with the bottom of the cylinder *W*.

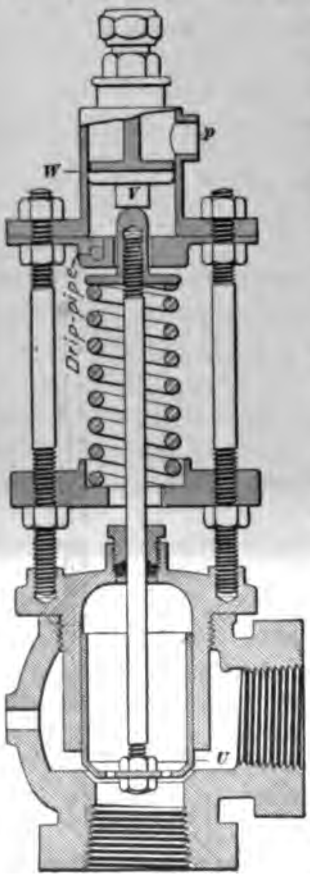


FIG. 18.

44. Ford Rheostat Regulator.—A device much used in connection with electric pumps and manufactured by

Thomas P. Ford is illustrated in Fig. 19. The purpose of this apparatus is to obtain a comparatively large movement, which is necessary for operating the switch and rheostat of the electric motor.

As in the apparatus shown in Fig. 18, the pressure pipe is connected to a small cylinder *W* in which works a piston *V*

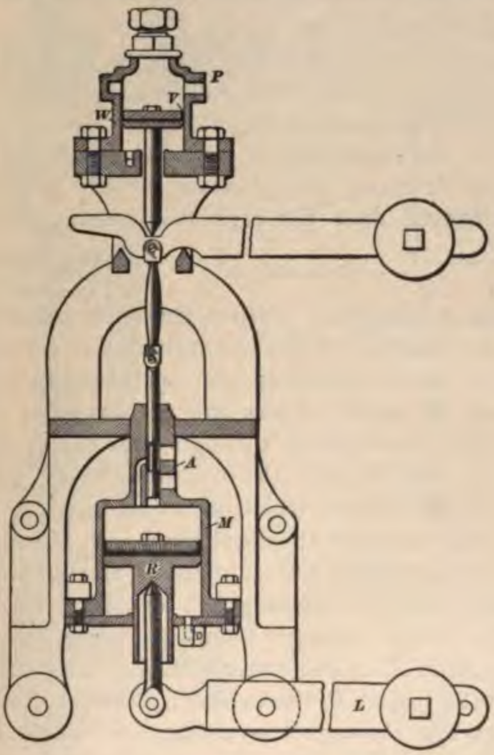


FIG. 19.

against a weighted lever. This lever is, however, not connected directly to the stopping and starting arrangement, but to the piston of an auxiliary hydraulic valve *A*. This valve has an inlet connected to some constant water supply of moderate pressure (not less than 35 pounds per square inch) and a discharge outlet. When the pressure in the tank falls below the required amount, the piston V rises and carries with it the piston of the auxiliary valve; water is then admitted into the cylinder M of the main valve, causing the piston R therein to be forced down and the outward end of a long double-armed lever L attached thereto to be forced up. This lever is also weighted and to it is attached the lever of a motor starting box. As soon as the pressure in the tank increases, the piston V moves down; by this movement the cylinder M is put in communication with the discharge, whereupon the main-valve piston moves up and the end of the lever L down by virtue of the weight attached to it. It is recommended in connecting up this valve to have the water from the constant supply go through a mud-drum placed near the regulator before entering the same.

45. Mason Elevator Pump-Pressure Regulator.—Fig. 20 shows a regulating device much used in elevator work. Referring to the illustration, the operation is as follows: Steam from the boiler enters the regulator at the point marked "inlet" and passes through into the pump, which continues in motion until the required water pressure is obtained in the system, which acts through a $\frac{1}{4}$-inch pipe connected at a and upon the diaphragm D. This diaphragm is raised by the excess water pressure and carries with it the weighted lever L, opening the auxiliary valve A and admitting the water pressure from the connection b to the top of the piston P, at the same time opening the exhaust port under the piston P, thus allowing the water under this piston to escape through the passage a' shown in dotted lines into the drip pipe d, thereby pushing down the piston, which closes the steam valve and stops the pump.

As soon as the pressure in the system is slightly reduced, the lever L, on account of the reduced pressure under the diaphragm, is forced down by the weights W, carrying with it the auxiliary valve A and thus opening the exhaust from the top of the piston, and at the same time admitting the water pressure under this piston, which is now forced up and opens the steam valve, again starting the pump. This action is repeated as often as the pressure rises above and falls below the required amount.

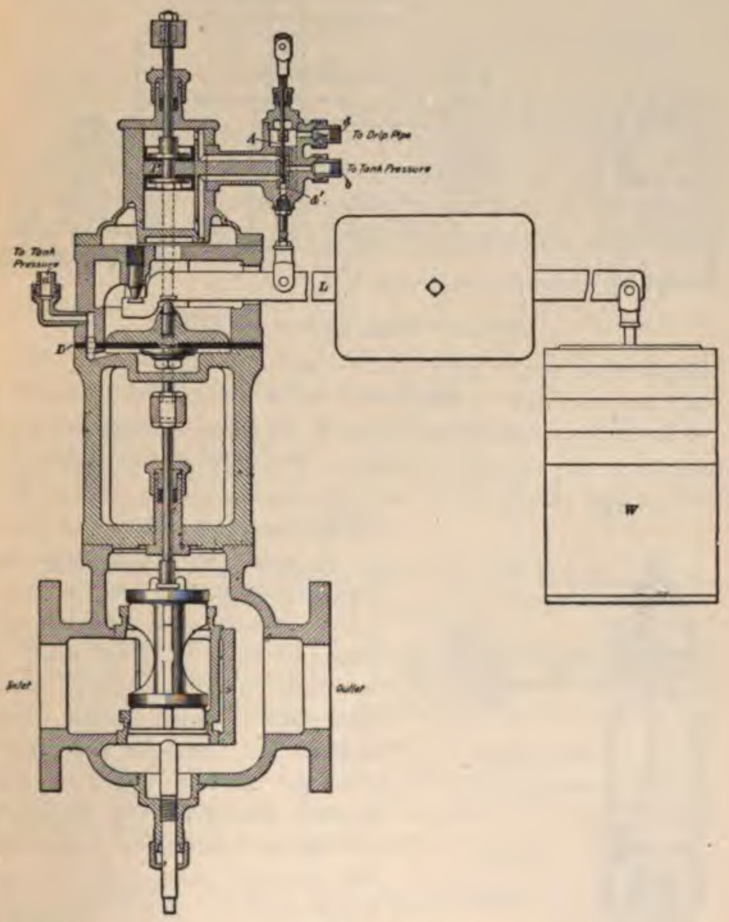


FIG. 20.

46. The Mason regulator is easily adapted for use in connection with a switch and rheostat for regulating electrically driven pumps. Fig. 21 shows such an arrangement comprising a solenoid rheostat and snap switch as made by the Elektron Manufacturing Company, a Perret motor, and a Mason regulator.

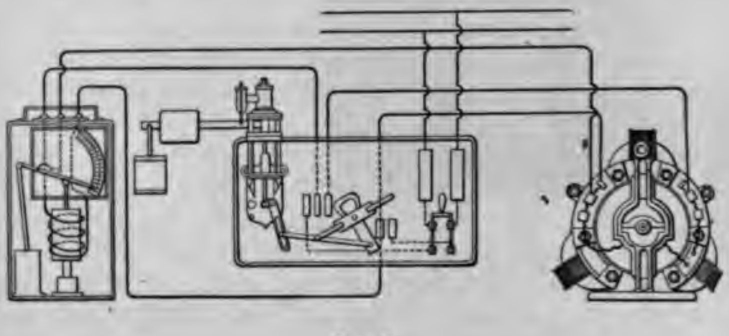


FIG. 21.

BY-PASS VALVE.

47. When the elevator service is quite continuous and regular it proves advantageous in many cases, especially with pumps driven electrically or by gas engines, to have the pump run continually and thus to do away with the more or less complicated automatic-valve switch and rheostat arrangements. In such cases a **by-pass valve** is installed near the pump, which opens communication between the delivery and suction pipes of the pumps whenever the pressure in the tank becomes excessive. By elevator men, such an arrangement is called a **closed system.**

FIG. 22.

Fig. 22 is an illustration of the Ford by-pass valve. Its construction is similar to that of the regulator described in Art. **44,** and it is connected up in the same manner, a mud-drum being preferably placed near the valve to free the water from any impurities before it enters the auxiliary valve.

SAFETY VALVE.

48. To provide for the emergency, should the regulating devices described in the previous articles stick, and should an excessive pressure accumulate in the tank, a **pipe** *s* fitted with a **safety valve** *m* (see Fig. 17) and leading from the pressure tank to the discharge tank is generally provided.

WATER GAUGE AND VALVES.

49. Besides the fixtures already mentioned, there is provided a water gauge *w* on the pressure tank and various globe valves *n*, *n'*, and *n''*, Fig. 17, which are used in starting and stopping the plant.

OPERATION AND MAINTENANCE OF HYDRAULIC ELEVATOR PLANTS.

50. Water.—The water to be used in hydraulic elevators should be clear and free from sediment. It should enter the system through a strainer, so as to exclude all foreign matter likely to damage the valves and pistons. The water should be changed at least every three months and the whole system should then be cleaned by washing and flushing. This requires closing down the plant completely.

51. Starting Up and Running.—With all parts supposedly in good working order, joints tight, stuffingboxes, pistons, and valves properly packed, guides, sheaves, and other moving parts well oiled, start the pumps and partially fill the pressure tank; in doing so, the air in the tank will be compressed, but there will not be sufficient air in the tank to give the required pressure. Therefore, when the tank is about half full of water, open the air vent in the suction pipe of the pump, thus introducing air with the water until the proper pressure is reached, when the gauge shows about one-third of air and two-thirds of water, this being the proportion upon which tanks are generally based to amply supply the necessary amount of water for the cylinders. When an extra air pump is provided, fill the tank two-thirds full of water and supply the air pressure afterwards. The water level indicated above should be carefully maintained by the engineer in charge during the operation of the plant by opening the vent in the suction pipe of the pump occasionally or starting the air pumps, respectively, whenever the water level rises higher through loss of air by leakage or absorption. It is better to have a little more air in the tank than too little, since too small an air volume is apt to cause considerable fluctuation of the pressure during each stroke of the elevator piston.

After the necessary pressure has been reached, slowly open the stop-valve between the tank and the controlling valve, which stop-valve is generally and preferably a gate valve. Next, slowly open the controlling valve, *all air valves or cocks having been previously opened*, to allow the air contained in the cylinder to escape; the air cocks are shown at *a*, Fig. 6, and at *b*, Fig. 8. For the first filling of the cylinder, the controlling valve must be set for *going up*. After all the air is expelled, which can be ascertained by water running from the air cock into the funnel of the drip pipe *m*, Figs. 6 and 8, close the air cock. The elevator is now ready for running.

52. Absorption and Discharge of Air.—As already mentioned, the air will be absorbed by the water to a certain extent. This air frees itself in the cylinder and may form a cushion. It is, therefore, occasionally necessary to remove the air. In vertical-cylinder (circulating) systems such an air cushion can form above and below the piston. Air below the piston is automatically removed in the Otis vertical elevator by a piston air valve *c*, Fig. 6, provided for the purpose, which lets the air into the space above the piston, whence it can be removed through the air cock *a*. When there is air in the cylinder, this will cause the car to spring up and down in stopping. When the quantity of air is small, it can generally be let out by opening the air cock and running a few trips. This should, therefore, be done occasionally. If there is much air in the cylinder the car must be run to the top and the controlling valve set for *going down*. While the car and valve are in this position, open the air cock and allow the air to escape. This may have to be repeated several times before the air is all removed. If the absorption of the air by the water is found to be considerable, it may effectually be prevented by the introduction into the tank of a layer of heavy oil about 4 inches thick. This expedient will, however, have to be resorted to but seldom.

53. Settling of Car.—After all the air is removed close the air cock, as otherwise the car will settle, that is, slowly creep down at the landings. If the air cock is properly closed and the car still shows a tendency to settle, the cause is probably that the piston or valve is leaking and needs repacking. Another cause for settling may be that the piston air valve *c*, Fig. 6, does not properly seat.

54. Groaning Noise in the Cylinder.—If a groaning noise is heard, it may be taken as a sign either that the two piston rods (in the vertical type) are not drawing alike or that the piston packing is worn out and needs renewal. If it is believed that the fault lies with the rods, this may be ascertained by trying to turn the rods with the hands; if one of them will turn, it needs tightening up. If the packing is at fault, the car will settle.

55. Stretching of Cables.—The cables should not be allowed to stretch enough to prevent the car from reaching the top landing, because of the danger of the piston striking the bottom cylinder head.

56. Hand Cable, Limit-Stop Buttons, Back-Stop Buttons.—The hand cable, or shipper rope, as we have called it, should be properly adjusted, neither too tight nor too loose. The limit-stop buttons should be so adjusted that the car will stop at a few inches beyond either extreme landing and before the piston strikes the head of the cylinder. The back-stop buttons should be so adjusted that the valve cannot be opened either way more than to give the car the required speed. In the case of auxiliary, or pilot, valves, the stops on the shipper sheave serve instead of the back-stop buttons.

57. Lubrication.—The plungers in plunger elevators should be kept well greased and clean. A good way to clean and grease the plunger, suggested by the Plunger Elevator Company in connection with their "elevator grease," is to stand at the bottom floor and to run the elevator slowly up while *wiping the plunger dry.* On running the car down again, cover the plunger with a thin coat of grease, rubbing it on and spreading it even with the hands. The plunger should be dry when the grease is applied; otherwise the grease will not stick. The inside of the cylinder should be lubricated about every two weeks with cylinder oil. Oil cups are generally provided for this purpose. The Otis Elevator Company, of Chicago (Crane Elevator Company), say the following in regard to lubrication: "The most effectual method of lubricating the internal parts of hydraulic-elevator plants, where pumps and tanks are used, is to carry the exhaust-steam drips from the foot of the pump-exhaust pipe to the discharge tank, thus saving the distilled water and cylinder oil. This system is invaluable when water holding minerals in solution is used, as these minerals greatly increase corrosion."

Horizontal machines operated by city pressure are best lubricated with a heavy grease, applied either mechanically or by means of a piece of waste on the end of a pole. The former method serves as a constant lubricator, while in the latter case greasing is often neglected and, in consequence, packing lasts but a short time.

Mr. Ford recommends as a lubricant for his valves, described in Arts. **43, 44,** and **47,** common soap applied once a month.

58. Bushing Sheaves.—If the traveling-sheave bushing is worn so that the sheave binds or if the bushing is nearly worn through, turn it half around and thus obtain a new bearing. If it has been turned before, put in a new bushing.

59. Precautions Against Freezing.—Precaution must be taken against the water freezing in any part of the system. If the cylinder and connections must be located in an exposed place, they should be protected against frost by building an air-tight box, open at the bottom, around them; a small gas jet should be kept burning at the bottom, or when steam is available a coil should be placed near the cylinder. Plunger-elevator cylinders are exempt from the danger of freezing. Supply pipes outside of the building are best protected by burying them in the ground below the freezing line, say 6 feet. If this cannot be done, they should be covered with non-conducting material, the same as is used for steam pipes. If in cold weather the elevator service is to be stopped for any length of time the water must be drained off, care being taken that this is done thoroughly. This applies especially to small pipe connections for drips, vents, etc., which should be free from bends, loops, or sags in which water may be left to freeze after the system has been drained.

60. Closing Down Hydraulic Elevators.—We will imagine that for some purpose, as prevention of freezing, change of water, etc., the plant is to be closed down. After removing the lower limit-stop button, run the car slowly to the bottom. Next shut off the supply by closing the valve provided for the purpose in the supply pipe, as the valve *n* in Fig. 17. In the plunger type of elevator machine, the valve and connections only are thus drained, the cylinder remaining full of water around the plunger, which, however, does no harm, since being far underground the water will not freeze. In the horizontal machines, running the car down and closing the supply leaves both cylinder and valve free of water. In the vertical (circulating) machine, however, the cylinder and circulating pipe are still full of water when the car is down and must be drained. For this purpose, open the air cock and the drain-pipe valve *d*, Figs. 6 and 8. Throw the valve for *going up* to empty the cylinder through the discharge pipe. Next throw the valve for *going down* to empty the circulating pipe through the drain pipe. After all water is drained off, grease the cylinder with heavy grease if the machine is of the horizontal type, and grease the piston rods if of the vertical type.

61. Packing Plunger and Piston Rods and Stuffingboxes.—Stuffingboxes that must be repacked from time to time occur in the plunger type, the vertical type, and the horizontal tension type of hydraulic elevators. For repacking the stuffingboxes, it is neither necessary nor expedient to drain the system.

For packing plunger stuffingboxes, run up the car sufficiently to be enabled to work conveniently in the pit, shut the three-way controlling valve and the supply stop-valve between the tank and the controlling valve. Block up the car, then remove the gland of the stuffingbox and renew the packing; replace the gland, screwing up the bolts just tight enough to prevent leaking, open the supply stop-valve and then slowly the controlling valve, setting it for *going up*. Remove the blocking.

62. Various materials are used for packing plunger stuffingboxes. For the smaller sizes, such as sidewalk-elevator plungers, fibrous packing, such as hemp, flax, or cotton, is used exclusively. For large plungers cup leathers are probably the best packing. But since the cup-leather ring must be split open in order to introduce it into the box, much of its value is impaired; therefore, fibrous packing is much used.

63. To retain the cup-leather principle and at the same time to avoid the objection to the butt joint, multiple cup leathers may be used. Fig. 23 shows a plan that is said to have proved very satisfactory. The packing consists of split leather rings, or even of ring sections, of V-shaped cross-section. The edges of these rings are cut down sharp, in consequence of which they act in much the same manner as cup leathers. The single sections are, of course, introduced so as to break joints. This kind of packing is very tight, but is likely to create a great deal of friction.

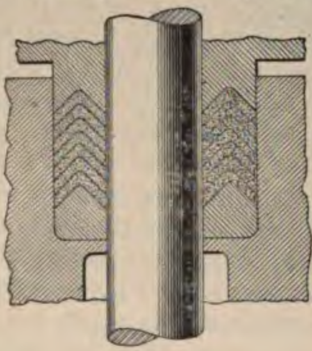


FIG. 23.

64. A much better arrangement is shown in Fig. 24. This packing, known as **Wright's elevator packing,** consists of a split rubber ring *l* of cup-shaped cross-section and a split leather ring *r* of L-shaped section. Both rings are placed in the stuffingbox so as to break joints.

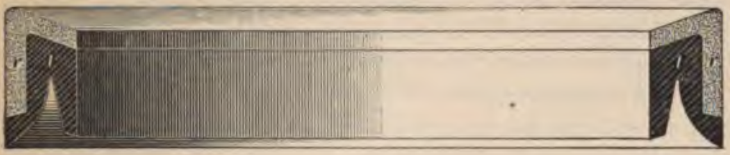


FIG. 24

65. For packing piston-rod stuffingboxes, close the supply stop-valve and open the air cock to make sure that there is no pressure in the cylinder; remove the followers and glands of the stuffingboxes and renew the packing. Screw down the followers only tight enough to prevent leaking. Fibrous packing is used exclusively.

66. Packing Vertical Cylinder Pistons.—In some designs of vertical elevators the piston can only be packed from the top, as in the elevators shown in Figs. 5 and 8. In others, provision is made for packing the piston either from the top or bottom, as in the Otis elevator shown in Fig. 6. In others, again, the piston can be packed only from the bottom, as in the elevator shown in Fig. 11.

67. To pack a vertical cylinder piston from the top, run the car to the bottom and close the stop-valve in the supply pipe. Open the air cock at the head of the cylinder and also keep open the valve in the drain pipe from the side of the cylinder long enough to drain the water in the cylinder down to the level of the top of the piston. Now remove the top head of the cylinder, slipping it up the piston rod out of the way and fastening it there. If the piston is not near enough to the top of the cylinder to be accessible, attach a rope or small tackle to the *main cables* (not the counterbalance cables) a few feet above the car and draw them down sufficiently to bring the piston within reach. Remove the bolts in the piston follower by means of a socket wrench. Mark the exact position of the piston follower before removing it, so that there will be no difficulty in replacing it.

In the elevators shown in Figs. 5 and 8 fibrous hemp packing is used. In the design shown in Fig. 6, a combination of cup-leather and duck packing is used. On removing the follower of this piston, a leather cup *l* is found turned upwards, with coils *u* of $\frac{5}{8}$-inch square duck packing on the outside. This duck packing should be removed and the dirt cleaned out; also clean out the holes in the piston through which the water acts on the cup. If the leather cup is in good condition, replace it and on the outside place three new coils of $\frac{5}{8}$-inch square duck packing, being careful that they break joints and also that the thickness of the three coils up and down does not fill the space by $\frac{1}{4}$ inch, as in such a case the water might swell the packing sufficiently to cramp it in this space, thus destroying its power to expand. If too tight, strip off a few thicknesses of canvas.

Replace the piston follower and let the piston down to its right position. Replace the cylinder head and gradually open the gate valve in the supply pipe, first being sure that the operating valve is on the center. As soon as the air has escaped, close the air cock and the elevator is ready to run.

68. To pack vertical-cylinder pistons from the bottom, remove the top limit-stop button and run the car up until the piston strikes the bottom head of the cylinder. Secure the car in this position by passing a strong rope under the girdle, or crosshead, and over the sheave timbers. When secured, close the gate valve in the supply pipe, open the air cock at the head of the cylinder, and throw the controlling valve for the car to *go up*. Also open the valve in the drain pipe from the side of the cylinder and from the lower head of the cylinder, thus allowing the water to drain out of the cylinder. When the cylinder is empty, throw the valve for the car *to descend*, in order to drain the water from the circulating pipe.

In cases of tank pressure, where the level of water in the lower tank is above the bottom of the cylinder, the gate valve in the discharge pipe will have to be closed as soon as the water in the cylinder is on the level with that in the tank, allowing the rest to pass through the drain pipe to the sewer. When all water is drained off, proceed as directed in the previous article in renewing the packing. To refill the cylinder after packing, close the valves in the drain pipes, leave open the air cock at the head of the cylinder, leave the controlling valve in the position to descend, and open the gate valve in the discharge. Slowly open the gate valve in the supply pipe, allowing the cylinder to fill gradually and the air to escape at the head of the cylinder. When the cylinder is full of water, close the air cock and put the controlling valve on the center. The car can then be untied, the limit-stop button reset, and the elevator is ready to use.

69. Packing Horizontal Hydraulic Elevators.—In a compression-type elevator run the car to within 1 foot of the extreme top and secure it to the overhead beams with a chain or rope. Close the gate valves in the supply and discharge pipes and open the air cock and valve in the drain pipe, emptying the cylinder. Remove the buffer across the front (open) end of the cylinder and slide it along the piston rod out of the way. Remove the follower of the piston. With a hooked piece of wire remove the old packing. Raise the piston head until it is in the center of the cylinder. If the cylinder is found to be in good condition, cut off four rings of square lubricated fibrous packing 9 inches longer than the circumference of the cylinder. Place the two ends of a ring together and form tucks with the balance. Force in these tucks one at a time with a hardwood stick until all are level against the head. Proceed in the same manner with the remainder of the packing. Arrange the packing so that the joints in the different rings do not come together.

