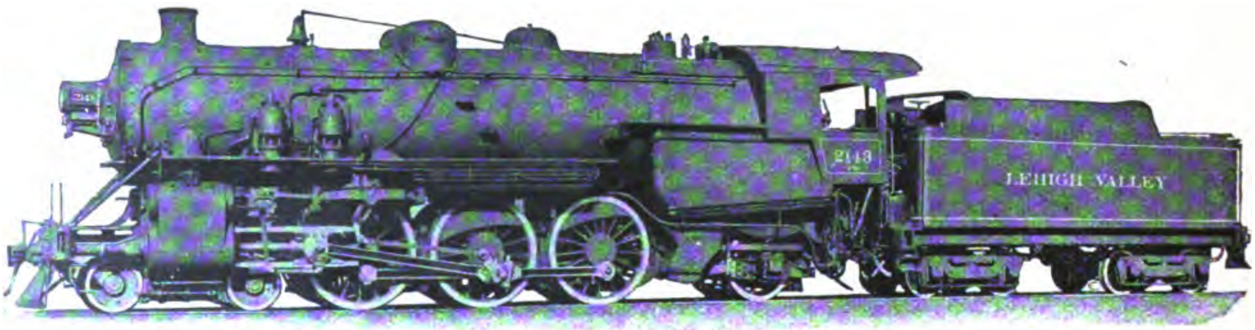


MODERN PASSENGER LOCOMOTIVE

Lehigh Valley Railroad.

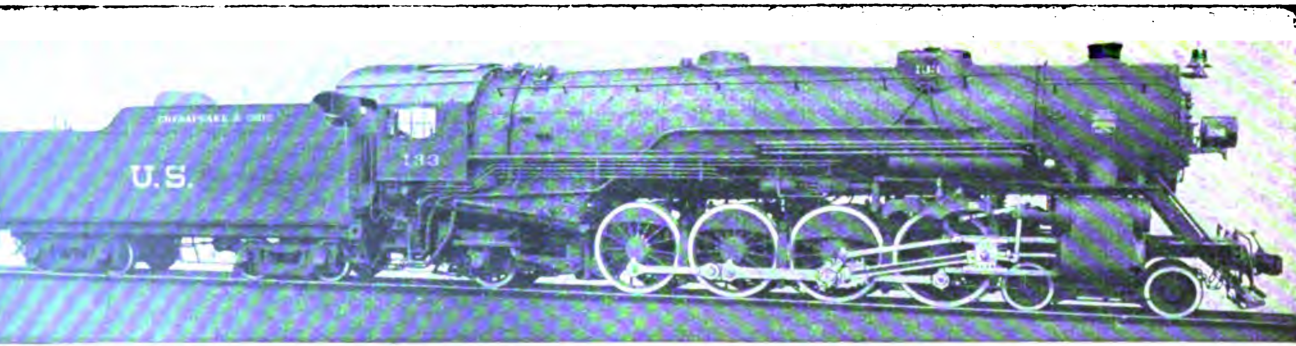
Built by

Baldwin Locomotive Works

Weight in working order 469,000 pounds. Maximum tractive power 48,700 pounds.

Type 4-6-2 Diameter of Drivers 73″ Cylinder 27x28″

Superheat.



MODERN FREIGHT LOCOMOTIVE

United States Railroad Administration

Built by

American Locomotive Company

Weight in working order 896,700 pounds. Maximum tractive power 57,900 pounds.

Type 4-8-2 Diameter of Drivers 69″ Cylinder 28x30″

Superheat.

REVISED EDITION.

THE

LOCOMOTIVE UP TO DATE.

BY

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CHAS. McSHANE

ILLUSTRATED.

Revised
By Chas. L. McShane.

PRICE, $5.00

GRIFFIN & WINTERS,
NEW YORK LIFE BUILDING,
CHICAGO, ILL.

PREFACE TO THE SECOND EDITION.

More than twenty years have elapsed since this work was originally published, and now the second edition is issued to meet the continuous demand with which the book has been honored, and to embrace the many changes and improvements that have been made in locomotive construction.

In the preparation of the present edition, no pains, nor expense, have been spared which seemed to promise for the book a high degree of accuracy and of usefulness in meeting the requirements of the present time. The text throughout has been carefully revised. Much of it has been entirely rewritten, in order to present certain subjects with greater fulness or in new aspects, as seemed, by the course of recent developments, to be rendered necessary, and a great deal of new material has been added. But the basic plan remains much the same—to present in a practicable manner useful information regarding the construction and operation of the locomotive up to date.

To enable the reader to obtain additional information regarding the various appliances described in the book, in which he may be interested, we have appended the names and addresses of the manufacturers of the appliances in the text.

Special care has been taken to use as plain, common sense language and grammatical construction, as the technical character of the work will permit—so that it may be understood by anyone who can read the English language. Accuracy has also been our aim, but it is almost impossible to edit a book of this nature entirely free from error, and it is issued with a realization that it is perhaps faulty in some instances. However, if the reader will kindly direct the publisher's attention to such errors of commission or omission as may be found, corrections will be gladly made in future editions.

It is a pleasure to the author, and it is most certainly his duty, to make public recognition of his indebtedness to those who have rendered him assistance in the prosecution of his labors. Personal acknowledgments of indebtedness have been made to various contributors, inventors and manufacturers, yet we feel that our obligation would not be fully discharged without publicly expressing our thanks to the American Locomotive Company and the Baldwin Locomotive Works; and to Mr. Thomas J. Pembroke and Mr. Frank McGrayel, of Chicago, Ill., for carefully reading the proofs, and also to the 100,000 purchasers of the first edition who made this work possible.

C. L. McS.

Chicago, Ill., April 15, 1920.

PREFACE.

It is unnecessary for me to point out in detail what I regard as the most valuable features of this work, as my judgment may be warped by the circumstances that I have found certain subjects more interesting than others. Indeed, in my judgment, every work must stand or fall upon its own merits, and nothing that an author can say in reference to it can avail to change the ultimate verdict of those who subject it to the decisive practical tests that they are called upon to apply in the discharge of their duties.

I desire to express my sincere thanks to the Railway Public for the generous reception accorded my former efforts, and trust that in this work I may be deemed, in some degree at least, to have made good my pledges to the gentlemen who requested the preparation by me of a more comprehensive work on the same subject.

The number of contributors, inventors, locomotive works and mechanical journals who have rendered assistance are too numerous to admit of my thanking them by name, but I beg here to renew to them all the acknowledgments which have already been made to each in person.

C. McS.

Chicago, January 21, 1899.

Brief History of the Locomotive.

The first self-moving steam engine of which we find any record was built by Nicholas Cagnot in France in 1769. It was, of course, of very crude form, being mounted on a carriage and run upon the public highway, but it was from this insignificant little engine that we are able to trace locomotive construction and development down to the present monsters of the rail. It is true that Isaac Newton has received credit for being the original inventor of the steam engine, but the boiler constructed by him in 1680, and called an engine, cannot properly be so termed, because it failed to move, and therefore failed to develop any power. It consisted of a spherical boiler mounted on a carriage and the intended propulsion was through the force of escaping steam against the atmosphere; naturally it proved a complete failure.

To James Watt, more than to any other man, is due the honor of first controlling and utilizing steam for power and perfecting the steam engine, although Newcomer and others had used steam for lifting water, etc., long before Watt's time. To Trevithick of England is due the honor of first applying the steam engine to rails, or tramways. His engine bore his name and was first run on the Merthyr Tydvil Tramway in South Wales, February 1, 1804. It was a pronounced success, although it appeared that Mr. Trevithick had built a machine which he could not control, as the engine ran off the track and was badly wrecked the first day. The presumption is, however, that this was due to his inexperience as an engine driver. Many other locomotives were soon afterward built in England, several of which have since become famous. American engineers were not inactive, however, while all this experimenting was taking place across the water. Nathan Reed, of Salem, Mass., built an engine as early as 1790, it being the first locomotive ever built in America; like the Cagnot engine it was also mounted on a carriage and was run on the public highway.

The first locomotive to run on rails in America was the "Tom Thumb," built by Peter Cooper, of New York, in 1829. The "Stourbridge Lion," being imported from England the same year, was the first locomotive to cross the Atlantic ocean. It was at about this time that locomotive construction actively began on both sides of the Atlantic. The most important factor in the success of the locomotive was found to be the mechanism employed to distribute and control the steam, which was called the valve gear.

In the early days of locomotive construction many locomotives were imported to this country, but the tide of importation soon turned and our builders have for many years been shipping American locomotives to all parts of the world. This is due to the fact that we build the fastest, most powerful and best locomotives in the world; the American locomotive being noted for its simplicity, convenience, speed and power. Modern types of freight and passenger locomotives, built in this country, with their general dimensions, are shown on the first pages of this volume.

LOCOMOTIVE VALVES.

Introduction.

The functions of a steam valve used in the locomotive cannot well be overestimated, and its importance has been aptly compared to that of the heart of the human body.

While the construction and operation of valves are comparatively simple, and can readily be understood, the subject has generally been surrounded with mystery, and the principal difficulty encountered by the student has been the false impression that the subject is necessarily a difficult one. If, however, the beginner can bring himself to believe that the slide valve is nothing more complex than a flat faced plain metal slide, adapted to work to and fro upon a flat faced seat which includes steam ports, to be alternately opened and closed by the forward and backward motion of the metal slide, he will have made a satisfactory beginning.

THE PLAIN SLIDE VALVE.

The slide valve in a crude form, Fig. 1, was invented by Matthew Murray, of Leeds, England, toward the end of the eighteenth century. It was subsequently improved by James Watt, but the long *D* slide valve, which, in improved form, is used at present, is credited to Murdock, an assistant of Watt's. It came into general use with the introduction of the locomotive, although Oliver Eames, of Philadelphia, Pa., appears to have perceived its actual value earlier, for he applied it to engines of his own build years before the locomotive era. But it was upon the locomotive that it clearly demonstrated its real value; its simplicity of construction and durability, together with the high speed at which it could be operated, at once commended it to the designers of locomotives of those days, and, although repeated efforts have

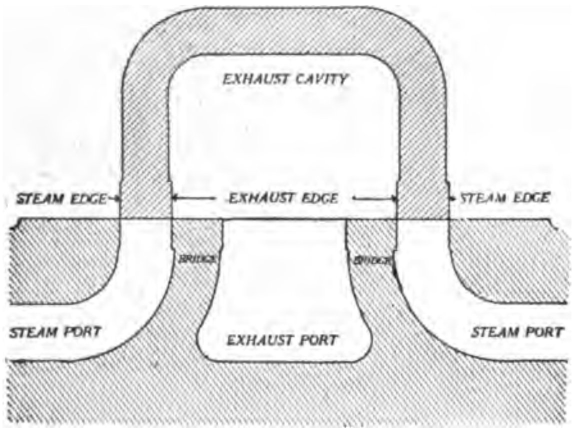

FIG. 1.

been made to displace it, it is still employed in one or another of its many forms on locomotives at the present time.

Elementary Principles.

The efficient and economical operation of a steam valve requires that a certain amount of steam, ample for the load to be moved, must be allowed to enter and escape from the cylinder at fixed predetermined points of the piston travel; for example:

First: Steam must not be admitted at both ends of the cylinder at the same time;

Second: The exhaust of steam from the cylinder should commence a little before, but never later than, the admission of steam at the opposite end, and

Third: The exhaust edges of the steam ports should be covered completely when the valve stands in its central position, so as to prevent steam from passing from the steam chest into the exhaust port.

A valve which will not fulfill these requirements cannot be used with success or economy.

Satisfactory results can only be obtained from a properly constructed valve, and its size is governed solely by the proportions of the valve seat upon which it slides.

For this reason, to assist the reader in understanding the designs of the valves now in use, we present a general outline of the valve seat.

Construction of the Slide Valve Seat.

The surface upon which the valve rests and slides, called the valve seat, is, generally, cast on top of the cylinder, and must be planed perfectly smooth, to avoid friction as much as possible, and to insure a steam tight fit. It is provided with two steam channels, or ports—one in each end, which also serve as exhaust passages; and one exhaust cavity, or port, situated in the center between the two end ports, all of which terminate in the smooth flat surface called the valve seat, or base.

Fig. 2 shows a top view, and a cross-section, of the valve seat. The steam ports communicate with the boiler and the exhaust cavity, and port, alternately, while the exhaust cavity, and port, terminate with the atmosphere, through the exhaust nozzle and stack, for the discharge of steam which has been used.

To insure good service, both the valve and the valve seat should be made of hard cast iron. It occasionally happens, however, that the cylinder casting is made of softer metal than it should be, and, as a result, the valve seat wears rapidly. In such cases a false seat must be applied, just as is done when the valve seat is faced down to its limit. The practice of substituting false seats should be avoided when possible, for they are expensive, and are almost sure to give trouble unless the work is performed in the best possible manner.

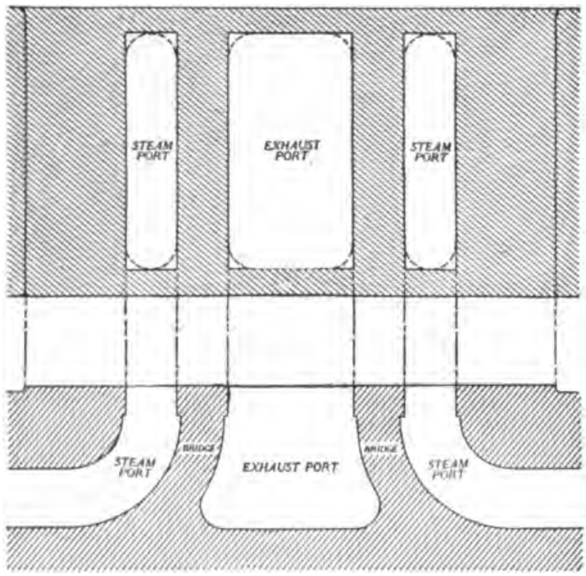

FIG. 2.

The valve seat should always be raised a little, from 1½ to 1¾ of an inch, above the surrounding metal of the cylinder, and should be a trifle shorter than the points of extreme travel of the valve, to prevent the wearing of a shoulder on the valve seat.

The longitudinal width of the valve seat is not so important, except where special valves of the Allen type are used, but when possible it should be made wide enough to allow a surface for the valve equivalent to the width of one bridge when the valve is at extreme travel position, as hereafter shown in Fig. 9, unless such allowance would result in the wearing of a shoulder on the valve seat when the engine is hooked-up in the working notch.

Bridges.

The walls, or partitions, which separate the steam-ports from the exhaust cavity, referred to as bridges, should be made as thin as possible, to reduce the pressure required to move the valve, but of sufficient strength to resist the highest pressure to which they might be subjected. The thickness of the cylinder walls is generally considered a safe measurement for that of the bridges. They are, however, usually made a little wider, but the face may be beveled without materially affecting its strength: however, it should be remembered that a reduction of 1/8 inch in its width will reduce the width of the valve 1/4 inch, and result in a decrease of the area on top. Over-travel should also be considered, and sufficient surface be provided for the valve, when it is in extreme travel position, to secure a steam tight joint, about 1/4 inch being sufficient. The wear must also be taken into consideration, for too narrow a bridge would not maintain a steam-tight joint.

As a general rule the width of the bridge is less than the width of the steam port, and on American locomotives it varies from 15/16 to 1 1/4 inches.

Port Opening.

The port opening is the width of opening, to or from a cylinder, when the valve is at its extreme positions.

An important task in designing a valve is to provide proper port areas for the admission and exhaust of steam. If the cross-sectional area of the port is too small, the steam will be choked, or throttled, its velocity increased, and its pressure reduced. This result is generally referred to as *wire-drawing,* the effect of which will be considered later on. On the other hand, if too large a channel be provided, the clearance volume will be unnecessarily increased, and economy of the engine reduced proportionally.

The proper size is that which will insure the largest port opening practicable at the running cut-offs, while reducing the preadmission in full gear to a negligible quantity. Any port opening desired at the working cut-off may be secured by giving the necessary lead to the valves. But to reduce preadmission, and render it negligible, so far as starting is concerned, steam lap should be added. The more lap given a valve, the less faulty steam distribution is to be expected, and vice versa.

Even though changes in speed may not affect the extent of port opening, the length of time of its opening will be reduced as the speed is increased, and, as a result, a smaller amount of steam will be admitted per stroke, depending, of course, upon the amount the speed is increased.

In some cases the throttling of steam is unavoidable, but it can be diminished by a quick opening and closing of the valve.

Location of Port Openings.

It is advisable, when possible, to use a small valve, and to place the ports as close together as practicable, for friction and weight are important considerations, and much work is required to accelerate and retard the moving masses employed; besides, the wear and tear of the valve gear will be in proportion to the acting forces. The unbalanced *D* slide-valve, as will be hereafter shown, has steam pressure over its entire back, or top, and the friction and weight are necessarily increased in proportion to its size.

Steam Ports.

No general rule governing the size of the steam ports can be stated, for considerable difference of opinion exists in regard to this, and it has not been determined with any degree of accuracy. But it may be said that the size depends, to a considerable extent, upon the speed of the piston and the dimensions of the cylinder.

Allowing the area of the steam ports to be about one-twentieth of the area of the cylinder, and the piston to be moving at about 300 feet per minute, saturated steam would travel through the ports at nearly its maximum velocity of 6,000 feet per minute. The ports should be arranged to suit this speed.

When the admission of full boiler pressure steam to the cylinder is desired, for high-speed engines, large ports are necessary, to secure free admission and exhaust, but small ports are more satisfactory and preferable, when they can be used, for they keep the valve motion within more practical limits.

The admission of live steam does not require as large a port opening as is necessary for the exhaust of the expended steam, and if the port is large enough for the exhaust to pass out without appreciable back pressure, it will be ample for the admission of steam. In high-speed engines there is seldom any difficulty experienced in getting steam into the cylinder, but occasionally trouble is encountered in getting the steam out of the cylinder fast enough.

It has been demonstrated that for a piston speed of 600 feet per minute, a good exhaust will be secured when the area of the steam port is 1/10 the area of the piston, if the steam is in an ordinary state of dryness. Of course, for slower piston speeds the steam port area will be less, and will be proportionally larger for increased piston speed; in fact the rule of proportion may be stated as follows:

Given piston speed in feet per minute $\times .1 \div 600 =$ the port area in fractional parts of the piston area.

. In other words, the specified piston speed in feet per minute multiplied by 1/10, and divided by 600, will equal the port area in fractional parts of the piston area.

Now, if we multiply the area in square inches of the piston by the port area, in terms of the piston area, we can ascertain the number of square inches that the steam port must contain; for example:

To find the steam port area for a cylinder 24 inches in diameter, and a piston speed of 650 feet per minute, we may put our data in the following form:

$$\frac{650 \times .1}{600} = .108$$

that is, the port area must be equal to 108/1000 part of the piston area. The area of a piston 24 inches in diameter is 452.39 square inches, hence $452.39 \times .108 = 48.85812$. This means that the steam port must be 48.85 square inches for this particular piston speed. Of course, for a slower piston speed, this port area should be less, for instance, if the piston speed is to be 500 feet per minute, and the diameter of the cylinder 24 inches, as in the previous example, we have—

$$\frac{500 \times .1}{600} = .083$$

and the piston area $452.39 \times .083 = 37.4837$ square inches for port area.

To ascertain the proper area for a steam port it is only necessary to multiply the area of the piston in square inches by the number opposite the piston speed shown in the following table, which has been found to give good results:

Speed of piston in feet per minute.	*Multiplied by*
100	.016
150	.025
200	.033
250	.041
300	.050
350	.058
400	.066
450	.075
500	.083
550	.091
600	.100
650	.108
700	.116
750	.125
800	.133
850	.141
900	.150
950	.158
1000	.166

Another rule for ascertaining the area of the steam port is this: multiply the square of the diameter of the cylinder by .078. The result, however, is applicable at about 600 feet per minute piston speed.

The length of the steam port is usually made equal to, but should never be less than three-fourths of, the diameter of the cylinder. Better results are obtained from long ports, as they increase the openings for admission and release, reduce the travel necessary to obtain a full port opening, and reduce the area on the back of the valve, so that the valve can be moved with less power.

Assuming that the length of a steam port, with square ends, has been decided upon, the necessary width of the port can be ascertained by dividing the area of the steam port by its length. If predetermined width is specified, divide the area of the port by the width, and the quotient will be the length of the port.

It is a general rule, in practice, to make the port opening coincide with the passage, or channel. It is also considered good practice to have the ends of the steam ports form a semi-circle, as shown by the dotted lines at the end of the ports in Fig. 2, for ports with square ends are liable to wear grooves and ridges, although they admit and cut off steam along the entire edge of the port at the same moment. Another advantage derived from constructing the ends of the ports semi-circular is the additional strength given to the bridges.

Exhaust Port.

The exhaust port is the opening, or cavity, in the valve seat, in which the exhaust passages terminate.