If the cylinder is badly worn, use square pure-rubber packing for the first and last ring, and make these but 1 inch larger than the circumference of the cylinder. This rubber insures a backing for the fibrous packing. After putting the packing in position, replace the follower and screw on the nuts with the fingers until the follower is close to the packing. On two of the studs opposite each other will be found jam nuts. Set these out against the follower and tighten with a wrench. Replace the buffer. Close the drain valve and set the controlling valve for going up. Open the gate valve in the supply pipe and fill the cylinder. When the cylinder is filled, close the air cock. As the car in the first place was not at the extreme top, the pressure in the cylinder will run the piston head against the buffer and the car will ascend to the extreme top. The fastenings may then be removed. Throw the controlling valve on the center and open the discharge. The elevator is then ready to descend. Do not make any trips until the cylinder is thoroughly greased. Continue greasing twice a week.

In the course of time, leaks will occur in the cylinder. Loosen the jam nuts back of the follower and set up the nuts on the studs equally until the leak is stopped. Then retighten the jam nuts.

70. In the tension type of horizontal hydraulic elevator, the procedure is exactly the same, with the exception that there is no buffer to be removed, the open end of the cylinder being at the back.

71. Packing the Controlling Valves.—Run the car to the bottom, close the supply valve, and drain the system as previously described. When the water is all drained off take off the cap. After marking the exact position of the various parts in relation to one another, remove the valve proper and renew the packings, placing the new ones in the same position as the old ones. Before refilling the cylinder close the valves in the drain pipes, but leave the air cock at the head of the cylinder open and be careful that the valve is in the position for the car to go down. Gradually open the gate valve in the supply pipe. When the cylinder is filled, close the air cock and open the gate in the discharge pipe.

72. Packing Material.—Fibrous packing is furnished by the trade in the form of a square braided fiber impregnated with a greasy substance. The material used is hemp, flax, or cotton. It is claimed by some that cotton is a more suitable material, being more elastic, softer, and more absorbent for grease. In using it, it is important that it should be well soaked in boiling tallow for several hours to exclude with certainty all air from the pores.

73. Leather for cups should be of the best quality, of an even thickness, free from blemish, and treated with a waterproof dressing. The cups should be of sufficient stiffness to be self-sustaining when passing over the perforated valve lining. Elevator builders generally make and furnish packings to fit their machinery, and it is recommended to get supplies from them. When ordering cups, the pressure of water carried should be specified, as the stiff cups intended for high pressure would not set out against the valve lining when low pressure is used.

ELEVATORS.

(PART 4.)

CAR SAFETIES.

PURPOSE.

1. The term **car safeties** applies to safety devices that in cases of emergency prevent the car from falling unretarded to the bottom of the shaft. All these devices, with the exception of air cushions, consist primarily of catches in the shape of wedges, pawls, etc. that lock the car to the guides. They differ, however, in the means by which these catches are set in operation. In some designs of car safeties, only the breaking of the hoisting cable or cables or its becoming slack through a temporary sticking of the car in its descent will operate the safety catches. In other designs, excessive speed of the car is relied on to operate them.

SLOW-SPEED CAR SAFETIES.

2. Fig. 1 shows the simplest form of a car safety intended to operate at the breaking of the hoisting cable or its becoming slack through a temporary arrest of the car. The hoisting cable is attached to a bolt *F*, which is free to slide in its hole in *d*, but has an enlarged head on the bottom through which the curved spring *c* passes. The lower end *c* of the bolt is slotted to receive one end of the bell-crank levers *E*, *E*, which are pivoted to the uprights of the car. The other ends of the bell-crank levers carry pawls *f*, *f*, which are spring-actuated and adapted to enter between suitable ratchet teeth on the guides. The pawls are normally held out of engagement with the teeth, the spring *e* being compressed by the load. Should the cable break or become slack for any reason, such as a temporary arrest of the car in its descent, the tension in the spring *e* would be relieved and the pawls would consequently engage the ratchet teeth, preventing the car from falling.

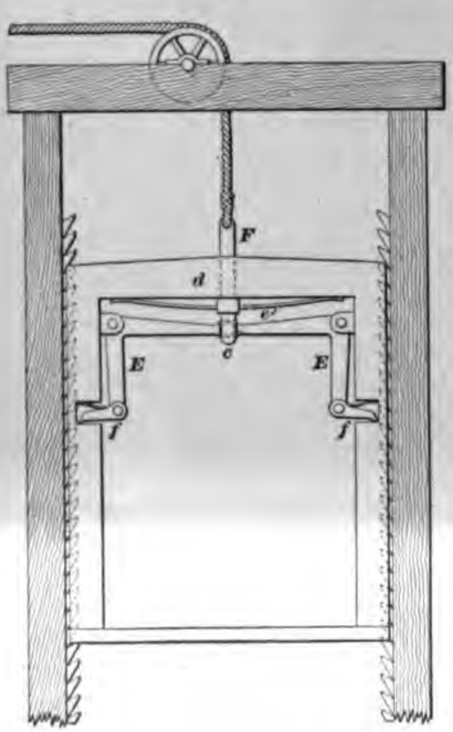


FIG. 1.

3. Pawl-and-ratchet arrangements, such as are shown typically in Fig. 1, are now but seldom used; they are suitable only for slow elevators. The pawl is generally replaced by a wedge that enters between the guides and the guide shoes. Fig. 2 is an example of a car safety of this kind. The cable is attached, as in the previous case, to a spring-actuated bolt or stirrup *F* carrying an **S**-shaped plate *S*, to which the links *L*, *L* are attached. These, in turn, are connected to levers *E*, *E*. When the cable breaks, the helical springs surrounding the legs of the stirrup force down the plate *S*, lifting up the outer ends of the levers E, E. These levers, in turn, then press on serrated wedges W, W contained in pockets of the guide shoes in such a manner that ordinarily they remain

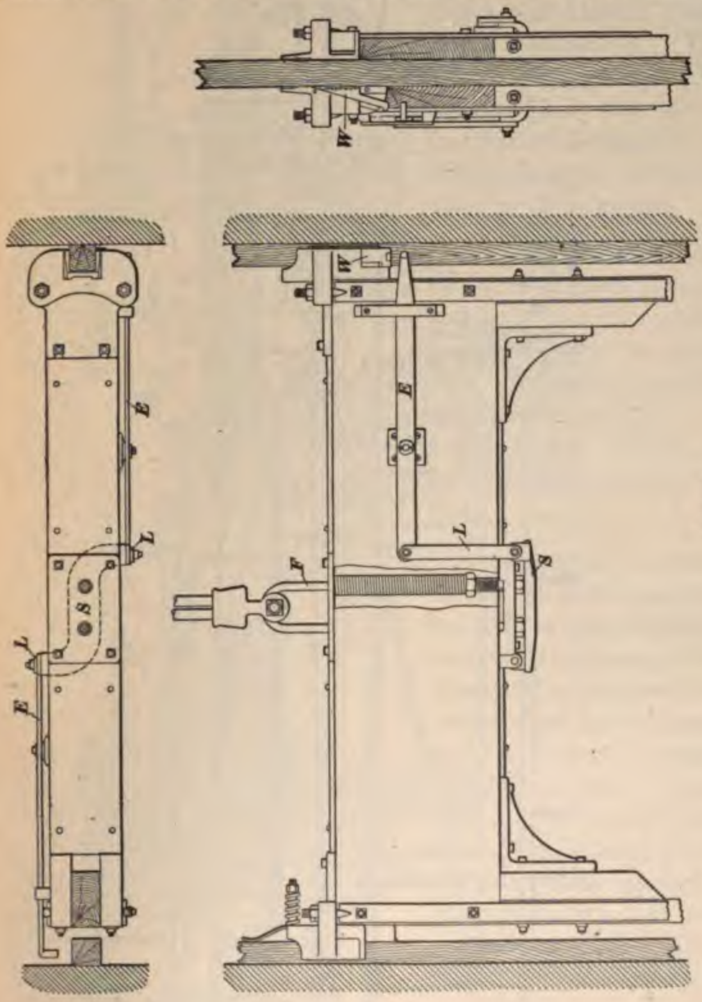


FIG. 2.

by gravity out of contact with the guides. The pressure of the lever ends forces them against the guides and the downward motion of the falling car wedges them tight, the

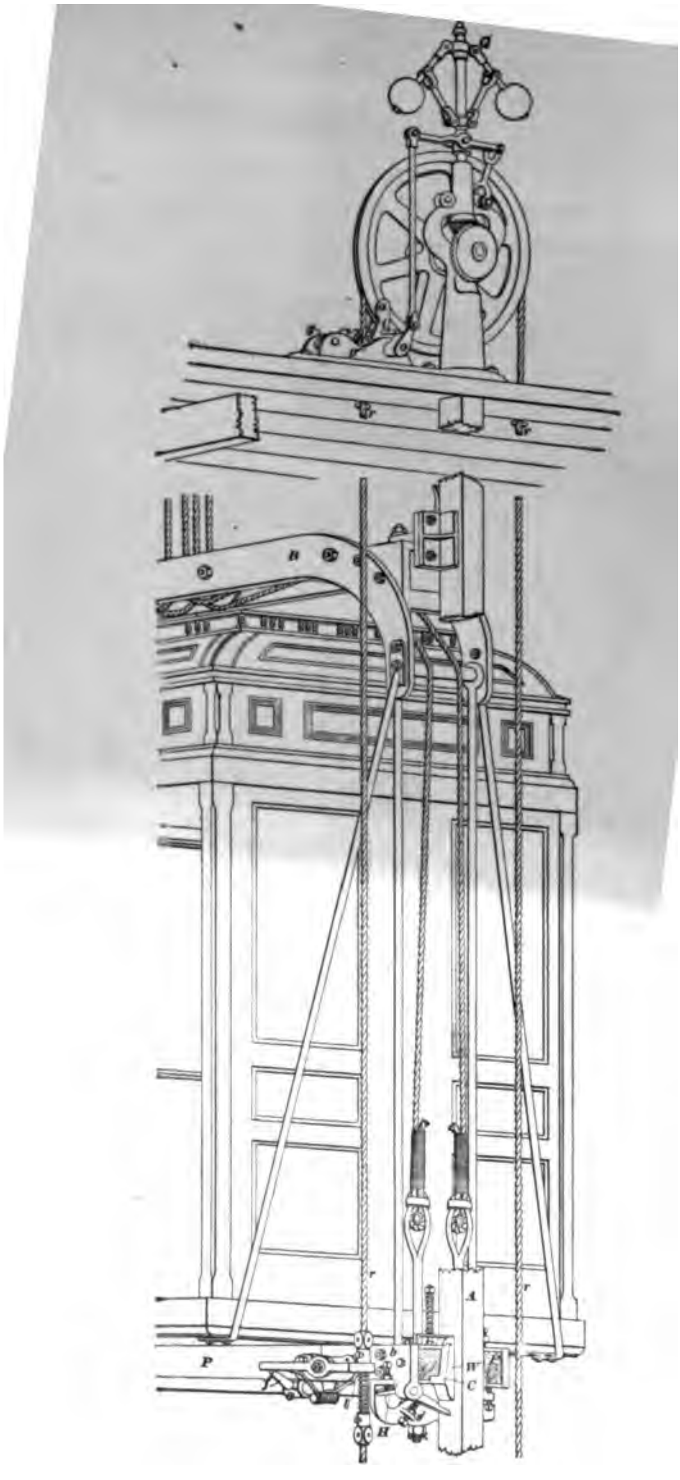


FIG. 3.

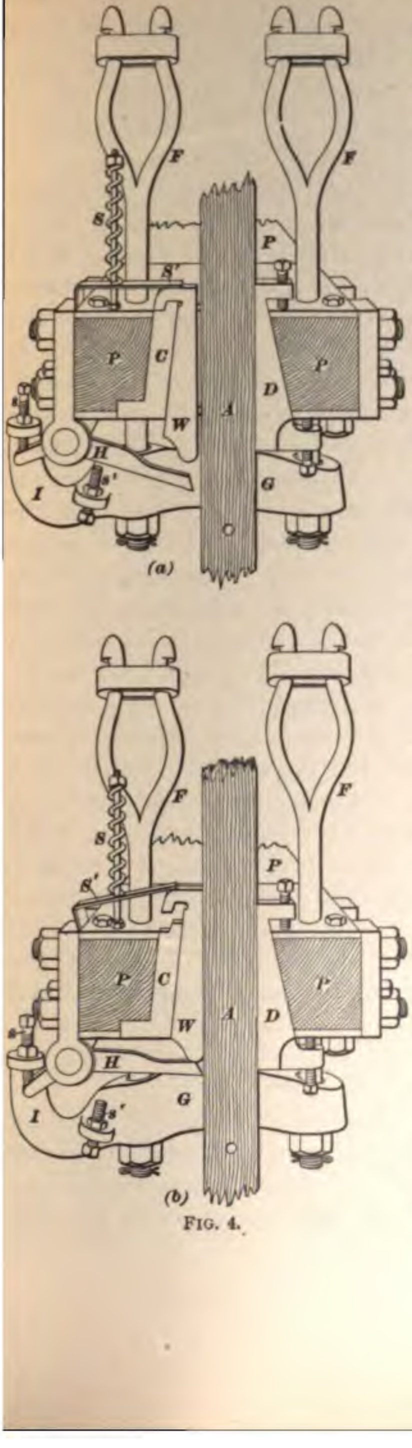


FIG. 4.

serrations or teeth burying themselves into the wood of the guides. When the car is again lifted, the wedges disengage themselves. This safety is regularly used by the A. B. See Manufacturing Company on their freight elevators with wooden guides.

4. It will be noticed in the arrangement shown in Fig. 2 that two cables are used, but in order that the safety mechanism should operate, both cables must break or become slack at the same time.

5. In order to make the safety device respond to the breaking, slacking, or even stretching of one of several cables used, the cables must be independently attached to wedge-operating levers. Figs. 3 and 4 show an arrangement of this kind known as the **Otis gravity-wedge safety.** It consists of a so-called **safety plank** P of hardwood placed under the platform of the car, and into the ends of which are let the guide shoes, each of which consists of a fixed jaw *C* and an adjustable one *D*, the latter being very clearly shown in Fig. 4 (*a*) and (*b*). Between the fixed jaw and the guide *A* is inserted a wedge *W*, which is normally held by gravity in such a position that the shoe can slide freely over the guide. The cables are attached, in the manner shown in Fig. 3, to the **shackle rods** *F*, Fig. 4, which, in turn, pass through an **equalizing lever** *G* pivoted in a suitable manner to the safety plank *P*. From the shackle rods the cables are carried upwards over rollers in a wrought-iron **girdle** *B*, Fig. 3, at the top of the car. By virtue of the equalizing levers, each of the four cables carries an equal strain, and as long as all cables are equally sound the equalizing levers will remain in their original position, that is, horizontal, as shown in Fig. 4 (*a*). As soon, however, as one of the cables breaks or even stretches more than its neighbor, the equalizing lever will tilt, as shown in Fig. 4 (*b*). An arm *I* of the equalizing lever carries a setscrew *s*, which is so adjusted that when the lever tilts down it will strike the end of a **finger** *H* mounted on a shaft under the safety plank. This finger then presses against the wedge *W*, making it engage the guide. Another setscrew *s'* is provided on the arm of the equalizing lever and has the same effect on the finger *H* in case the lever tilts the other way. It is thus seen that in case any of the cables become slack, stretched, or broken, the car will be stopped. The car may then be lifted by the other cables, but it cannot be lowered until the damaged cable is replaced. The spring *S* acting on the spring plate *S'* keeps the wedge *W* in place and prevents it, under normal conditions, from being drawn into engagement by mere sliding contact with the guide.

HIGH-SPEED CAR SAFETIES.

6. Safeties operated by the breakage, slacking, or stretching of a hoisting cable are today not considered sufficient except for very slow-speed elevators. In all high-speed elevators the catches are set into operation by excessive speed of the car, and the most generally adopted plan to effect this is the employment of a centrifugal governor placed either on top of the hoistway or carried on the car, and operated by an endless rope attached to the car or some fixed point. Such an arrangement is often found in addition to safeties to be operated by breaking cables, notably when city ordinances demand the latter.

7. An example of such a safety device is given in the Otis elevator shown in Fig. 3. The finger shaft mentioned in Art. **5** can also be operated by a rope r attached to a lever l, which, in turn, presses on a finger f on the finger shaft. The rope r passes around the pulley of a centrifugal governor G, Fig. 3, on top of the hoistway and an idler at the bottom. The idler is mounted on a crosshead that slides vertically in short guides and is weighted so as to give the rope r the proper tension. The centrifugal governor, by the outward motion of the balls, operates a clutch consisting of two eccentrics g and g', between which the rope r passes and which are geared together so as to grasp and pinch the rope when the balls move out too far, owing to excessive speed. The shaft of the eccentric g' has a crank o connected by a rod to the operating lever of the ball governor G. The eccentric g' is, however, loose on its shaft and has fastened to it an arm a having a stop pin i, against which the crank o strikes at excessive speed of the governor, bringing the eccentrics together so as to just bite the rope r. The continuing motion of the rope then pulls the eccentrics over fully, finishing the grip. The governor thus only starts the gripping of the eccentrics. It will be easily understood that reversing the motion of the car will throw the eccentrics back into their original position. The gripping of the rope causes the descending car to turn the lever l left-handed; this, in turn, rotates the finger shaft through engaging the finger f, and the finger H then operates the wedge W; the guides are thus gripped.

8. Another governor-operated device that is extensively used by the Otis Elevator Company in connection with steel guides is shown, together with the whole car frame, in Fig. 5, and in detail in Fig. 6. It will be noticed from the drawing that the four hoisting cables are not connected in any way

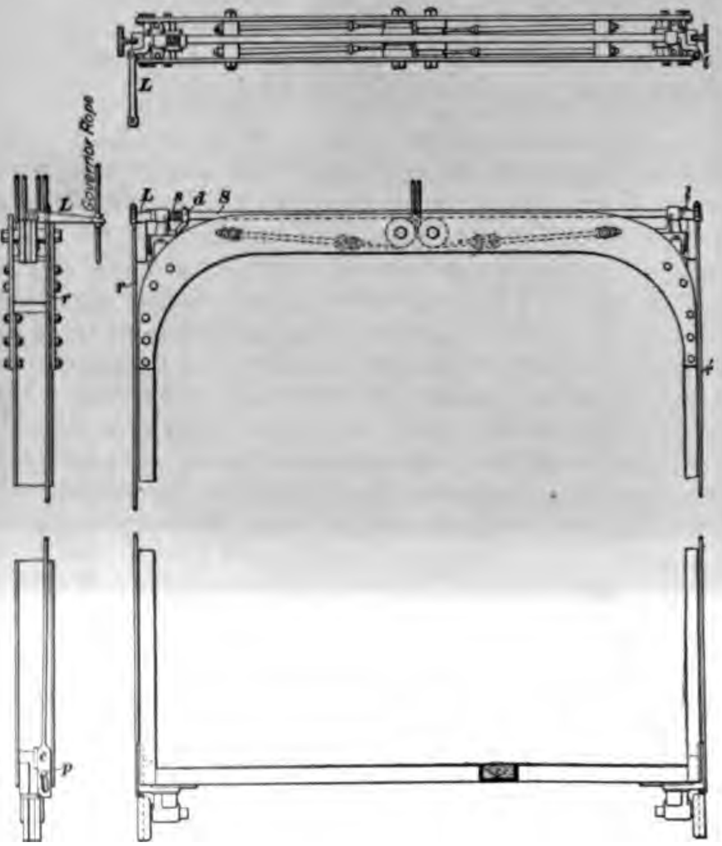


FIG. 5.

with the safety device, but that the latter is solely operated by the governor rope. The rope is attached to a lever L, which is fastened to a shaft S running across the top of the car frame. This shaft is held, normally, in a fixed position by a helical spring s and a stop-collar, or dog, d, resting against the guide-shoe casting. A little nearer the fulcrum of the lever L a rod r is attached to this lever; and to a separate lever l on the other end of the shaft S a similar rod r' is attached. These rods extend downwards to the safety plank, where they have flattened ends e, Figs. 5 and 6. A slot in each of these flattened ends serves to guide the rods by means of a pin p. On the under side of the flattened end is a shelf f, Fig. 6, that supports a loose roller r, serrated on

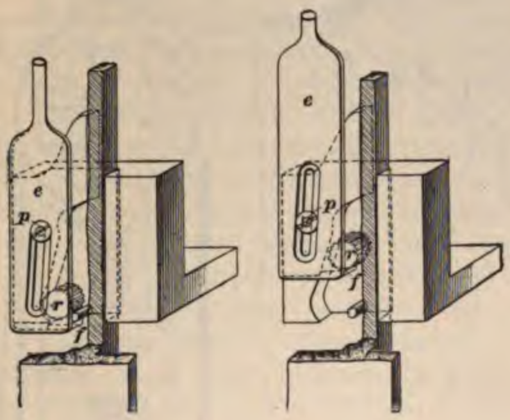
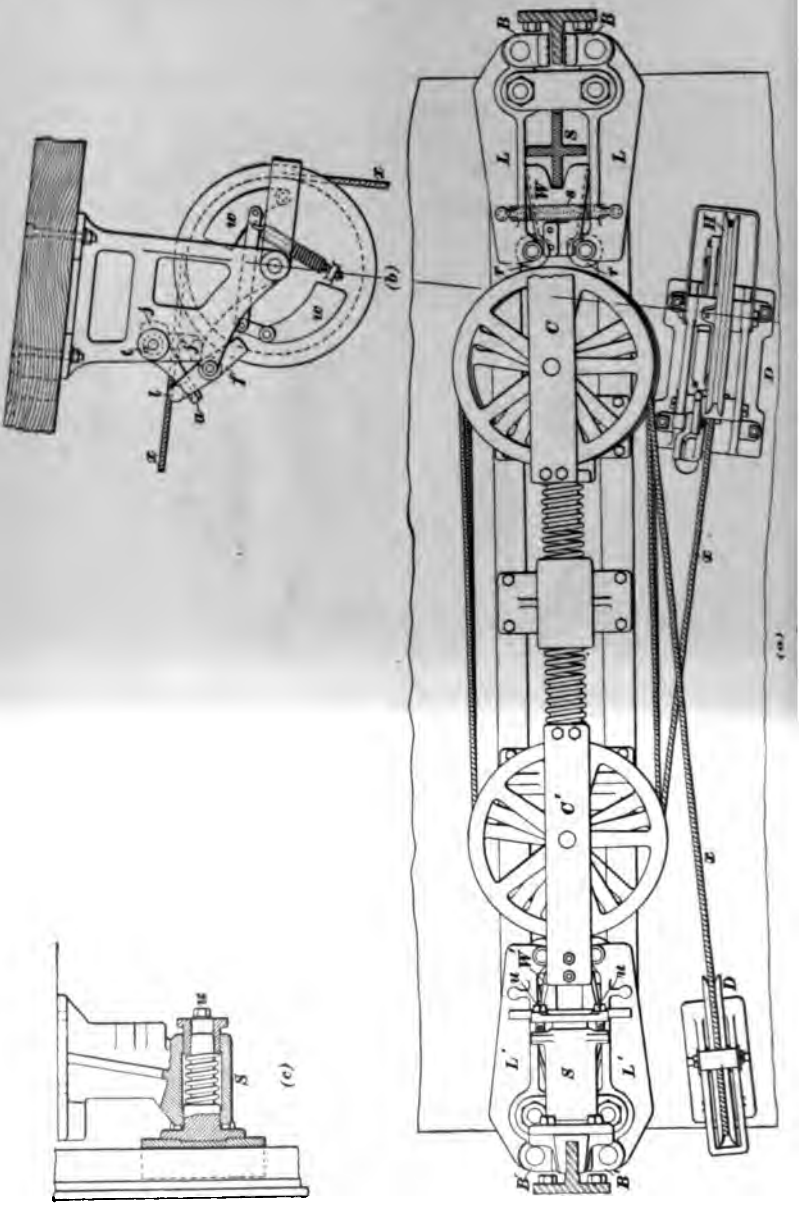


FIG. 6.

its cylindrical surface and contained in a pocket formed in the casting on the end of the safety plank. This pocket is so formed that if the roller r be lifted, it will be wedged in between the back wall of the pocket and the side of the T-shaped guide rail. The operation of this arrangement will be easily understood from the above brief description and Fig. 6, which shows the clamping roller in action and out of action.