The exhaust port should be made more than twice the width of the steam port, especially with over-travel, to secure the free escape of steam, and to reduce the back pressure as far as practicable; if made smaller it would throttle or choke the steam, as shown by Fig. 9. On the other hand, it should not be made too wide, or it would unnecessarily add to the size of the valve and increase the pressure on it, thereby increasing friction and the wear and tear on the valve gear. In addition, it may be noted here that the size of the exhaust port has no material influence on the valve. With unbalanced valves, however, it is considered good practice to reduce the exhaust cavity as much as possible; in such a case the exhaust cavity opening at the extreme end of the valve travel may be reduced to about five-eighths of the width of the steam port area, as a result of the valve overlapping.

The general rule for ascertaining the width of the exhaust port may be stated as follows: Add the width of one steam port to one-half the travel of the valve, and from that amount subtract the width of one bridge. Another rule for determining the area of the exhaust port, is to multiply the square of the diameter of the cylinder by .178.

Assuming now that the reader has a general idea of the form of the valve seat, the question arises: Of what size shall we make the valve?

Construction of the Valve.

We do not believe that an analysis of the various theories advanced in favor of a large or small sized valve would be of any benefit to the reader at this time; therefore, we shall proceed to construct a valve for the valve seat previously illustrated.

The face of the valve seat is here shown by the line A B in Fig. 3, and we will now draw the perpendicular line C D through the center of the line A B, to locate the center of the exhaust port and the center of the valve when it stands in mid-position.

If the valve be constructed without exhaust lap, or exhaust clearance, its exhaust edges must coincide with the exhaust edges of the steam ports, and its inside width must equal that of both bridges and the exhaust port, or be $1\frac{1}{4}+1\frac{1}{4}+3=5\frac{1}{2}$ inches.

Next we will draw the heavy base lines from the outer edges of the bridges to the outer edges of the steam ports, a distance of $1\frac{3}{8}$ inches.

This will give us a valve which is termed *line and line,* as shown in Fig. 1, but such a valve would not be satisfactory, for reasons we shall soon make clear; therefore, we must give the valve steam lap, and, without stopping at this time to state the rule for determining the proper amount of lap for any particular valve, we will add the width of one bridge, or $1\frac{1}{4}$ inches, at each end, so that our heavy base line is now $1\frac{1}{4} + 1\frac{3}{8} = 2\frac{5}{8}$ inches, at each end.

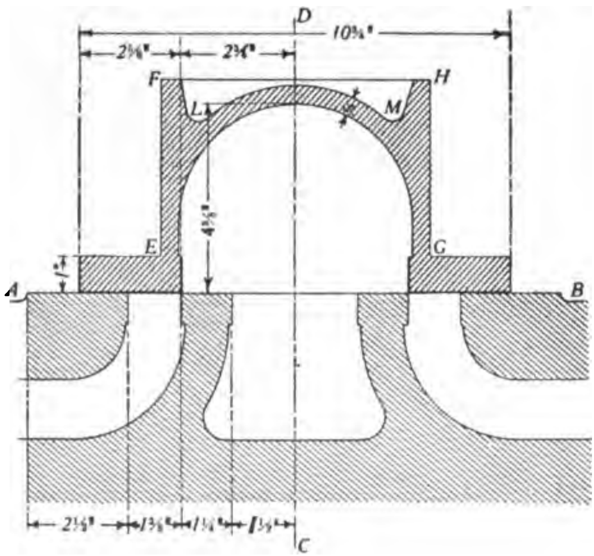

FIG. 3.

Now, in determining the thickness of the metal necessary to support the valve face we must allow something for wear, and its depth should be enough to insure a proper support for the valve yoke, so we will make the flange twice the thickness of the sides or top of the valve, or 1 inch.

Now, to lay out the exhaust cavity, we will draw two heavy lines 1 inch high, to correspond with the outer edges of the valve flanges, from the inner edges of the steam ports, making the two lines perpendicular to the line A B. The depth of the exhaust cavity is sometimes made a trifle less, but it should not be greater, than the width of the exhaust port plus one steam port. We believe, however, the larger size will give better results so we will make it $3+1\frac{3}{8}=4\frac{3}{8}$ inches.

Now we must decide upon the height of the outer walls of the valve, and upon the shape of the top of the valve, for some valves are constructed with a round top, as illustrated in Fig. 1, while others are made with a square top.

If the flange of a valve face is allowed ½ inch for wear, then the thickness of the top and sides for either form of valve may be made the same, for they do not come in contact with wearing surfaces, and are not subject to wear.

Without attempting to decide upon the advantages to be derived from the adoption of either form of top, or crown, we shall assume the round top is, for some reason, undesirable.

Now, returning to Fig. 3, we will draw a light line, from the top of the outer end of the metal flange supporting the valve face, to a point ½ inch from the inner edge of the exhaust arch.

It is obvious that the outer sides, designated E F and G H, must extend upward a distance in excess of the depth of the exhaust cavity ($4\frac{3}{8}$ inches), and the thickness of the exhaust arch (½ inch), or a total of $4\frac{7}{8}$ inches, to secure a square top, so we will draw the two outer lines to a point 5 inches from the valve face.

The two depressions, or recesses, indicated by the letters L M, are solely for the purposes of saving metal and dispensing with unnecessary weight of the valve.

A flat top for the valve is preferable because the increased height of the valve provides a larger surface against which the valve yoke may bear, and it gives a form which may be laid on its back and secured to the planer without difficulty, when the valve face is to be planed. The flat top also furnishes a convenient base for the insertion of balance springs and strips, when it is desired to convert a plain slide valve into a balanced valve.

Unless the reader thoroughly understands the meaning of the terms commonly used in describing the construction and operation of the valve, it will be difficult, if not an impossibility, for him to intelligently follow our discussion of the valve and the valve events. Therefore, to assist the student, we will first explain the meanings of the principal terms, and, afterwards, the construction of the valve.

Valve Events.

The valve events are the periods during its travel when the valve and port edges register. They are the points of the piston at which five distinct events occur, in the distribution of steam, during one revolution of the crank, namely: Admission, Cut-off, Expansion, Release, or Exhaust, and Compression. The events enumerated are all important and may be defined as follows:

Admission.—Is the entrance of steam, through the steam port opening, to the cylinder, to be used in applying positive pressure to the piston. The admission of steam begins when the steam edge of the valve uncovers the steam edge of the steam port, at the point indicated by the small arrow, in Fig. 4, when the valve is moving in the direction of the arrow B, and continues until the valve reaches the end of its travel and returns to the same point—at which admission commenced—when the supply of steam to the cylinder is cut off.

Cut-off.—Is the stopping, or cutting off of, the admission of live steam to the cylinder, before the piston has completed its stroke. The point of cut-off is reached when the steam edge of a valve arrives at the same position as for admission, but is moving in the opposite direction, and completely closes the steam port, as shown by the small arrow in Fig. 4, when the valve is traveling in the direction of the arrow A.

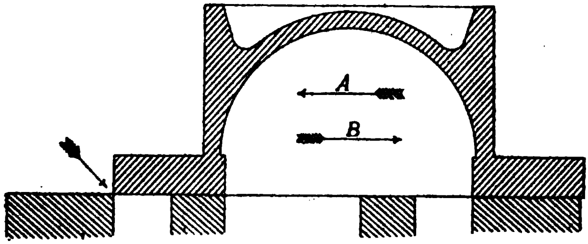

FIG. 4.

Expansion.—Is the continuation of a positive pressure on the piston, beyond an admission period, which occurs in a cylinder when the steam supply is cut off between the steam chest and the cylinder by valve lap, while the piston continues its motion. Expansion begins at the point of cut-off, and continues to the point of release, or exhaust, shown by the small arrow in Fig. 5, when the valve moves in the direction of the arrow A.

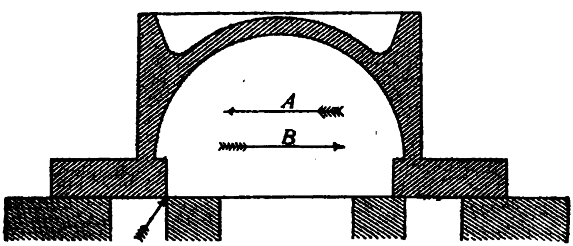

FIG. 5.

Release, or Exhaust.—Is the reduction of pressure on the piston, and is accomplished by the release of the expanded steam from the cylinder to the atmosphere, and is caused by exhaust port opening. The point of release is reached when the exhaust edge of the valve uncovers the exhaust edge of a port and allows the steam which has performed its work in driving the piston, to escape through the exhaust port. The point of release is shown by the small arrow in Fig. 5, when the valve is traveling in the direction of the large arrow A, and it continues until the valve reaches the end of its travel and returns to the same point—when release ceases and compression begins.

Compression.—It is accomplished by suspending communication from a cylinder to the exhaust passage, before the piston has completed its stroke. The point at which compression begins is reached when the exhaust edge of the valve has completely closed the port and cut off the escape of exhaust steam, as shown by the small arrow in Fig. 5, when the valve is traveling in the direction of the arrow B, and the piston is approaching the end of its stroke to the left, and it continues until preadmission. The positions of the valve for compression and release are the same, but the valve is moving in opposite directions for the two events.

The resistance of the exhaust during the period steam is being released is called "back pressure," and it is sometimes designated as an "event" of the stroke, which may be defined as follows:

Back Pressure.—It is the pressure on the exhaust side of the piston, caused by the steam left in the cylinder after the exhaust opening, which opposes the advancing motion of the piston during its return stroke, and causes a reduction of the work accomplished by the piston through the action of live steam.

It is also important for the reader to note what details of the valve, and connected parts, govern the valve events enumerated, and what relation they bear to each other, by relative changes in any of them. Those positions of the valve which control the distribution of steam are steam lap and exhaust lap—the former governing admission and cut-off, and the latter release and compression.

Lap.—The distance the steam edge of a valve overlaps the steam port, when the valve stands in its central position, is called *steam-lap,* or simply *lap.* Steam lap is equal to the distance the valve is displaced from mid-position when admission or cut-off occurs. The distance by which the exhaust edge of a valve over-

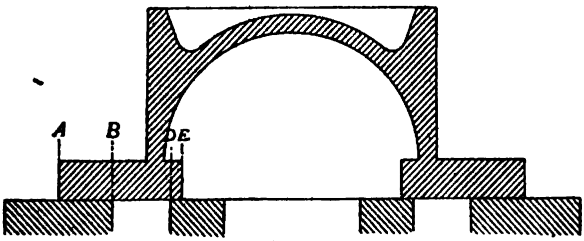

FIG. 6.

laps the steam port, when the valve stands central upon the valve seat, is known as *exhaust-lap,* or *cover.* Exhaust lap is equal to the distance the valve is displaced from mid-position at the point of release, or compression. Steam-lap is indicated by the distance between the lines A and B, while exhaust-lap is shown by the distance between the lines D and E, in Fig. 6.

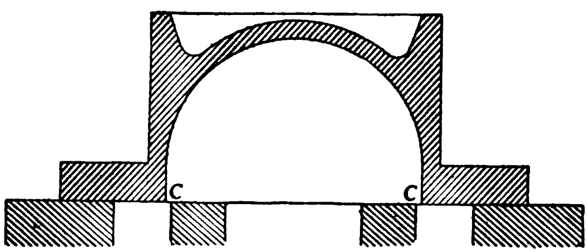

FIG. 7.

Exhaust Clearance.—Is the distance between the exhaust edge of the valve and the exhaust edge of the port—the width of port uncovered by the exhaust edges of a valve when the valve is in mid-position. It is also called *negative exhaust lap,* and is indicated by the letters C and C, in Fig. 7.

Lead.—Is the width of steam port opening at the beginning of the piston stroke. It is not a part of the valve proper, but is indicated by the letter A, in Fig. 8.

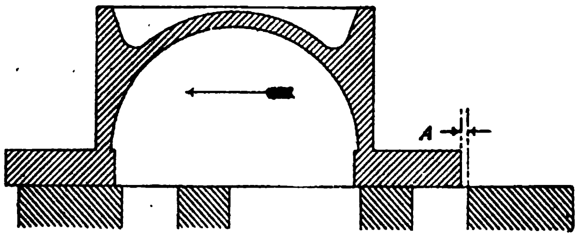

FIG. 8.

Valve Travel.—Valve travel is the distance the valve travels when moved from one extreme position to the other; its maximum movement, unless otherwise designated. It is twice the eccentricity, or throw of the eccentric.

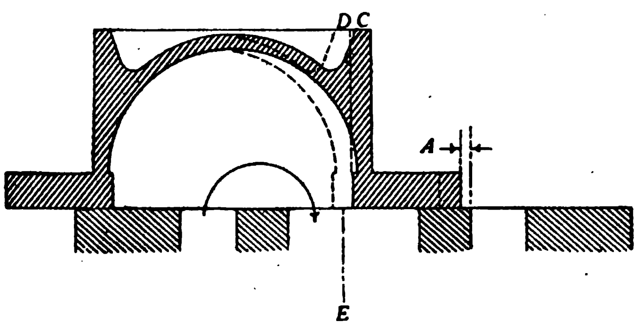

FIG. 9.

Over-Travel.—The distance the valve travels after the steam port is wide open, as indicated by the distance between the steam edge of the valve and the exhaust edge of the steam port, shown by the letter A, in Fig. 9.

Admission.

Admission is accomplished by establishing communication between a steam chest and cylinder, as a result of steam port opening. It is a continuation of preadmission, and its duration is from the beginning of preadmission until the valve reaches the end of its travel and returns to the cut-off position.

During this period of admission, it is essential that a sufficient supply of steam be taken into the cylinder to move the piston for the work required during the entire stroke.

Cut-Off.

The cut-off of the admission of live steam to the cylinder from the valve is one of the most important events of the piston stroke. It does not take place at the same point on the forward and backward strokes, respectively, because the slant, or obliquity, of the connecting rod to the line of stroke results in the piston being more advanced in the forward than in the backward stroke than it should be to correspond precisely with the position of the crank. This disparity will be explained later on.

The percentage of the piston's stroke at which cut-off occurs depends entirely upon the position of the reverse lever in the quadrant. It is measured by the amount of piston travel completed at the time cut-off occurs, and it varies with the amount of valve travel. If the stroke of the piston is 32 inches, economy of steam can be obtained by placing the reverse lever at a point which will cause the valve to cut-off at about 8 inches, which would be one-quarter of the piston stroke, and allowing the expansion of steam in the cylinder to supply the force for the remaining 24 inches of the piston stroke. If, however, the desired speed cannot be obtained by such an arrangement, the stroke may be increased by moving the reverse lever farther away from the center of the quadrant. In fact, the point of cut-off is generally controlled by local conditions, and it generally ranges from 6 to 20 inches, or from 25 to 60 per cent of the piston's stroke.

The point of cut-off may be changed by altering the lap of the valve, and this may also be accomplished by changing the angular advance of the eccentric, and its throw, without changing the lap of the valve.

An early cut-off of the steam causes a restriction of the steam port opening, especially near the point of cut-off, where the current of the steam is rapid, and is followed by an early release and exhaust closure, which causes considerable increased compression.

An early cut-off is of material advantage, when an engine is operated at high-speed, because it allows more time for the exhaust steam to escape during the quick movement of the valve.

The increased compression resulting from an early cut-off may be considered an unavoidable evil and an inherent defect, or as a desirable feature and an incidental advantage, of the combination slide-valve; but the point of view depends entirely upon existing conditions and varying theories.

An increase of the travel of the valve would generally be considered the most effective remedy, or correction, for compression, but where an unbalanced valve is used, an extension of the valve will increase the unbalanced area and the friction proportionally. When a contraction, or limitation, of the valve travel is considered necessary, duplicate or supplementary ports may be used to advantage.

The contraction of the steam port opening is not of much importance unless the steam is cut-off at an early point in the stroke of the piston, and its most objectionable feature is an early, or premature, release of steam.

Expansion.

Following the point of cut-off, the piston will be moved only by the expansive force of the steam. That is, the pressure of the steam confined in the cylinder, exerting pressure on the head of the piston, will force it forward, or backward, as the case may be, thus economically utilizing the expansive properties of the steam. This will increase the space in which the steam is confined, and there will be a fall in pressure approximately proportioned to the increase in volume. Expansion will continue from the time cut-off begins until the valve opens for exhaust, that is, for only a small portion of the stroke; a portion equal to the total of the exhaust and the steam lap of the valve.

It must be apparent that if the opening of the valve for exhaust could be delayed so that expansion would continue until the pressure of the steam dropped to zero, practically all of the available energy of the steam would be utilized, and converted into draw-bar pull. But such an arrangement would be impracticable for an ordinary cylinder, and in order to free the cylinders, the steam must be released earlier, to allow the steam sufficient *time* to escape from the cylinder chamber before the piston begins its stroke in the opposite direction.

Compression.

Compression is measured by the amount of *unfinished* piston travel when the exhaust ceases, and it varies with the cut-off. Its object is to economically produce initial pressure for the return piston stroke and consequently reheat the piston and cylinder surfaces.

The desire to eliminate compression, or to reduce it to a minimum, is prompted by the fact that compression shortens the effective power stroke of the engine. The advantage of eliminating this resistance is generally admitted, for, although a portion of the energy exerted in causing compression may be regained later on, it is evident that *only a part* of it can reasonably be expected; but even if all of the power were returned there would be a useless trading of energy from which no benefits would be obtained.

Of course, compression assists the lead opening, because it partly fills the clearance space with steam pressure, helps to maintain the heat of the cylinder, prevents loss from steam condensation, acts as a cushion in bringing the piston to rest at the end of the stroke, and also obviates the shock which would otherwise be caused by the admission of high-pressure steam for the return stroke. It must be remembered, however, that no advantage is derived from pressure on the crank pins when the piston is within two inches of the end of its stroke, for it only increases friction and performs no useful work. In fact, compression represents negative horse power, developed by the closing of the exhaust cavity.

In high-speed engines, compression is generally considered advantageous and essential, unless it produces a pressure equal to or higher than the initial pressure, but this occurs only a part of the time, usually when the engine is running at the higher speeds with cut-off at ¼ stroke or less. At other times compression does not fill the clearance spaces and the shortage must be provided by live steam from the boiler. But it should be remembered that excessive compression, even in high-speed service, will cause an engine to ride hard, and, if there be any wear in the rods and boxes it will pound.

Release, or Exhaust.

The point of release is just as important as the points of expansion and compression, and may be considered an aid of the two events, for each depends upon the others. If release occurs too early the benefits of expansion are sacrificed and compression will be delayed; on the other hand if release occurs too late the beneficial effect of expansion will be off-set by back pressure.

The valve should open for exhaust a little before the piston reaches the end of its stroke, to give the exhaust steam ample *time* to escape from the cylinder before the piston begins its return stroke. If the exhaust steam were not discharged before the piston commenced its return stroke, the cylinders would become choked, and each end would continually be working against the other end.

Back-Pressure.

As all of the steam employed in driving the piston cannot immediately escape through the port when the valve opens for exhaust, a certain portion of it remains in the cylinder for a time, and acts as an obstruction to the piston when it begins its return stroke; the amount depending, to a considerable degree, upon the dimensions of the port and of the exhaust nozzle, together with the presence or absence of exhaust clearance of the valve.

There is always some back pressure in a locomotive cylinder, and a certain amount of this steam is necessary to create the draft for the fire, in passing through the exhaust nozzle, but an excessive amount will cause a serious loss of efficiency, and its presence can be readily detected by the increased amount of fuel consumed.

The retarding force of back pressure against the piston in locomotives with simple cylinders is about 7 to 9 pounds per square inch, but it may increase slightly as the speed increases, for the amount of condensation is not exactly constant at all speeds. It should not, however, be greater than 10 pounds at a speed of 20 miles per hour, or 15 pounds at 50 miles per hour.

Now that the reader understands the terms defined, an explanation of the reasons why the valve is given these functions is necessary.

Valves Without Lap.

In the early days of locomotive construction valves without lap, as shown by Fig. 1, or with only a trifling amount of steam lap, were used to secure a prompt and free admission of steam; but when the beneficial results of working the steam expansively were thoroughly understood, such valves were discarded.

Such a valve, Fig. 1, is described as *line and line;* that is, when the valve stands on the center of the valve seat the exhaust and steam edges of the steam ports and the exhaust and steam edges of the valve coincide. By moving this valve, either to the right or left, steam will be admitted into one port, and, at the same time, the opposite port will be opened to the exhaust of steam. It is evident that such a valve will admit steam during the entire stroke of the piston, or, in other words, follow full stroke, and release the steam from the opposite end of the cylinder at the same time. A valve of this kind is extremely wasteful, because it does not use the steam expansively. It was also found to be unsatisfactory, because the steam pressure in the cylinder, during the greater part of the piston's stroke, remained at about the same pressure as that in the boiler, and for the further reason that at the commencement of the stroke the advancing piston encountered a full cylinder of useless steam, which caused considerable back pressure.

The use of valves without steam lap was discontinued long ago, and they are not in use at present.

Valves with Lap.

• Lap is given a valve to cause it to cut off steam at predetermined points of the piston's stroke.