9. In some governor-operated safeties the governor is carried under the platform of the car. Fig. 7 shows an arrangement of this kind as built by the A. B. See Manufacturing Company, and is intended for use with steel guides. Fig. 7 (a) is a bottom view, while Fig. 7 (b) is an



elevation of the governor and cable-gripping device. To the guide-shoe castings S, shown in side view in Fig. 7 (c), shown complete at the left of Fig. 7 (a), and at the right with the guide shoe proper and its sleeve removed, are pivoted the levers L, L and L', L'. The short arms of these levers carry grip blocks B, B and B', B', which are intended to close upon the guide rails in case of excessive speed. The long arms of the levers L, L' carry rollers r, r. Each pair of levers is connected by a spring s that normally holds the grip blocks off the guide rail. The governor rope x passes up from the bottom of the elevator shaft over the governor sheave H to the first one of a set of sheaves mounted in a crosshead C', thence to the first one of another similar set of sheaves mounted in a crosshead C, thence back and forth over the other sheaves of these sets, and finally over an idler D up to the top of the hoistway. The crossheads C and C' are properly guided and held by springs a certain extreme distance apart under normal conditions. To the crossheads are bolted cast-iron wedges W, W', which are so designed as to enter between the rollers r, r on the ends of the long arms of the levers L, L', and thus to push the same apart, closing the grip blocks down on the guide rails. These wedges enter between the rollers when the governor rope is arrested by the gripping device on the governor, since then the two sets of sheaves mounted on the crossheads C, C' will be pulled together by the rope shortening between them.

10. The action of the governor will be easily understood from Fig. 7 (b). The governor rope x coming from the governor sheave H passes between two jaws j and j', the former of which is pivoted to the governor frame and is actuated by a helical spring t that gives it a tendency to bear down on the rope against the other jaw j', which is fixed. The movable jaw j has an arm a attached to it, over which hooks the lug l on one end of a double-armed lever or finger f. The other end of the lever f projects into the paths of the governor weights w, w so as to be struck by them when they fly out too much, owing to excessive speed. In this case the lug *l* releases the jaw *j*, the rope *x* is locked to the frame, and the safety is put into action.

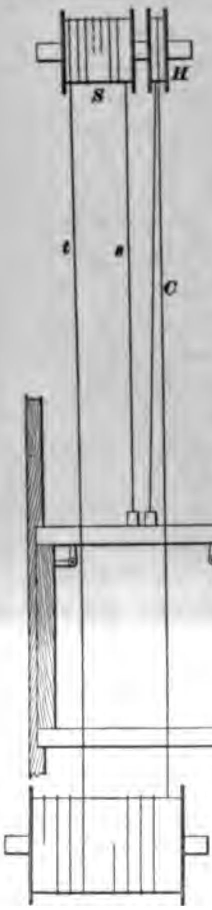


FIG. 8.

SAFETY DRUM.

11. Fig. 8 is a diagram of an arrangement often met with on Otis steam elevators. A so-called safety drum *S* is placed on the same shaft as the overhead sheave *H* for the hoisting rope. Attached to this safety drum are two ropes; one, the **safety rope** *s*, runs down to the levers of a suitable car safety on the car, and the other one, *t*, which is wound the reverse way on the drum, runs down to the hoisting drum; this rope is called the **take-up rope.** When the car is ascending, the take-up rope winds the safety rope on the drum *S*. If the hoisting cable *C* should break, the weight of the car would come on the safety rope and thus throw the car safety into action. The hoisting rope is generally also connected to an independent car safety.

12. In connection with the safety drum, a governor-controlled brake is generally used, which, if the hoisting rope should break, insures a gradual fall of the car, thus giving the safety time to act without a sudden shock.

The governor and brake are shown in diagrammatic form in Fig. 9, where *S* is the safety drum, *B* the brake pulley, and *G* a spur gear driving a pinion *P*. From the shaft of this pinion motion is transmitted to the governor spindle by bevel gears, as shown. The sleeve of the governor operates a bell-crank lever *L* having a projection *l*, on which is supported, by a hook *h*, the brake lever *W*. It is easy to see

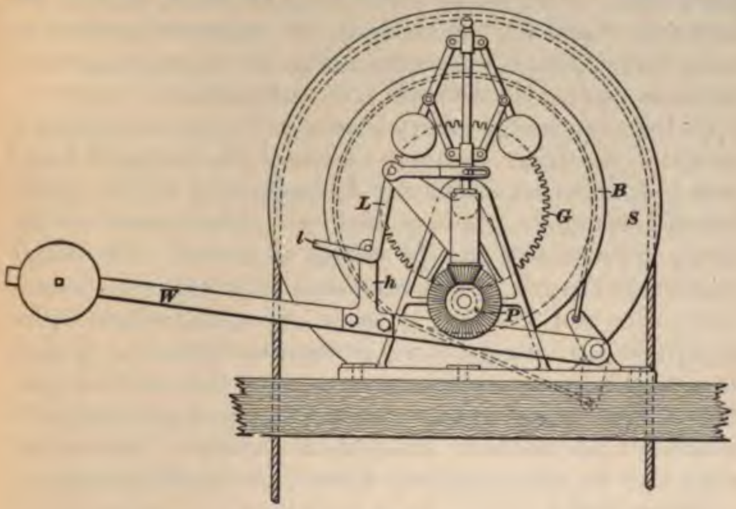


FIG. 9.

that when the governor balls fly out owing to the excessive speed of the car, the arm *l* will pass from under the hook *h*, and the weight on the brake lever *W* will apply the band brake.

13. The different designs of car safeties in actual use are very numerous, but a person understanding the operation of those here described will be able to understand the operation of most of them.

CARE OF CAR SAFETIES AND GUIDES.

14. The importance of keeping car safeties and guides clean and well lubricated, so that they will promptly do their duty when called upon, cannot be emphasized too strongly. Car safeties need adjustment from time to time.

15. When the guide shoes are adjustable, as most of them are, they should be so adjusted that the car will not wabble, but they should not be tight enough to bind on the guide rails. With spring-actuated guide shoes, such as are shown in Fig. 7 (*c*), for instance, the proper adjustment is easily accomplished by manipulating the screw bolts *u* in the same manner as the bolts of a stuffingbox.

In the Otis wedge safety shown in Fig. 4, the spring *S* must be just tight enough to prevent the wedge *W* being pulled upwards when the car is descending by the guide rail *A* coming in contact with it. A weakness of the spring *S* frequently causes wedges to rattle. The wedge should move perfectly free and should be frequently examined to see that it does. If, when the safety wedges move freely and the springs *S* are sufficiently tight, the wedges are still thrown into action or rattle when the car descends, the probability is that one of the cables has stretched or is broken. Care must be taken that all cables draw alike; when they do, the equalizing lever *G* should be horizontal, as shown in Fig. 4 (*a*). In this position the setscrews *s*, *s'* should not touch the finger *H*, but should be so adjusted as to touch and move the finger when the lever *G* is tipped a certain amount either way. The governor should not be too sensitive to harmless variations in car speed. For this reason, the governor rope *r* acts on the lever *l* through the intermediary of a spring, as shown in Fig. 3. This spring should be just tight enough to prevent the wedges from rattling when the car is moving at its normal speed, but not tighter, or the usefulness of the governor will be destroyed.

16. Guides should not be allowed to become gummy, for in this condition they are apt to cause much trouble; they frequently cause the safety wedges to stick, to be thrown into action unnecessarily, or, at least, to rattle. The governor should be examined frequently.

17. In case the safety has acted and has stopped the car, it is of the greatest importance to see, before unlocking the safety, that there is no slack in the hoisting cable. If there is slack, carefully take it up very slowly, reversing the motion of the motor and running it slowly. In hydraulic elevators, this can be done generally by carefully opening the controlling valve; in electric elevators, it is better to turn the worm-shaft by hand. After the slack has been taken up, unlock the safety catches. Most safeties are so arranged that they unlock automatically when the car is moved upwards. Thus, in the Otis gravity-wedge safety the wedges will drop back by gravity. In the safety shown in Fig. 5, the grip roller will readjust itself. In the safety shown in Fig. 7, the governor rope will automatically release itself when the car is going up, but the tripping device must be readjusted by hand. A hole in the car floor is provided for that purpose.

In case the car has been stopped above the top landing, it may become necessary to remove the limit-stop button on the shipper rope, so that the car may be raised high enough to unlock it. If this should prove insufficient, it may even become necessary to raise the car by a tackle.

AIR CUSHIONS.

18. The car safeties treated in Arts. **2** to **17** are designed to act immediately after the slacking or the breaking of a cable, or at the attainment of an excessive car speed. If, when the cable breaks, the car safety should fail to work, owing to neglect or some other cause, the car will drop unretarded to the bottom of the hoistway, causing destruction of property and the probable death of the passengers. An always-ready means of preventing such serious accidents is the **air cushion.** This may be formed by extending the hoistway below the lowest landing in the form of a pit, which has a cross-section at its top somewhat larger than the platform of the car and which gradually tapers towards the bottom to nearly the same cross-section as the platform. When the car falls into this pit, the air within it is compressed and is forced out gradually around the platform of the car, thus letting the car down gradually.

19. Air-cushion pits, in order to be effective, should have a depth equal to one-fifth the whole lift of the car, that is, 20 feet for each 100 feet of hoistway. The walls of the pit must be air-tight, and great care must be used in their construction. Owing to local conditions, it is not always possible to extend the pit far enough below the ground to make it efficient, in which case it may be formed by making the lower part of the hoistway air-tight, say for one or two stories, and providing it with air-tight doors. The engineer in charge of the plant can only see that the pit is not filled with rubbish and when there any doors that they close air-tight.

ACCESSORIES.

SAFETY APPLIANCES.

ELEVATOR ENCLOSURES.

20. The question of **elevator enclosures** is largely a matter of city ordinances. In general, it may be said that every possible means should be taken to prevent accident to passengers on the elevator, as well as persons whose duty brings them near elevator shafts and hatchways. Whatever means are taken by the builders, either of their own account or in compliance with city ordinances, it is the duty of the engineer in charge to see to it that all enclosures are kept in proper condition. He should be constantly on the lookout for improvements in this line.

Whenever possible, elevator enclosures should extend from floor to ceiling, to prevent anything that is being carried on the car catching between its platform and the ceiling. No projections whatever should extend into the hoistway. If full enclosures are not practicable and goods are carried that are liable to stick out, such as rods and similar articles, a car should be used that is enclosed on at least three sides.

Full enclosures need not necessarily be solid walls or partitions, but can be made of lattice, or grille, work substantially braced. As a matter of fact, solid walls for elevator shafts, while recommended by some engineers, are of doubtful value. An elevator shaft so constructed will act, in case of fire, as a chimney, and will carry the flames from one floor to another. Besides, such shafts are apt to be dark unless windows are arranged in them, which make the shaft more dangerous in case of fire. The windows in such shafts should be securely fastened and preferably covered with wire screens. Latticework enclosures will admit plenty of light. In case enclosures are not carried up to the ceiling, they should be at least 5 feet high. Many an accident has occurred by people bending over too low enclosures to look for the car, which then struck them while coming down. Passenger-elevator enclosures are usually made of artistically formed wrought iron and are intended as an ornament to the building in addition to their usefulness. They are generally expensively varnished and should, therefore, be treated with care. They should be cleaned with a feather duster and soft rags. The use of gritty substances, soap, or oil should be avoided. They should be revarnished from time to time, especially after repairs have been made.

ELEVATOR DOORS.

21. Requirements.—Elevator doors should always be, if possible, sliding doors or gates so hung that they will operate very freely. They should be provided with latches or locks that can be opened only from the *inside* of the shaft, but they should open easily; that is, without requiring much exertion on the part of the operator. Self-closing doors are to be preferred. The operator should not, however, rely on these self-closing devices, but should always make sure that the door is closed before he leaves the landing with his car. He will and should be held strictly responsible for accidents due to doors having been left open.

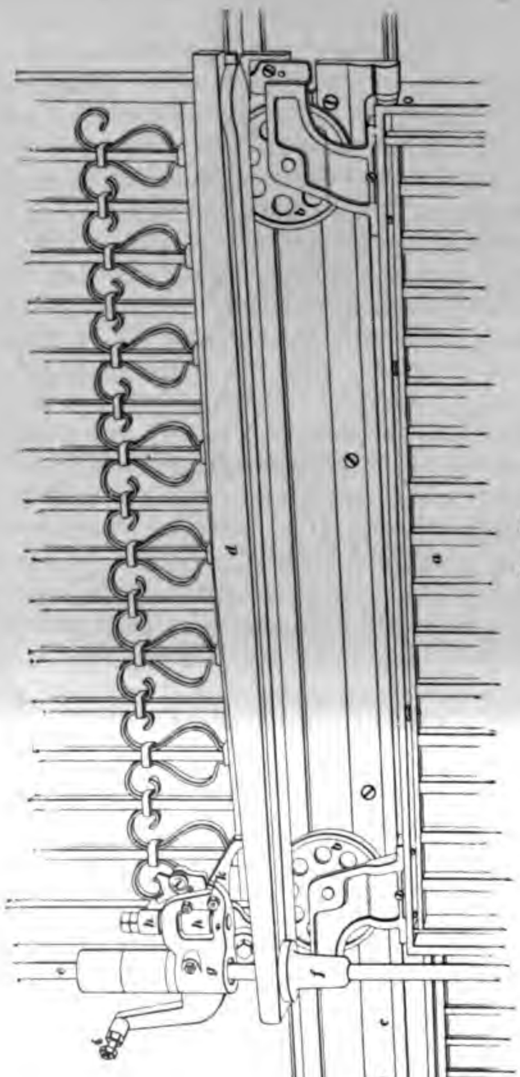


FIG. 10.

22. Self-Opening and Self-Closing Elevator Doors. Various devices are used by different manufacturers to make an elevator door self-opening and self-closing. These devices, in general, have for their object the automatic closing and locking of the door immediately upon the elevator car leaving a landing, and, in addition, are so designed that the operator can open or close the door at will without touching it while the car is at one of its landings and at rest.

23. The elevator-door operating device made by the Winslow Brothers Company, Chicago, Illinois, is shown in Fig. 10. The operation of this device is purely mechanical, the door being moved either way by a friction cone engaging either side of a suitable bar rigidly connected to the door. The construction of the device is as follows: The door *a* is supported by rollers *b*, *b* upon a level track *c* having a **V** groove planed in it to receive the **V**-shaped rollers. This arrangement prevents any side motion of the door. The so-called **traction plane** *d* is rigidly attached to the two door hangers that carry the rollers. A vertical shaft *e* carrying a friction cone *f* and also a cone-operating device at the top of each landing extends from the top to the bottom of the elevator shaft and is continually revolved by a small electric motor, or from some other source of power by belting. The so-called swing bar *g* is pivoted to a bracket *h* that is rigidly fastened to the transom above the door; the swing bar carries a bushing so fitted as to allow it to swing a little. The revolving shaft *e*, which owing to its length is quite flexible, passes through the bushing of the swing bar, the said bushing forming a journal for the shaft. The free end of the swing bar carries the adjustable buffer *i* intended to come in contact with a vertical shoe placed on top of the car. This vertical shoe can be thrown forwards so as to press against the buffer, and hence can be made to swing the swing bar around its pivot by a treadle in the car operated by the foot of the operator.

The traction plane is slotted, the slot being beveled and wider at the bottom; by pressing the buffer *i* away from the car the friction cone will be pressed against the side of the slot nearest the transom and the revolving cone will thus open the door.

As soon as the operator removes his foot from the treadle, the shoe on the top of the car will move away from the buffer *i* and the shaft will spring back, bringing the friction cone against that side of the slot in the traction plane that is farthest from the transom; the revolving cone will then, by its friction against the surface with which it engages, cause the door to close.

As has just been explained, the door closes whenever the shoe on the top of the car is moved out of contact with the buffer *i*. This shoe is quite short; consequently, should the operator forget to remove his foot from the treadle in the car when starting the elevator, the movement of the car will very quickly take the shoe vertically out of engagement with the buffer *i*; the revolving shaft *e* will then immediately spring back to its normal position and the door will be closed automatically.

The door is held open automatically while the car is at a landing by virtue of a recess in the end of the traction plane into which the friction cone passes after opening the door. The door after closing is locked automatically by a catch *k*.

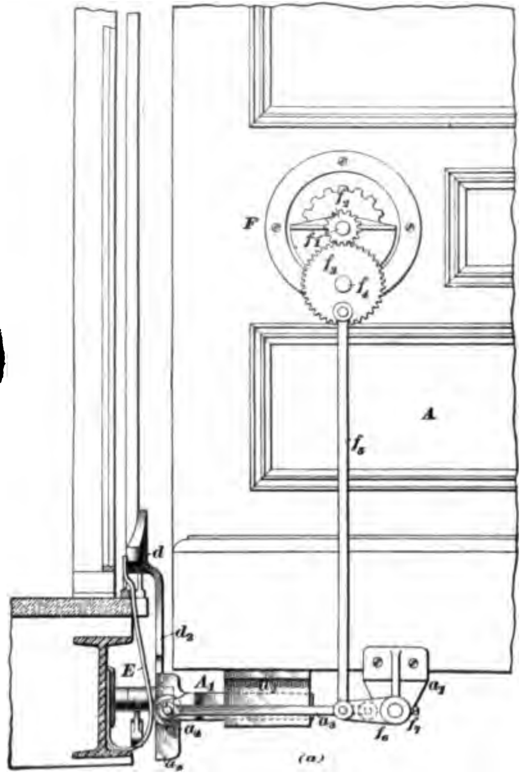
24. The Burdett-Rowntree Manufacturing Company use a horizontal pneumatic ram at each landing to automatically open and close the door. The piston of the ram is attached by a link to a long swinging lever connected to the door, and as the ram piston moves one way or the other it carries the door with it. The device is so designed that the door is always held closed until the car is at a landing, when the operator, by pressing on a treadle, throws a movable vertical shoe against a suitable part of the valve gear. This operation unlocks the door and admits air under pressure to one side of the piston in the ram cylinder, at the same time opening the other side to the exhaust. The door now opens, and when wide open can be kept so by a finger lock as long as the car is at rest. Whenever the operator removes his foot from the treadle, or unlocks the finger lock, or starts the car either way without having closed the door, the door closes automatically by reason of the valve gear operated by the shoe on the car returning immediately to its normal position.

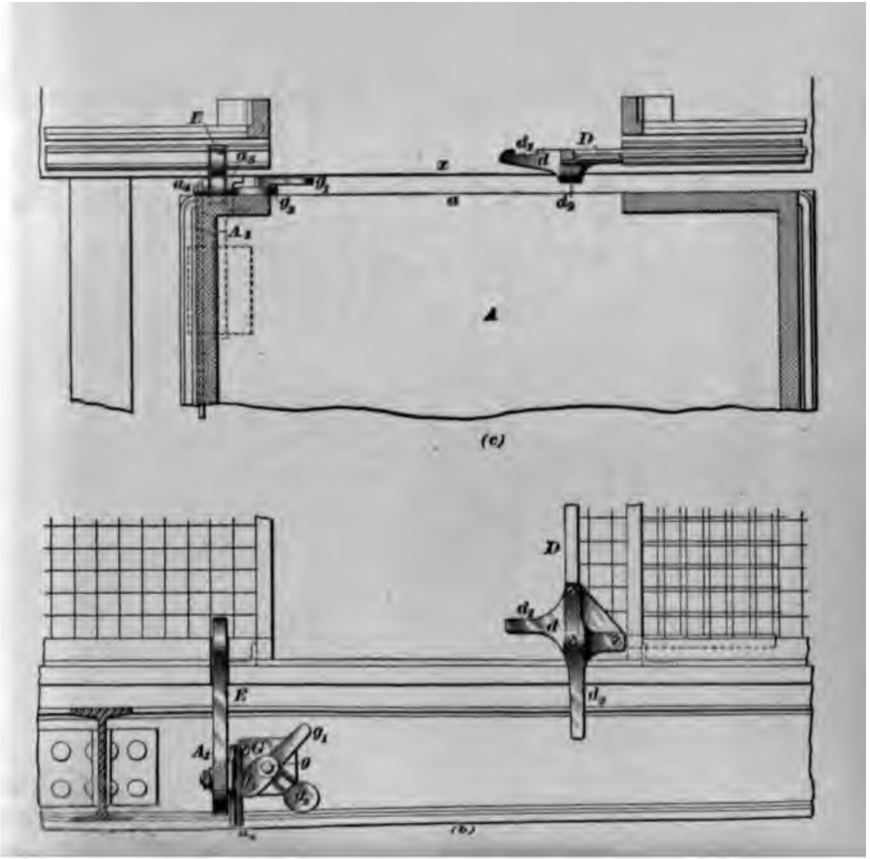
25. Car-Locking Device.—With elevator doors that are operated directly by hand by the operator, a **car-locking** device is sometimes used that automatically holds the car in position at its landings and only releases the car when the door is fully closed. While such devices are called car-locking devices, it must not be inferred that they lock the car itself to the landings or to the guides; instead they lock the operating device in the car so that the operator cannot move it to start the car in case the door has been left open.

26. Fig. 11 shows the car-locking device designed by Messrs. I. S. Muckle and W. H. B. Teamer. In Fig. 11 (*a*), the car *A* is shown at one of its landings and at rest, in which position the operating device *F* occupies its central position. The door *D* is unlatched and opened, as shown in Fig. 11 (*b*) and (*c*); the operating device in the car is then locked.

The following description of the device is partially taken from the patent specifications: Secured to one of the floor-beams within the elevator well is a spring latch *E*, which is bent as shown in Fig. 11 (*a*), and extends up into the path of an arm *d* secured to the door *D*. This arm is notched at d_1 to receive the spring latch *E* when the door is closed. When the latch is in the notch of the arm of the door, the latter cannot be moved until the latch is pushed out of the notch by the mechanism carried by the car; the door will then be free to be opened.

A pinion f_2 is keyed to the shaft f_1 of the operating device in the car and meshes with a gear f_3 turning on the stud f_4. A crankpin on this gear f_3 is connected by a rod f_5 to the lever f_6 pivoted at f_7 to a bracket a_1 fastened to the bottom of the car. A bearing a_2 on the bottom of the car carries a


(a)


(b)
(c)

slide A_1, and this slide is connected to the lever f_6 by a rod a_3. It is readily seen that, by virtue of the manner in which the parts are connected, the slide A_1 will be in its extreme outer position when the operating device is in its central position, as shown in Fig. 11 (*a*). The slide A_1 carries a roller a_4 that engages with the spring latch E and forces it out of the notch d_1 of the arm d carried by the sliding door, releasing the latter.

It is seen from the above description that the combination of the slide A_1 with the operating device constitutes a mechanism adapted to release the sliding door whenever the operating device is moved to stop the car, that is, is moved to its central position.