By the use of steam lap, steam can be worked expansively, by which a great saving in fuel is obtained, and the tendency has been to increase the lap, until at present it varies from about ¾ of an inch to 1¼ inches. The proportion between maximum (full gear) valve travel and steam lap is, in most cases, about 6, or 7, to 1.

When the Indicator, as described in the following pages, was invented, and applied to the locomotive, it was shown that additional work was derived from the steam by working it expansively, due to steam lap.

A long steam lap is economical, but it is not always desirable—for the service often requires a sacrifice in economy to obtain the desired results. The amount of steam lap to be given depends, of course, upon the character of the service the engine is required to produce, and it varies from ½ to 1½ inches. The tendency at the present time is to increase the amount of steam lap, with a corresponding increase in travel. An increase in steam lap will affect the distribution of steam as follows:

Admission is later; ceases sooner.

Cut-off is earlier; expansion longer.

Release is unchanged; exhaust period unchanged.

Compression is unchanged; period unchanged.

Travel must be greater for a given maximum port opening.

Exhaust Clearance.

Exhaust clearance is given a valve to reduce the resistance to the movement of the piston, and it should be sufficient to take full advantage of the nozzle opening, and must be greater for high speed than for slow speed engines.

One of the early objections to giving a valve exhaust clearance was the fact that it placed the steam ports in communication with each other, as well as with the exhaust port, and it was thought that the steam would pass through and be wasted. It was soon discovered, however, that the intercourse between the steam ports did not produce the anticipated loss of live steam, for both of the steam port channels act only as exhaust ports when they are in communication.

The old practice of giving valves exhaust lap has been discontinued almost altogether, and instead valves are now made with exhaust edges, line and line, or are given from 1/8 to 3/16 of an inch exhaust clearance. The amount depends, however, upon the nature of the service required, and the character of the roadbed, and it can be safely said that 1/4 inch exhaust clearance is excessive for ordinary conditions.

The objectionable features resulting from exhaust clearance are the sacrifice of water and fuel for increased speed. It is also undesirable at long cut-offs, because it exhausts the steam earlier, with the result that more steam is exhausted without performing its proportion of the work expansively.

Exhaust clearance is certainly an advantage, however, to fast running engines, or engines with small wheels running at a moderate rate of speed, because, at high piston speeds, it is often more trouble to get the exhaust steam out of the cylinder than it is to get the live steam in. An increase in exhaust clearance will affect the steam distribution as follows:

Admission is unchanged.

Cut-off is unchanged; expansion period shorter.

Release is earlier; exhaust period longer.

Compression is later; period shorter.

Lead.

Lead causes preadmission, and is given to increase the pressure in the clearance space of the steam port channel and to insure high initial steam port opening for an abundance of steam pressure, to assist expansion of the compressed steam behind the piston, at the very beginning of the piston's stroke—when full steam pressure in the cylinder is most advantageous; it also helps to maintain a satisfactory supply of steam throughout the period of admission.

In starting, and working at slow speeds, it can readily be seen that lead is undesirable, and should be sacrificed as much as possible, for instead of making an engine "smart," or quick, it has a tendency to retard the movement of the piston at the completion of its stroke.

It may also be noted that a proper reduction of the amount of lead will give a more efficient steam distribution, because such a reduction of lead will not necessarily reduce the maximum port opening at short cut-off sufficiently to materially impair the admission of steam by wiredrawing, for the speed of the piston at the quarter stroke is comparatively slow.

On the other hand, as the speed of the engine is increased, the time allowed for the admission of steam to the cylinder is diminished, and a certain amount of lead is necessary to give a larger port opening for the admission of steam in the short cut-offs. An increase in lead will affect the steam distribution as follows:

Admission is earlier; period unchanged.
Cut-off is earlier; expansion period unchanged.
Release is earlier; exhaust period unchanged.
Compression is earlier; period unchanged.
Maximum port opening is earlier.

Valve Travel.

The travel of the valve is of necessity the most important feature of the valve operation, because a change in the travel must be accompanied with changes in all of the functions of the valve.

As the minimum amount of travel necessary to give a full port opening must equal twice the width of the steam port, an increase in lap will require a corresponding increase in the travel. But the total width of the lap of the steam port and the bridges combined, plus the over-travel, if any, should not be more than one-half of the travel. If over-travel causes cut-off to occur too late, it can be remedied by increasing the steam lap, while the delay in the exhaust caused by over-travel can be neutralized by giving the valve exhaust clearance.

The greater the travel the longer the steam port will remain open for the admission of steam, but increased travel requires more power to operate the valve, with greater wear of the valve and its seat, but these disadvantages are partly overcome by the benefits derived from better steam distribution. An increase in the travel of the valve will affect the steam distribution as follows:

Admission is earlier; continues longer.
Cut-off is later; expansion period shorter.
Release is later; exhaust period shorter.
Compression is later; period shorter.

Over-Travel.

The over-travel of a valve, as previously defined, tends to choke the exhaust, but insures sufficient port opening with an early cut-off, and may be considered advantageous from the standpoint of steam distribution, but the gain obtained, or loss sustained, cannot be calculated by any fixed rule, for the results depend upon circumstances.

If over-travel causes the cut-off to occur too late, the evil effects can be overcome by increasing the steam lap, while delayed compression can be neutralized by increasing the exhaust lap; if exhaust occurs too late, as a result of over-travel, the exhaust lap may be sacrificed, but if there is no exhaust lap the valve may be given exhaust clearance.

In Fig. 9 it will be observed that the slide valve is shown in two positions on the valve seat. When in the position marked C the valve has traveled to the left beyond the point necessary to give the full steam port opening, and is about to commence its return travel to the right, but the inner edge of the valve has not traveled to the left beyond the center of the exhaust port. On the other hand, the position of the valve marked D shows that the inner edge has not only passed the center of the exhaust port, but the line marked E, which is drawn a distance from the left hand bridge equal to the width of the steam port. When a valve is given sufficient over-travel to bring it beyond the position corresponding to the line E, or, in the position of the valve marked D, it is evident that the exhaust port will be contracted, and this will prevent the free escape of exhaust steam. Over-travel will affect the steam distribution as follows:

Admission is earlier; continues longer.

Cut-off is later; expansion period shorter.

Release is later; period longer.

Compression is later; period shorter.

Clearance.

It may be well for the reader to remember that there is a difference between *clearance,* and the terms *engine-clearance,* and *piston-clearance,* for they must not be confounded, but we shall here treat clearance as all of the steam channel space between the valve and the piston when the piston is at the beginning of its stroke.

The compression of steam by the advancing piston will be increased or decreased, and its expansive force is raised or lowered, in proportion to the amount of clearance present between the valve and the piston.

A reduction of the clearance space is a distinct advantage, aside from its connection with proper cushioning, for a reduction in this space adds to expansion, reduces the surface for steam condensation, and will prove economical, for the clearance space must be filled with live steam from the boiler, to a considerable extent. While direct and straight ports to the cylinder will, to some extent, reduce the clearance, it is almost impossible to design, or construct, a large locomotive cylinder without allowing at least 8 or 9 per cent cylinder clearance.

Incidentally it may be said that *piston-clearance* is the distance between a piston, when at either end of its stroke, and its striking position against its nearest cylinder head. It is given to prevent the piston from striking, and bursting, either cylinder head, when the brasses on the connecting-rod wear and lost motion develops, or when water accumulates in the cylinder. In locomotive cylinders, piston clearance varies from 1/4 to 1/2-inch; it is generally 3/8 of an inch.

Preadmission.

Preadmission is the point at which the steam edge of a valve begins to open the port, for the admission of steam, before the piston has completed its stroke. It is measured by the amount of unfinished piston movement, and varies with the amount of valve travel.

Preadmission is always subject to change, even when the lead is constant, and is, under ordinary conditions, the greatest factor in compression, for it commences when the valve port opens for lead and ends when the crank pin is on center, or at the point where lead begins.

The accompanying outline drawing, Fig. 10, clearly shows the commencement and brief duration of preadmission.

By referring to the illustration it may be seen that preadmission commences in the right-hand port when the valve begins to uncover the rear port, when the crank-pin center, which is connected to the piston, is in the position shown; and it ends when the crank-pin is on dead center, that is, when the piston reaches the end of its stroke to the right.

This small movement of the piston is just sufficient to move the valve to the left to give full lead opening in the right-hand port.

Of course, preadmission is caused by lead, and is fixed by the amount of lead, but, during the time the main pin is traveling from the preadmission to the lead position, the movement of the valve is so rapid, while that of the piston is so slow, that preadmission may be considered negligible; at least, so far as starting is concerned. In fact, with certain valve gears, lead in full gear does not give more than 1/32 of an inch preadmission. But, on the other hand, when the reverse lever is moved toward the center of the quadrant (hooked-up), from 7/8 to 1¼ of an inch preadmission may be secured, but it may be as short as 5/8 of an inch with some valve gears.

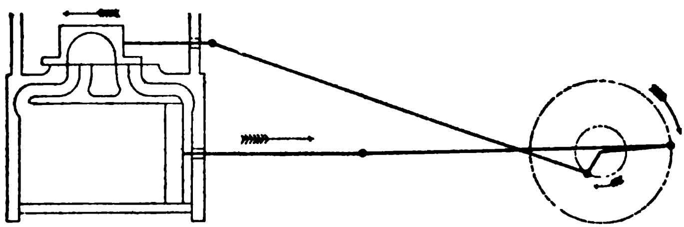

FIG. 10.

There is, however, considerable difference of opinion regarding the beneficial results to be obtained by preadmission, and many able men who have given the matter serious consideration are inclined to believe that preadmission is of no special benefit, for the reason that compression is fairly well developed before preadmission commences.

WIREDRAWING.

Wiredrawing results from reducing the area through which the steam may flow, thus materially reducing its pressure after passage through the constricted opening.

As the amount of space available for the steam ports of a locomotive is limited, their areas are necessarily small, even when they are fully opened, and their outlines are far from being straight; as a result the passageways for the admission of steam from the boiler to the cylinder are restricted so that wiredrawing is unavoidable, to a certain extent. In such cases the steam is said to be throttled on its way from the boiler to the cylinder, and, in consequence, its pressure behind the piston is reduced as the piston proceeds on its stroke, until, at the point of admission, it falls below the boiler pressure.

It is generally believed that wiredrawing in locomotive service is, to any extent, objectionable, and a thing to be avoided, but, as a matter of fact, it is always present, and renders useful service in certain classes of work. It aids in the maintenance of uniform boiler pressure for various rates of speed and is, therefore, beneficial from an economic standpoint.

For example, take an engine traveling up a grade at the rate of 20 miles per hour, with the throttle and reverse-lever in a position to require a moderate working load on the boiler; when the apex, or top, of the grade is passed, the load will be reduced in proportion to the grade, and the speed will increase twice or three times that maintained in ascending the grade, but the throttle and reverse-lever may be allowed to remain in their former positions, for the wiredrawing action will answer the demand, and prevent the cylinder from demanding more steam than the boiler can supply, without reducing the speed, or the capacity of work.

In fact, wiredrawing results in economy of steam in proportion to the increase of speed from about 20 to 50 miles per hour, but for greater speeds the evil effect is increased, for it produces increased steam consumption.

RELATIVE POSITIONS OF THE PISTON AND CRANK PIN.

As the connection between the crosshead and the piston is rigid, their positions correspond exactly; hence, if we know the motion of one of them, we also know the motion of the other.

We will therefore dispense with a view of the piston in our illustrations for the present, and ignore the crosshead connection; proceeding on the theory that the piston is connected to the crank by a link. This link is generally termed a connecting rod, and is employed in changing the reciprocating rectilinear motion of the piston into the circular motion of the crank pin.

If the ~~eccentric~~ connecting rod were of infinite length, or if its obliquity were neglected, then, with the crank pin at any position between the front and back dead centers, the corresponding position of the piston could be ascertained by drawing a perpendicular line upon the diameter of the crank pin circle, which may be said to represent the stroke of the piston. For example, when the crank had traveled one-quarter of a revolution, the piston would be in the center of its stroke, etc. In practice, however, the changing angles of the connecting rod during the different periods of the crank pin revolution must be taken into consideration.

Any irregularities imparted by the crank pin to the motion of the piston will be conveyed to the motion of the valve, but the throw of the eccentric is generally so small in comparison with that of the crank pin that it is inappreciable and may be disregarded, for the eccentric rod is proportionally longer than the connecting rod, and it follows that the distortions in the motion of the valve are necessarily much less than those in the motion of the piston.

The obliquity, or angularity, of the connecting rod can be more forcibly impressed upon the reader's mind by the use of an illustration than by any written description we might give, so we shall proceed to construct Fig. 11, to show the travel of the piston, crank pin and connecting rod. The length of the connecting rod, which is measured from the center of the crank pin to the center of the crosshead pin, varies in practice from four to eight times the length of the crank, so, for the purpose of this illustration, we will use a connecting rod six times the length of the crank.

First we will draw the circle, *a p b d,* which is the crank pin circle, with the diameter equal to the travel of the piston. Let *C* represent the center of the crank shaft, and the horizontal line, *A b,* which passes through the center of the circle *C,* the center line of motion of the crosshead, piston rod and piston. Now from the point *a,* of the circle as a center, and with a radius equal to the length of the connecting rod, strike an arc cutting the line *A b* in the point *A.* This point will be the center of the crosshead pin at the forward end of its travel. From the point *b* as

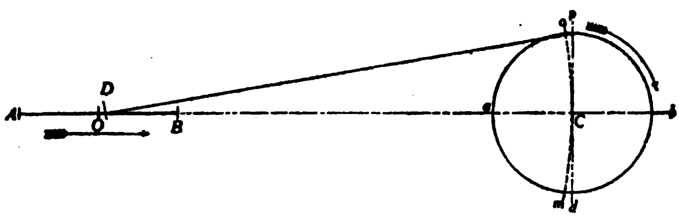

FIG. 11.

a center, and with the same radius, strike another arc cutting the line *A b* at the point *B,* and this point will be the center of the crosshead pin at the back end of its travel. Since the distance between the points *A* and *B* represent the full length of the piston stroke, we may find the center of the stroke by striking an arc on the line *A b* from the center *C* of the circle, using the same radius, which will equally divide the distance between *A* and *B;* and we will mark this point *O* on the line *A b.* Now from the point *O,* and with a radius equal to the length of the connecting rod, we will draw the arc *o C m,* and mark the points *o* and *m* where it intersects the crank pin circle; the two points *o* and *m* will be the centers of the crank pin when the piston is at the center of its stroke.

By referring to the illustration again it will be noticed that the end of our connecting rod has reached its quarter stroke and is at the point *p*, so, with a radius equal to the length of the connecting rod, we will strike an arc cutting the line *A b* in the point *D*. The point *p* bisects the semi-circumference which extends from *a* to *b*, but it may be readily observed that the point *D* does not bisect the stroke *A B;* in fact the point *D* is much nearer the *B* end of the stroke than it is to *A*.

From this we see that during the first half stroke of the piston the crank pin travels from *a* to *o*, and during the second half stroke of the piston the crank pin travels from *o* to *b*. Since the motion of the crank must be uniform, the average speed of the piston during the first half stroke will be a little greater than during the second half stroke; and this difference in the speed of the piston is due to the obliquity of the connecting rod.

Now, assuming that our illustration of the variation in the piston's travel, between the first and the second half of its stroke, is clear, the reader may find it profitable to ascertain the movements in the different periods of the stroke.

To assist in the continuation of our examination, let us make a new illustration, making the crank pin circle and piston travel the same as in Fig. 11, but on a slightly larger scale. We shall, however, employ, in Fig. 12, a connecting rod three times the length of the crank, instead of six times its length, as used in Fig. 11. A connecting rod of this length could not be used in locomotive service, and it is only adopted here to emphasize the movement of the piston at different positions.

Our next step will be to divide the piston travel and the upper semi-circumference of the crank pin circle into eight equal parts, which we will number to correspond. Advancing the short connecting rod from the point *a*, which represents the front dead center, to the point 1 on the semi-circumference of the crank pin circle, we strike an arc cutting the line *A b* in the point *c*, which will indicate the position of the crosshead pin, and, in a similar manner, from each of the other figures on the semi-circumference of the crank pin circle we strike similar arcs and mark with letters to correspond, using as a radius, throughout, the length of the connecting rod.

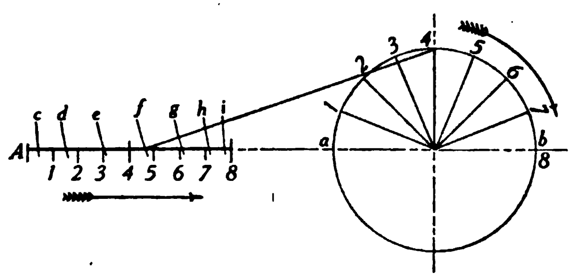

FIG. 12.

The crank pin moves at a symmetrical rate of speed through the divisions we have marked on the semi-circumference of the crank pin circle, yet we find from an examination of Fig. 12, that the movements of the crosshead pin, and therefore of the piston, from *c* to *d*, and from *e* to *f*, on the line *A b*, are not equal. In fact, it shows that the piston will, at the commencement of its stroke, move comparatively slow, and increase its speed as it approaches the center of the stroke, and when the piston is moving away from the center of the stroke, its speed is constantly decreasing.

This variable motion of the piston is caused chiefly by changing the rectilinear, or straight, motion into a uniform circular motion, and to some extent, by the angle formed by the center line of the connecting rod and the horizontal line *A b*, an angle which is constantly changing during the stroke.

Various expedients have been adopted to overcome, or neutralize, the evil effects due to the angularity of the connecting rod, but it has been found that fairly good results can be obtained, at small expense, by setting the link saddle pin back. This remedy will be explained later on.

Position of the Eccentric for a Valve Without Lap.

In practice the eccentric is always mounted on the crank-shaft, and, if the valve has no lap and no lead, the center line of the eccentric must be perpendicular to the center line of motion of the valve gear when the piston is at the end of its stroke and the valve stands in its central position.

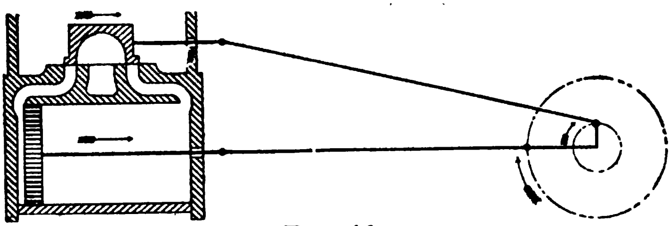

FIG. 13.

The eccentric must, in all cases, travel ahead of the crank, and when the connection between the valve and the eccentric is direct, as in Fig. 13, and the piston is at the end of its stroke, the crank is on dead center; therefore, the valve must be in its central position, and the center of the eccentric must be midway between the two dead centers. (Here we have replaced the eccentric with a crank of equal throw, for clearness of illustration.)

In such a case, when the piston begins to move toward the right-hand end of the cylinder, the center of the eccentric will turn, with the crank, to the right, and move the valve in the same direction, so that the left-hand edge of the valve face will open the left-hand steam port. This admits steam to the left-hand steam port, behind the piston, driving it in the desired direction, while the right-hand edge of the valve will uncover the right-hand steam port, and permit the steam in front of the advancing piston to escape.

If the center of the eccentric were placed one-quarter behind the crank, midway between the two dead centers, it is apparent that when the piston commenced its stroke to the right the valve would be moved to the left, thereby uncovering the right-hand steam port for the admission of steam in *front* of the piston. If, however, we turn the crank in the opposite direction, that is, to the left, the valve will uncover the left-hand steam port and admit steam *behind* the piston and the engine will move in the opposite direction. In other words, if the center of the eccentric is placed 90 degrees *ahead* of the center line of motion of the valve gear the shaft will rotate to the *right;* if placed 90 degrees *behind* the center line of motion of the valve gear the shaft will turn to the *left.*

We shall assume that the parts, shown in Fig. 13, are about to move in the direction of the arrows. When the crank reaches the top quarter, the eccentric center will be on the back dead center, the valve will have reached the end of its travel to the right, and the piston will have almost reached the center of its stroke. The piston would be exactly at the center of its stroke but for the angularity of the connecting-rod, which has been explained. When the parts have moved another quarter the crank will be on the back dead center, the eccentric will be on the lower quarter, the piston will have reached the end of its stroke to the right, and the valve will have moved to the left, so that it stands central. Another quarter movement will place the crank on the lower quarter, the eccentric center on the forward dead center, the valve at the end of its travel to the left, and the piston almost in the center of its stroke. The final quarter movement will place the parts in the positions they originally occupied in the illustration. (It must be obvious that a valve of this pattern will not admit steam into the cylinder to *start* the movement of the engine from a state of rest, for the valve opens just *after* the stroke of the piston commences.)