It will now be shown how the operating device is rendered inoperative, i. e., how the operator is prevented from starting the car while the door is open. On the face of the elevator well, to one side of the spring latch E, is a plate G carrying a stud g on which is hung a three-armed lever. The arm g_1 of this lever extends in the path of an arm d_2 depending from the door, so that the opening of the door allows the lever, under the influence of the weight g_2, to turn to the position shown in Fig. 11 (*b*). In this position the arm g_3 of the lever has passed behind a flange a_5 of the slide A_1 and prevents the slide from being drawn towards the car. Consequently, the operator cannot move his operating device to start the car, since this can only be moved when the slide A_1 is free. On closing the door, the dependent arm d_2 of the door engages the arm g_1 and turning the lever about its fulcrum g moves the arm g_3 out of the way of the flange on A_1, thus unlocking the slide and hence the operating device.

TRAP DOORS.

27. In many instances it is impractical to erect enclosures of any kind, as, for instance, when the elevator is located in the center of a warehouse and must be accessible from all sides. In such a case, the holes in the floors through which the car passes must be kept covered and must be uncovere(only to let the car pass. This is best done automatically i(some such manner as is shown in Fig. 12. The car is provide(

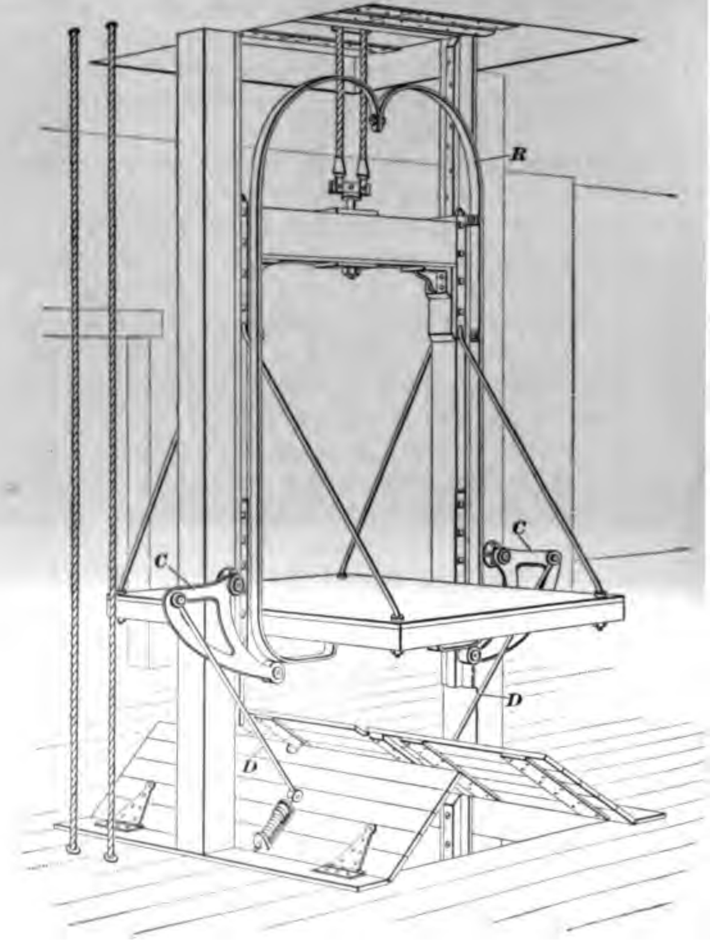


FIG. 12.

with an iron rail R. The arch-shaped upper part of this 1 gradually opens the trap doors when the car ascends, a the curvature of the under part lets them down gently wl the car descends. To open the trap doors when the car descends, the rail R strikes with its lower portion bell-cranks C, C that are suitably connected to the door by rods D, D.

INDICATORS AND SIGNALS.

INTRODUCTION.

28. Signals must be considered in many cases as a necessary element of safety, especially in freight elevators with insufficient enclosures or trap-door elevators. Electric bells, one on each floor, so arranged that they commence and continue to ring while the elevator passes the floor, are excellent safeguards; they not only warn persons against the approaching car, but tend towards the prevention of any attempt being made to operate the elevator from two floors at the same time.

29. For passenger service, a signal is necessary to communicate with the operator in the car from each floor. This is done very simply by means of a so-called **annunciator** placed in the car and a push button on each floor near the elevator door. Where the traffic is but slight, this means of communication is satisfactory enough; but where the service is rapid, it proves insufficient. Generally in such cases there are, at least, two elevators running all the time, one going up, the other down, and the would-be passenger should know which one to signal. For this purpose, so-called **indicators** have been devised, which show on each floor simultaneously the whereabouts of the car and whether it is going up or down.

MECHANICAL INDICATOR.

30. A simple mechanical device of this kind is shown in Fig. 13. On the shaft A of the overhead sheave is mounted a worm D meshing with a worm-wheel E that is mounted on a shaft F. This shaft carries a chain wheel I, from which motion is transferred by a chain N and rods T down the elevator shaft to each floor. The rods T are guided in plates W, one on each floor, and carry arms Z, Z. From these arms cords are carried over idlers X, X mounted on the

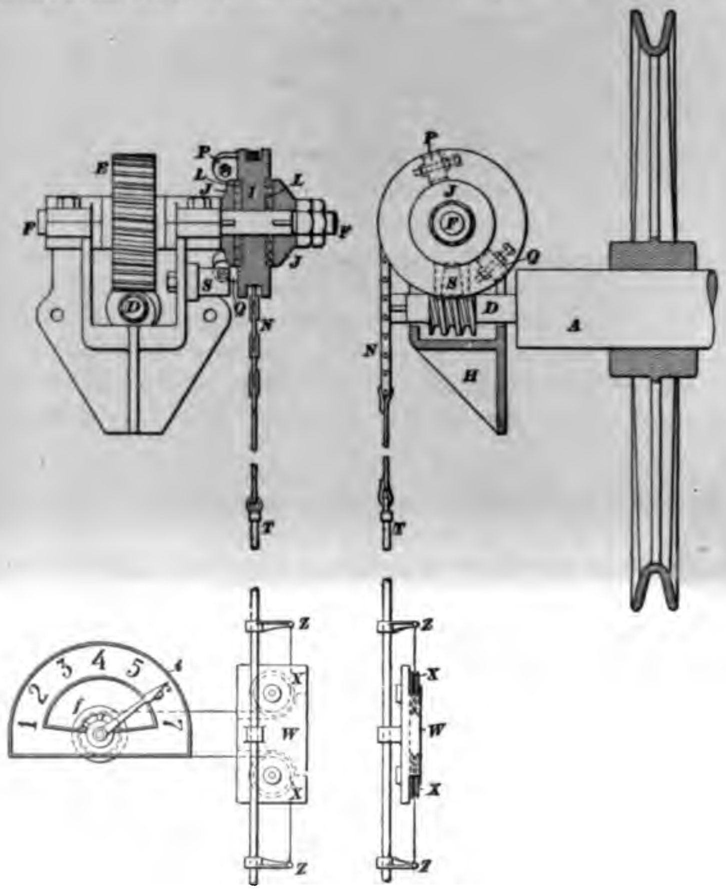


FIG. 13.

plates W and around small sheaves f in dial plates i attached at conspicuous places near the elevator doors. It will be understood that as the car travels up or down, the dial hand will move over the figures displayed on the dial and thus indicate the position of the car. The apparatus is made self-adjusting to rectify any disarrangement due to slipping of the chain.

The wheel *I* only makes a part of a revolution. It is provided with lugs *P* and *Q* that strike a stop *S* fixed to the frame of the machine as the car reaches its uppermost or lowermost positions, respectively. In case the apparatus has become deranged and indicates wrong, the one or the other of the lugs *P*, *Q* will strike the stop *S* before the car reaches its extreme point of travel and will bring the chain wheel *I* to a stop. On the return trip, the apparatus will then be readjusted. The chain wheel proper is mounted loosely on its shaft *F* and is clamped thereto by friction disks *J*, *J* fast to the shaft and leather washers *L*, *L*.

ELECTRIC SIGNALS AND INDICATORS.

31. The enormous traffic that has to be handled in the large office buildings has called for still more elaborate means of signalling than those afforded by annunciators and indicator dials. In such buildings the service is practically continuous and very swift; the operator has no time to consult an annunciator to find out on which floor passengers are waiting. On the other hand, a passenger standing in front of a row of swift-running elevators and wishing to get the next car would have, if he were to consult indicator dials, to patrol up and down in front of the elevator doors, and would be likely to miss several cars running in the direction in which he wants to go.

32. The usual plan followed in such cases is to provide a signal which, when operated by the passenger, will be noticed by the operator on every car of the series early enough for him to stop at the particular floor where the signal was given. The first car conductor answering the signal then destroys all the signals in the other cars. This plan has been successfully carried out in the Armstrong system, handled by the Elevator Supply and Repair Company, of

New York. This system operates as follows: There are several push-button plates of two buttons, the one marked *up* and the other *down*, conveniently located on each floor. Over each elevator door is a double-light electric lantern, one light marked *up* and the other *down*. A passenger desiring to signal the first car of a bank of elevators, pushes either the "up" or "down" button. This sets the signal, and when the first car moving in the direction the passenger wishes to go reaches a point about three floors distant from that on which he is standing, the lamp in the "up" or "down" compartment of the signal lantern on the outside of the elevator enclosure is automatically illuminated. When the first car approaching the waiting passenger going in the direction he wishes, either up or down, reaches a point about one floor distant, the "operator's signal" is flashed, giving him ample time to stop his car before running past the floor. The operator's signal is a small lamp inside the car constantly in sight. The lamps in both the lantern and car fixture remain illuminated until the car has left the floor from which the signal was given.

There can be no confusion of signals, because the operator can never have but one signal at a time. Moreover, the system is entirely automatic. It allows the operator the free use of his hands and he can thus give all his attention to the control of the car and the safety of his passengers. When no signal light appears in the car, the operator can run at full speed, knowing that no passengers are waiting. Should the first car that receives the signal be fully loaded and therefore unable to stop for more passengers, the operator may transfer the signal to the next car by pushing a button.

All this is accomplished by means of so-called commutators, one for each elevator, placed at the top of the shaft and run by a belt or chain from a pulley on the overhead sheave shaft, in connection with a number of electromagnets corresponding to the number of floors in the building. We forego a detailed description of the apparatus and the electrical connections thereof, since once installed, the apparatus

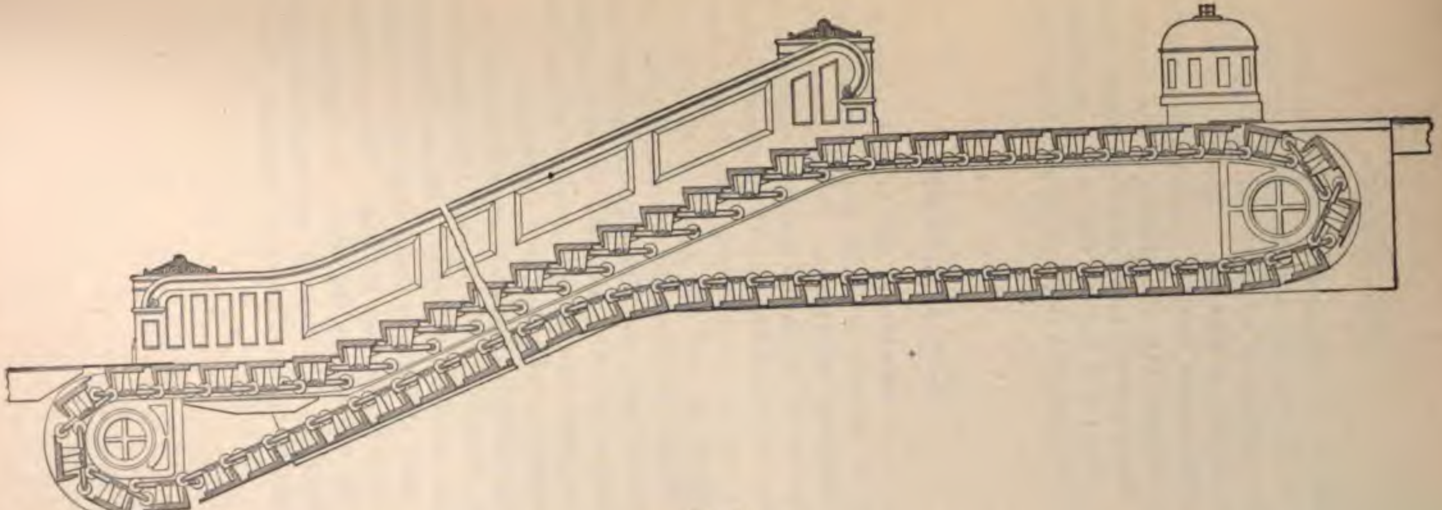


FIG. 14.

needs never to be disturbed. The engineer in charge should see to it that the contacts are kept clean and that the mercury cups used to make the various circuits have the proper amount of mercury. The current for the push-button circuits is furnished by a small motor-dynamo transforming an ordinary 110-volt lighting circuit to one of about 10 volts. This motor-dynamo, of course, needs an occasional inspection, just the same as the other machinery. The current for the lanterns is taken from the lighting circuit direct.

ESCALATORS.

33. The name **escalators** has of late appeared in the terminology of elevator practice for what are commonly known as **moving stairways.** These moving stairways are, properly, not to be classed among elevators, being constructed upon entirely different principles and are mentioned here only for sake of completeness and for the reason that they are destined to take the place of elevators in many instances. Thus it has been found that for short lifts, say one or two stories high, and where great numbers of people are to be transported, that adequate elevator capacity can be had only at great expense and sacrifice of floor space out of keeping with the profits accruing therefrom.

The moving stairway consists of an endless chain, to which are attached steps in such a manner that they form steps like those of an ordinary stairway. By an arrangement of cams, guide rails, and rollers these steps form a plane surface at the bottom and top landing. The accompanying sketch, Fig. 14, will convey the idea. It represents one of the latest designs of this class of passenger-transportation machinery built by the Otis Elevator Company, New York.

A SERIES

OF

QUESTIONS AND EXAMPLES

RELATING TO THE SUBJECTS
TREATED OF IN THIS VOLUME.

It will be noticed that the Examination Questions that follow have been divided into sections, which have been given the same numbers as the Instruction Papers to which they refer. No attempt should be made to answer any of the questions or to solve any of the examples until that portion of the text having the same section number as the section in which the questions or examples occur has been carefully studied.

THE STEAM ENGINE.

EXAMINATION QUESTIONS.

(1) Define the terms head end and crank end of a steam-engine cylinder.

(2) What is meant by the stroke of an engine?

(3) What are the stationary parts of an engine?

(4) Define *valve gear.*

(5) Explain the difference, if there is any, between piston clearance and clearance volume.

(6) What is understood by the throw of an eccentric?

(7) Explain the difference between outside lap and inside lap of a **D** slide valve.

(8) Is it possible to cut off when a **D** slide valve operated by an eccentric has no outside lap?

(9) What is meant by angle of advance?

(10) Suppose you had a plain slide-valve engine and you wished to make the cut-off earlier, what would you do? The port opening is to be the same as before.

(11) What is the effect of increasing the inside lap of a **D** slide valve?

(12) Define *lead.*

(13) With an ordinary slide valve and an engine running under, is the eccentric set behind or ahead of the crank?

(14) How can the valve be given a travel greater than the throw of the eccentric?

(15) If a reversing rocker is used with an ordinary slide valve, will the eccentric occupy the same position as with a direct rocker?

(16) How does the angularity of the connecting-rod affect compression?

(17) What is the object of the passage cored in an Allen valve?

(18) State in your own words how to set the valve of a plain slide-valve engine.

(19) A 14″ × 28″ engine has a clearance volume of 247 cubic inches. Express the clearance in per cent.

Ans. 5.73 per cent.

(20) In a 36″ × 60″ engine the steam is cut off when the piston has moved over 21 inches of its stroke. The clearance being 2 per cent., find the real cut-off.

Ans. 36.27 per cent.

(21) What is the ratio of expansion of the engine given in question 20? Ans. 2.76.

(22) Suppose that the outside lap of a D slide valve is decreased, but that the valve travel and angle of advance remain the same as before. Investigate the effect of this with the aid of the Bilgram valve diagram and state your conclusions and explain how they were reached.

(23) A 44″ × 80″ engine is to run at 75 revolutions per minute. What actual diameter of steam and exhaust pipe should be used?

Ans. $\begin{cases} \text{Steam pipe} = 18 \text{ in.} \\ \text{Exhaust pipe} = 22 \text{ in.} \end{cases}$

(24) What should be the area of the steam port for the engine given in question 23 if the steam port is short?

Ans. 202.7 sq. in.

THE INDICATOR

EXAMINATION QUESTIONS.

(1) What is meant by the scale of an indicator spring?

(2) About what scale of spring is usually selected for a given boiler pressure?

(3) What is a reducing motion?

(4) What is the principal objection to the lazy-tongs and pantograph reducing motions?

(5) In Fig. 9, find the length of the arm UV so that the diagram may be 3 inches long, the stroke of the engine being 24 inches, and the length of the arm UW 40 inches.

Ans. 5 in.

(6) What precautions should be taken before attaching an indicator to an engine?

(7) What is the vacuum line and how is its position located?

(8) What are the distinguishing characteristics of indicator diagrams taken from Corliss engines as compared with diagrams taken from high-speed engines?

(9) (*a*) In a plain slide-valve engine, how would you remedy too early admission? (*b*) What effect would this remedy have on the other events of the stroke?

(10) If one end of a cylinder with a slide valve is found to be doing more work than the other, how can the fault be remedied?

(11) (*a*) How is the amount of compression influenced by the speed of the engine? (*b*) What should be the amount of compression for high-, low-, and medium-speed engines?

(12) What may be inferred (*a*) when the steam line falls abruptly? (*b*) when the back-pressure line is much above the atmospheric line? (*c*) when the actual expansion line rises above the theoretical expansion line?

(13) (*a*) What faults in steam distribution are shown by the diagram, Fig. I, which is taken from a plain slide-valve

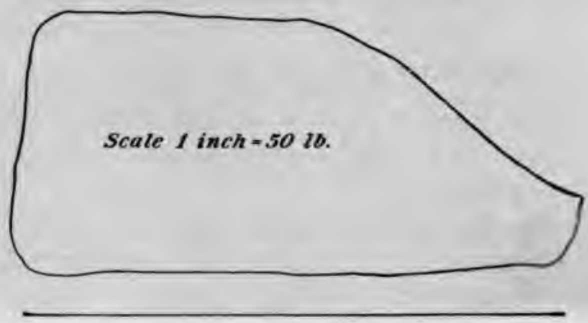


FIG. I.

engine? (*b*) How may they be partially remedied?

(14) Criticize the indicator diagrams shown in Fig. II.

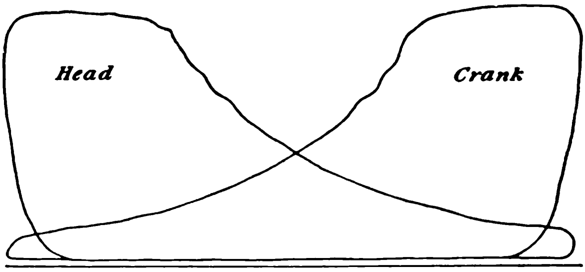


FIG. II.

(15) Why does the actual expansion line generally rise above the theoretical expansion line near the end of the stroke?

(16) What is the most general method of determining the point of cut-off on a diagram taken from a high-speed engine ?

(17) (*a*) To what are wavy lines on a diagram generally due ? (*b*) Expansion lines that drop by a series of steps ?

(18) Criticize the diagrams shown in Fig. III.

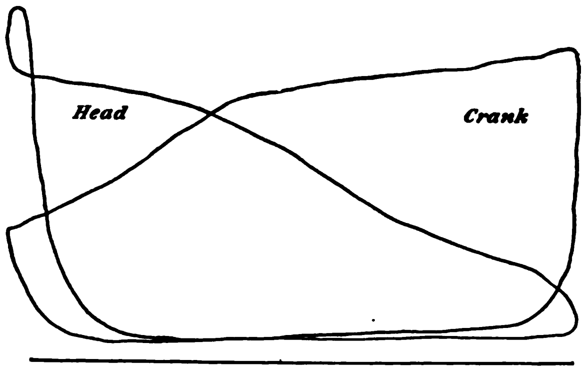


FIG. III.

(19) What is the cause of the difference in the shape of the loop above the steam line in Figs. 20 and 22 ?

(20) If the actual expansion line follows the theoretical expansion line closely, is that a positive indication that the valves and piston do not leak ?

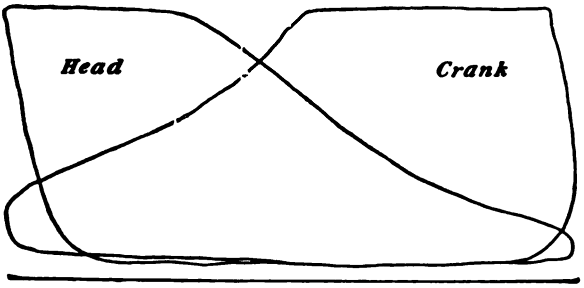


FIG. IV.

(21) Criticize the diagrams shown in Fig. IV.

ENGINE TESTING.

EXAMINATION QUESTIONS.

(1) Define (*a*) adiabatic expansion; (*b*) isothermal expansion.

(2) What relation exists between the amount of work done in compressing a gas isothermally or adiabatically and the amount of work done by the gas when expanding under similar conditions?

(3) Why is not the relation of volume and pressure of steam when it expands as simple as in the case of a perfect gas?

(4) If the net pressure on the piston of an engine is 45.6 pounds per square inch and the volume swept through by the piston at each stroke is 6.3 cubic feet, (*a*) how much work is done at each stroke? (*b*) The engine makes 76 strokes per minute; what horsepower does it develop?

Ans. $\begin{cases} \text{41,368.32 ft.-lb.} \\ \text{95.27 H. P.} \end{cases}$

(5) What is (*a*) the mean ordinate of an indicator diagram and (*b*) how is it found?

(6) A diagram like that shown in Fig. 7 is 3 inches long and has an area of 6.75 square inches; the vertical scale of pressure is 50 pounds per inch; the cylinder from which the diagram was made has an area of 1 square foot and a length of 2 feet. (*a*) Find the horizontal scale of volumes and (*b*) the work per stroke of piston.

Ans. $\begin{cases} (a) & \frac{2}{3} \text{ cu. ft. per in.} \\ (b) & \text{32,400 ft.-lb.} \end{cases}$

(7) How can the net horsepower of an engine be approximately obtained without the use of a dynamometer?

(8) The I. H. P. of an engine running under full load is 176.8. When running light the I. H. P. is 25.6. What is the efficiency of the engine? Ans. 85.5 per cent.

(9) The area of a diagram is 2.75 square inches and the length is 3.15 inches. A 50-pound spring was used. Find the M. E. P. Ans. 43.65 lb. per sq. in.

(10) In finding the area of a diagram with a planimeter, how may the accuracy of the work be easily checked?

(11) If the M. E. P. of a diagram with loops is to be found by the use of ordinates, how can the mean ordinate be found?

(12) Find the approximate M. E. P. of a non-condensing engine cutting off at $\frac{5}{8}$ stroke and making 300 revolutions per minute. The boiler pressure is 75 pounds gauge.

Ans. 57.69 lb. per sq. in.

(13) (*a*) What is meant by piston speed? (*b*) An engine with a 24-inch stroke runs at a speed of 180 revolutions per minute. What is the piston speed? Ans. 720 ft. per min.

(14) What is an engine constant?

(15) (*a*) What is the engine constant for a uniform speed of rotation of an $18'' \times 24''$ engine running at a speed of 185 R. P. M.? (*b*) What is the I. H. P. of the engine when the average M. E. P. for a pair of indicator diagrams is 53.8 pounds per square inch?

Ans. $\begin{cases} (a) & 5.706. \\ (b) & 306.98. \end{cases}$

(16) (*a*) Find the engine constant for a uniform scale, number of ordinates and piston speed of a $24'' \times 36''$ engine running at 150 revolutions per minute when 20 ordinates are used and the scale of the spring is 60. (*b*) What is the I. H. P. of the engine when the sum of the 20 ordinates is 16 inches?

Ans. $\begin{cases} (a) & 37.013. \\ (b) & 592.2. \end{cases}$

(17) What is meant by the brake horsepower of an engine?

(18) In Fig. 17 the distance from the center of the shaft to the point of support of the brake arm on the scale is 4 feet. When the brake is not in operation the scale balances at 14.5 pounds. What horsepower is developed by the engine when it is running at 225 revolutions per minute and the scale balances at 274.5 pounds? Ans. 44.55 H. P.

(19) In Fig. 18 the diameter of the pulley is 47 inches and the diameter of the rope is 1 inch. When the engine is running at 350 revolutions per minute, the weight W is 241 pounds and the spring balance A indicates 10 pounds. What horsepower is developed? Ans. 30.78 H. P.

(20) Why is the steam consumption, as calculated from an indicator diagram, always less than the actual steam consumption?

(21) How can an idea of the amount of cylinder condensation be obtained?

(22) The following measurements were taken from a diagram like Fig. 19: $a\,m = .70$ inch, $O\,m = 3.41$ inches, $b\,n = .62$ inch, $O\,n = .36$ inch, and $c\,h = 3.25$ inches. The diagram was taken from an engine having a 16″ × 20″ cylinder and running at 160 revolutions per minute. The area of the diagram is 2.41 square inches and the scale of the spring is 45 pounds. Find the steam consumption per I. H. P. per hour. Ans. 26.25 lb.

(23) The following measurements were obtained from a diagram (see Fig. 20) taken from a 24″ × 36″ engine: $a\,m = .71$ inch, $l = 2.93$ inches, $L = 3.36$ inches, M. E. P. = 37.5 pounds, spring 50. What is the steam consumption per I. H. P. per hour? Ans. 27.79 lb.

CONDENSERS.

EXAMINATION QUESTIONS.

(1) Why is it that a perfect vacuum cannot be formed in a condenser?

(2) What is the function of the air pump attached to a condenser?

(3) State the principle of condensing the steam (*a*) in a jet condenser and (*b*) in a surface condenser.

(4) If a condenser becomes flooded, what is liable to happen to the engine?

(5) When does a condenser get hot?

(6) Briefly explain the theory of the condenser.

(7) If an engine exhausts into a condenser, what is the only sure way of telling if the steam valves and pistons are tight?

(8) Mention two ways in which the amount of steam generally used in large plants to run the independent air and circulating pumps may be reduced.

(9) In a siphon condenser, how is the vacuum formed?

(10) What is the object of contracting the pipe of a siphon condenser into a neck, or throat?

(11) (*a*) A siphon condenser, like the Baragwanath, is placed 34 feet above the hotwell. Is it necessary for the pump to force the injection water to this height? (*b*) Why?

(12) (*a*) If the supply of injection water is impure, what type of condenser should be used? (*b*) Why?

(13) (*a*) How many pumps are required to operate a surface condenser? (*b*) What is the function of each?

(14) What are some of the advantages and disadvantages of the surface condenser compared with the jet condenser?

(15) What is the use of the snifting valve attached to a surface condenser?

(16) Where a surface condenser is used, why is it objectionable to take feedwater from the circulating side of the condenser to supply that lost by leakage, blowing-off, etc.?

(17) (*a*) What is the cause of loss of efficiency in surface condensers? (*b*) How may the efficiency be restored? (*c*) How may the loss of efficiency be prevented?

(18) (*a*) What is a good composition for condenser tubes? (*b*) How should the surface of the tubes be protected?

(19) (*a*) How are condenser tubes generally fastened in the tube-sheets? (*b*) Why are they so fastened?

(20) How may split condenser tubes and leaky tube packings be detected?

(21) How will air leaks manifest themselves in a condenser?

(22) What is the object of tinning condenser tubes both inside and outside?

(23) On what principle do all devices for cooling condensing water operate?

(24) When water is freely exposed to the air, on what factors does the amount of evaporation depend?

(25) Explain how the evaporation of a portion of the condensing water cools the remaining portion.

(26) Briefly describe the principle of the cooling tower.

(27) (*a*) In what three ways does the warm water falling through a cooling tower lose its heat? (*b*) Which has the greatest cooling effect?

(28) Briefly describe the Linde system of cooling condensing water.

(29) If the valves of a circulating pump which is driven from the main engine slam on account of too little water being pumped, how may the slamming be stopped?

(30) How is it usual to provide against the breaking of a cylinder head should the air pump of a jet condenser suddenly refuse to work and allow the water to back up towards the cylinder?

(31) On what does the amount of water required to condense a pound of steam depend?

(32) The temperature of the water entering a surface condenser is 55° and on leaving its temperature is 105°. The pressure of the steam at release is 5 pounds absolute and the temperature of the condensed steam as it enters the air pump is 135°. How many pounds of condensing water are required per pound of steam? Ans. 20.57 lb., nearly.

(33) If the vacuum in a jet condenser is less than it should be and the hotwell temperature is higher, what is probably the trouble?

(34) The exhaust enters a jet condenser at a pressure of 3 pounds absolute. The temperature of the condensing water is 65° and the temperature of the mixture as it enters the pump is 130°. How much condensing water is used per pound of steam? Ans. 15.8 lb.

(35) In practice about how much vacuum can be obtained?

COMPOUND ENGINES.

EXAMINATION QUESTIONS.

(1) Mention some of the mechanical advantages that the compound engine has over the single-cylinder engine of equal power.

(2) Why cannot a high ratio of expansion be economically used in a single cylinder?

(3) State briefly why the division of the temperature range between several cylinders tends to reduce cylinder condensation.

(4) Is it probable that the steam condensed in the high-pressure cylinder does any work in the low-pressure cylinder?

(5) (*a*) Which requires the heavier flywheel for equal steadiness, a tandem compound engine or a cross-compound engine? (*b*) Why?

(6) Briefly distinguish between the Woolf compound type and the receiver compound type of steam engines.

(7) Explain why a receiver is necessary for a cross-compound engine having cranks set 90° apart.

(8) To what do the terms compound, triple-expansion, and quadruple-expansion engine refer?

(9) Mention some of the advantages claimed for the so-called triangular connecting-rod patented by John Musgrave & Sons.

(10) What do you understand by the term "drop" when applied to the pressure in a receiver?

(11) What effect has the drop in a receiver on the quality of the steam that enters it?

(12) (*a*) How does a change in the high-pressure cut-off affect the receiver pressure? (*b*) Give reasons for your answer.

(13) How may the drop in a receiver be regulated?

(14) If the receiver pressure is raised by changing the low-pressure cut-off, what effect will it have on the relative amount of work done in the two cylinders?

(15) What objection is there to governing a compound engine by changing only the high-pressure cut-off?

(16) Why should a pop safety valve be fitted to the receiver?

(17) (*a*) In a cross-compound engine having duplicate piston rods, connecting-rods, crankpins, and crossheads, the power developed by the two cylinders is the same. Will the stresses in the duplicate parts necessarily be the same? (*b*) Why?

(18) In practice how is it probably best to distribute the work between the cylinders?

(19) (*a*) What is the object of the steam jacket? (*b*) Will anything be gained by using a steam jacket when superheated steam is used? (*c*) Why?

(20) In a horizontal compound engine should the high-pressure or the low-pressure valves have the most lead?

(21) (*a*) What is a reheater? (*b*) What is its purpose?

(22) In using a reheater the temperature of the exhaust is found to be greater than that corresponding to its pressure. What does this indicate?

(23) If a reheater is constructed similar to a tubular boiler, why is it a good plan to take the live steam for the reheater from a connection placed between the engine and throttle?

(24) (*a*) To what is the ratio of expansion of a multiple-expansion engine equal? (*b*) Does the volume of the intermediate cylinders affect this ratio?

(25) The volume of the low-pressure cylinder of a compound engine is 12,460 cubic inches. The volume swept through by the high-pressure piston, including clearance, is 3,115 cubic inches. The ratio of expansion of the high-pressure cylinder is 3. What is the total ratio of expansion?

Ans. 12.

(26) In what two ways may the ratio of expansion of a multiple-expansion engine be expressed?

(27) The volume of the high-pressure cylinder of a compound engine up to the point of cut-off is 800 cubic inches; the volume of the low-pressure cylinder up to release is 8,800 cubic inches. What is the total ratio of expansion?

Ans. 11.

(28) Estimate the probable horsepower of a triple-expansion engine having cylinder diameters of 15, 26, and 39 inches. The common stroke is 24 inches and the speed is 180 revolutions per minute. The boiler pressure is 180 pounds, the engine is condensing and is fitted with slide valves.

Ans. 511 H. P.

ENGINE MANAGEMENT.

(PART 1.)

EXAMINATION QUESTIONS.

(1) Explain how water hammer is caused by suddenly opening a stop-valve or throttle valve and allowing steam to enter a cold steam pipe or cylinder.

(2) What precaution should be taken with regard to the throttle valve before steam is allowed to enter the main steam pipe ?

(3) How should an engine be cared for after it has been stopped ?

(4) Describe briefly the manner of warming up the steam pipes and engine preparatory to starting up.

(5) If the boilers are fired up just previous to starting the engine, how may the engine and steam pipe be warmed up without the use of steam ?

(6) From what position of the crank is it easiest to start a horizontal engine ?

(7) In starting a slide-valve non-condensing engine, why should the throttle be opened quickly ?

(8) If a cracking noise is heard in the cylinder soon after starting, what is probably the trouble ?

(9) Mention one advantage of allowing an engine to cool down with the cylinder drain cocks closed.

(10) How may the cylinder of an engine that is fitted with a reversing gear be warmed up?

(11) Why is it advisable to have the air and circulating pumps of a condenser driven independent of each other?

(12) Describe briefly the manner of starting a slide-valve condensing engine that is fitted with a surface condenser.

(13) (*a*) Why must the injection valve of a jet condenser be opened at the same moment that the engine is started? (*b*) If the condenser gets "hot," how may it be started?

(14) (*a*) What is a snifting valve and (*b*) what is its purpose?

(15) Briefly describe the operation of starting a Corliss engine.

(16) What is the principal difference in the manner of stopping a simple Corliss engine and a simple slide-valve engine?

(17) In general, how may the low-pressure cylinder of a compound engine be warmed up?

(18) How will too high or too low a pressure in the receiver of a compound engine affect its starting?

(19) (*a*) For what class of compound condensing engines is the use of an independent vacuum engine particularly advantageous? (*b*) Why?

(20) If a reversible engine is fitted with an adjustable cut-off gear, how should the gear be set as soon as the engine is stopped?

(21) What relation should the center lines of the connecting-rod brasses bear to each other and to the center line of the connecting-rod?

(22) Briefly describe the method of stretching a line coincident with the center line of a cylinder, when an engine is being lined up.

(23) Suppose that it is desired to level up one of the lines used in lining up an engine; how may this be done with the aid of a plumb-line?

(24) (*a*) In lining up new shaft-bearing brasses for a horizontal engine, how is the height of the center line of the shaft usually left with regard to the center line of the cylinder? (*b*) Why is this done?

(25) Describe the method of fitting the shaft to the brasses after the latter have been lined up.

(26) In lining up an engine, suppose the crank-shaft to be in position and that it is desired to further test the shaft for being at right angles to the center line of the cylinder. How may this be done by means of the line stretched through the center of the cylinder?

(27) Describe one method of testing the crankpin for parallelism with the crank-shaft.

(28) How may it be ascertained whether or not the center lines of the connecting-rod brasses are in the same plane?

(29) What is probably the most frequent cause of pounding in engines?

(30) How may the necessity of stripping brass-bound boxes be provided against?

(31) Why is a loose piston nut very liable to cause a breakdown?

(32) Explain how too little compression may cause an engine to pound.

(33) (*a*) Will too late a release cause an engine to pound? (*b*) Give a reason for your answer.

(34) In case it is desirable not to allow sufficient water to enter the circulating pump to stop its pounding, how may the pounding be stopped?

ENGINE MANAGEMENT.

(PART 2.)

EXAMINATION QUESTIONS.

(1) What are the arguments for and against (*a*) solid bearings? (*b*) adjustable bearings?

(2) If a bearing shows an inclination to heat, how should it be treated?

(3) What is the objection to pouring cold water on hot bearings?

(4) If a bearing becomes exceedingly hot and the engine cannot be stopped long enough to allow the bearing to cool, what should be done in order to continue running the engine?

(5) What is meant by "wearing down a bearing"?

(6) If the brasses of a large journal are removed, why is the bearing very apt to heat up after the brasses have been replaced?

(7) (*a*) May loose brasses cause a bearing to heat? (*b*) How?

(8) Describe two methods of setting up bearings.

(9) What is the remedy for warped brasses?

(10) What is usually the cause of the chronic heating of bearings?

(11) (*a*) Explain why brasses that have been quickly and excessively heated are liable to pinch the journal near their edges. (*b*) How may this be prevented?

(12) How should the heating of bearings due to gritty or dirty oil be guarded against?

(13) (*a*) What one property in particular should oil for large bearings possess? (*b*) Why?

(14) In large bearings how should the bearing surfaces of the brasses be finished in order to aid the oil to penetrate between the journal and brasses?

(15) On what does the pressure that a bearing will sustain per square inch of rubbing surface without heating depend?

(16) Explain why an overloaded engine may cause the bearings to heat.

(17) Why is a little side play in a journal a good thing?

(18) On what does the value of a lubricant depend?

(19) Mention some of the desirable features of a good lubricating oil.

(20) (*a*) Can the lubricative qualities of an oil always be judged by its specific gravity or its viscosity? (*b*) Why?

(21) (*a*) What is the best animal oil for lubricating machinery? (*b*) What is probably the best vegetable oil?

(22) (*a*) What are the sources of mineral oils, and (*b*) how are they graded?

(23) Why should a compounded oil having a mineral oil base not be used as a cylinder lubricant?

(24) How are boiled, or cup, greases made?

(25) How does a lubricant prevent the rubbing surfaces becoming hot?

(26) Why are mineral oils especially adapted to lubricating pistons and slide valves working under a high steam pressure?

(27) How does the temperature of a bearing affect the lubricating power of an oil?

(28) How may the comparative viscosity of greases be approximately judged?

(29) What is meant by the flashing point of an oil?

(30) If the purity of a mineral oil is doubtful, how may it be determined if animal or vegetable oils have been mixed with it?

(31) What property of an oil is increased by adulterating it with paraffin, waxes, gums, etc.?

(32) If a mineral oil is darkened in color after it has been heated to 300° F. for a few minutes, what adulterant is probably contained in the oil?

(33) (*a*) Explain the principle of the water-displacement lubricator. (*b*) What are the objections to the use of this lubricator?

(34) (*a*) Should a double-connection lubricator ever have one connection attached to the steam pipe between the throttle and boiler and the other between the throttle and engine? (*b*) Why?

(35) (*a*) Into what three classes may steam lubricators be divided? (*b*) On what does the operation of the hydrostatic lubricator depend?

(36) What is the principal difference between double- and single-connection lubricators?

ENGINE INSTALLATION.

EXAMINATION QUESTIONS.

(1) (*a*) What are the two principal features that dictate the use of vertical engines? (*b*) Why are these the controlling features?

(2) Mention some of the most important advantages the horizontal engine has over the vertical engine.

(3) (*a*) Can the automatic cut-off engine be made to give as good steam economy as the releasing-gear engine? (*b*) If so, what is the type of the automatic cut-off engine?

(4) Are there any throttling engines built that give a fairly economical steam consumption?

(5) What two leading factors determine the use of compound engines?

(6) In deciding between the use of a simple or compound engine, what elements, besides first cost and economy of fuel, should be considered?

(7) What is the principal disadvantage of tandem compound engines?

(8) Explain why the mechanical efficiency of a cross-compound engine is greater than the efficiency of a tandem compound engine of equal power.

(9) What is the average proportion existing between the volume of the cylinders of tandem and cross-compound engine and the volume and reheating surface of the reheating receiver?

(10) Why do tandem compound condensing engines generally run smoother than cross-compound condensing engines of equal power ?

(11) Mention the principal advantages and disadvantages of high-speed engines.

(12) (*a*) What two forms of valves are principally used on high-speed engines ? (*b*) What are their relative advantages and disadvantages ?

(13) How do high- and low-speed engines compare in regard to closeness of regulation ?

(14) Explain one manner in which the fast rotative speed of high-speed engines is conducive to economy in steam consumption.

(15) Mention some of the advantages of high-speed engines for direct-connected work.

(16) In the slow-speed engine, what means are taken to secure extreme economy of steam and high mechanical efficiency ?

(17) (*a*) What is the principal advantage of high-speed engines for direct-connected electrical work ? (*b*) Why are they not used for large direct-connected units ?

(18) If an engine of 150 horsepower is to be direct-connected to an electric generator, what type of engine should be selected to give a maximum efficiency ? The steam pressure is to be 120 pounds; there is plenty of water available, but the cost of fuel is high.

(19) Why is the compound condensing engine better adapted to continuous running with varying loads than the compound non-condensing engine ?

(20) Why is a simple non-condensing engine usually selected when the power required is continuous and uniform, but is liable to be increased owing to a growth in the business ?

(21) What influence has the cost of fuel on the type of engine to be selected for any particular service ?

(22) If superheated steam is used, will jacketing the high-pressure cylinder increase the economy of the engine ?

(23) What is the general principle of the cooling tower ?

(24) Mention some of the advantages gained by installing a number of small engines instead of one large one in a manufacturing plant.

(25) How are the vibrations of engines used on the upper floors of buildings sometimes absorbed ?

(26) (*a*) Of what materials are engine foundations usually made ? (*b*) What kind of mortar should be used ?

(27) (*a*) What is the objection to building an engine foundation directly on solid rock ? (*b*) How is this objection usually overcome ?

(28) How are engine foundations usually constructed where it is necessary to go to a great depth in order to find a sufficiently hard bottom to support the load ?

(29) What is the object of setting the capstone for the outboard bearing of an engine lower than the actual figures called for ?

(30) How are the foundation bolts for an engine located and held in position while the masonry is being built ?

(31) After an engine is in place on its foundation, how is the irregular space between the bed and foundation usually filled ?

ELEVATORS.

(PART 1.)

EXAMINATION QUESTIONS.

(1) How are elevators usually classified with reference to the motive power used?

(2) What is meant by a corner-post elevator?

(3) What do you understand by the term "drum type" of elevator?

(4) In the drum type of elevator, how is the rope, as it winds upon the drum, prevented from jumping the grooves of the drum by its deflection?

(5) (*a*) What do you understand by overbalancing an elevator? (*b*) What type of elevator cannot be overbalanced? (*c*) Why?

(6) What is the advantage of overbalancing an elevator?

(7) How may the change in the counterbalancing due to the weight of the rope when the car is in different positions be compensated?

(8) What are the objections to the simple shipper rope for operating an elevator?

(9) Into what two classes may safety devices be divided?

(10) In what two forms is the motor of the hand elevator usually represented?

(11) Are hand elevators usually overbalanced or underbalanced, if they are balanced at all?

(12) Why should all elevators be started and stopped gradually?

(13) (*a*) Why are wire ropes used in elevator work made with hemp centers? (*b*) When should a wire rope be condemned as dangerous?

(14) Mention three preparations for lubricating wire ropes.

(15) In fastening the rope to the drum, what precaution should be observed in order to reduce the stress at the point of fastening?

(16) Why should not the guides be allowed to become gummy?

(17) What do you understand by the term belt elevator?

(18) Why are worm-geared belt elevators usually overbalanced while spur-geared ones are not?

(19) How are the limit stops on the shipper rope of belt elevators usually made?

(20) Briefly describe the principle of the most common form of motor limit stop.

(21) What provision is usually made on elevators to prevent the cable from unwinding should the car stick in its descent?

(22) In worm-geared elevators how may the end thrust due to the use of a worm be avoided?

(23) What is the maximum speed at which belt elevators should be run?

(24) In worm-geared elevators what lubricant should be used for the worm bath?

(25) In general, what precautions should be taken in the maintenance of belt elevators?

(26) Explain briefly the principle of the mechanism for reversing the engines of an Otis spur-gear steam elevator.

(27) What is the general principle of the slack-cable safety provided on all steam elevators?

ELEVATORS.

(PART 2.)

EXAMINATION QUESTIONS.

(1) Mention (*a*) the different kinds of electric motors used in elevator service, and (*b*) the service to which each is particularly adapted.

(2) Why should the main switch be closed with all the starting resistance in the armature circuit?

(3) Briefly describe the operation of the rheostat shown in Fig. 1 of the text.

(4) In the reversible switch shown in Fig. 5 of the text, how may sparking at the clips connected to the shunt field be prevented when the circuit is being opened?

(5) (*a*) What do you understand by a solenoid rheostat? (*b*) What is one advantage of this type of rheostat?

(6) What two important conditions must be fulfilled by the motor of a direct-connected electric elevator?

(7) What are the only alternating-current motors that are satisfactory for direct-connected electric elevators?

(8) Are direct-connected electric elevators of the drum type over or under counterbalanced?

(9) In electric elevators, by what different means may the brake be operated?

(10) (*a*) Briefly describe the simple controller used by the Elektron Manufacturing Company. (*b*) What is the

reason for turning the shipper sheave through such a wide angle in order to reverse the motor ?

(11) In the electrical-mechanical brake used by the Elektron Manufacturing Company, how is the rapidity of action of the brake controlled ?

(12) Briefly describe what takes place when the above elevator is started or stopped.

(13) Describe the principle of the dynamic brake made by the Elektron Manufacturing Company.

(14) (*a*) When a dynamic brake is used, why is the field kept excited after the armature circuit is broken and the armature short-circuited ? (*b*) How is this done in the Elektron Manufacturing Company's brake ?

(15) (*a*) What is the peculiarity of the step bearing used on the A. B. See elevator shown in Fig. 17 of the text ? (*b*) What is the advantage of this arrangement ?

(16) What motor safeties are applied to the A. B. See elevator ?

(17) (*a*) In the Otis single-worm elevator, how is an increased thrust-bearing surface obtained without increasing the size of the shaft ? (*b*) How is the pressure between the bearing surfaces equalized ?

(18) What is the object of the safety magnet used on the Otis elevator ?

(19) (*a*) In the Otis high-speed elevators, what provision is made for stopping the elevator almost instantly when the limits of travel are reached ? (*b*) How is the operating device arranged to prevent accidental reversal of the motor in stopping ?

(20) What is the use of the potential switch made by the Otis Company ?

(21) Briefly describe the potential switch as it is made when it only operates when the current falls below the normal.

(22) (*a*) Give the general features of the magnet-control method of operating elevators. (*b*) What are its advantages over the rheostat method ?

(23) Explain the operation of the Otis G. S. controller when the handle of the car controlling switch is moved to the "fast down" position.

(24) (*a*) What is the main difference between the Otis G. S. magnet controller and the No. 6 controller? (*b*) Explain the action of the No. 6 controller when the handle of the operating switch is moved to the "down" position.