The primitive form of valve illustrated, which just covers the steam ports when in mid-position, can not be used with economy, as it allows steam to follow the piston for the whole of the stroke, and does not permit the use of its expansive properties, for the simple reason that, at the instant the admission of steam ceases, the exhaust of the same body of steam must immediately commence, thereby sacrificing the beneficial results obtained from expansion.

As it has been shown that this elementary valve is extremely wasteful, and therefore impractical, let us modify it by extending its face so that when the valve is in mid-position it will overlap the steam ports. Thus we may see what effect the addition of lap to the valve will have upon the position of the eccentric.

Position of the Eccentric for a Valve With Lap.

Now that we are about to examine the positions of the eccentric for a valve with steam lap, it may be well to define the terms "linear advance" of the valve, and "angle of advance" of the eccentric.

The *linear-advance* of the valve is the distance the valve is moved from its central, or mid-position, when the piston reaches the end of its stroke, or, the distance by which the port is open when the piston begins its stroke.

The *angle of advance* represents the distance the center of the eccentric is advanced, toward the crank-pin, from a line drawn through the center of the axle at right angles to the center line of motion. Or, the combined angle of lap and lead through which the valve must move from its central position to the position it should occupy when the crank is on dead center. It is sometimes designated *angular advance.*

It will be found that the positions of the two are closely related, and, in a sense, dependent upon each other, yet the distinction between the terms is of great importance and should be thoroughly understood.

When the valve is given lap, that is, when the face of the valve is lengthened, it will be necessary for the linear travel of the valve to be increased a distance equal to the steam lap, for the admission of steam to take place. The valve must travel the length of the lap plus the width of the port, if the port is to be fully opened, and, of course, increased linear travel means increased throw, or advance of the eccentric, which will require a change in the position of the eccentric on the crank shaft.

In Fig. 14 we have shown the valve in two positions, and have placed the eccentric circle above the same, so that the reader may readily observe the effect that a movement of either the valve or the eccentric will have upon the other. The dotted lines represent the valve standing in the center of its travel, and, for reference, we have marked it *A*. The other view shows the valve at the commencement of the piston stroke, and is marked *B*.

If we assume that the piston is at the beginning of its stroke to the right, it is obvious that the valve *A* is not in a position to admit steam to the left-hand port; in fact to admit steam into the cylinder behind the piston the valve *A* must be moved to the right from its central position until it has opened the left-hand port, and, if the valve is given lead, an additional amount equal to the lead. As a result the center of the eccentric, at the beginning of the piston's stroke, cannot remain in the position shown in Fig. 13, that is, at right angles to the crank, but must be advanced (in this case, to the right) an amount equal to the lap and the lead combined, as shown in the illustration. When the necessary amount of linear advance is given to the valve *A*, it will occupy the position of the valve *B* and admit steam behind the piston at the beginning of its stroke to the right.

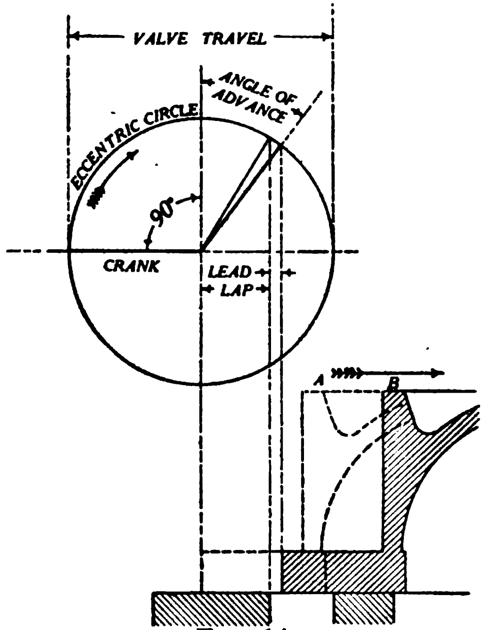

FIG. 14.

As a result of the change in the position of the eccentric all of the events effected by the valve will be completed earlier; that is, the port for the admission of steam will be partly opened at the beginning of the piston's stroke, and it will be closed before the end of the stroke, so that the steam will be allowed to expand during the period from cut-off to release. The exhaust will also be closed before the end of the stroke, and the undischarged steam trapped in the cylinder will undergo compression.

Relative Positions of the Valve and Piston.

Now that the reader is familiar with the various positions of the valve on its seat for the events of the stroke, and the positions of the eccentric and crank for a valve without lead or lap, let us compare the different positions of a valve with steam lap to the corresponding positions of the piston.

The motion of the piston is not symmetrical, as has already been explained, but we shall here disregard this slight error due to the angularity of the connecting-rod.

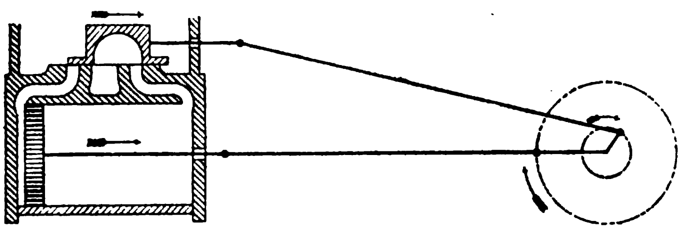

FIG. 15.

Admission, or Lead Opening. In Fig. 15 the slide valve is shown at the point of lead opening, that is, the left-hand port is open to the admission of steam into the head-end of the cylinder. The eccentric arm, instead of being at right angles to the crank at the beginning of the stroke, as shown in Fig. 13, is set ahead an amount equal to the angular advance. The reason for this is, as previously explained, that the valve, instead of standing central as it would if it had no lap or lead, must be advanced, or set ahead an amount equal to the steam lap plus the lead, in order that steam may be admitted into the cylinder at, or a little before, the commencement of the stroke of the piston. The crank pin is on the forward dead center, and the piston is about to begin its stroke, so the admission of steam behind the piston will now begin, and continue, through the left-hand port, behind the piston, until the valve returns to close the left-hand port.

Cut-off. When the valve has traveled to the end of its stroke to the right, in order to fully open the left-hand port to admission, it must return in the direction of the arrow to the position shown in Fig. 16. The eccentric arm will have moved from its position in Fig. 15 to that in Fig. 16, in the direction indicated by the arrow. During this period a corresponding movement of the crank takes place, causing the piston to assume the position shown in Fig. 16. It will be noticed that the piston has already completed more

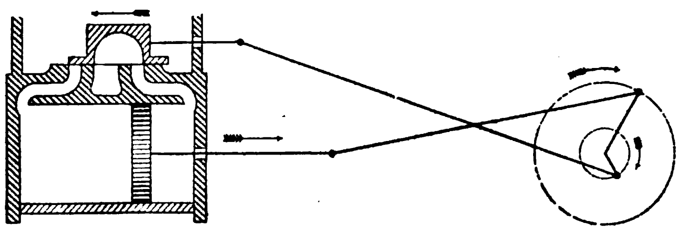

FIG. 16.

than half of its stroke to the right. A further movement of the valve in the direction of the arrow will close the left-hand port to the admission of steam, that is, cut-off takes place. Expansion will now commence in the head-end of the cylinder, and the force of the expanding steam will cause the piston to continue its movement to the right.

Compression. As the position of the piston in Fig. 17 is reached, the valve will have moved to cover the right-hand port, as shown in the illustration. The unexhausted steam trapped in the crank end of the cylinder will be compressed by the expansive force of the steam previously admitted to the head-end of the cylinder acting upon the piston and moving it to the right. Expansion, theoretically, should cease immediately at the point of release, but in practice there is a gradual, although rapid, decrease in pressure on the piston, causing a slight prolongation of expansion.

Release. Because the valve shown in these illustrations is line-and-line with the steam ports, that is, it has no exhaust lap or clearance, the point of the commencement of compression in the crank-end corresponds to the point of release in the head-end.

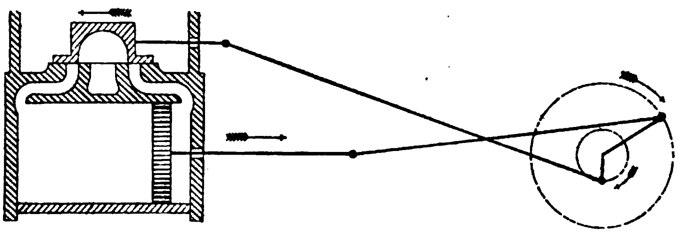

FIG. 17.

If the valve had exhaust lap, release would follow compression; if it had clearance, release would take place before compression began. This is readily understood by referring to Fig. 17. Exhaust will continue from the point of release shown in Fig. 17 at the left-hand port, until the point of compression on the return stroke of the piston.

Admission—Crank-end. Compression will continue from the point shown in Fig. 17 to that shown in Fig. 18, where the valve

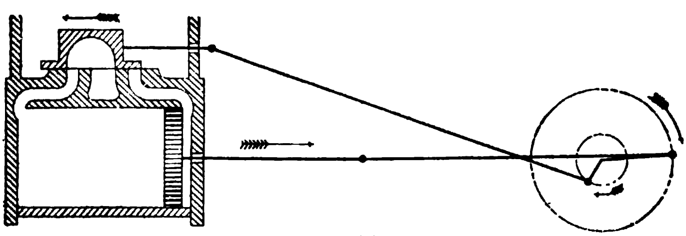

FIG. 18.

is just about to uncover the right-hand port to admission. It may be seen from the position of the crank that the piston has not yet reached the end of its stroke. The reason for this is that preadmission commences a little before the point of full lead opening, and therefore before the completion of the stroke, is reached. The eccentric arm, too, must travel an amount equal to the lead opening of the valve in order to reach a point exactly opposite that shown in Fig. 15.

Valve With Rocker Arrangement.

The reader should bear in mind that the foregoing illustrations are primarily intended to show the corresponding positions of the piston and a valve with steam lap, and the various positions of the eccentric and crank are shown only for that purpose.

It may be observed in the illustrations that the valve is not in line with the eccentric, and for this reason, if for no other, the

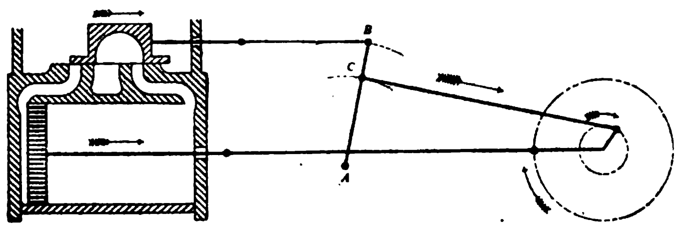

FIG. 19.

direct connection between the valve and eccentric would be impracticable. As a result of this angle between the eccentric rod and the center line of motion, the speed of the valve would vary during the stroke, even though the eccentric rotated at a constant speed. For this reason a rocker, of the type shown in Figs. 19 or 20, is interposed between the eccentric and the valve, and is used merely to connect the eccentric rod and the valve rod. While it affects the position of the eccentrics, and often the travel of the valve, the rocker will not necessarily affect the rules regarding the valve construction.

In Fig. 19 the rocker is pivoted at A, and the rotating motion of the eccentric imparts a backward and forward motion to the rocker through the connection C. This reciprocating movement is applied to the valve through the valve stem connected to the rocker at B.

It is often more convenient to place the pivot of the rocker between the valve stem connection and the connection with the eccentric rod, as shown in Fig. 20, at the point A. Now, in order to impart to the valve the same motion as that in Fig. 19, the eccentric must be set ahead 180 degrees, for it will be seen, as indicated by the arrows, that in this illustration the valve stem and eccentric rod move in opposite directions. Thus, even though the eccentric rod moves in the direction opposite to that in Fig. 19, the valve rod moves in the same direction as formerly, be-

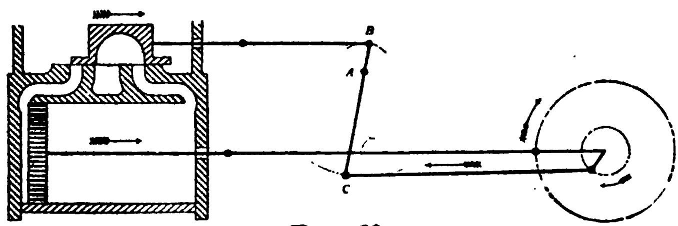

FIG. 20.

cause the pivot A of the rocker is placed *between* the two connections. Either type of rocker may therefore be used with the same results.

The rocker may also be used so as to make the throw of the eccentric less than the travel of the valve, for, as the distance between the points B and C, Fig. 19, is increased, the throw of the eccentric for a given travel of the valve is decreased.

Direct and Indirect Valve Motion.

The rocker arm is usually the means by which the motion imparted to the eccentric is reversed. With the Stephenson link motion, if both arms extend in the same direction from the rocker shaft, the motion is direct. However, if they extend in opposite directions from the shaft, one up and one down, the motion is reversed, and the valve is indirect.

With the more modern radial valve gears, the motion is direct when the link block is working below the center of the link, when it has the same direction of motion from the eccentric to the valve. But when the link block is working above the center of the link, the motion is indirect. Therefore, reversing the engine, or placing the link block above the center of the link, changes the valve motion from direct to indirect.

Inside admission valves, as the modern piston valve, also influence the valve motion. When the valve admits steam from the inside, it must move in the opposite direction to the outside admission valve, that is, in a direction opposite to that of the piston at the commencement of the piston stroke. To accomplish this, the valve may be driven direct, without a rocker, leaving everything else the same. The rocker will give the desired motion, however, if both eccentrics are moved half way round the axle.

The following is a simple means of distinguishing between direct and indirect motion:

If the crank pin and eccentric are on the same side of the driving shaft, and a rocker is provided to reverse the motion, the valve is indirect, outside admission. But if the crank pin and eccentric are together without a rocker arm, the valve has inside admission, direct.

If the crank pin and eccentric are on opposite sides of the driving shaft, with a rocker to reverse the motion, the valve has inside admission, and is indirect. With crank pin and eccentric opposite without a rocker, however, the valve has direct motion, and is outside admission.

The distinction may be stated briefly as follows: When the eccentric rod moves forward, and produces a foreward movement of the valve, the motion is direct. But, if when the eccentric moves forward, the valve moves back, the motion is indirect.

Problems Relating to the Slide Valve.*

To find the point of cut-off when the lap and travel of the valve are given, the valve to have no lead.

Example.—Lap of valve is one inch; travel, 5 inches; no lead; stroke of piston, 24 inches. At what part of the stroke will the steam be cut off?

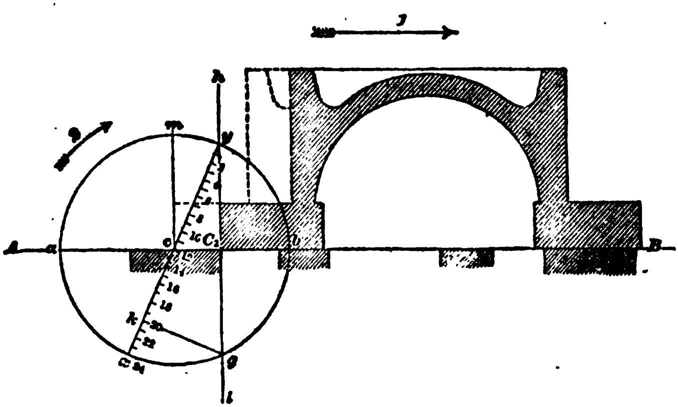

FIG. 21.

We must first find the center *c*, Fig. 21, of the circle ***a b m***, whose circumference represents the path of the center of eccentric, and this is found by placing the valve in a central position, as shown in dotted lines in this figure. Then the edge *c* of the valve will be the center of the circle. The valve drawn in full lines shows its position at the commencement of the stroke of piston; and since the valve is to have no lead, the edge C_2 will coincide with the outer edge of the steam port. Through the edge C_2 draw the line *i h* perpendicular to the line *A B;* the line *i h* will intersect the circumference ***a b m*** in the point *y*, and this

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point will be the center of eccentric when the piston is at the beginning of its stroke. Now, assume that the circumference *a b m* also represents, on a small scale, the path of the center of the crank-pin; then the diameter *y x* of this circle will represent the length of the stroke of the piston; the position of this diameter is found by drawing a straight line through the point *y* (the center of the eccentric when the piston is at one end of its stroke) and the center *c*. Also assume that the point *y* represents the center of the crank-pin when the piston is at the beginning of its stroke. To make the construction as plain as possible, divide the diameter *y x* into 24 equal parts, each representing one inch of the stroke of piston, and for convenience number the divisions as shown. The arrow marked 1 shows the direction in which the valve must travel, and arrow 2 indicates the direction in which the center *y* must travel. Now it must be evident, because the points *y* and C_2 will always be in the same line, that during the time the center *y* of the eccentric travels through the arc *y g*, the valve not only opens the steam port, but, as the circumference *a b m* indicates, travels a little beyond the port, and then closes the same, or, in short, during the time the center of eccentric travels from *y* to *g*, the port has been fully opened and closed; and the moment that the center of eccentric reaches the point *g*, the admission of steam into the cylinder is stopped. We have assumed that the point *y* also represents the position of the center of crank-pin at the beginning of the stroke; and, since the crank and eccentric are fastened to the same shaft, it follows that during the time the center of eccentric travels from *y* to *g* the crank-pin will move through the same arc, and when the steam is cut off the crank-pin will be at point *g*. Therefore, through the point *g* draw a straight line *g k* perpendicular to the line *y x*; the line *g k* will intersect the line *y x* in the point *k*, and this point coincides with the point marked 20; hence the steam will be cut off when the piston has traveled 20 inches from the beginning of its stroke.

Lead Will Affect the Point of Cut-Off.

In Fig. 21 the valve had no lead; if, now, in that figure, we change the angular advance *m y c* of the eccentric so that the valve will have lead, as shown in Fig. 22, then the point of cut-off will also be changed. How to find the point of cut-off when the valve has lead, is shown in Fig. 22.

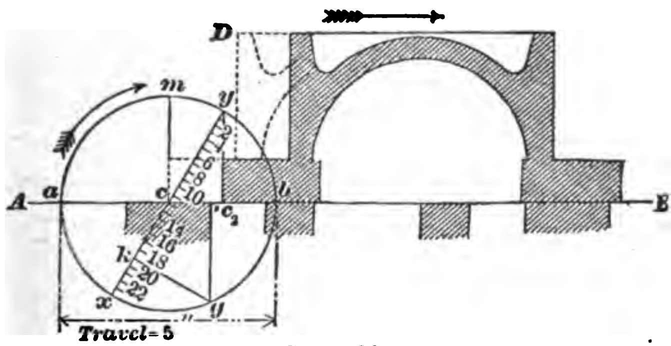


FIG. 22.

Example.—The lap of valve is 1 inch, its travel 5 inches; lead ¼ of an inch (this amount of lead has been chosen for the sake of clearness in the figure); stroke of piston, 24 inches; at what part of the stroke will the steam be cut off?

On the line *A B*, Fig. 22, lay off the exhaust and steam ports; also on this line find the center *c* of the circle *a b m*, in a manner similar to that followed in the last construction, namely, by placing the valve in a central position, as shown by the dotted lines, and marked *D*, and then adopting the edge *c* of the valve as the center of the circle *a b m;* or, to use fewer words, we may say from the outside of the edges *s* of the steam port, lay off on the line *A B* a point *c* whose distance from the edge *s* will be equal to the lap, that is, 1 inch. From *c* as a center, and with a radius of 2½ inches (equal to one-half of the travel), describe the circle *a b m*, whose circumference will represent the path of the center of eccentric. The lead of a valve in a locomotive ranges from 0 to as much as ¼ of an inch, usually about 1/16, when the valve is in full gear, but for the sake of distinctness we have adopted in this construction a lead of ¼ of an inch in full gear. Draw the section of the valve, as shown in full lines, in a position that it will occupy when the piston is at the beginning of its stroke, and consequently the distance between the edge c_2 of the valve and the edge *s* of the steam port will, in this case, be ¼ inch. Through c_2 draw a straight line perpendicular to *A B*, intersecting the circumference *a b m* in the point *y*; this point will be the center of the eccentric when the piston is at the beginning of its stroke, and since it is assumed that the circumference *a b m* also represents the path of the center of the crank-pin, the point *y* will also be the position of the crank-pin when the piston is at the commencement of its stroke. Through the points *y* and *c* draw a straight line *y x*, to represent the stroke of the piston, and divide it into 24 equal parts. Through the point *s* draw a straight line perpendicular to *A B*, intersecting the circumference *a b m* in the point *g*, and through *g* draw a straight line perpendicular to *y x*, and intersecting the latter in the point *k*; this point will be the point of cut-off. If, now, the distance between the point *k* and 19 is about ⅛ of the space from 19 to 20, we conclude that the piston has traveled 19⅛ inches from the beginning of its stroke when the admission of steam into the cylinder is suppressed.

Here we see that when a valve has no lead, as in Fig. 21, the admission of steam into the cylinder will cease when the piston has traveled 20 inches; and when the angular advance of the eccentric is changed, as in Fig. 22, so that the valve has 1/4 of an inch lead, the point of cut-off will be 19 1/8 inches from the beginning of the stroke, a difference of 7/8 of an inch between the point of cut-off in Fig. 21 and that in Fig. 22. But the lead in locomotive valves in full gear is usually only about 1/16 of an inch, which will affect the point of cut-off so very little that we need not notice its effect upon the period of admission, and, therefore, lead will not be taken into consideration in the following examples.