(25) (*a*) Describe briefly the operation of the Otis automatic elevator with older style floor controller when the car is at the first floor and the button on the fourth floor is pressed. Also, when the passenger enters the car and pushes the button to descend to the second floor. (*b*) Show how the circuits are arranged so that the movements of the elevator cannot be interfered with when it is already in use.

(26) Describe briefly the operation and distinctive features of the Sprague-Pratt electric elevator.

(27) Explain the operation of the Fraser differential-speed elevator.

ELEVATORS.

(PART 3.)

EXAMINATION QUESTIONS.

(1) What are some of the advantages and disadvantages of hydraulic elevators?

(2) (*a*) What do you understand by a plunger elevator? (*b*) For what kinds of service are they mostly used?

(3) Why cannot plunger elevators be overbalanced?

(4) In a balanced plunger elevator, what would happen if the car became loose from the plunger?

(5) How is it that the controlling valve in a hydraulic elevator acts as a power control and brake at the same time?

(6) How may the rapid descent of a plunger elevator be provided against should the controlling valve fail to work?

(7) What is the principal advantage of the piston elevator over the plunger elevator?

(8) In vertical piston elevators in which the cylinder is always full of water, why is it preferable to put as much of the counterweight as is possible directly on the piston or piston rods?

(9) (*a*) What is the object of making the water circulate from the top to the bottom of the piston in vertical piston elevators? (*b*) Explain your answer.

(10) What is the object of placing a relief valve in the discharge pipe between the cylinder and the controlling valve ?

(11) (*a*) What is the purpose of the pilot valve? (*b*) Why is its use necessary in high-speed elevators ?

(12) Explain the use of the throttle placed between the upper and lower pistons of the main controlling valve.

(13) (*a*) What do you understand by a double-power hydraulic elevator ? (*b*) Where are they used ?

(14) In hydraulic elevators in which the ratio of car travel to piston travel is very high, the water is admitted to but one side of the piston. Why is this done ?

(15) What are the principal advantages of horizontal hydraulic piston elevators over vertical hydraulic piston elevators ?

(16) What do you understand by the terms "tension type" and "compression type" as applied to horizontal hydraulic piston elevators ?

(17) In the horizontal tension type, why is the short distance required between the sheaves a decided advantage ?

(18) When a closed pressure tank is used, how is the pressure within the tank kept practically constant while the elevator is in operation ?

(19) (*a*) In what ways may the air escape from closed pressure tanks ? (*b*) How is it replenished ?

(20) Briefly describe the Ford regulating valve.

(21) How is the Ford regulating valve modified so as to operate the switch and rheostat of an electric motor ?

(22) What is the object of using a by-pass valve to open a communication between the suction and delivery pipes of the pressure pump ?

(23) If the absorption of the air by the water in the pressure tank is excessive, how may it be prevented ?

(24) In filling the pressure tank, how may sufficient air be introduced to give the required pressure ?

(25) If a large quantity of air collects in the cylinder, how may it be removed ?

(26) In what manner will a worn or leaking piston packing indicate itself ?

(27) Briefly describe an effective method of lubricating the internal parts of elevator plants.

(28) In the vertical circulating hydraulic elevator, how may the water be removed from the cylinder and circulating pipe ?

(29) Briefly describe the necessary steps in packing a vertical cylinder piston from the top.

(30) If the cylinder of a horizontal hydraulic elevator is badly worn, how should the piston be packed ?

(31) If the packing used is made of cotton, how should it be treated to remove the air from the pores ?

ELEVATORS.

(PART 4.)

EXAMINATION QUESTIONS.

(1) Mention some of the means by which car safeties are set in operation.

(2) (*a*) For what kind of elevators is the pawl-and-ratchet safety suitable? (*b*) By what arrangement is the pawl and ratchet usually replaced?

(3) Why is it necessary to attach each cable to a separate wedge-operating lever when a gravity-wedge safety is used?

(4) (*a*) In high-speed elevators, what else besides a broken or a slack cable is usually made to operate the safety devices? (*b*) By what apparatus is this usually accomplished?

(5) Briefly describe how the governor on an Otis elevator applies the car safety.

(6) When a safety drum is used, what is the object of also having a governor-controlled brake?

(7) Why should not the guides be allowed to become gummy?

(8) Before unlocking the safety after it has been set, what precaution should be taken?

(9) If a car has been stopped above the top landing by a wedge-safety device, how would you proceed to lower the car?

(10) Briefly describe the air-cushion safety and its operation.

(11) (*a*) In the air-cushion safety, what should be the depth of the pit compared with the height of the lift? (*b*) In high lifts, how is this depth of pit obtained?

(12) (*a*) Mention some of the objections to solid walls or partitions for elevator shafts. (*b*) If the partitions are not carried to the ceilings, how high should they be?

(13) Mention some of the requirements of elevator doors.

(14) In some elevators, how is the car prevented from starting before the door is closed?

(15) In passenger service, how is the movement of the car sometimes indicated to the would-be passengers?

(16) For what class of service are escalators particularly adapted?

A KEY

TO ALL THE

QUESTIONS AND EXAMPLES

CONTAINED IN THE

EXAMINATION QUESTIONS

INCLUDED IN THIS VOLUME.

The Keys that follow have been divided into sections corresponding to the Examination Questions to which they refer, and have been given corresponding section numbers. The answers and solutions have been numbered to correspond with the questions. When the answer to a question involves a repetition of statements given in the Instruction Paper, the reader has been referred to a numbered article, the reading of which will enable him to answer the question himself.

To be of the greatest benefit, the Keys should be used sparingly. They should be used much in the same manner as a pupil would go to a teacher for instruction with regard to answering some example he was unable to solve. If used in this manner, the Keys will be of great help and assistance to the student, and will be a source of encouragement to him in studying the various papers composing the Course.

THE STEAM ENGINE.

(1) The head end of a cylinder is the end farthest away from the crank-shaft; the crank end is the end nearest the crank-shaft. See Art. **4.**

(2) The distance passed over by the piston in moving from one extreme position in the cylinder to the other. See Art. **4.**

(3) The cylinder with its heads and steam chest, the steam pipe and exhaust pipe, the guides, the bed, and the bearings. See Art. **6.**

(4) The mechanism by which the steam is distributed, when considered as a whole, is termed the valve gear. See Art. **6.**

(5) The linear distance between the piston and the cylinder head when the former is at the end of its stroke is the piston clearance. The volume of this space added to the volume of the steam port leading to it is the clearance volume. See Art. **7.**

(6) The diameter of the circle described by the center of the eccentric is generally defined as the throw, but some people consider the eccentricity as the throw. See Art. **12.**

(7) By outside lap is meant the amount that the outside edge of the valve projects beyond the edge of the steam port when in mid-position; by inside lap is meant the amount that the inside edge of the valve projects beyond the edge of the steam port when in mid-position. See Art. **15.**

(8) No. See Art. **16.**

(9) The angle the eccentric radius makes with the position it would occupy if the valve had neither outside lap nor lead. See Art. **20.**

(10) Increase the outside lap of the valve, increase the valve travel, and the angle of advance. See Arts. **29, 60, 62,** and **63.**

(11) The exhaust port will be opened later and closed earlier. See Art. **30.**

(12) Lead is the distance the steam port is opened when the piston is at the end of its stroke. See Art. **31.**

(13) Ahead, irrespective of whether the engine runs under or over. See Art. **32.**

(14) By using a rocker-arm. See Art. **34.**

(15) No. The eccentric must be placed behind the crank. See Art. **35.**

(16) The exhaust port will be closed later on the forward stroke than on the return stroke, and hence there will be more compression at the head end. See Art. **41.**

(17) To double the port opening. See Arts. **42** and **43.**

(18) See Art. **46.**

(19) The volume swept through by the piston is $14^2 \times .7854 \times 28 = 4{,}310$ cubic inches. Then, the clearance is $\frac{247 \times 100}{4{,}310} = 5.73$ per cent. Ans. See Art. **51.**

(20) The apparent cut-off is $\frac{21 \times 100}{60} = 35$ per cent. Applying rule **1**, Art. **55,** we find the real cut-off to be $\frac{(35 + 2) \times 100}{100 + 2} = 36.27$ per cent. Ans.

(21) The ratio of expansion, by Art. **56,** is $\frac{100}{36.27} = 2.76$. Ans.

(22) Since the lap circle is smaller, without having changed the position of its center, the point *d* of Fig. 25 of

the text will be farther to the right, and, consequently, the perpendicular dropped from it will also be farther to the right, thus showing that cut-off will take place later. Since the circumference of the lap circle is at a greater distance than before from the line $a\ b$, the lead line $g\ h$ is also farther away from it, which shows that the lead will be greater. The points of release and exhaust closure will not be affected at all, but the port opening will be greater. In case the port was opened originally its full width, the port opening obviously cannot be greater now than the width of the port; in this case it shows that the valve overtravels, i. e., moves in the same direction in which it moved to open the port after the port is wide open. The amount of overtravel will be equal to the difference in the width of the port and the distance of the circumference of the lap circle from the center o. See Arts. **60**, **61**, and **62**.

(23) The piston speed is $\frac{80}{12} \times 75 \times 2 = 1{,}000$ feet per minute. The area of the piston is $44^2 \times .7854 = 1{,}520$ square inches, nearly. Applying rule **2**, Art. **66**, we get

$$c = \frac{1{,}520 \times 1{,}000}{6{,}000} = 253.3 \text{ square inches}$$

as the area of the steam pipe. The corresponding diameter is $\sqrt{\frac{253.3}{.7854}} = 18$ inches, nearly. By applying rule **3**, Art. **67**, we get

$$b = \frac{1{,}520 \times 1{,}000}{4{,}000} = 380 \text{ square inches}$$

as the area of the exhaust pipe. The corresponding diameter is $\sqrt{\frac{380}{.7854}} = 22$ in., nearly. Ans.

(24) The piston speed is 1,000 feet per minute. Applying rule **4**, Art. **68**, we get

$$a = \frac{1{,}520 \times 1{,}000}{7{,}500} = 202.7 \text{ sq. in.} \quad \text{Ans.}$$

THE INDICATOR.

(1) The pressure per square inch required to raise the pencil 1 inch on the paper drum. See Art. **3.**

(2) About equal to one-half the boiler pressure. See Art. **3.**

(3) A reducing motion is a mechanism used to communicate the motion of the crosshead, on a reduced scale, to the paper drum of an indicator. See Art. **10.**

(4) Their joints are apt to become loose and thus distort the diagram. See Art. **21.**

(5) By Art. **19,** $UV = \frac{40 \times 3}{24} = 5$ in. Ans.

(6) See that it is clean and in working order. The joints of the various links should be free but not slack enough to allow the pencil to shake. Also, see that there is no backlash between the piston and spring. See Art. **27.**

(7) The vacuum line is the line of no pressure and is located below and parallel to the atmospheric line at a distance equal to the atmospheric pressure divided by the scale of the spring used to make the diagram. See Art. **31.**

(8) Corliss diagrams have the events of the stroke sharply defined and the expansion and compression curves are comparatively even. Diagrams from high-speed engines do not show the events so distinctly and the expansion and compression lines are usually irregular. See Art. **34.**

(9) (*a*) Decrease the angular advance of the eccentric. See Art. **36.**

(*b*) Cut-off, release, and compression will be later. See Art. **36.**

(10) Change the length of the valve stem so as to make the cut-off occur earlier on the end doing the greater amount of work. See Art. **44.**

(11) (*a*) The higher the speed, the more compression is needed. See Art. **49.**

(*b*) $\frac{8}{10}$ the initial pressure with high-speed engines, $\frac{5}{10}$ with medium-speed engines, and from $\frac{2}{10}$ to $\frac{3}{10}$ with slow-speed engines. See Art. **49.**

(12) (*a*) There is probably some restriction in the steam passage leading from the boiler to the cylinder. See Art. **56.**

(*b*) The free escape of the exhaust is prevented. See Art. **57.**

(*c*) The valve leaks and allows steam to enter after cut-off. See Art. **53.**

(13) (*a*) Admission, release, and compression are all too late. The back pressure is also excessive.

(*b*) Shift the eccentric ahead on the shaft and make the exhaust port or exhaust pipe larger.

(14) The rounding of the corners at the beginning of the stroke indicates that admission is a trifle late, otherwise the diagrams are very good. See Art. **37.**

(15) An account of the reevaporation of some of the condensed steam near the end of the stroke. See Art. **52.**

(16) By prolonging the expansion line and noting where it leaves the actual line of the diagram. See Art. **54.**

(17) (*a*) To the vibrations of the pencil motion when there is a sudden change of pressure. See Art. **58.**

(*b*) To the sticking of the indicator piston. See Art. **59.**

(18) Compression is too early on the head end, as is shown by the compression line extending above the steam line. There is also an unequal distribution of power in the two ends of the cylinder, as shown by the cut-off being much later on the crank-end diagram, which shows that the valve gear has been displaced. Release is too late on the crank-end diagram. See Arts. **42** and **44.**

(19) The valve on the engine from which Fig. 20 was taken had no lead, while that from which Fig. 22 was taken had lead. See Art. **42.**

(20) No. The steam might leak out as fast as it leaks in. See Art. **53.**

(21) Admission, cut-off, and release are too late on the crank end of the cylinder, thus giving an excess of power above the head end.

ENGINE TESTING.

(1) (*a*) When a gas expands and does work and no heat is added to it from an outside source the expansion is said to be adiabatic. See Art. **7.**

(*b*) When the temperature of a gas is kept constant during expansion, the expansion is said to be isothermal. See Art. **8.**

(2) The work is the same. See Art. **11.**

(3) If dry, saturated steam expands adiabatically, some of it will be condensed, and the relation between pressure and volume will then depend on the proportion of water present in the mixture. If, on the other hand, the expansion is isothermal, the steam will be superheated. See Art. **17.**

(4) (*a*) By rule 1, Art. **20,** the work is

$W = 144\,P\,V = 144 \times 45.6 \times 6.3 = 41{,}368.32$ ft.-lb. Ans.

(*b*) The number of foot-pounds per minute is $41{,}368.32 \times 76$, and the horsepower developed is $\frac{41{,}368.32 \times 76}{33{,}000} = 95.27$ H. P. Ans.

(5) (*a*) The mean ordinate is the ordinate whose length is the average of all the ordinates of the diagram. See Art. **24.**

(*b*) It is found by dividing the area of the diagram by its length. In case the area is not known, divide the length of the diagram into a number of equal parts, and half way between these parts or points of division draw vertical lines extending from the upper to the lower lines of the diagram. To find the mean ordinate, add the lengths of these ordinates or vertical lines and divide by their number. See Arts. **24** and **26.** It may also be found by a planimeter.

(6) (*a*) Volume of cylinder = area × length = 1 × 2 = 2 cubic feet. The horizontal scale of volumes

$$= \frac{\text{volume of cylinder in cubic feet}}{\text{length of card in inches}} = \tfrac{2}{3} \text{ cu. ft. per in.} \quad \text{Ans.}$$

See Art. **21.**

(*b*) Work per stroke of piston = area × horizontal scale × vertical scale × 144 or work $= 6.75 \times \frac{2}{3} \times 50 \times 144 = 32{,}400$ ft.-lb. Ans. See Art. **26.**

(7) By taking the difference between the indicated horsepower when the engine is running loaded and the indicated horsepower when it is running light. See Art. **32.**

(8) The approximate net H. P. = I. H. P. − friction H. P. $= 176.8 - 25.6 = 151.2$. The efficiency by rule **2**

$$= \frac{100 \times 151.2}{176.8} = 85.5 \text{ per cent.} \quad \text{Ans.}$$

See Art. **33.**

(9) $\frac{2.75}{3.15} \times 50 = 43.65$ lb. per sq. in., M. E. P. Ans.

See Art. **36.**

(10) By passing around the diagram two or three times and noting the reading at the return to the starting point, each time. The difference between the readings should be the same each time or the last reading divided by the number of readings should equal the first reading. See Art. **41.**

(11) By subtracting the sum of the lengths of the ordinates of the loops from the sum of the lengths of the ordinates of the main part of the diagram and dividing by the number of ordinates. See Art. **45.**

(12) Using rule 3 and the table, Art. **46,** $75 + 14.7 = 89.7$. From the table the constant for $\frac{5}{8}$ cut-off is .904, and

.904 × absolute pressure = .904 × 89.7 = 81.1. M. E. P. = (81.1 − 17) × .9 = 57.69 lb. per sq. in. Ans.

(13) (*a*) Piston speed is the distance traveled by the piston in 1 minute.

(*b*) By rule **5**, Art. **48**, $S = \frac{24 \times 180}{6} = 720$ ft. per min. Ans.

(14) It is a number obtained by combining into a single factor all the constant horsepower factors for that engine. See Art. **50.**

(15) (*a*) The length L of the stroke is $\frac{24}{12} = 2$ feet; the area A of the piston is $18^2 \times .7854 = 254.47$ square inches, and the number of strokes N is $2 \times 185 = 370$. Substituting these values in the formula corresponding to rule **6**, we have

$$C_u = \frac{\frac{24}{12} \times 254.47 \times 185 \times 2}{33{,}000} = 5.706. \quad \text{Ans.}$$

(*b*) Multiplying the engine constant by the M. E. P., we have I. H. P. = 5.706 × 53.8 = 306.98. Ans. See Art. **51.**

(16) (*a*) Scale of spring $s = 60$; length L of stroke is $\frac{36}{12} = 3$ feet; area A of piston $= 24^2 \times .7854 = 452.39$ square inches, and the number N of working strokes is $2 \times 150 = 300$ per minute. The number n of ordinates is 20. Substituting these values in rule **10**, we have

$$C_o = \frac{60 \times 3 \times 452.39 \times 300}{33{,}000 \times 20} = 37.013. \quad \text{Ans.}$$

(*b*) The I. H. P. = 37.013 × 16 = 592.2. Ans. See Art. **56.**

(17) The horsepower measured by some type of absorption dynamometer. See Art. **60.**

(18) The net pressure on the scale = 274.5 − 14.5 = 260 pounds. Substituting, in rule **11**, Art. **61**, we have

$$\text{H. P.} = \frac{260 \times 4 \times 225 \times 6.2832}{33{,}000} = 44.55. \quad \text{Ans.}$$

(19) The lever arm in this case is $\frac{\frac{47}{2} + \frac{1}{2}}{12} = 2$ feet and the net pull is $241 - 10 = 231$ pounds. Substituting in rule **11**, Art. **61**, we have

$$\text{H. P.} = \frac{231 \times 2 \times 350 \times 6.2832}{33{,}000} = 30.78. \quad \text{Ans.}$$

(20) When fresh steam enters the cylinder, part of it is condensed and is not taken account of by the indicator diagram. See Art. **65.**

(21) By calculating the water consumption at cut-off and then at release. See Art. **67.**

(22) Length of stroke $= \frac{20}{12} = 1\frac{2}{3}$ feet; hence, each inch of length of diagram equals $\frac{1\frac{2}{3}}{3.25} = .51$ foot of stroke. As a 45-pound spring was used to make the diagram, $a\,m = .70 \times 45 = 31.5$ pounds and $b\,n = .62 \times 45 = 27.9$ pounds. Also, $O\,m = 3.41 \times .51 = 1.74$ feet and $O\,n = .36 \times .51 = .18$ foot. Area of piston $= \frac{16^2 \times .7854}{144} = 1.396$ square feet. The volume of steam in the cylinder when the piston is at the point represented by a is $1.74 \times 1.396 = 2.429$ cubic feet. The volume when the piston is at b is $.36 \times 1.396 = .50256$ cubic feet. The weight of a cubic foot of steam at an absolute pressure of 31.5 pounds is .07536 pound, and at a pressure of 27.9 pounds the weight is .06931 pound. The weight of steam in the cylinder is $.07536 \times 2.429 = .183049$ pound and the weight saved by compression is $.06931 \times .50256 = .034832$ pound. The steam used per stroke is $.183049 - .034832 = .148217$ pound. See Art. **66.**

The M. E. P. $= \frac{2.41 \times 45}{3.25} = 33.37$ pounds. See Art. **36.**

The I. H. P. is $\frac{33.37 \times 1\frac{2}{3} \times 16^2 \times .7854 \times 160 \times 2}{33{,}000} = 108.4.$ See Art. **47,** rule **4.** The water consumption per I. H. P. per hour is $\frac{.148217 \times 160 \times 2 \times 60}{108.4} = 26.25$ lb. See Art. **66.**

(23) The pressure at a is $50 \times .71 = 35.5$ pounds, absolute. The weight of a cubic foot of steam at this pressure is .086916 pound. Using rule **12**, we have

$$Q = \frac{13,750 \times 2.93 \times .086916}{37.5 \times 3.36} = 27.79 \text{ lb. per I. H. P. per hr.}$$

See Art. **68.** Ans.

CONDENSERS.