The Travel of the Valve Will Affect the Point of Cut-Off.

Fig. 23 represents the same valve and ports shown in Fig. 21, but the travel of the valve in Fig. 23 has been increased to 5 3/4

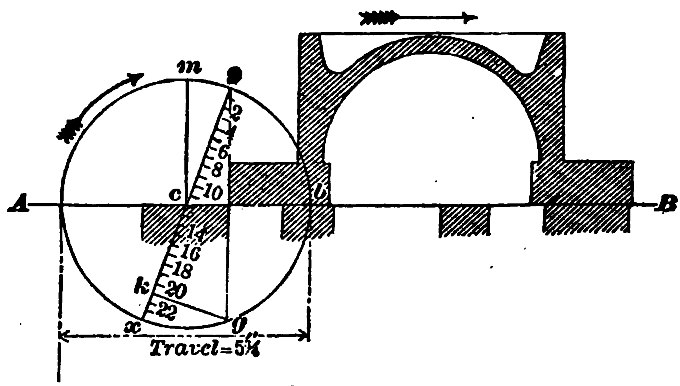


FIG. 23.

inches. The point of cut-off *k* has been obtained by the same method as that employed in Figs. 21 and 22, and we find that this point *k* coincides with point 21. Now notice the change caused by an increase of travel; when the travel of the valve is 5 inches, as shown in Fig. 21, the admission of steam into the cylinder will cease when the piston has traveled 20 inches from the commencement of its stroke, and when the travel of the same valve is increased ¾ of an inch, as shown in Fig. 23, the admission of the steam will not be suppressed until the piston has traveled 21 inches. Here we notice a difference of 1 inch between the two points of cut-off. But it must be remembered that when the travel of a valve for a new engine is to be found or established, the point of cut-off does not enter the question; we simply assign such a travel to the valve that steam ports will be fully opened, or give it slightly greater travel when the valve is in full gear. The point of cut-off is regulated by the lap and position of the eccentric.

In order to find the point of cut-off it is not necessary to make a drawing of the valve, as has been done in Fig. 22. The only reason for doing so was to present the method of finding the point of cut-off to the beginner in as plain a manner as possible. In order to show how such problems can be solved without the section of the valve, and, consequently, with less labor, another example is introduced.

Example.—Lap of valve is 1⅜ inches; travel 5½ inches; stroke of piston, 24 inches; width of steam port, 1¼ inches; find the point of cut-off.

Draw any straight line, as $A\ B$, Fig. 24; anywhere on this line mark off 1¼ inches, equal to the width of the steam port. From the edge s of the steam port lay off on the line $A\ B$ a point c, the distance between the points s and c being 1⅜ inches; that is, equal to the amount of lap. From c as a center, and with a radius equal to half the travel, namely, 2¾ inches, draw a circle $a\ b\ m$; the circumference of this circle will represent the path of the eccentric, and also that of the crank-pin. Through s draw a straight line $i\ h$ perpendicular to $A\ B$; this line $i\ h$ will intersect the circumference $a\ b\ m$ in the points y and g. Through the points *y* and *c* draw a straight line *y x*; the diameter *y x* will represent the stroke of the piston. Divide *y x* into 24 equal parts; through the point *g* draw a straight line *g k* perpendicular to *y x*, and intersecting *y x* in the point *k*, and this point is the point of cut-off. Since *k* coincides with the point 18, it follows that the piston had traveled 18 inches from the beginning of its stroke when the flow of the steam into the cylinder ceased.

Now we may reverse the order of this construction and thus find the amount of lap required to cut off steam at a given portion of the stroke.

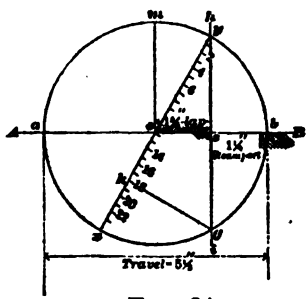


FIG. 24.

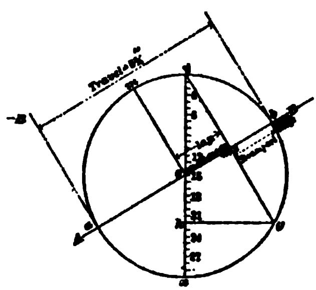


FIG. 25.

Example.—Travel of valve is 5¾ inches; stroke of piston, 30 inches; steam to be cut off when the piston has traveled 22 inches from the beginning of the stroke; find the lap.

Draw a circle *a b m* whose diameter is equal to the travel of the valve, viz., 5¾ inches. Through the center *c* draw the diameter *y x*. In Fig. 25 we have drawn the line *y x* vertically. for convenience; any other position of this line will answer the purpose equally well. The circumference *a b m* represents the path of the center of the eccentric, also that of the crank pin; the diameter *y x* will represent the stroke of the piston, and, therefore, is divided into 30 equal parts. The steam is to be cut off when the piston has traveled 22 inches from the beginning of the stroke, therefore, through the point 22, draw a straight line *g k* perpendicular to *y x*, the line *g k* intersecting the circumference *q b m* in the point *g*. Join the points *y* and *g* by a straight line. Find the center *s* of the line *y g*, then, through *s* and perpendicular to the line *y g*, draw the line *A B*; if the latter line is drawn accurately it will always pass through the center *c*. The distance between the points *s* and *c* will be the amount of lap required, and in this example it is 1-7/16 inches.

It sometimes occurs, in designing a new locomotive, and, often, in designing stationary or marine engines, that only the width of steam port and point of cut-off are known, and the lap, and the travel of the valve are not known. In such cases both of these can be at once determined by the following method:

Example.—The width of the steam port is 2 inches; the stroke of piston, 30 inches; steam to be cut off when the piston has traveled 24 inches from the beginning of its stroke; find the lap and travel of the valve.

Draw any circle, as *A B M*, Fig. 26, whose diameter is larger than the travel of the valve is expected to be. Through the center *c* draw the diameter *y x*, and, since the stroke of piston is 30 inches divide *y x* into 30 equal parts. Steam is to be cut off when the piston has traveled 24 inches; therefore through point 24 draw a straight line *g k* perpendicular to the diameter *y x*, intersecting the circumference *A B M* in the point *g*. Join the points *y* and *g* by a straight line; through the center *s* of the line *y g* draw a line *A B* perpendicular to *y g*. So far, this construction is precisely similar to that shown in Fig. 25, and in order to distinguish this part of the construction from that which is to follow, we have used dotted lines; for the rest full lines will be used. It will also be noticed by comparing Fig. 26 with Fig. 25 that, if the diameter *A B* had been the correct travel of valve, then $c\ s$ would have been the correct amount of lap. But we commenced this construction with a travel that we knew to be too great; hence, to find the correct travel and lap, we must proceed as follows: Join the points B and y. From s towards B, lay off

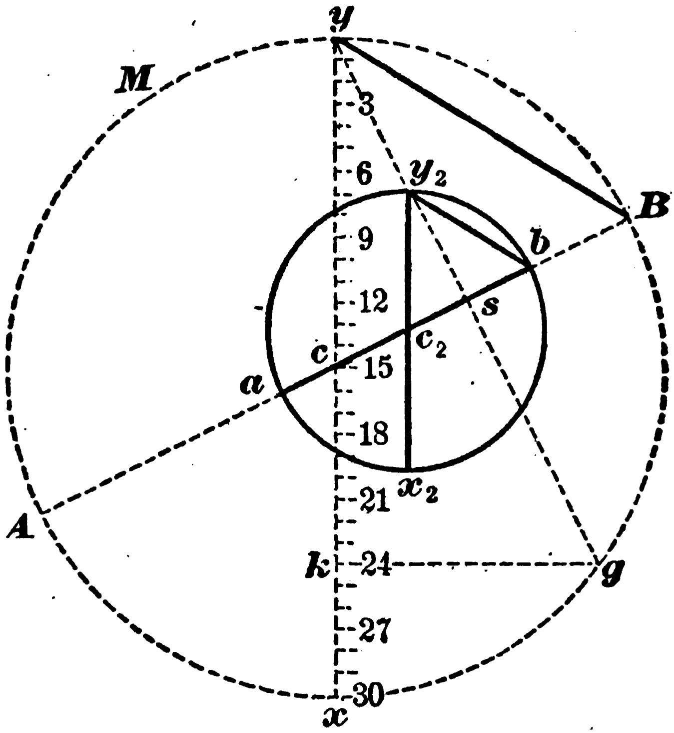


FIG. 26.

on the line $A\ B$ a point b; the distance between the points s and b must be equal to the width of the steam port plus the amount that the valve is to travel beyond the steam port, which, in this example, is assumed to be ⅛ of an inch. Therefore the distance from s to b must be 2⅛ inches. Through b draw a straight line $b\ y_2$ parallel to $B\ y$, intersecting the line $y\ g$ in the point y_2. Through the point y_2 draw a straight line $y_2\ x_2$, parallel to the line $y\ x$, and intersecting the line $A\ B$ in the point c_2. From c_2 as a center, and with a radius equal to $c_2\ b$, or $c_2\ y_2$, describe a circle $a\ b\ y_2$. Then $a\ b$ will be the travel of the valve, which, in this case, is 7⅝ inches, and the distance from c_2 to s will be the lap, which, in this example is 1-11/16 inches.

The Events of the Distribution of Steam.

The outside edges c_2 and c_3 of the valve, and the outside edges o and o_2 of the steam ports, will regulate the admission and suppression of steam; the inner edges i and i_2 of the valve and the inner edges s and s_2 of the steam ports control the release and compression of steam. The parts of the stroke of the piston during which these events will happen can be found by the following methods:

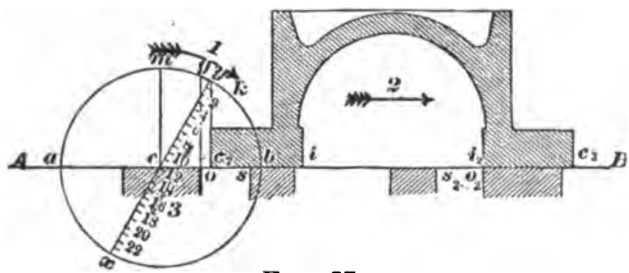


FIG. 27.

Example.—Travel of valve, 5 inches; lap, 1 inch; lead, ¼ of an inch; stroke of piston, 24 inches; no inside lap or clearance. Find at what point of the stroke the admission, suppression, release, and compression will take place.

In Figs. 27, 28 and 29 the valve occupies different positions, but the sections of the valve in these figures are exactly alike, because they represent one and the same valve. In Fig. 27 the distance between the edge c_2 of the valve and the edge o of the steam port is ¼ inch, which is the amount of lead given in our example; hence, this position of the valve indicates that the piston is at the beginning of its stroke, and the angle *m c y* is the angular advance of the eccentric. In Fig. 28 the edge c_2 of the valve and the edge *o* of the steam port coincide, and, since the valve is moving in the direction indicated by arrow 2, the suppression commences, or, in other words, the valve is cutting off steam when it is in the position as here shown. In Fig. 29 the inside edge *i* of the valve coincides with the inner edge *s* of the steam port, and, since the valve is moving in the direction indi-

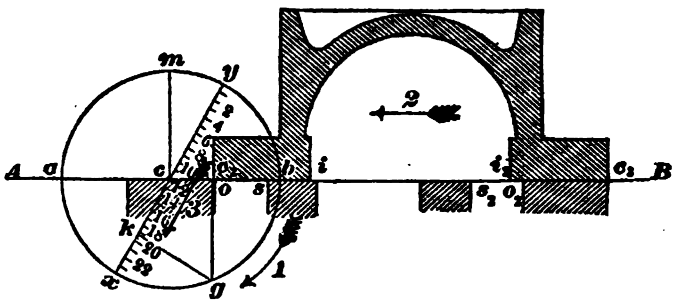


FIG. 28.

cated by arrow 2, the release must commence when the valve arrives in the position here shown.

In Figs. 27, 28 and 29 the distances from the outside edge *o* of the steam port to the center *c* of the circle *a b m* are equal; that is, the points *c* and *o* are one inch apart, which is the amount of lap. The diameters of the circles *a b m* are all five inches, which is the travel of the valve given in the example, and the circumference of each circle represents the path of the eccentric, and also the path of the center of the crank-pin. The point *y* in these figures represents the position of the center of eccentric when the piston is at the beginning of its stroke. The distance between the point *y* and *m* is the same in all figures, and consequently the angles formed by the lines *y x* and *m g* are equal and represent the angular advance of the eccentric.

When the valve occupies the position as represented in Fig. 27, the center line of crank will coincide with the line *A B;* and since the piston will then be at the beginning of its stroke, it follows that the line *A B* will indicate the direction in which the piston must move. In order to compare the relative position of the piston with that of the valve with as little labor as possible, we shall assume that the direction in which the piston moves is

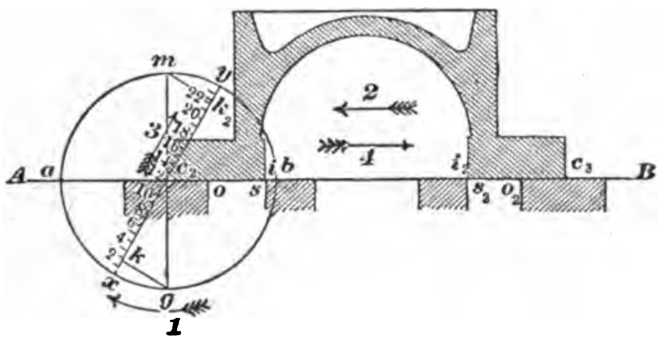


FIG. 29.

represented by the line *y x,* instead of the line *A B;* hence the point *y* will not only show the position of the center of the eccentric, but it will also indicate the position of the center of the crank-pin when the piston is at the commencement of its stroke. If these remarks are thoroughly understood, there will be no difficulty in comprehending that which is to follow.

Now let us trace the motions of the valve and piston and thus determine at what part of the stroke the events (previously named) will take place. When the crank-pin is moving in the direction as indicated by the arrow marked 1, Fig. 27, the center of eccentric will move through part of the circumference, *a b m,* and the valve will travel in the direction indicated by the arrow

2, thus opening the steam port wider and wider until the end *b* of the travel is reached; then the valve will commence to return, and as it moves toward the center *c*, the steam port gradually closes, until the valve reaches the position as shown in Fig. 28; then the steam port will be closed and steam cut off. To find the position of the piston when the valve is cutting off steam, we draw through the edge c_2 of the valve, Fig. 28, a straight line c_2 *g*, perpendicular to *A B*, intersecting the circumference *a b m* in the point *g*; through this point draw a line perpendicular to *y x* intersecting the latter in the point *k*, and this point *k* being 19⅛ inches from *y* indicates that the piston has traveled 19⅛ inches from the beginning of its stroke before the steam is cut off, and that steam has been admitted into the cylinder during the time the piston traveled from *y* to *k*. As the piston continues to move toward the end *x* of the stroke, the valve will move in the direction of the arrow 2, Fig. 28, and the steam port will remain closed so that no steam can enter the cylinder or escape from it; hence the steam that is now confined in the cylinder must push the piston ahead by its expansive force, but the moment that the valve reaches the position as shown in Fig. 29 the release of steam will commence. To find the corresponding position of piston we draw through the edge c_2 of the valve, Fig. 29, a line c_2 *g*, perpendicular to *A B* intersecting the circumference *a b m* in the point *g*. Through this point draw a line *g k* perpendicular to *y x*, intersecting the latter in the point *k*, and this point *k* being 22⅜ inches from the beginning of the stroke indicates that the piston has traveled through this distance when the release of steam commences. Now notice, the steam is cut off when the piston has traveled 19⅛ inches, and the release of steam commences when the piston has traveled 22⅜ inches, consequently the steam is worked expansively during the time the piston moves 3¼ inches of its stroke. The steam port will remain open to the action of the exhaust during the time the piston completes its stroke and moves through a portion of its return stroke. In the meantime the valve will move to the end *a* of the travel and return as indicated by arrow 4, and the moment that the valve again reaches the position shown in Fig. 29, the release of steam will be stopped. To find the corresponding position of the piston, draw through the edge c_2 of the valve, Fig. 29, a straight line c_2 *m* perpendicular to *A B*, intersecting the circumference *a b m* in point *m*. Through this point draw a straight line *m* k_2 perpendicular to *y x*, and intersecting the latter in the point k_2. Since the distance between the points *x* and k_2 is 22⅜ inches, it follows that the piston has moved through 22⅜ inches of its return stroke, by the time that the release of steam will cease. As the valve continues its travel in the direction of arrow 4, Fig. 29, the steam port will remain closed until the edge c_2 of the valve coincides with the outer edge *o* of the steam port, and during this time the steam which remained in the cylinder is compressed, but as soon as the edge c_2 of the valve passes beyond the steam port edge *o*, the admission of steam into the cylinder will commence. To find the corresponding position of the piston, draw through the outer edge *o* of the steam port, Fig. 27, a straight line *o g* perpendicular to *A B*, and intersecting the circumference *a b m* in the point *g*; through this point draw a line *g k* perpendicular to *y x*, intersecting the latter in the point *k*, and since the distance between the point *x* and *k* is 23⅞ inches, we conclude that the piston has moved through 23⅞ inches of its return stroke before the admission of steam will begin. Here we see that steam will be admitted into the cylinder before the return stroke of the piston is completed, and that is the object of lead, as has been stated before, Notice once more: the compression of steam will commence when the piston has traveled 22⅜ inches of its return stroke, and will cease when the piston has traveled 23⅞ inches of its return stroke, hence the steam is compressed during the time that the piston travels through 1½ inches.

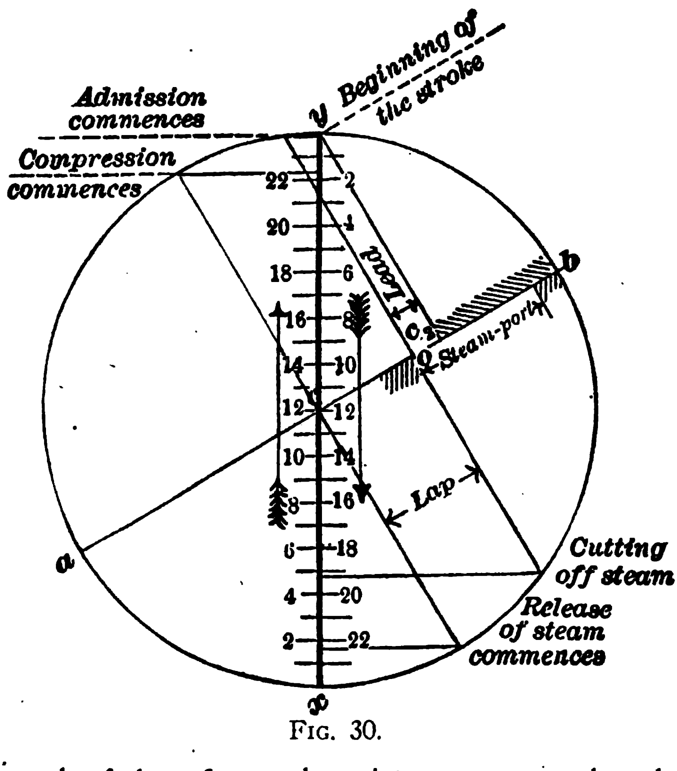


FIG. 30.

In each of these figures the point g represents the relative position of the center of eccentric to that of the valve. The point g will always be found in the circumference $a\ b\ m$ and in a straight line $c_2\ g$ drawn perpendicular to $A\ B$, the former passing through the outer edge c_2 of the valve.

The reason why the point g should in all cases be found in the straight line $c_2\ g$ drawn through the outside edge c_2 of the valve is this: the center c of the circle $a\ b\ m$ has been placed on the line $A\ B$ in such a position (as shown in these figures), that the distance between the center c and the outside edge o of the steam port is equal to the lap, therefore the center g of the eccentric and outer edge c_2 of the valve will always lie in the same straight line drawn perpendicularly to $A\ B$. If the distance between c and the outer edge o of the steam port is greater or less than the lap, then the center of the eccentric and outside edge of the valve will not lie in the same straight line drawn perpendicular to the line $A\ B$. Here, then, we can conceive the necessity of placing the center c of the circle $a\ b\ m$ in the position as shown in these figures. The correctness of these remarks must be evident to the reader if the explanations in the previous examples have been understood. Again, since we have assumed that the point g not only represents the center of the eccentric, but also the center of the crank-pin, it follows that, in order to determine how far the piston has moved from the beginning y of its stroke when the crank-pin is at g, we must draw a straight line through the point g perpendicular to $y\ x$, as has been done in these figures.