(1) On account of the air contained in the exhaust steam and the air that leaks in around the piston rod and valve stems, and also on account of the vapor from the water. See Art. **9.**

(2) To remove the air, vapor, and condensed steam and sometimes, also, the condensing water from the condenser. See Art. **11.**

(3) (*a*) In a jet condenser the exhaust steam is condensed by mingling with the condensing water. See Art. **15.**

(*b*) In a surface condenser the exhaust steam is condensed by coming into contact with metallic surfaces that are kept cool by water constantly flowing over them. See Art. **16.**

(4) Water is apt to be drawn into the cylinder and cause the blowing out of a cylinder head. See Art. **21.**

(5) When the condenser is deprived of injection water. See Art. **21.**

(6) A cubic inch of water converted into steam at atmospheric pressure occupies 1,646 cubic inches; therefore, if 1,646 cubic inches of steam contained in a closed vessel at atmospheric pressure are condensed into water, a vacuum will be formed. Thus, by making the engine exhaust into

a closed vessel and condensing the exhaust steam, the pressure on the exhaust side of the piston is made less and consequently the net pressure on the piston will be increased. See Arts. **9** and **11.**

(7) By means of an indicator card. See Art. **12.**

(8) By compounding the steam cylinders of the independent air and circulating pumps; or, in case of multiple-expansion engines, by running the exhaust of the independent engines into the receiver of the low-pressure cylinder of the main engine. See Art. **13.**

(9) By a column of water flowing downwards through a vertical pipe not less than 34 feet long and having its lower end immersed in the water of the hotwell. See Art. **29.**

(10) To increase the velocity of the falling water, which improves the action of the condenser. See Art. **30.**

(11) (*a*) No. If there is a vacuum of 24½ inches, the pump is required to force the water but 7 feet. See Art. **31.**

(*b*) Because the vacuum assists the circulating pump. See Art. **31.**

(12) (*a*) A surface condenser. See Art. **40.**

(*b*) Because the exhaust steam does not come into direct contact with the impure injection water. See Art. **40.**

(13) (*a*) Two. See Art. **41.**

(*b*) The circulating pump forces the injection water through the condenser tubes and the air pump removes the air, vapor, and water of condensation from the condenser. See Art. **41.**

(14) The surface condenser is more complicated, costs more, and requires more attention than the jet condenser, but the pure feedwater obtained more than compensates for these disadvantages. See Art. **42.**

(15) It relieves the condenser of any excess of steam, air, or vapor that may accumulate within it, and it allows

the engine to be run non-condensing should the air pump become inoperative. See Art. **43.**

(16) Because the water in the boiler will soon become as impure as the injection water and it will then be necessary to blow off some of the very impure water and to replace it with less impure water, thus causing a serious loss of heat. See Art. **45.**

(17) (*a*) The coating of the tubes by the grease carried from the cylinder by the exhaust steam. See Art. **46.**

(*b*) If animal or vegetable oils are used, the condenser may be cleaned by boiling it out with a solution of caustic soda or caustic potash. If mineral oils are used, the tubes must be removed and the grease scraped from them by hand. See Art. **46.**

(*c*) By introducing a grease extractor in the exhaust pipe. See Art. **48.**

(18) (*a*) Copper, 70 per cent.; zinc, 29 per cent.; and tin. 1 per cent. See Art. **50.**

(*b*) They should be tinned both inside and outside. See Art. **50.**

(19) (*a*) By a screw packing gland that prevents leakage between the tube and tube-sheet and yet allows the tube to freely change its length. See Art. **51.**

(*b*) Because if they were rigidly fastened to the tube-sheets, they would become greatly distorted through unequal expansion and contraction. See Art. **52.**

(20) By removing the condenser bonnets and filling the steam side of the condenser with water. See Art. **53.**

(21) By a falling vacuum. See Art. **54.**

(22) To prevent the formation of a galvanic current between the copper of the tubes and the iron of the condenser casing, feedpipes, and boilers. See Art. **55.**

(23) On the principle that the evaporation of a part of the water undergoing the cooling process extracts the heat from the remaining part. See Art. **57.**

(24) The amount of surface exposed, the condition of the air with regard to the moisture it contains, the temperature of the air, and the amount of air brought in contact with the water. See Arts. **59** and **60.**

(25) To evaporate a pound of water at the atmospheric pressure requires 966 B. T. U., and as this amount of heat must come from the condensing water, the remaining water must be cooled. See Art. **61.**

(26) The cooling tower consists of a tower about 30 feet high, to the top of which the water to be cooled is delivered. As the water descends from the top of the tower, it meets an ascending current of air and also with obstructions so placed that it falls in a fine spray or thin sheets, thus exposing a large area of evaporating surface to be acted upon by the air. See Art. **62.**

(27) (*a*) By radiation, by contact with the cool air, and by evaporation. See Art. **63.**

(*b*) Evaporation. See Art. **63.**

(28) The Linde system of cooling discharge water consists of a number of horizontal thin metallic cylinders immersed to one-third their diameters in the condensing water. By revolving the cylinders a thin film of water, which adheres to the surface of the cylinders, is brought into contact with a current of air and thus produces a cooling effect. See Art. **71.**

(29) By allowing a certain amount of air to enter the barrel of the circulating pump at each stroke, or by means of a regurgitating valve. See Arts. **75** and **76.**

(30) By the use of a vacuum breaker. See Art. **82.**

(31) On the initial and final temperature of the steam and on the initial and final temperature of the condensing water. See Art. **83.**

(32) From the Steam Tables we find H to be 1,131.462. Applying rule **1**, we get

$$W = \frac{1{,}131.462 - (135 - 32)}{105 - 55} = 20.57 \text{ lb., nearly.} \quad \text{Ans.}$$

(33) Not enough condensing water is being used. See Art. **87.**

(34) From the Steam Tables, H for 3 pounds absolute is 1,125.144. Applying rule **1**, we get

$$W = \frac{1{,}125.144 - (130 - 32)}{130 - 65} = 15.8 \text{ lb.} \quad \text{Ans.}$$

(35) About 2 inches less than the theoretical vacuum. See Art. **88.**

COMPOUND ENGINES.

(1) The pressures, and consequently the strains, are less, and are more evenly distributed throughout the stroke; the turning moment on the shaft is more uniform, which allows lighter parts to be used; the flywheel can also be lighter, and thus there is less friction. See Art. **9.**

(2) On account of initial cylinder condensation. See Art. **3.**

(3) By expanding the steam successively in several cylinders the range of temperature in each cylinder is reduced, as is also the fluctuation in temperature of the cylinder walls of each. As the rate of heat transmission is proportionally much smaller for a small than for a large fluctuation of temperature, the sum of the condensation losses in the several cylinders will be smaller than the condensation losses in a single cylinder having the same temperature range between the initial and final temperature. See Art. **7.**

(4) The greater part of the condensed steam in the high-pressure cylinder is probably reevaporated during the exhaust period and enters the low-pressure cylinder as steam, and thus does some work. See Art. **8.**

(5) (*a*) A tandem compound engine. See Art. **14.**

(*b*) Because the turning moment of the tandem compound engine is not as uniform as that of the cross-compound engine. See Art. **14.**

(6) In the Woolf compound type the pistons commence and complete the stroke at the same time. The pistons may operate upon one crank or upon two or more cranks that are either in line or placed 180° apart. In the receiver compound type the high-pressure cylinder exhausts into a separate vessel in order to allow the cranks to be placed at any desired angle other than 0° or 180° apart. See Arts. **16** and **17.**

(7) Assuming that cut-off occurs at the same time in both cylinders, the high-pressure exhaust will be compressed by the high-pressure piston after cut-off occurs because the exhaust has no place to go to. This compression represents a waste of work, and to overcome it a receiver must be used. See Art. **18.**

(8) To the number of different stages in which the steam is expanded. See Art. **19.**

(9) The weights of the two sets of reciprocating parts balance each other. There are no dead centers and the turning effort is the same as if two cranks set nearly 90° apart were used. Also, the stresses on the crankpins are gradually changed around the pins and not suddenly reversed, as is the case with an ordinary engine. See Art. **27.**

(10) By drop is meant the difference in pressure at release and in the receiver. See Art. **29.**

(11) Owing to the free expansion of the steam, the drop in a receiver tends to superheat the steam and thus make it drier. See Art. **30.**

(12) (*a*) Making the high-pressure cut-off later increases the receiver pressure; making the cut-off earlier decreases it.

(*b*) If the low-pressure cut-off remains the same, the volume of steam removed from the receiver per stroke remains constant regardless of the amount of steam entering the receiver. But as the high-pressure cut-off is changed, the amount of steam entering the receiver is changed, and thus the pressure must vary, owing to a constant volume being taken from it. See Art. **31.**

(13) By changing the cut-off in the low-pressure cylinder. See Art. **33.**

(14) Less work will be done in the high-pressure cylinder and more in the low-pressure cylinder. See Art. **34.**

(15) When the high-pressure cut-off occurs late, the low-pressure cylinder does the greater share of the work, and when the cut-off occurs very early, the high-pressure does the greater part of the work, and in some cases the low-pressure cylinder may then act as a drag on the engine. See Art. **35.**

(16) To relieve it of undue pressure, which would result should the low-pressure cut-off be set too early or should the low-pressure admission valves fail to open when a releasing gear is used. See Art. **37.**

(17) (*a*) No. See Art. **39.**

(*b*) Because the initial load is entirely independent of the relative amount of work done in the two cylinders. See Art. **39.**

(18) So as to obtain the highest economy in conjunction with a satisfactory mechanical operation. See Art. **42.**

(19) (*a*) To prevent cylinder condensation. See Art. **44.**

(*b*) No. See Art. **44.**

(*c*) Because if the steam is sufficiently superheated there will be no cylinder condensation. See Art. **44.**

(20) The low-pressure valves should have the most lead. See Art. **38.**

(21) (*a*) A reheater consists essentially of a receiver containing a nest of pipes through which high-pressure steam circulates, and around which the working steam must circulate before entering the cylinder. See Art. **47.**

(*b*) To thoroughly reheat the steam before it enters the cylinder in which it does its work. See Art. **47.**

(22) It indicates that the reheater is wasting heat. See Art. **47.**

(23) Because the steam will then be admitted to both sides of the reheater as soon as the engine is started, and thus its expansion will be more uniform. See Art. **49.**

(24) (*a*) To the ratio between the volume of steam admitted to the high-pressure cylinder and the volume of steam in the low-pressure cylinder at low-pressure release. See Art. **51.**

(*b*) No. See Art. **52.**

(25) By rule **2,** we have

$$E = \frac{12,460 \times 3}{3,115} = 12. \quad \text{Ans.}$$

(26) As a ratio of expansion by volume, or as a ratio of expansion by pressure. See Arts. **52** and **53.**

(27) By rule **1,** we have

$$E = \tfrac{8800}{800} = 11. \quad \text{Ans.}$$

(28) The initial absolute pressure is approximately $(180 + 14.7) - 5 = 189.7$, say 190 pounds per square inch. The terminal pressure may be taken as 9 pounds (see Art. **59**). The ratio of expansion by pressure is $\tfrac{190}{9} = 21.1$, say 21. By Table I, the factor for a ratio of expansion of 21 is .192. The back pressure, by Art. **59,** may be estimated at 3 pounds. The factor to be taken from Table II is .60.

Applying rule **3,** we get

$$p_m = (190 \times .192 - 3) \times .60 = 20.03,$$

say 20 pounds per square inch.

The probable indicated horsepower is

$$\frac{20 \times \frac{24}{12} \times 39^2 \times .7854 \times 2 \times 180}{33,000} = 511.2, \text{ say } 511. \quad \text{Ans.}$$

ENGINE MANAGEMENT.

(PART 1.)

(1) When the steam enters the cold pipe or cylinder, it condenses and forms a partial vacuum that causes a still larger volume of steam to enter the pipe or cylinder. This action is repeated until a mass of water collects, which will rush through the steam pipe with the steam, and on striking a bend or other obstruction cause water hammer. See Art. **19.**

(2) The valve should be eased on its seat to prevent its getting stuck by the unequal expansion of the valve casing. See Art. **20.**

(3) All oil should be wiped from the bright and painted parts before it has time to set; rusty spots on the bright work should be removed. All oil holes not fitted with oil cups should be carefully plugged and any dirt about the engine or engine room should be removed. See Arts. **22** and **23.**

(4) First, thoroughly warm the steam pipe by slightly opening the stop-valves and allowing the steam to flow until it issues from the drain cock near the throttle; then close the drain cock and slightly open the throttle or the by-pass around the throttle, thus allowing the steam to enter the steam chest and cylinder. The drain cocks on the cylinder and steam chest should also be opened, and if the cylinders are steam-jacketed, steam should be turned into the jackets

and the jacket drain cocks opened. To further warm the cylinder, steam should be admitted to both ends by means of a by-pass valve or by turning the engine over by hand. See Art. **17.**

(5) By allowing the hot air coming from the boilers before steam is generated to circulate through the cylinder and steam pipe. See Art. **18.**

(6) From the upper or lower half center. See Art. **24.**

(7) To jump the crank over the first center, after which the flywheel will carry it over the other centers. See Art. **25.**

(8) There is probably water in the cylinder. See Art. **27.**

(9) If the drain cocks are closed, the steam will condense inside the engine and thus allow it to cool off slowly, which will lessen the danger of cracking the cylinder. See Art. **29.**

(10) By simply throwing the link from one side to the other. See Art. **30.**

(11) In order to allow the amount of circulating water to be readily adjusted to suit variations in the temperature of the water and to suit the degree of vacuum desired. See Arts. **33** and **35.**

(12) The cylinder and steam pipe should first be warmed up, and just previous to starting the engine the injection and delivery valves of the condenser should be opened and the air and circulating pumps started. The engine may then be started by simply opening the throttle. After the engine has been running a few minutes, the speed of the air and circulation pumps and the admission of injection water should be regulated. See Arts. **34** and **35.**

(13) (*a*) If the injection valve is not opened at the moment the engine is started, the condenser will be filled with air and steam, which will prevent the injection water entering. See Art. **36.**

(*b*) When the condenser gets hot, it is necessary to pump cold water into it or to cool it by playing cold water upon it before it can be started. See Art. **36.**

(14) (*a*) The snifting valve is used on jet condensers. It is similar to a safety valve, but is held to its seat simply by its own weight and the pressure of the atmosphere. See Art. **37.**

(*b*) Its purpose is to relieve the pressure on the inside of the condenser when it gets hot. See Art. **37.**

(15) The eccentric rod is unhooked from the wristplate and a starting bar inserted in the wristplate. The throttle is then opened and the wristplate vibrated back and forth, by hand, by means of the starting bar. In this manner the steam and exhaust valves are operated and the engine started. After the engine has made several revolutions, the eccentric rod is hooked onto the wristplate and the starting bar unshipped. See Art. **43.**

(16) The eccentric rod is unhooked and the Corliss engine stopped in the desired position by the operation of the wristplate by hand. The slide-valve engine is stopped in the desired position by operating the throttle. See Art. **48.**

(17) If pass-over or starting valves are provided they may be opened, or the steam may be worked into the low-pressure cylinder by operating the high-pressure valves by hand. See Art. **49.**

(18) If the pressure is too high or too low in the receiver, the engine will not start. See Art. **50.**

(19) (*a*) Reversible compound engines. See Art. **52.**

(*b*) Because they can be started much easier when there is a vacuum in the condenser. See Art. **52.**

(20) It should be set at the greatest cut-off in order that the engine may be started promptly. See Art. **56.**

(21) They should be parallel to each other and perpendicular to the center line of the connecting-rod. See Art. **62.**

(22) Secure a strip of wood across the head end of the cylinder by means of the stud bolts and through the strip of wood bore a 1-inch hole approximately in line with the center of the cylinder. At the crank end of the bed erect a standard with a hole similarly located. Through these holes tightly stretch a fine cord, fastening the cord in such manner that it can be shifted about in the holes. Now locate the cord central with the head end of the cylinder by shifting it about until it measures the same distance from all sides of the cylinder. In a similar manner locate the cord central with the crank end of the cylinder. The location of the cord should then be verified for both ends of the cylinder. See Arts. **64, 65,** and **66.**

(23) By dropping a plumb-line from overhead and touching the line to be leveled and then shifting the line until it is perpendicular to the plumb-line. See Art. **73.**

(24) (*a*) The center line of the shaft is usually left a little higher than the center line of the cylinder. See Art. **75.**

(*b*) So that when the brasses and journals wear to a bearing, the center line of the shaft will be nearly level with the center line of the cylinder. See Art. **75.**

(25) The journals are given a coat of red or black marking material and the shaft is placed in position and rocked back and forth, the lower brasses alone being in position. The shaft is then removed and the high spots of the brasses scraped down. This operation is repeated until the desired bearing surface is obtained; then the upper brasses are fitted in a similar manner. See Art. **79.**

(26) It may be done by turning the shaft until the crankpin touches the line stretched through the center of the cylinder and measuring the distance from the line to one of the shoulders on the crankpin. Then turn the shaft so that the crankpin touches the line on the other side of the center of the shaft, and if the line is the same distance from the shoulder of the crankpin as before, the shaft is set correctly. See Art. **80.**

(27) Place the crank on one dead center, connect the connecting-rod to the crankpin, leaving it free at the crosshead end, and key up the brasses snugly; then measure the distance from some point on the crosshead end of the connecting-rod to one of the crosshead guides. Place the crank on the opposite center and repeat the measurement. If both measurements are alike, the crankpin is parallel to the shaft in the horizontal plane. The same operation should then be repeated with the crankpin at the upper and lower half centers. See Art. **83.**

(28) The crankpin may be given a thin coat of Prussian blue or red-lead paint and the rod then connected snugly to the wristpin, but not so snugly to the crankpin. After turning the crank through one revolution, examine the crankpin brasses, and if the coloring matter has spread evenly over them, their correct adjustment may be assumed. See Art. **86.**

(29) Loose journal brasses. See Art. **89.**

(30) By reducing the two halves of the brasses so that a number of thin strips of brass may be placed between them. One or more of these strips of brass may then be removed when the boxes become brass bound. Or, keepers of cast brass, cast iron, or wood may be placed between the boxes instead of the brass strips. See Arts. **90** and **91.**

(31) Usually there is but little space between the piston-rod nut and the cylinder head, so that the former cannot back off very far before it will strike and break the cylinder head. See Art. **95.**

(32) If there is too little compression, the piston rod, connecting-rod, and crank-shaft will be suddenly thrown forwards at the dead centers, where the direction of pressure is suddenly reversed, thus causing pounding, as there is always some lost motion at the pin and shaft bearings. See Art. **101.**

(33) (*a*) Yes. See Art. **105.**

(*b*) Pressure is retained so long on the driving side of the piston that there will not be sufficient compression to stop the piston gradually. See Art. **105.**

(34) By allowing sufficient air to enter the barrel of the pump to form a cushion for the piston. See Art. **112.**

ENGINE MANAGEMENT.

(PART 2.)

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(1) (*a*) If it were not for the wearing of the bearings and journals, the solid bearing would be the ideal bearing, as it cannot be tampered with by careless persons. Its main disadvantage is that it cannot be adjusted, which is necessary on account of wear. See Arts. **3** and **4.**

(*b*) The advantage of adjustable bearings is that the wear may be taken up as fast as it occurs, but through carelessness or ignorance they are liable to be taken up too much and thus cause the bearing to heat. See Art. **4.**

(2) If increasing the oil feed does not stop the heating, mix some flake graphite, flour sulphur, or powdered soapstone with the oil and feed the mixture into the bearing. Aqua ammonia will also sometimes stop the heating. See Art. **6.**

(3) The brasses are liable to warp or crack by unequal contraction. See Art. **8.**

(4) Stop the engine and slack off the brasses; then keep the inside of the bearing deluged with a mixture of oil and graphite, sulphur, or soapstone. If necessary, cold water may be applied to the outside of the bearing. See Art. **11.**

(5) By wearing down a bearing is meant the running of the bearing until the rough rubbing surfaces of the brasses and journal have become smooth and are in their normal working condition. See Art. **15.**

(6) It is difficult to replace the brasses exactly as they were before removal; consequently the brasses do not bear evenly on the journal, which causes heating. See Art. **16.**

(7) (*a*) Yes. See Art. **20.**

(*b*) By the hammering of the journal against the brasses. See Art. **20.**

(8) The brasses may be set up solid on the journal and then slacked off until, by trial, it is found that they work properly, or thin strips of metal may be put between the brasses and the bearing set up tight. Enough strips should be used to cause the brasses to set loosely on the journal; then by removing a pair of strips at a time the brasses can be set up to the proper point. See Arts. **21** and **22.**

(9) If not too bad, the brasses may be filed, scraped, or chipped to fit the journal, but if too greatly distorted, they will have to be replaced by new ones. See Art. **23.**

(10) Badly fitting brasses. See Art. **25.**

(11) (*a*) When the brasses are heated quickly, they tend to expand along the surface in contact with the journal; which would tend to make the bore of larger diameter; but this expansion is prevented by the cooler portions of the brasses and by the outer part of the bearing. The layer of metal near the journal thus receives a permanent set, so that the brasses close on the journal when they become cooler. See Art. **26.**

(*b*) By chipping off the brasses at their thin edges parallel to the journal. See Art. **27.**

(12) All oil should be strained or filtered before it is used and all oil cups, oil cans, and oil channels should be frequently cleaned. See Art. **30.**

(13) (*a*) It should have a high degree of viscosity. See Art. **33.**

(*b*) Because oil having a high degree of viscosity offers more resistance to being squeezed from between the journal and brasses than does thin oil. See Art. **33.**

(14) By chipping oil channels in the brasses. See Art. **33.**

(15) On the materials of which the brasses and journals are composed, the fineness of their finish, the accuracy of their fit, the adjustment of the brasses, and the lubricant used. See Art. **38.**

(16) Overloading an engine increases the pressure on the journals. This increased pressure may cause the practical limit of pressure allowable on the bearings to be exceeded and thus cause the bearings to heat. See Art. **41.**

(17) Because it promotes a better distribution of oil and prevents the journal and brasses wearing into concentric parallel grooves. See Art. **48.**

(18) On the amount of greasy particles that it contains. See Art. **52.**

(19) It should reduce friction to a minimum. It should be free from acids, alkalies, and disagreeable odors. It should not be altered by exposure to the air and should stand a low temperature without solidifying or depositing solid matter. It should be free from grit and all foreign matter. Cost is also a consideration. See Art. **52.**

(20) (*a*) No.

(*b*) Because the specific gravity of an oil is not an indication of its viscosity, neither is the viscosity of an oil an indication of its specific gravity. See Art. **53.**

(21) (*a*) Pure lard oil. See Art. **57.**

(*b*) Olive oil. See Art. **58.**

(22) (*a*) Mineral oils are distilled from bituminous shale and from the residuum of crude petroleum after the volatile oils and illuminating oils have been distilled off.

(*b*) According to their specific gravity. See Art. **59.**

(23) Because they are liable to have a low flashing point. See Art. **61.**

(24) Boiled, or cup, greases are made by saponifying fats and fatty oils with lime and dissolving the soap in mineral oil. See Art. **67.**

(25) The lubricant spreads itself in the form of a thin film between the rubbing surfaces and thus prevents their coming in direct contact with one another. See Art. **75.**

(26) Because they are not carbonized by the high temperature nor are they decomposed by heat, as are animal and vegetable oils, which break up into acids that attack the metal of the pistons, valves, etc. See Art. **76.**

(27) The higher the temperature of the bearing, the less is the lubricating power of the oil. See Art. **78.**

(28) By rubbing the greases between the forefinger and thumb or in the palm of the hand. See Art. **83.**

(29) The point or temperature at which the vapor rising from the oil will ignite with a flash. See Art. **84.**

(30) Boil about a pint of the oil, into which there has been placed 1 or 2 ounces of caustic soda or concentrated lye, for ½ hour and allow it to cool. If, when cool, the surface of the oil is covered with soap, the oil contains animal or vegetable fats; otherwise it is pure mineral oil. See Art. **86.**

(31) The viscosity is increased. See Art. **87.**

(32) Sulphur. See Art. **89.**

(33) (*a*) Steam is allowed to enter the reservoir containing the oil and as it condenses, the water, being heavier than the oil, sinks to the bottom, thus raising the level of the oil until it overflows into a suitable passage. See Art. **92.**

(*b*) The flow of oil cannot be readily controlled and there is no means of telling when the lubricator stops working. See Art. **102.**

(34) (*a*) No.

(*b*) Because on closing the throttle there will be full steam pressure on the condenser and none on the sight-feed glass. In consequence, the lubricator will be rapidly emptied. See Art. **111.**

(35) (*a*) Mechanical, water-displacement, and hydrostatic. See Art. **92.**

(*b*) On the pressure of the head of water furnished by condensation of steam. See Art. **92.**

(36) In a double-connection lubricator the steam is admitted through one pipe and the oil leaves through the other pipe. In a single-connection lubricator the oil passes through the same pipe that admits the steam. See Art. **104.**

ENGINE INSTALLATION.

(1) (*a*) The available floor space and the size of the engine. See Art. **2.**

(*b*) The first reason is self-evident. As to the second reason, the piston of large horizontal engines, and particularly the low-pressure piston of compound engines, becomes so massive that it is almost impossible to support it in such manner that it will not seriously wear the lower side of the cylinder. See Art. **2.**

(2) Accessibility for repairs, inspection, and oiling, lower cost, the cylinders are easily drained, and in general it requires much less physical strength to properly take care of it. See Arts. **4, 5, 6,** and **7.**

(3) (*a*) Yes.