From these constructions we can obtain our answer to the example illustrated by Fig. 28, namely:

Steam will be cut off, or, in other words, suppression will commence when the piston has traveled 19⅛ inches from the beginning of its stroke, and steam will be admitted into the cylinder during the time that the piston travels through this distance. The steam will be released when the piston has traveled 22⅜ inches from the beginning of its stroke, consequently the steam will be worked expansively during the time the piston travels through 3¼ inches. The release of steam will continue until the compression commences, which will occur when the piston has traveled 22⅜ inches of its return stroke. The compression will cease, and the admission of steam commence, when the piston has traveled 23⅞ inches of its return stroke.

The same answer to our example could have been obtained with less labor by a construction as shown in Fig. 30, which is nothing else but a combination of the three preceding figures; the methods of finding the different points in Fig. 30 have not been changed, and therefore an explanation in connection with this figure is unnecessary.

THE ALLEN SLIDE VALVE.

The Allen ported valve, which is sometimes referred to as the Trick valve, is, in general design, similar to the plain D slide valve, with the exception that it has a supplementary port, or passageway, cast into it, which passes over and through the crown forming the exhaust arch which ends in the two ports in the valve face, as shown in Fig. 31.

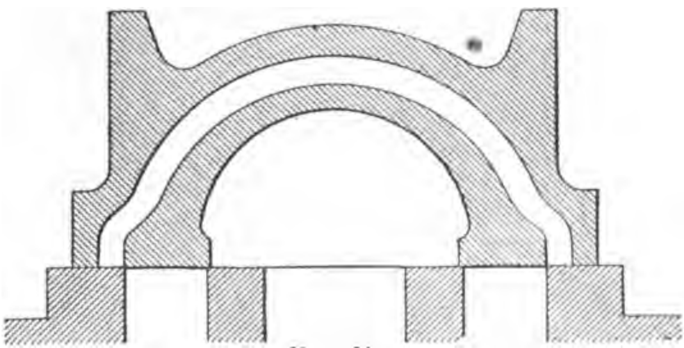


FIG. 31.

The Allen valve was designed to overcome the defects of the plain slide valve, which does not permit the use of full boiler or steam chest pressure at the beginning of a stroke, when it is most needed, without giving excessive lead, which would produce a premature cut-off and otherwise impair the operation of the valve. The Allen valve was also designed to give a larger area for the admission of steam, and for a portion of the valve travel the area of steam port opening to the cylinder is, in fact, doubled.

The valve and seat are so arranged that they give two openings to steam for the same amount of movement necessary for a plain slide valve to give one opening; that is, there is the same amount of opening for live steam with one-half the travel of a plain slide valve. As soon as the steam edge of the valve begins to open the steam port, the supplementary passage also begins to receive steam, thereby giving a double opening for the admission of steam. As the travel of the valve is always short, when an engine is running at high speed, the advantage of the double opening is evident, because it admits the steam at the beginning of the stroke, and maintains a full pressure on the piston till the point of cut-off.

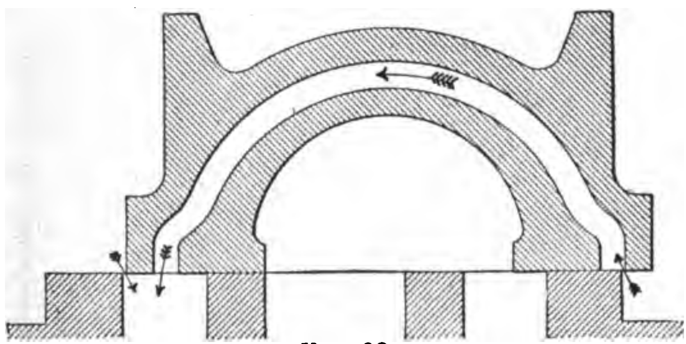


FIG. 32.

Fig. 31 shows the valve in central position upon the valve seat, and, the reader will observe, both steam ports are completely closed, the same as with the plain slide valve. Therefore the points of release or of compression will not be affected unless the lead is changed, when they will take place earlier or later in the stroke.

Fig. 32 shows the same valve moved off of its central position and the reader may note that the left steam port is receiving steam from each side of the valve at the same time. It may also be observed that the steam edge of the valve and edge of the supplementary port open simultaneously, and, therefore, must cut-off at the same time.

The chief advantage derived from the use of this valve is at high speed when the engine is worked at a short cut-off; the advantage secured results from the increased admission of steam into the cylinder, thus enabling the engine to develop increased power.

It has been claimed by many men in positions to know the facts that a saving of coal amounting to about seven per cent is obtained by the use of this valve, while others in just as good positions to know do not consider that this valve gains anything along economical lines.

One undesirable feature of the valve is the fact that when operated at short cut-offs there is double the port area for admission that there is for exhaust—because the auxiliary port is used only for admission.

While the value of this valve has always been more or less a debatable question, it has often been condemned where the evil results could more properly be attributed to its improper adjustment or application.

The best results can only be secured from the use of this valve by designing the ports and bridges so they will exceed the full travel of the valve by at least 1/8 inch. It may also be said that full advantage cannot be obtained if the valve is given as much lead as a plain slide valve.

The rule providing that lead should be reduced in proportion to the length of the port must be modified for the Allen valve, and the lead must be further reduced to about one-half that given to plain slide valves. It may also be noted here that the Allen valve should be given enough exhaust clearance to permit a proper exhaust period, due to the fact that at short cut-offs there is double the port area for admission that there is for the exhaust, and without exhaust clearance the compression may become excessive.

The original design of the Allen valve proved impracticable when used as a slide valve for locomotives, until it was balanced, for the excessive pressure bearing against its frail shell-like form caused a springing of the valve face and rapid wear of the valve and its seat.

We will have occasion to refer to this valve again when we reach the subject of Balanced Valves.

Power Required to Move a Valve.

To determine the power required to move a valve, multiply the area of the valve face by the steam pressure upon it, and then deduct one-third for back pressure from steam port and exhaust port. Friction between two smooth surfaces well lubricated varies from 1/10 to 1/14 of the pressure; the weight of the valve itself being so slight that it need not be considered.

As an illustration, let us refer to a valve 10 ins. by 20 ins. with 165 lbs. of pressure per sq. in. Then $10\times20=200\times165=33{,}000$ from which deduct one-third, and the result is 22,000 lbs. This amount, if divided by 10, will be 2,200 lbs., the amount of power in pounds which is required to move the valve. This strain on the valve gear will cause it to wear rapidly. This amount is decreased by the proportions of the reverse lever, in proportion to the distance between the lower connection of the reverse lever and the point at which the reach rod is connected to the lever, in comparison with the distance to the end of the lever. Yet the effort required of the engineer, in moving such a valve, is very great.

It is on this account that the plain slide valve, as distinguished from the balanced valve, has rapidly lost favor, and although a great many modifications of an improved type of slide valve have been invented, and tested, the plain slide valve has been almost entirely discarded.

BALANCED VALVES.

As we have said, the plain slide valve used in the early days of locomotive construction answered all purposes then required, but with the increase in size of the locomotive, together with the increased boiler pressure, it was almost impossible for an engineer to reverse an engine, and a great deal of power was wasted in moving a valve of this kind on account of the enormous pressure on the back of the valve.

Early History.

We have no doubt that the problem of removing the excessive pressure from the back of the valve challenged alike the skill of practical mechanics and the professional engineers of early days, and we presume many devices were constructed to accomplish the purpose, but the first patent issued by the United States Government covering a balanced valve was granted to Mr. Hiram Strait, of East Nassau, N. Y., on June 25, 1834, and the second patent was secured by Mr. John Kirkpatrick, of Baltimore, Md., on July 10, 1834, but neither invention proved to be of practical advantage.

The oldest form of balance valve, which was found to be practicable, was invented by Mr. George Richardson, of Springfield, Mass., and he secured a patent covering the same on October 28, 1862.

We are unable to say exactly when or where the Richardson balance valve was first used, but about 1870 the Boston & Albany Railroad Company tried it out on an engine and the results obtained were so satisfactory that the company applied the invention to the majority of their locomotives.

Countless other forms of balancing valves have made their appearance and have departed since, but we shall confine our discussion to the various types which have stood the test and which have been found to be practical, together with a few new inventions which may demonstrate their worth in the future.

Object of Balancing.

Almost all forms of balanced slide valves are constructed on the same principle, and are intended to prevent the steam chest pressure from exerting its force on the greater portion of the back, or upper surface, of the valve, thus enabling the valve to be easily moved, without excessive friction on the seat or excessive effort to lift it against the pressure which holds it down.

A slide valve may be considered balanced when the pressure on its top is just enough in excess of the pressure under it to insure the valve remaining on its seat ordinarily, but not sufficient to prevent the valve from relieving excessive pressure, or accumulation of water in the cylinder, by lifting from its seat.

In the design of the valves, the balance is equally as important as the efficient distribution of steam in the cylinders, for an improper, or defective, balance will impose enormous stress upon the valve gear and connecting rods, and greatly impair the work they are required to perform. In fact, it is necessary for the designer to consider, when calculating the dimensions of stems and rods, the increased work that would be imposed upon them in the event of an accident to the balancing strips and lubricating apparatus.

Area of Balance.

The size and shape of the protected area on the top of the valve varies in different forms of balancing, but it should equal the sum of the area of one steam port, the exhaust port, and two bridges, plus eight per cent of this sum for plain valves, or five per cent for Allen-ported valves. The balanced area should be measured from the outside of the strips, or the bottom outside edge of the inner ring when the outer ring is raised.

It may be said generally that good results may be obtained if about 65 per cent of the upper surface of the valve is protected, and 35 per cent exposed, to counteract the upward pressure of the steam against the face of the valve, which is not at all times uniform.

Methods of Balancing.

The original method of balancing a slide valve, which was first employed by Richardson, is generally followed, and the object is accomplished by fitting four iron strips ½x1½ inches, which are called balance, or packing strips, in suitable grooves cut in the top of the valve near its outer edges, thus forming a rectangular enclosure on top of the valve. The strips are supported on coil or semi-elliptic springs to hold them against the pressure plate which is attached to the cover of the steam chest.

A later form of balancing consists of a conical ring, cut through at one point, and fitted to a taper bearing on the top of the valve. No springs are necessary to support the ring since its reaction on the taper bearing, due to its elasticity and the steam pressure, tend to lift it against the pressure plate.

Hole in Top of Balanced Valve.

One or more small holes are drilled through the top of balanced valves to permit any small volume of steam which might get on top of the valve, as a result of defective balance strips, to pass into the exhaust and thence to the atmosphere, in this manner maintaining the balance feature of the valve. This hole in the top of the valve is generally referred to as the release port.

If an escape for the steam which leaked by the balance strips were not provided, such steam would accumulate on the top of the valve, and eliminate the benefits secured from properly balanced valves.

Advantages of Balancing.

The chief advantage of a balanced slide valve is the fact that it is easily moved, without excessive friction, and that it can relieve itself of excess pressure, or the accumulation of water in the cylinder, by lifting from its seat. As a result it is unnecessary to provide means for relieving excess pressure in the cylinders.

Disadvantages of Balanced Valves.

When cylinders of large dimensions are used, with high steam pressure, the slide valve becomes unduly large for a proper length of port opening and, even when the valve is well balanced, creates an excessive amount of friction when it is moved on its seat. A slide valve when used on a very large cylinder gives undue cylinder clearance, due to the increased length ot the ports, and the large steam chests necessary, and causes more or less steam condensation. This probably accounts for the high water rate of engines with very large valves and steam chests.

Another objection often advanced is that the end strips cut into the balance plate, and therefore, it is uncommon for a locomotive to run more than 25,000 miles before the valves require refacing. This difficulty may, however, be overcome by dispensing with springs under the front and back strips, by allowing the end strips to overlap and receive their support from long side strips. When such an arrangement is used shoulders should be provided on the bottom of the side strips, so that they cannot work out of their proper positions.

While slide valves have been operated successfully with a high degree of superheat, they are not as reliable as the piston valve under high temperature steam.

The Richardson Balanced Slide Valve.

The Richardson balanced valve, briefly referred to heretofore, is similar to the plan slide-valve, but it has a certain amount of space on the top of the valve enclosed by four rectangular pack-

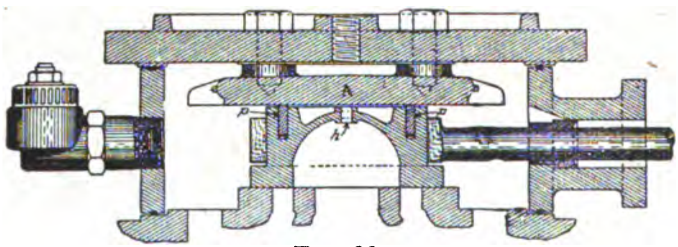


FIG. 33.

ing, or balance strips, ½x1½ inches, of cast iron, which are held tightly against the pressure plate by the steam pressure, and by semi-elliptic springs placed in the grooves beneath them. The two shorter packing strips have gib-shaped ends to retain them in their proper positions, and are arranged so that no packing strip of the balance device will travel over, or rub, the path of travel of any other strips, thereby insuring a steam-tight joint at various travels. The balance strips are marked *p p* on the illustration, Fig. 33.

There is also provided a pressure plate, A, which is bolted to the inside of the cover of the steam chest, but this may be cast in a single piece. The steam acts against the outside of the strips, making them steam tight, and, as a result, excludes steam from the top of the valve. A small amount of clearance between the valve and the pressure plate is allowed, so that excessive cylinder pressure may be relieved by allowing the valve to lift slightly from its seat. A small hole, *h,* is drilled through the top of the valve, so that any steam that escapes past the packing strips may pass out to the exhaust without exerting pressure on the top of the valve.

The proper area to be enclosed within the four balance strips may be determined by adding the area of the exhaust cavity to the area of the one steam port. If a greater area be enclosed there is danger of overbalancing the valve, and a smaller area will often give satisfactory results.

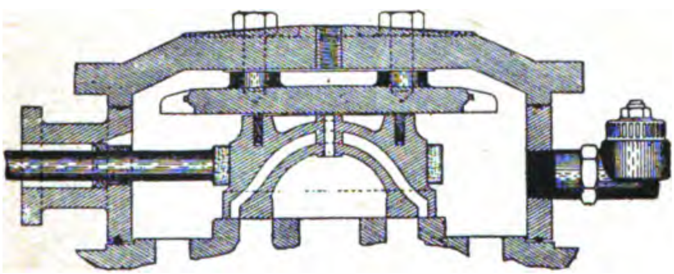


FIG. 34.

The reliability of this type of balanced valve, together with its simplicity of construction, and the fact that the balance feature can be used with various forms of slide-valves, have resulted in its general adoption, and it is now-used extensively.

The Allen-Richardson Balanced Valve.

This well known type of balanced valve is simply a combination of the Allen ported valve and the Richardson system of balancing. which is shown in the sectional view presented in Fig. 34.

We pointed out in the preceding pages, when analyzing the Allen valve, that it was not practical for locomotive work, until it was balanced, but when used in connection with the Richardson system of balancing it gives very good results, and thousands of this combination type are now in service. This valve is manufactured by H. G. Hammett, of Troy, N. Y.

American Balance Valve.

This form of balance valve is used quite extensively in locomotive service, and its popularity is due, no doubt, to its simplicity of construction, positive action, economy of maintenance, and large area of balance.

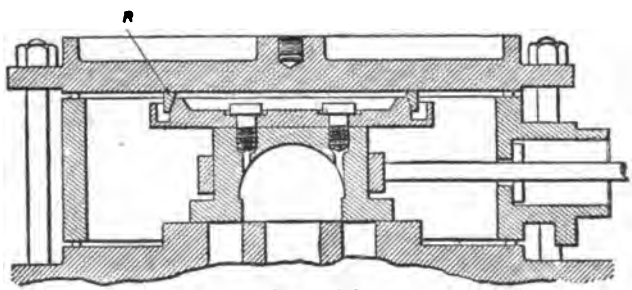


FIG. 35.

The valve is balanced by a single or double cone, or disk, cast on the back of the valve, the single cone being preferable where steam chest space will permit its use. These cones have a beveled face, are fitted on the outside by taper rings, marked R in the illustration, Fig. 35, of the same bevel as the cone rings, made ¼ inch smaller than the bevel of the cone, and are expanded over the cone so that it does not require a spring.

The conical ring is cut through at one point, in order that the ring may expand, and the opening is protected by an L-shaped plate of the same depth as the ring, which fits against the cone and insures a steam tight joint.

Owing to the natural elasticity of the ring and its expansion over the cone, a tension is placed on the ring, the function of which is to close the ring on the cone, which necessarily moves upward. The ring is therefore self-supporting and self-adjusting.

For the valve to lift from its seat it is necessary to force the cone up into the taper ring; and since the ring is held, by the steam chest pressure, from opening, the valve cannot lift without first overcoming the entire friction of the beveled face, besides opening the ring against the steam chest pressure. Of course, if the taper was made large enough, say 45 degrees, the action of the steam chest pressure on the circumference of the ring would wedge it between the cone and the steam chest cover and, as a result, exert an enormous pressure on the valve, so that it would not work satisfactorily. Extensive experiments, however, have been made, with the taper from 9 to 24 degrees, and, from actual tests, the proper degree of taper has been ascertained whereby the ring will rise under all conditions without crowding the balance plate more than is necessary to give good results.

The rings are made of hard, close-grained cast iron from standard gauges, which are fitted on the lathe in place of a caliper, or rule, and as they are made 1 inch deep, they can easily wear 3/8 inch and still adjust themselves. As the ring is turned 1/4 inch smaller than its working diameter, it can be readily seen that 1/32 inch in the diameter of the ring either way from the size specified would in no wise interfere with the service of the valve.

The rings are all bored smaller than the diameter at which they are to work; therefore, when a ring is set on its cone, it will stand higher than in its working position. The face of the pressure plate must not be closer than ⅛ inch to the top of the cone after the steam chest cover has been bolted down, for, in placing the steam chest cover in this position, the ring is expanded over the cone until its inside diameter at the bottom is the proper balancing diameter.

Sufficient allowance must be made for the removal of the valve-yoke, and, as a result, a detachable disk is used, but the makers recommend an arrangement whereby the valve yoke may be carried on the steam chest at the ends of the valve.

When the valve is in position, and the steam chest cover has been bolted down, there must be ⅛ inch space between the face of the upper bearing (in this case the pressure plate), and the top of the disk, or cone. This arrangement will allow the valve to lift from its seat ⅛ inch, which it will do as soon as steam is shut off while the engine is in motion, if it is not held down by the valve-yoke.

A casual inspection of the illustration may lead the reader to believe this valve is overbalanced, but the impression will disappear when proper consideration is made for the reaction from the surface of the bevel ring, and friction of the conical joint.

The valve is manufactured by The American Balance Valve Company of Jersey Shore, Pa.

The Miller Double Acting Slide-Valve.

A double acting slide-valve designed to embrace the advantages derived from inside clearance, and obviate the disadvantages resulting from excessive outside lap, by extending the period of exhaust and delaying compression until substantially the point of admission, without delaying admission or hastening cut-off, was patented January 13, 1914, by Mr. Joseph Miller, of Chicago, Ill.

The principal feature of the improvement is the employment of an auxiliary valve, sliding on and above the main slide valve, alternately opening and closing the two small auxiliary ports—one at each end—which extend through the main slide valve.

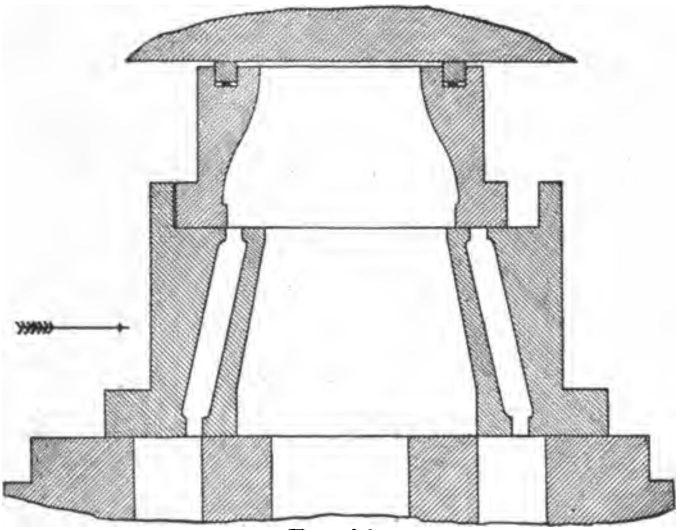


FIG. 36.

In Fig. 36 it may be observed the main valve is moving to the right, and the auxiliary valve is being moved in the same direction, by special operating mechanism, and the upwardly projecting portions, or buffers, of the main valve, against which the auxiliary valve abuts. It will also be seen by inspection that the center exhaust passageways extending vertically through the center of the main and auxiliary valve are always in communication.

The auxiliary exhaust passages, which extend entirely through the body of the main valve, are positioned to register with the cylinder ports in the valve seat after the inside edges of the main valve have passed inwardly beyond the ports, and delay compression by providing means of escape for the steam which would normally be trapped in the cylinder ahead of the advancing piston.

This is manifest by an inspection of the left-hand port shown in the illustration, for the point of exhaust has been reached in the right-hand port while exhaust continues in the left-hand port; in fact, the left-hand port will remain open for exhaust until ¼ inch of the port opening for steam admission.

At the same time it will be seen that premature release in the right-hand port is prevented by the auxiliary valve closing the upper end of the vertical auxiliary passageway extending through the main valve, so as to prevent release until the inner edge of the main valve has moved to a position to open the right-hand cylinder port.

The auxiliary valve travels back and forth in the steam chest with, and in the same direction as, the main valve, as the latter approaches the end of its stroke.

The two valves are balanced in the usual manner by the use of springs, or other resilient means, interposed beneath packing strips which fit in grooves and are forced against the pressure plate, to produce a seal between the interior of the valves and the live steam compartment.

The main valve is lubricated by means of steam, trapped in a series of transverse grooves, or steam pockets, cut in the valve face beneath the valve, as the latter slides thereover. The expansive force of the trapped steam in these pockets tends to raise the valve from its seat, thereby reducing friction and eliminating the necessity of artificial lubrication. In fact, it is claimed that this system of lubrication permits the valve to be used with success on superheated engines of very high temperature.

It has been demonstrated that one of these double acting valves having 1 inch steam lap, and ⅜ inch exhaust lap, will give 40% increase of expansion. With 1 inch lap, and exhaust line and line, however, a decrease of compression of about 75% over the plain slide valve, may be obtained.

The Wilson Balanced Slide-Valve.

This valve was designed for high-pressure, and, as it is of the "Grid Iron" type, it is very light, and, being the only moving part, the reciprocating weight is at the minimum.

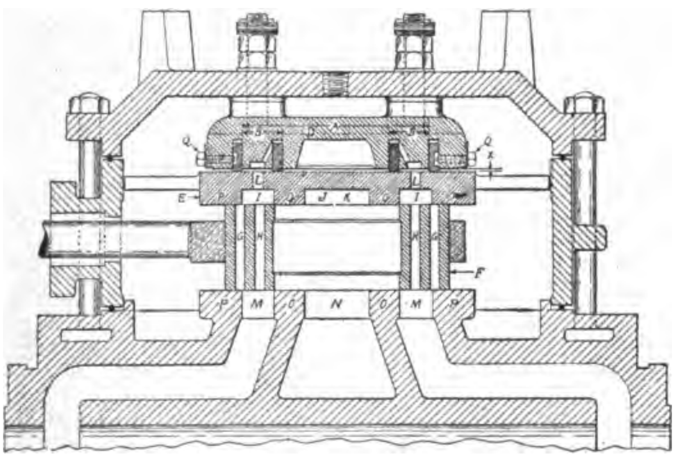


FIG. 37.

It will be observed that the valve has two faces which are parallel and of identical construction; that the lower surface of the balance plate is of the same area and shape as the raised valve seat, that the pockets I I are of the same area as the steam admission ports to the cylinder; that the "floating" balance plate can rock slightly and lift to relieve the cylinder when the engine is drifting, inasmuch as the plate is located a short distance from the under surface of the pressure plate; and that the packing strips are stationary.

The inner area of the top surface of the balance plate bounded by the end packing strips and outer strips, which is to be enclosed from the pressure of steam within the chest, is determined in any given case by experiment. However, the main requirement is that the area of the balance plate exterior of these pack-

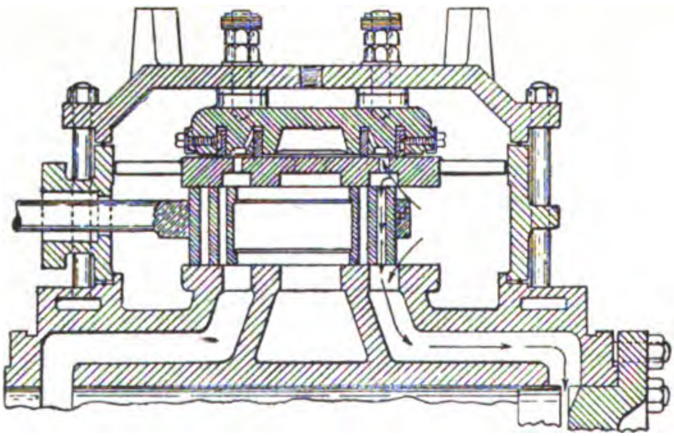


FIG. 38.

ing strips, which is subjected to steam pressure, should be sufficient to hold the valve in place and allow it to move from its central position when moderate force is applied to the valve stem.

When the valve is moved from mid-position, Fig. 37, to the opening position, Fig. 38, steam passes directly downward through the admission passage M to the cylinder. At the same time steam passes to the enclosed area B at the top of the balance plate. Steam pressure upon the top and bottom faces of the valve, where it laps the port, and the top and bottom faces of the balance plate at the edge adjacent the port, is thus equalized. In the wide open position, the balance is preserved, inasmuch as the top face of the valve at the left has passed out from under the balance plate the same distance its lower face has passed over the edge of the valve seat. When the valve covers the port, the pressure upon the top and bottom faces and upon the top and bottom surfaces of the balance plate is obviously constant. As the valve moves farther toward the right, and reaches the position of exhaust, steam passes directly from the port, and also by way of the exhaust passages to the exhaust cavity. At the same time steam in the inclosed space between the balance strips above the port passes out into the exhaust cavity, thus preserving the balance which otherwise would be destroyed by the top surface of the valve excluding from steam pressure the under surface of the balance plate adjacent to the pocket therein.

The balance area varies with each stroke, or movement of the valve, an amount sufficient to correspond to the changed position of the valve on its seat at the different points of its travel. It also provides a double exit, and admission, for the release and admission of steam.

This valve is manufactured by the American Balance Valve Co., of Jersey Shore, Pa.

Gould Balance Valve.

The Gould balance valve was designed to overcome the tendency of balance valves, when slightly worn, to suffer from steam leakage; and also with the idea in view of obtaining a valve that will take up its own wear, so that this difficulty is very largely overcome.

This valve is constructed like the plain D slide valve, with the same face measurements, and it is intended to work on the same ports and has the same admission, cut-off and release as the common D slide valve. Fig. 39 shows this valve with cover displaced and rings raised.

This valve is semi-circular or half round in shape, and the flat portion travels on the valve seat; while in operation it travels under a half round, or semi-circular, pressure plate which rests on the valve seat; it has expansion half rings, four in all, two at each end of the valve, and these rings expand, with steam, against the pressure plate, thus preventing steam from entering or escaping at each end of the valve.

FIG. 39.

Between these rings at each end is a groove of exactly the same length as the packing grooves and it extends around the oval portion of the valve; when the rings are expanded this groove becomes a steam chamber, as steam is taken from the ports in passing over them, through holes bored in the face of the valve to this chamber or circular groove. The semi-circular pressure plate keeps the steam from pressing downward upon the back of the valve proper; the valve is balanced against port pressure by steam taken from the ports through to the chamber previously described; the pressure plate is kept from moving endwise by pins in each end of the plate which rest against the walls of the steam chest, and is held down firmly by a set screw through the steam chest cover.

An X-ray view of the valve through its semi-circular casing, showing its interior construction, and the sleeve, or valve rod connection, is presented in Fig. 40.

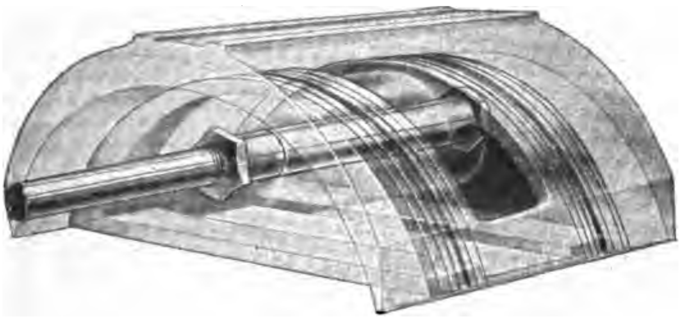


Fig. 40.

The advantage of the Gould valve is that there is no boiler pressure on the back of the valve, as the cover fully protects the valve from live steam, and the rings exclude the steam from passing beyond the ends of the valve, as there is no opening for steam to get between the cover and the valve. The ends of the valve are exposed to live steam, but, the pressure on both ends being equal, there is no end friction. The communicating ports always allow the same pressure on the back of the valve that is on the face of the valve, thus keeping it perfectly balanced at all times. This valve is manufactured by the Gould Balance Valve Company, of Kellogg, Iowa.

PISTON VALVES.

The first patent issued by the United States Government covering a piston valve was secured by Mr. M. P. Mack of Canandaigua, N. Y., in March, 1866, about 32 years after a patent was first issued for the balanced valve. We have pointed out, however, that the early balanced valves proved to be impractical, and it was not until the year 1862 that a balanced valve of merit was produced. We may say, therefore, practically speaking, that the piston valve is as old as the balanced type.

The piston valve, like the balanced valve, was not generally adopted for many years after its invention, but there were more serious obstacles in its path than confronted the early balanced valves. For example, the piston valve was obliged to overcome the popularity of the balanced valve which had been before the public for some years, and had answered all requirements of the low pressure engines then in use. The piston valve was criticised because its use resulted in the rapid wear of the valve cage at the port openings, which was partly due to the absence of bridge strips in the part openings, as ring bearers for the piston. Another objectionable feature was that the tallow used in early days for lubrication prevented the piston valve from giving good results.

In some of the earlier forms of piston valves the steam was admitted at the outer ends of the valve and, as a result, the action of the valve, in such cases, was precisely the same as that of the plain slide-valve. Valves of this type are manufactured and used at present, but are not so common as the inside admission type.

After some of the objectionable features of the piston type of valve had been eliminated, and various improvements had been made in its construction, the valve was, in time, adapted to marine engines, where it rendered good service. As a result, it has grown into popular favor in locomotive construction, and is now used very extensively.

The piston valve is of cylindrical form, and derives its name from the fact that it consists of two pistons, one to each port, which are connected by an ordinary spindle. The valve works in a cylindrical valve seat, in which port openings are made completely around the valve.

This arrangement allows greater port area, and requires less space for the operation of the valve, for this type of valve is generally made long enough to bring the two faces, or working edges, of the valve close to the ends of the cylinder, thereby reducing the clearance space between the valve face and the piston.

This type of valve has been well described, by numerous authors, as an ordinary slide valve with its plane surface and rectangular steam ports rolled into cylindrical form, and we shall adopt this definition as satisfactory.

General Design.

As a general rule the piston valve is made hollow, with as large a passage through the center as possible, to secure a large area for the exhaust at the instant release occurs, so that the back pressure will be reduced to a minimum. The exhaust produces more or less pressure on the end of the valve rod and, as a result, the follower ribs of the valve should have a long bevel.

The idea that the effect of a contracted exhaust nozzle is more apparent, and detrimental, with the piston type of valve than with a slide valve, is erroneous, and may be explained by directing attention to the fact that back pressure from the exhaust acts on the outer ends of the piston valve, and, as a result of short ports, it exerts more force on the pistons.

The Built-Up Type.

The most common type of built-up piston valve is constructed of two follower plates made of cast steel or malleable iron, two cast iron skeleton or bull rings, and a single connecting piece called a spool.

The Solid Type.

As a general rule the solid type of piston valve is now made hollow, and is supplied with three light section snap rings—about 3/8 by 1/2-inch—on each end. From an economical standpoint the solid type is preferable, for it can be manufactured and maintained at lower cost, but in case of an accident the entire valve must be replaced, while a renewal of the damaged part is all that is necessary with the built-up type of valve.

Size of the Piston Valve.

There is a great difference of opinion regarding the proper size of the piston valve that will give the best results for large high pressure engines, but the prevailing opinion at present appears to be that the circumference of the valve minus the space occupied by the bridges, must, to a considerable extent, exceed the length of the slide-valve port for a similar bore of cylinder. This is accounted for by the resistance offered by the bridges, and by the fact that the portion of the port diameter opposite the cylinder cannot be so effective as that nearest it. The port ordinarily should be so proportioned that the area around the outside of the bushing at any point is equal to the combined area of all openings above this point. Formerly the use of a piston valve 55 per cent. of the diameter of the cylinder was considered good practice, but of late there has been a tendency toward a smaller diameter of valve, and it has been demonstrated that a 12-inch diameter is large enough for cylinders up to 27 inches in diameter, and, even, 8-inch valves for 24-inch pistons.

The size of the valve may be changed by increasing or decreasing the thickness of the valve bushing, and it is evident that a reduction in the size of the valve will reduce its weight and its stress on the valve motion, especially at high speed.

Without doubt, it is possible to use a smaller diameter of valve with superheated steam than with saturated steam, and this is probably due to the fact that the velocity of superheated steam through the ports is much greater.

The only point of similarity between the piston and slide-valve is that they are both employed to control the admission and release of steam. In all other respects they are different, but we shall here confine our examination to the difference in operation between the two.

Operation of the Piston-Valve.

The piston valve is generally of the inside admission type, with outside exhaust; that is, the inner edge of the piston valve is the steam edge, while the outer edge controls the exhaust. As a result of such arrangement, all of the events of the stroke are just the reverse of the slide valve events. This is, as previously explained, because the slide valve has outside admission, and inside exhaust, and may be readily understood from the following.

When the piston valve moves in the direction of the arrow A, Fig. 41, it will uncover the right-hand port, while a similar movement of a slide valve would mark the commencement of exhaust, at the same port, but, with the piston valve steam is admitted from the inside of the valve at the point indicated by the letter T. Admission begins here and will continue until the

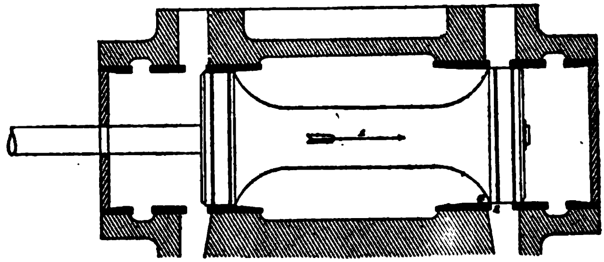


FIG. 41.

valve reaches the end of its travel to the right, and returns, in the opposite direction, to cut-off—the position it occupies in the illustration—where the admission of steam to the cylinder ceases, and cut-off takes place at this point. With the slide valve compression would now begin.

Now directing our attention to Fig. 42, we will assume the valve continues its travel in the direction of the arrow until the outer edge of the valve coincides with the outer edge of the

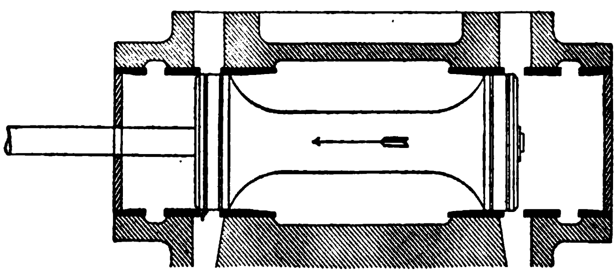


FIG. 42.

steam port indicated by the letter V. It must be apparent that the steam which has not been exhausted from the left-hand port will be trapped in the cylinder, and must be compressed by the advancing piston which is also moving in the direction of the arrow. This point, compression, corresponds to the point of cut-off with the slide valve.

As the valve continues its movement to the left, the exhaust of steam will take place, in the right hand port, whereas, the movement of a slide valve in the same direction would allow the admission of steam in the right-hand port.

From this brief explanation of the difference in operation between the movements of a piston and a slide valve, the effects of over-travel, lap, lead, etc., with the piston valve, must be apparent.

Robinson's Piston Valve.

The illustrations we present herewith are of the Robinson piston valve, which was invented by John G. Robinson, of Man-

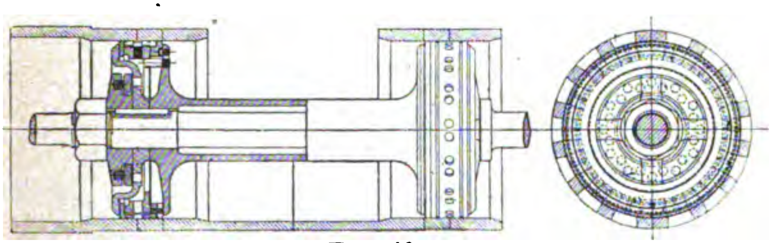


FIG. 43.

chester, England, in 1915. Fig. 43 is an inside admission valve, while Fig. 44 shows the outside admission type.

The illustrations show the valve fitted with the inventor's pressure release rings, the functions of which are:

To prevent sudden reversal of stresses in the motion when the engine is running without steam.

To release any undue pressure from the cylinders, from whatever cause arising, when that pressure increases above the pressure in the steam chest.

To provide for the circulation of air from one side of the pistons to the other and thereby prevent the influx of gases from the smoke-box through the blast pipe into the cylinders when coasting.

These valves are used quite extensively in England and her colonies, and are manufactured by the Superheater Corporation of London, England.

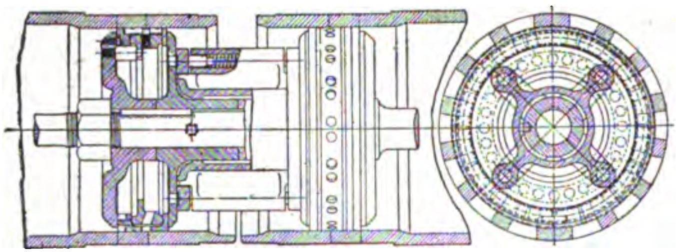


FIG. 44.

Double-Ported Piston Valve.

This is another form of piston valve which is beginning to come into use to the extent that a description of it will be advantageous. The double-ported piston valve is really a piston valve fitted with Allen ports, just as in the case of the Allen double-ported slide valve. However, it embodies added complications, in that it requires a special arrangement of the cylinder ports, as may be seen by referring to Fig. 45.

Three sets of packing rings are employed on each end of the valve, or six sets in all, and the steam chest incorporates double ports at each end, leading to the cylinder. The main packing rings, which open and close the ports as they pass over them, are designated A in the illustration. The sets B are employed to separate the steam from the exhaust cavity of the valve.

Steam is always present in the space C, as the illustration shows an inside admission valve. From C, the steam is free to pass to the space D, through the passages E in the valve.

By means of the hollow central portion of the valve, G, the spaces F are always open to the exhaust, outside the ends of the valve.

When the valve has been moved from its central position to the right, as shown, the right hand port H admits steam from the steam space C of the valve, into the cylinder, to act upon the head of the piston, moving it to the left. Simultaneously,

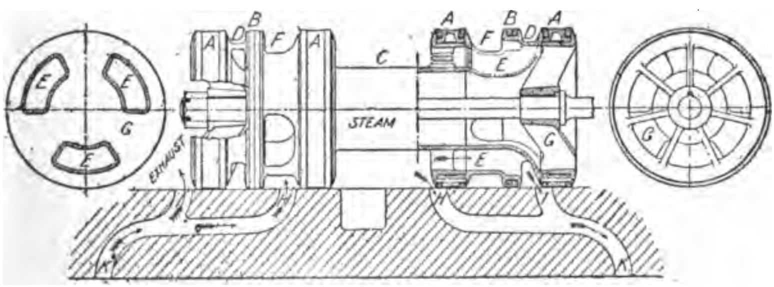


FIG. 45.

the right port I receives steam from the steam space, through the passages E and the auxiliary space D. This steam enters the cylinder at K, along with the steam admitted at H.

At the same time, and at the opposite (left) end of the cylinder, the valve has uncovered the ports I and H, and the steam is escaping, as the arrows indicate, directly from the port I, and indirectly from H, through the hollow portions of the valve, to the exhaust space at the end of the valve. In this manner, a double width of port opening, both for admission and release, is obtained.

PISTON VALVE CYLINDERS.

Piston valves may be located in almost any position with regard to the cylinders, and may be made of any length desired. As a result, the steam ports of piston valve cylinders should be made short and direct, and the steam chest placed close to the cylinder bore, leaving only sufficient room for the barrel flange to be turned for the head casting. Piston valve cylinders weigh less than the first class slide-valve cylinders, and are are more suitable to curved lines than the type used with slide-valves.

Universal Valve Chest.

An ingenious arrangement for the application of piston valves to cylinders primarily designed for slide valves, was invented July 14, 1914, by Mr. John E. Muhfeld, of Scarsdale, N. Y., and Mr. Hal R. Stafford, of Plainfield, N. J.

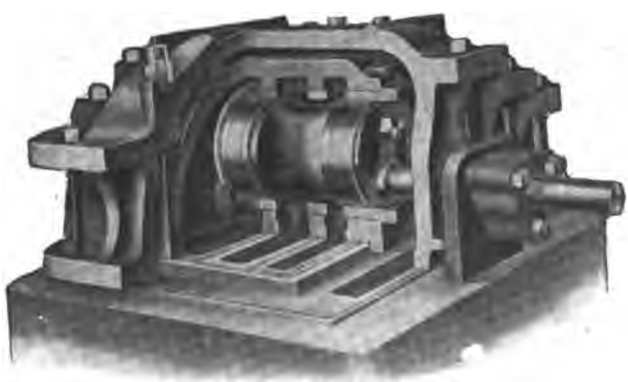


FIG. 46.