(*b*) They are of the four-valve type, in which the cut-off valve is mounted on the back of the main valve and is positively driven by a shaft governor. See Art. **12.**

(4) Yes; the throttling engine with a Meyer valve by which the cut-off can be varied from 0 to $\frac{3}{4}$ stroke is fairly efficient. See Art. **16.**

(5) The cost of fuel and the amount of power required. See Art. **17.**

(6) The compound engine is more complicated and hence more liable to break down, which requires more spare parts to be carried. In order to keep it in proper working order, a higher degree of skill is required and the boilers required

are more expensive, although there may not be as many of them. The facilities for repairs should also be considered. See Art. **21.**

(7) The inaccessibility of the cylinders and pistons for inspection or repairs. See Art. **23.**

(8) The friction of the crossheads and crankpins is the same for both classes of engines and the friction should also be the same for the pistons and piston rods of both engines, but owing to wear the friction is slightly greater in the tandem compound. The friction of the valve gear is in favor of the tandem compound, but the tandem compound requires a flywheel $1\frac{6}{10}$ times heavier than the cross-compound; this requires a heavier shaft and larger bearings, which greatly increases the frictional resistance. See Art. **25.**

(9) The volume of the receiver is made about equal to the volume of the low-pressure cylinder and there are about 50 square feet of tube-reheating surface for each cubic foot of steam exhausted by the high-pressure cylinder. See Art. **28.**

(10) There is difficulty of securing sufficient compression in the low-pressure cylinder to absorb the inertia of the reciprocating parts. See Art. **29.**

(11) Their advantages are comparatively low first cost and small space required; their principal objections are the extra care required to keep them in order and their wastefulness of fuel. See Art. **36.**

(12) (*a*) Piston valves and valves balanced by cover-plates or pressure plates. See Art. **37.**

(*b*) The piston valve gives fairly good results on vertical engines, but when used on horizontal engines it is liable to leak badly after running a short time. Owing to the method of regulation, the piston valve wears very unevenly, which further causes it to leak. Valves balanced by pressure plates require less clearance, and when properly fitted wear better than balanced valves. See Art. **37.**

(13) Owing to the high rotative speed, the regulation of high-speed engines is much closer than the regulation of slow-speed engines. See Art. **40.**

(14) Owing to the high speed, but little time is allowed for initial condensation or for change of temperature in the cylinder walls between the strokes. See Art. **45.**

(15) There is a direct saving due to the omission of transmission machinery, such as jack-shafts, belts, or gearing and bearings. The cost of lubrication, attendance, and repairs are also reduced and the foundations are less expensive. See Art. **46.**

(16) The valves are designed to give a minimum clearance volume and clearance surface, the cylinders and cylinder heads are steam-jacketed, the internal surfaces of the heads and the pistons are polished to prevent initial condensation, care is taken to free the cylinders of water, and the valve gears are carefully constructed to give a theoretical steam distribution. See Art. **48.**

(17) (*a*) Their principal advantage is that they allow a much smaller armature or revolving field to be used on the generator, which greatly reduces the first cost of the unit. See Art. **50.**

(*b*) After passing 200 or 300 horsepower, the saving in first cost is more than offset by the increased economy of the slow-speed engine; hence it is not usual to use high-speed engines for large direct-connected units. See Art. **50.**

(18) A compound condensing high-speed engine having separate steam and exhaust valves and having a governor controlling the admission valve only would be most efficient under the conditions named. See Art. **54.**

(19) Because the economical range of cut-off is much greater for the compound condensing engine than for the compound non-condensing engine. See Art. **56.**

(20) Because it can be readily converted into a duplex engine, a condensing engine, or a compound condensing

engine when the increased power required demands it. See Art. **57.**

(21) In general, if fuel is very cheap, a cheap simple engine is selected, but if fuel is dear, then a more expensive engine of the multiple-expansion type is selected in order to secure as small a steam consumption as possible. See Arts. **67** and **68.**

(22) No, probably not. Superheating and jacketing both tend to reduce initial cylinder condensation, so where one is used there is nothing gained by using the other also. See Art. **68.**

(23) The cooling tower as usually constructed consists of a round or rectangular tower so arranged that when the water from the condenser is delivered at the top of the tower, it is divided into a great number of fine sprays in its descent. An artificial current of air cools the water as it falls and thus renders it fit for injection again. See Art. **78.**

(24) In many manufacturing plants the various departments require widely varying speeds or are required to run overtime or all night, or they may require power for but a short time; under these conditions the subdivided power will give the best results. See Art. **79.**

(25) By suspending a heavy mass underneath the floor, but rigidly bolted to the engine base. See Art. **83.**

(26) (*a*) Engine foundations are usually made of brick, dressed stone, or concrete. See Art. **86.**

(*b*) If brick or stone are used, they should be laid in Portland cement mortar, as lime mortar disintegrates under vibration. See Art. **86.**

(27) (*a*) The vibrations of the engine will be transmitted directly to the rock and hence to adjoining property. See Art. **83.**

(*b*) The rock is usually covered with a layer of timber or rubble or layer of sand 2 or 3 feet deep; the foundation is then built on this footing. See Art. **88.**

(28) The foundation is usually supported on piles driven from 2½ to 4 feet apart from center to center. To form a footing for the foundation a timber grating is fastened to the top of the piles and a layer of concrete is deposited, or the space between the piles is filled with rubble, clay, or concrete. The foundation proper is then built upon the footing thus prepared. See Art. **87.**

(29) If the capstone is too low, the bearing can be easily shimmed up to the proper height, but if it is too high, it must be reset or chipped down to the proper height. See Art. **90.**

(30) By means of a templet constructed of wood, on which have been carefully marked the center lines of the engine and crank-shaft and the relative position of the bolt holes. The templet is suspended in its correct position above the foundation and the bolts are passed through holes bored in the templet. The bolts are located at the proper height by blocks placed on the templet, the height of the blocks depending on the height of the engine bosses. See Art. **95.**

(31) The space is usually filled with grouting, which may be made of iron borings mixed with cement, sal ammoniac, sulphur, and water, or melted sulphur or pure Portland cement may be used alone. See Art. **99.**

ELEVATORS.

(PART 1.)

(1) They are classified as hand-power elevators, belt elevators, steam elevators, electric elevators, and hydraulic elevators. See Art. **3.**

(2) An elevator in which the upright posts of the car are placed on diametrically opposite corners. See Art. **4.**

(3) One in which the transmitting devices include a drum and rope. See Art. **5.**

(4) By guiding the rope on the drum by means of a sheave that is caused to follow the motion of the rope back and forth across the drum. See Art. **6.**

(5) (*a*) Making the counterweight heavier than the full weight of the car. See Art. **11.**

(*b*) Hydraulic elevators. See Art. **10.**

(*c*) Because the power can only be applied on the up trip. See Art. **10.**

(6) If the elevator is overbalanced by an amount equal to the average load, no power except that necessary to start the machinery and overcome frictional resistances will be required when lifting the average load, thus enabling a smaller motor to be used. See Art. **11.**

(7) By using a chain attached to the bottom of the car and extending either to the bottom or to the middle of the shaft way, where it is fastened. In the former case the

chain must have the same weight per unit length as the rope to be balanced; in the latter case it must be twice as heavy per unit of length. See Art. **13.**

(8) There is no means of telling the exact position of the controlling device, hence it cannot be applied to motors requiring delicate adjustment. Also, the sliding of the rope through the hand of the operator is inconvenient and may be dangerous. See Art. **17.**

(9) Motor safeties and car safeties. See Art. **26.**

(10) By a shaft actuated through a rope sheave and an endless rope or by a crank driving a windlass. See Art. **29.**

(11) They are generally overbalanced. See Art. **33.**

(12) To avoid undue stress being thrown on the machinery. See Art. **39.**

(13) (*a*) To make them more pliable and thus more durable. See Art. **44.**

(*b*) When the wires commence cracking. See Art. **45.**

(14) Equal parts of linseed oil and Spanish brown or lampblack. Seven parts of linseed oil and three parts of tar oil. Cylinder oil, graphite, tallow, and vegetable tar also make a good preparation. See Art. **46.**

(15) The rope should encircle the drum several times when the elevator is in its lowest position. See Art. **50.**

(16) Because the elevator will be given a jerky motion and the car may then drop sufficiently far in some cases to break the rope. See Art. **51.**

(17) An elevator driven directly by belts from line shafting. See Art. **52.**

(18) Overbalancing spur-geared machines greatly increases the jerkiness of motion, while it has little influence that way on worm-geared ones. See Art. **54.**

(19) By clamping knobs or buttons to the shipper rope in such positions that the car will strike them and cause the belt to be shifted automatically when it reaches the limits of its travel. See Art. **58.**

(20) The most common form of motor limit stop consists of a gear-wheel having a thread cut in its hub and working loosely on a thread cut on an extension of the drum shaft or on a shaft positively driven from the drum shaft. This gear meshes with another of wide face attached to the shipper sheave, and as it is prevented from turning by the wide-faced gear, it travels back and forth on its shaft, its position depending on the position of the car. Should the car overrun its limit of travel in either direction, jaws on the hub of the loose gear engage with jaws fastened to the threaded shaft and thus the loose wheel is rotated. This causes the wide-faced gear to revolve and turn the shipper sheave, which reverses the motion of the elevator. See Art. **59.**

(21) Some form of slack-cable safety is provided that is operated by the slack cable and reverses the direction of motion of the drum. See Art. **60.**

(22) By using two worms on the same shaft, one being right-handed and the other left-handed. The two worms mesh with two worm-gears that are in mesh with each other. See Art. **62.**

(23) 60 feet per minute. See Art. **74.**

(24) Castor oil or a mixture of 2 parts of castor oil and 1 part of the best cylinder oil. See Art. **71.**

(25) The limit stops should be frequently tested; the brake should be adjusted whenever the car settles at the landings; the belts should not be allowed to become slack and they should not be subjected to the influence of steam, water, or oil. All bearings should be kept well lubricated, particularly the step bearing, and the worm-gearing oil bath should be occasionally renewed. See Arts. **66** to **72.**

(26) The principle of the reversing mechanism is that by means of a reversing valve the valves of the engine can be changed from direct to indirect valves. See Art. **80.**

(27) It consists of a rod extending across the under side of the winding drum and so arranged that the loose cable striking it will cause a spring or weight to be released, which will cause the steam to be shut off. See Art. **94.**

ELEVATORS.

(PART 2.)

(1) (*a*) Continuous-current constant-potential shunt-wound single-speed motors, alternating-current motors, polyphase-synchronous motors, and induction motors. See Art. **5.**

(*b*) For belt-shifting elevators, the continuous-current constant-potential shunt-wound single-speed motors are used. If the motor runs continuously, any kind of alternating-current motor may be used, but if the motor is to be started and stopped frequently, polyphase-synchronous motors or induction motors are used. See Art. **5.**

(2) To prevent a damaging rush of current in starting the motor. See Art. **6.**

(3) The contact bar of the rheostat shown in Fig. 1 of the text is attached to a rack that is driven by a two-toothed pinion, the pinion being on a shaft that is, in turn, driven from the main shaft of the motor. When the motor is started, the rack is drawn into contact with the pinion by means of an electromagnet that is energized by a coil in shunt with the motor circuit, and as the contact bar rises, it gradually cuts out the resistance. As soon as the current is broken, the contact arm drops back and the rack springs out of gear. See Art. **7.**

(4) By connecting across the shunt a series of incandescent lamps having a combined voltage of from 6 to 8 times that of the line current. See Art. **11.**

(5) (*a*) A rheostat in which a solenoid is used to operate the arm that cuts out the resistance. See Art. **13.**

(*b*) It enables the rheostat to be mounted separate from the switch and the switch alone to be operated by the hand rope. See Art. **13.**

(6) It must have a strong torque and it must get up speed rapidly, though gradually. See Art. **17.**

(7) Two-phase or three-phase induction motors. See Art. **17.**

(8) Overbalanced. See Art. **19.**

(9) By mechanical, electrical-mechanical, or wholly electrical attachments. See Art. **21.**

(10) (*a*) The controller consists of a double-throw switch attached to the hub of the shipper sheave and a solenoid rheostat placed near the machine. A casting forming one part of the switch is bolted to the frame of the machine and carries four sets of clips to which the line, field, armature, solenoid, and electric-brake connections are made. By rotating the shipper sheave, the switch blades attached to it may be brought into contact with either of the two sets of clips, thus reversing the motor. See Art. **27.**

(*b*) It gives the rheostat arm time to fall back into its starting position before the current in the armature can be reversed, and it also helps to reduce sparking and flashing at the clips. See Art. **27.**

(11) By means of a dashpot. See Art. **28.**

(12) When the shipper sheave is rotated, the brake magnet is energized and slowly releases the brake. The solenoid is also energized and cuts out the resistance from the armature circuit at such a rate that when the motor is up to speed, the resistance is entirely cut out. When the circuits are broken, the brake is applied and the resistance arm drops back to its original position. See Art. **30.**

(13) The principle of the dynamic brake is that the motor is made to act as a dynamo by means of a variable

resistance so arranged that the armature is short-circuited through it immediately after the line circuit is broken. This has the effect of slowing the motor down quickly but gradually. As the motor slows down, the resistance is gradually cut out, thus making the stop still more gradual. See Art. **31.**

(14) (*a*) In order to make the motor act as a dynamo or brake. See Art. **32.**

(*b*) The field in this case is kept constantly excited, and in order to use less current a resistance is inserted in the fields that are short-circuited when the elevator is started, thus giving the full torque available. When the elevator is stopped, the resistance is cut in, leaving the field current strong enough to get a dynamic-brake effect. See Art. **32.**

(15) (*a*) Both steps are on the same end of the shaft. See Art. **40.**

(*b*) It is easily accessible for inspection or adjustment. See Art. **40.**

(16) The usual traveling-nut, limit-stop, and clutch-operating slack-cable safety, and also a limit switch that brakes the current through the armature and brake solenoid at the limits of the car travel. See Art. **43.**

(17) (*a*) By dividing the pressure between the end surface of the shaft and the ring-shaped surface of a bushing placed around the shaft. See Art. **53.**

(*b*) By means of two small equalizing levers, which distribute the pressure equally over both surfaces. See Art. **53.**

(18) To automatically apply the brake should the current be interrupted in the system. See Art. **59.**

(19) (*a*) The brake is so arranged that it will be set in action by the limit stop much quicker and more effectively than by the ordinary device. See Art. **58.**

(*b*) The tripping device is given considerable lost motion, or backlash, which allows the motor to be stopped without danger of reversing it. See Art. **60.**

(20) To break the main current and thus release the safety brake when the current falls below the normal or when the current becomes excessive. See Arts. **61** and **62.**

(21) The potential switch has three blades with three corresponding double clips, of which the first two are connected to the line wires and the third to a wire leading to the starting resistance. The first two blades are connected to the motor circuit and one of them is also connected to the third blade. An electromagnet placed in shunt across the line and in series with the safety-brake magnet holds the first two blades in contact with their corresponding clips. A spring counteracts the magnet and causes the first two blades to leave their clips and the third blade to engage its clip when the current in the magnet windings falls below the normal. See Art. **61.**

(22) (*a*) In the magnet-control system of operating electric elevators, the starting, stopping, and reversing of the motor and the cutting out of the starting resistance are accomplished by a series of electromagnetic switches that are controlled by a car-operating switch on the car. The switches are usually in the form of solenoids or electromagnets that operate whenever current is made to flow through them by means of the car-operating switch. See Arts. **68** and **69.**

(*b*) It avoids the necessity of using the sliding arm and numerous contact plates that are necessary with a rheostat and that always give more or less trouble due to burning and cutting, especially with heavy currents. See Art. **69.**

(23) Give a description similar to that contained in Art. **81.** The action is the same as there described, except that the main switches operate so as to make the current flow through the armature in the reverse direction.

(24) (*a*) The main difference lies in the construction of the switches, the principle of operation being practically the same. Each switch consists of a solenoid arranged so as to draw up a core or plunger to which contact disks are

attached and which make the required connections by being brought into contact with fixed fingers mounted on the switchboard. See Art. **85.**

(*b*) The explanation required is similar to that contained in Art. **88,** except that the operating switch is supposed to be on the down position and, consequently, the direction-controlling switches operate so as to reverse the motor.

(25) (*a*) A description similar to that contained in Arts. **96, 97, 98,** and **99** is required. This can be made considerably shorter than that given in the text, but the path of the operating current and the main current should be described.

(*b*) See Art. **99.**

(26) See Art. **102.**

(27) See Art. **110.**

ELEVATORS.

(PART 3.)

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(1) Hydraulic elevators are safe, reliable, smooth-acting, and are under perfect control. The wearing parts are few and are easily and cheaply replaced. On the other hand, they require considerable space and usually require the installation of steam pumps, reservoirs, etc., which makes them expensive. See Art. **1.**

(2) (*a*) An elevator in which the car is placed directly on top of the piston or plunger. See Art. **3.**

(*b*) For freight and passenger service for short lifts. See Art. **2.**

(3) Because the power acts only on the up stroke of the elevator. See Art. **6.**

(4) The car would be jerked upwards against the overhead work. See Art. **5.**

(5) As a power control it shuts off the power at the will of the operator, and as a brake it shuts off the water gradually by throttling. See Art. **7.**

(6) By putting in the discharge pipe a throttle valve controlled by the pressure corresponding to the velocity of the exhaust. See Art. **10.**

(7) The cylinder can be made much shorter by introducing multiplying sheaves, and thus the water used per stroke is greatly reduced. See Art. **11.**

(8) Because the car will not tend to teeter up and down when the power is suddenly cut off if the counterweights are arranged in this manner. See Art. **15.**

(9) (*a*) To make the effective pressure on the piston the same at all points of the stroke. See Art. **16.**

(*b*) When the piston is at the top of the cylinder, the weight of the water above it is practically nothing, while the column of water below it exerts a suction corresponding to the height of the column, as long as the column is not higher than 34 feet. As the piston moves down, the weight of water above it increases, while the suction below it decreases by a corresponding amount. Thus the pressure remains constant for all positions of the piston. See Art. **16.**

(10) To avoid the water ram that would occur if the controlling valve was suddenly closed when the piston was descending. See Art. **18.**

(11) (*a*) To control the opening and closing of the main controlling valve. See Art. **19.**

(*b*) Because in high-speed hydraulic elevators the controlling valve cannot be operated directly without danger of producing violent shocks in starting or stopping. See Art. **19.**

(12) The throttle, if properly adjusted, deadens the noise occasioned by the circulating water and serves as a brake in descending. It also prevents the water from rapidly flowing out of the circulating pipe should the supply pipe break. See Art. **23.**

(13) (*a*) An elevator that can be connected either to a high-pressure or a low-pressure tank at the will of the operator. See Art. **25.**

(*b*) In office buildings where it is only occasionally necessary to lift heavy loads. See Art. **25.**

(14) Because the greater the ratio, the shorter the cylinder and hence the less becomes the head of water to be

counterbalanced, thus allowing the non-circulating system to be used. See Art. **26.**

(15) They occupy less valuable floor space and are more accessible than the vertical type. See Art. **28.**

(16) The terms apply simply to the condition of the stress in the piston rod, that is, to whether the rod is in tension or compression when the car is going up. See Arts. **29** and **30.**

(17) Because the whipping of the ropes is very much reduced and thus teetering of the car is prevented to a great extent. See Art. **30.**

(18) By having the tank partly filled with air, which expands as the water is withdrawn. See Art. **39.**

(19) (*a*) By leakage or by absorption in the water. See Art. **39.**

(*b*) By a vent in the suction pipe or by a separate air pump. See Art. **39.**

(20) The Ford regulating valve consists of a spring-actuated steam valve that is operated by a small water piston working in a cylinder that is connected to the pressure tank by a small pipe. As the pressure in the tank rises and falls the water piston rises and falls also, thus causing the steam valve to open or close. See Art. **43.**

(21) The spring-actuated steam valve is replaced by a small water piston valve that controls, by its movement, the amount of water allowed to enter or leave an auxiliary cylinder, to the piston of which is connected a lever operating the switch and rheostat. By this means a comparatively large movement is obtained. See Art. **44.**

(22) To allow the pump to run continuously. See Art. **47.**

(23) By introducing into the tank a layer of heavy oil about 4 inches thick. See Art. **52.**

(24) By opening the vent in the suction pipe when the tank is about half full or by filling the tank two-thirds full

and then pumping up the required pressure by the auxiliary air pump, if one is attached. See Art. **51.**

(25) Run the car to the top and set the controlling valve for going down. While the car and valve are in this position, open the air cock and allow the air to escape. See Art. **52.**

(26) By a groaning in the cylinder or by the car settling at the landings. See Art. **54.**

(27) Connect the exhaust-steam drips from the pump with the discharge tank, thus allowing the cylinder oil to be pumped with the circulating water, by which means all the internal parts of the plant are lubricated. See Art. **57.**

(28) When the car is down, open the air cock and drain-pipe valve and then throw the valve for going up. This will drain the cylinder. To drain the circulating pipe, throw the valve for going down. See Art. **60.**

(29) Run the car to the bottom and close the stop-valve in the supply pipe. Open the air cock at the head of the cylinder and drain the water in the cylinder down to the top of the piston. Remove the cylinder head, and if the piston is not near enough to the top attach a small tackle to the main cables and draw it up within reach. Now, remove the piston follower and renew the packing. After replacing the piston follower, let down the piston to its proper position and replace the cylinder head. Place the operating valve on the center, open the supply-pipe valve, and as soon as the air has escaped close the air cock and the elevator is ready to run. See Art. **67.**

(30) The first and last ring of packing should be of pure rubber cut about 1 inch longer than the circumference of the cylinder. The remaining rings should be of fibrous packing. See Art. **69.**

(31) It should be soaked in boiling tallow for several hours. See Art. **72.**

ELEVATORS.

(PART 4.)

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(1) The breaking of the cable or cables, the temporary sticking of the car, allowing the cable to become slack, or excessive speed of the car. See Art. **1.**

(2) (*a*) For slow-speed elevators. See Art. **3.**

(*b*) By a wedge that acts between the guides and guide shoes. See Art. **3.**

(3) In order to operate the safety should any one of the cables break, stretch, or become slack. See Art. **5.**

(4) (*a*) Excessive speed of the car. See Art. **6.**

(*b*) By means of a centrifugal governor driven by a rope attached to the car, the safety devices are brought into action when the speed exceeds a certain limit. See Art. **6.**

(5) The governor is operated by a rope that is attached to the finger shaft of the safety device. When the speed becomes excessive, the governor balls fly out and operate a clutch consisting of two eccentrics that grip the rope and hold it firmly. Consequently, as the car descends the tension on the rope becomes great enough to rotate the finger shaft and thus operate the safety device. See Art. **7.**

(6) To insure a gradual fall of the car, thus giving the safety time to act, should the hoisting rope break. See Art. **12.**

(7) Because they might cause the safety wedges to stick and be thrown into action. See Art. **16.**

(8) All slack in the hoisting cables should be taken up. See Art. **17.**

(9) Remove the limit-stop button on the shipper rope and raise the car enough to unlock the wedges. If this cannot be done, the car may be lifted by a tackle. See Art. **17.**

(10) The air-cushion safety consists of an extension of the hoistway below the lowest landing in the form of a pit. The cross-section of the pit is such that the car is gradually brought to rest by the escape of the air contained in the pit through the space between the sides of the car and the sides of the pit. See Art. **18.**

(11) (*a*) The depth of the pit should be about one-fifth the whole lift. See Art. **19.**

(*b*) By making the lower part of the hoistway air-tight. See Art. **19.**

(12) (*a*) In case of fire, the shaft would act as a chimney and carry the fire from floor to floor. The closed shaft is always dark, and if windows are placed in the walls the danger from fire is increased. See Art. **20.**

(*b*) About 5 feet, or high enough to prevent people bending over the enclosure to look for the car. See Art. **20.**

(13) Elevator doors should be sliding doors or gates that operate freely. They should be so locked that they can only be opened from the inside of the shaft. Self-locking doors are preferable. See Art. **21.**

(14) By means of a car-locking device that prevents the operation of the starting or operating device in the car while the door is open. See Art. **25.**

(15) By means of mechanically operated indicators that indicate the position of the car and whether it is going up or down. See Art. **29.**

(16) For that class of service where the lift is short and where great numbers of people are to be carried. See Art. **33.**