We understand that the device has been successfully adapted to every design of slide valve cylinder in service. In fact, it can be designed for every combination, for the application of the device permits the use of superheated steam with the old cylinder, at almost the same cost as cylinders originally designed for piston valves, with the same effect and power, for either inside or outside admission valves, and may be used with any type of valve gear in use at present.

There are two distinct types of this valve chest, viz.:

In type A, Fig. 46, a separate inner valve chamber is jointed to the valve seat, independent of the outer chest, which is seated on the cylinder apron in the usual way. Thus the regular inlet ports to the steam chest may be utilized. This type is also designed for use with either inside or outside steam pipes.

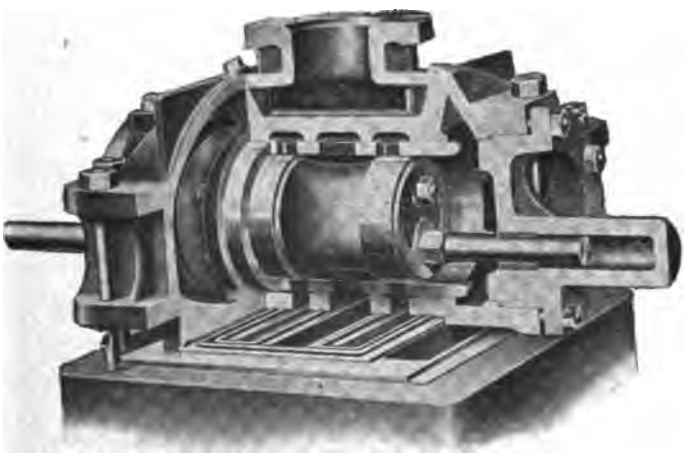
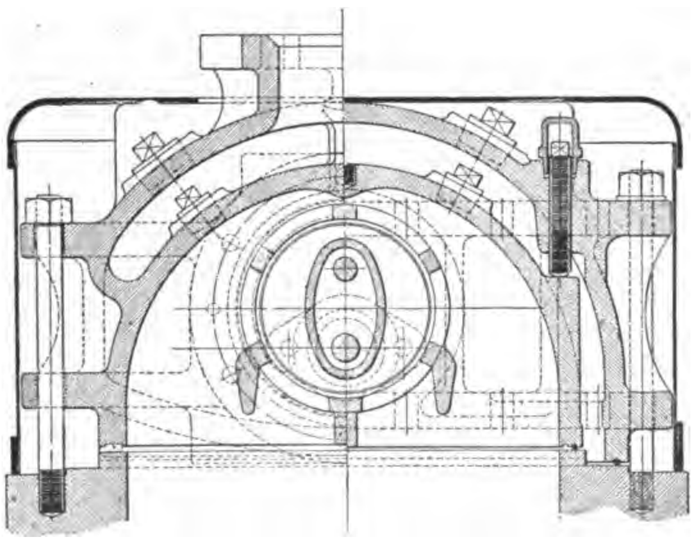


FIG. 47.

On the other hand, type B, Fig. 47, is designed for use where new outside steam pipes are to be applied. This type is made in one piece, is joined to valve seat only, and by a system of joint wires set in grooves, around the chest and each separate port, is rendered steam tight.

An end view, in cross-section, of both valves is shown in Fig. 48. On the left is shown the type B, Fig. 47, which calls for the application of new outside steam pipes. On the right, type A, Fig. 46, designed for use with either inside or outside steam pipes, is represented. This combination of the two figures clearly illustrates the difference in construction of the valve chest to be used with either type.



TYPE B **TYPE A**

FIG. 48.

A special feature claimed for the invention is that it permits the use of a valve of minimum diameter, because the steam ports are of such fo— that the entire circumference of the valve is effective.

The advantages gained by the use of the small valves will be readily appreciated by practical mechanics, for the decrease in size reduces friction and stresses proportionally, and results in a better steam distribution. The Franklin Railway Supply Co., of New York City, are the manufacturers of this device.

Streamline Cylinder Port.

A new method, or arrangement, of restricting the size of the valve for ordinary piston valve cylinders, in accordance with the established principles of the flow of gases, without sacrificing the amount of work generally secured by the use of a larger valve, is described and illustrated herewith.

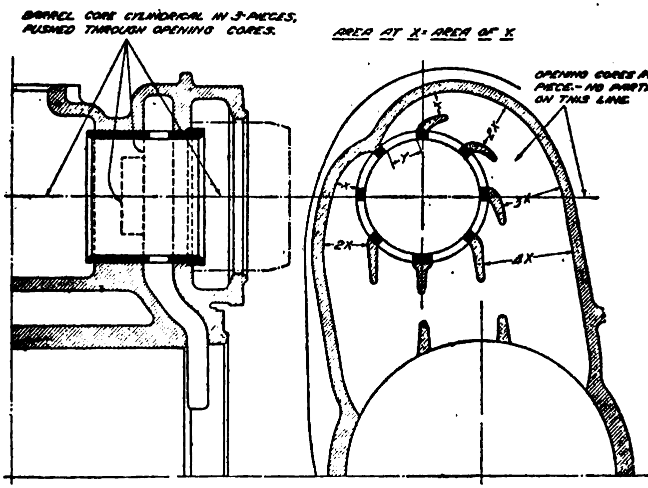


FIG. 49.

With the use of this device it is unnecessary to make the circumference of a piston any greater than the length of a slide valve port, for the beneficial effect is obtained by properly directing the flow of steam to and from the valve, so that every square inch of the bushing port will become effective. This permits the use of a valve only half the size commonly used, and, at the same time, facilitates the flow of steam to and from the cylinder to such an extent that the engine is much smarter and faster.

Fig. 49 shows the application of these principles to new power, where the design of the cylinders is under control. The letter

Y equals the area of each bushing port, while X equals Y in area. Thus it will be seen that the steam port around the bushing is so arranged that, during the admission period, as steam issues from the valve, it is divided into a number of streams, depending upon the necessary number of bridges in the bushing. These streams are directed by the ribs, so that they do not interfere with each other, but join into one smooth flowing whole when the main passage is reached. During the exhaust stroke the stream, as it flows from the cylinder to the valve, is divided by the ribs into a number of equal streams, one to each port in the bushing.

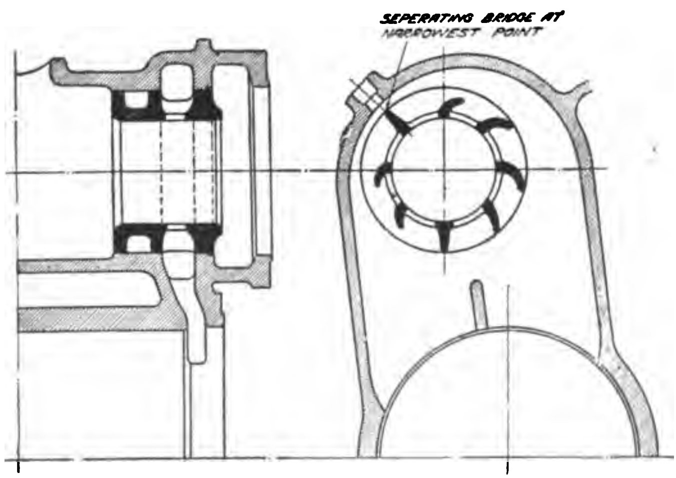


FIG. 50.

This absence of all quick turns, or other obstructions, speeds up the flow of steam, and, with an eight or ten inch valve, gives a higher initial pressure and a better admission line than can be obtained with the ordinary design of port with a fourteen or sixteen inch valve.

Fig. 50 shows the method of applying small valves to existing engines on which valves of much larger diameter were originally used. In this case a new bushing of special design, in which the directing ribs are cast as a part of the bushing bridges, is applied in place of the original bushing. This arrangement makes it possible to secure the advantages of the smaller and lighter valve, and gives a steam distribution at least equal, and in most cases superior, to that of the larger valve.

This device is licensed for use by the Franklin Railway Supply Co., of New York City, N. Y.

VALVE GEARS.

With the advent of the locomotive it became necessary to devise a means of reversing the engine, and numerous devices for that purpose, which are known as "valve gears," have been invented.

The first, or original, device employed to reverse a locomotive consisted of a single eccentric which could be turned by hand. When the steam was shut off, and the engine brought to a stop, the engineer could go forward and loosen a nut, turn the eccentric around to the position desired, screw the nut tight to clamp the eccentric in its new position, and proceed in the opposite direction.

This method of reversing was necessarily slow and unsatisfactory, so it was not long until someone conceived the idea of using two eccentrics, instead of one, to reverse the engine. This device consisted of two hooks, which could be raised or lowered and be brought in contact with the valve rod, and it was called "gab-hooks." Fig. 51 The arrangement was crude, but it enabled the engineer to change the motion of the engine without leaving the cab, and was considered a great improvement at the time.

Each eccentric-rod of this device has a notched hook in its forward end, and the rear end of the valve rod carried a pin with which the notched hooks could be brought into contact to actuate the valve. When the upper hook was lowered and connected with the valve rod pin the forward eccentric operated the valve, and vice versa.

Of course the device could only be used on locomotives operated at slow speed, and it only deserves notice here because it introduced the principle, employed by many inventors since, of using two eccentrics.

The next forward step toward the improvement of valve gears was the introduction of the curved, or straight, link, which is now used in connection with practically all reversing gears.

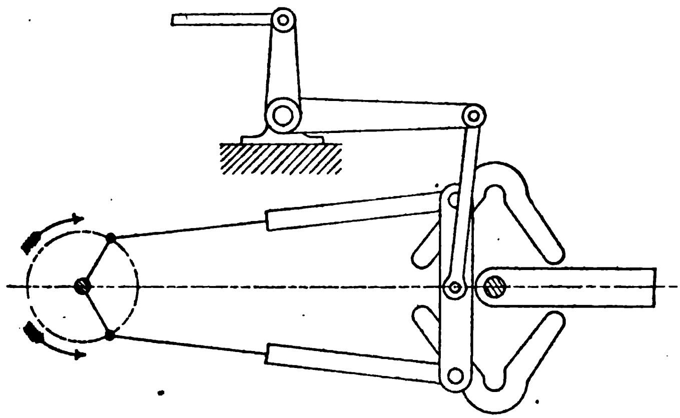


FIG. 51.

This was, indeed, a great improvement, not only in providing for the reversal of the locomotive, but because it provided means of allowing a variation in the arrangement of the valve mechanism, thereby permitting a more economical steam distribution in the cylinder.

As the Stephenson valve gear is one of the most worthy exponents of the shifting link type, and one of the oldest valve gears in use in America, we believe it would be appropriate to first discuss and describe its principal features.

LINK MOTION.

It was generally believed, for many years, that William Howe, of New Castle, England, invented and first applied the shifting link to a locomotive, to improve the method of reversing an engine. Howe was the head pattern-maker of the Robert Stephenson locomotive works, and it was claimed that he made a sketch of the link, which he explained to his employers, who were, in turn, favorably impressed with the idea, and allowed him to give the link a trial on a locomotive constructed for the Midland Railway Company, about 1842. Although Stephenson gave Howe the means of applying the device, Howe failed to perceive its value, and did not take out a patent, but sold the device to Stephenson for twenty guineas, or about one hundred and five dollars. Stephenson never claimed credit for the invention, but he secured a patent in his own name and the device has since been known as the Stephenson link motion.

Of all the link motions invented the Stephenson has been, no doubt, the most largely used on locomotives.

It always did seem odd to the writer that a man competent to invent such an important device could not appreciate its value, and protect his rights with a patent, but convincing evidence has been discovered of late which shows that Howe did not invent the link and had no right to sell the idea.

We are indebted to Mr. Fred H. Colvin,* a most profound student of the locomotive, for directing our attention to an able article by Mr. W. L. Campbell, in the *American Machinist,* of February 11, 1904, from which it appears: That Mr. Ralph L. Whyte, then eighty-four years old, was chief draftsman in the Stephenson locomotive works at the time the link was first suggested, and that the idea was brought to his attention by Williams, the youngest man in his department. He claims the original sketch connected the link directly to the eccentric, but Williams then said that it would be better to use rods between them. Following the usual course of that time, Whyte sent

*Editor of the *American Machinist.*

Williams to the pattern shop to have a model made, and was surprised, a few days later, to hear that Howe claimed the idea as his own. Whyte emphatically denied Howe's claim and supposed that Williams' rights would be protected, but he left for France soon afterwards, and severed his connection with locomotive affairs. Whyte later moved to Canada and it was only of recent years that he learned the credit due to young Williams all these years had been wrongfully accorded to Howe.

It is, indeed, strange that this voice from the past has remained silent so many years but, as the testimony does not appear to be prompted by base motives, and seems to be convincing, tardy recognition should now be accorded to Williams—as the real inventor of the link motion.

However, we are satisfied with a statement of the facts, and, to avoid confusion, we shall hereafter refer to the valve gear by its old name—the Stephenson valve gear.

THE STEPHENSON VALVE GEAR.

The general form in which the various parts of the Stephenson valve gear are arranged is clearly shown by Fig. 52, and the various parts are plainly distinguishable, so that the reader may become familiar with them and with their relative locations.

This valve gear is actuated by two eccentrics, which are generally keyed to the main shaft, or driving axle. The forward motion eccentric is shown at the top, and the backward motion eccentric at the bottom, of our illustration.

The eccentrics are supplied with straps, to which straps are attached eccentric blades, and the latter are bolted, or connected, near their opposite ends to a radial movable link; the forward motion eccentric blade being attached near the top of the link, and the backward motion eccentric blade near the bottom of the link.

The center of the link is spanned by a plate, called the saddle, on which is formed the pin or stud, attached to the lower end of the link hanger, which supports the link and eccentric rods. The top end of the link hanger being attached to the lower, or short, end of the tumbling shaft.

The motion imparted to the eccentrics is, in turn, transferred to the link by the eccentric straps and eccentric blades. The link is slidably connected to the link block which is attached to the back end of a transmission bar, and the latter is suspended by two hangers; in some cases the forward end of the transmission bar is supported by the lower end of the rocker arm. The top, or upper, rocker arm is attached to the valve stem, or rod, which controls the movement of the valve. On some engines the transmission bar is not used as a connection between the link

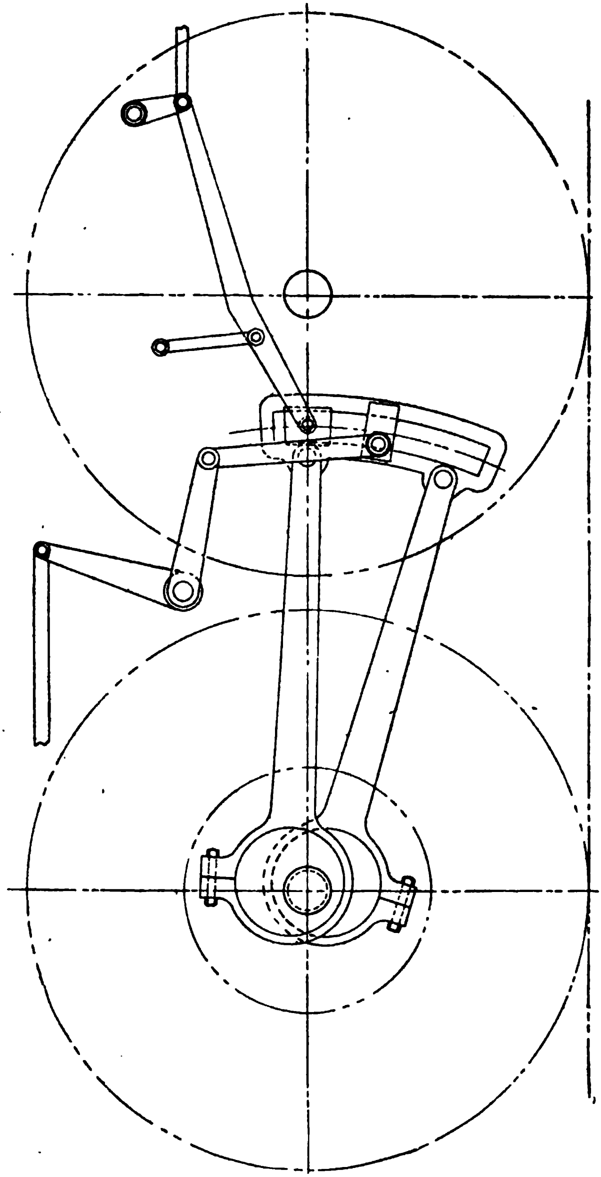


Fig. 52.

block and the rocker arm, but the link block is connected directly to the lower end of the rocker arm. The arms of the tumbling shaft are rigidly attached to each other, and they oscillate freely on a rigid tumbling shaft by which they are supported.

Referring again to Fig. 52, it will be observed that the link is shown in full forward gear, thus throwing the entire influence of the forward eccentric upon the valve motion, almost to the complete exclusion of the backward motion eccentric.

Now, if the reverse lever (not shown in the illustration) which is attached to the reversing rod, be moved half-way toward the center of the quadrant, the link, through the reversing rod, tumbling shaft arms, and link hanger, will be raised until the link saddle is near the link block. In this position the link saddle, receiving its motion from both of the eccentrics, will have less horizontal swing, and the valve travel and power of the engine will be reduced.

If, however, the reverse lever be moved to its full backward gear position, the lower, or backward, eccentric rod will be raised until it is in line with the link block. This movement will also cause the rocker to turn on the rock shaft, and the top arm of the rocker arm will move the valve, through the valve stem, to the right a sufficient distance to open the back steam port and reverse the motion of the engine.

Now that the reader has a partial knowledge of the operation of the shifting link motion, and the principles of its action, we shall proceed to a more thorough study of the construction of its various parts, together with an explanation of their influence upon the motion, and show how to lay out a link motion.

DETAILED CONSTRUCTION.

The Eccentric.

Literally, the word eccentric means, "out from the center." While there is not much similarity in the appearance of an eccentric and a crank arm, they are used for the same purpose, and give exactly the same results. It may also be said that there is in reality, nothing mysterious about an eccentric, for it is essentially a crank, whose length is equal to the radius of the throw of the eccentric with the pin increased to include the shaft.

A crank arm can only be used at the end of a shaft, while an eccentric may be attached at any point desired along the shaft, without reducing (by cutting or dividing) the strength of the shaft, and it is the latter feature which often decides in a selection between the two.

To make the subject more clear we present two views of the eccentric, and one of a common crank, on page 118. It will be observed that Fig. 54 shows the eccentric in full, while Fig. 53 shows it in section, and Fig. 55 shows a common crank.

In Fig. 54, the eccentric proper, *A A*, which rotates with the shaft and corresponds to the crank arm, is called the eccentric sheave. The eccentric sheave is surrounded by a metal strap, or band, commonly called the eccentric strap, *B B*, inside of which the eccentric sheave is free to turn. Both the sheave and the strap are made in two halves, so that they can be readily put on or taken off; the two pieces of the sheave are parted on a line passing through the center of the shaft at right angles to the horizontal center line of the eccentric, and are bolted together and rigidly keyed or fastened by studs to the shaft. The strap is grooved so as to prevent it from becoming displaced or getting off the sheave, and it is bolted together at its top and bottom.

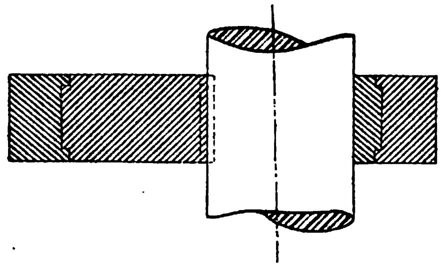


FIG. 53.

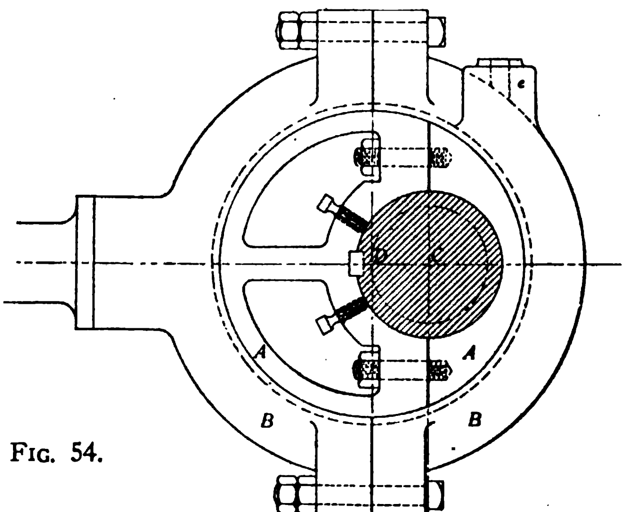


FIG. 54.

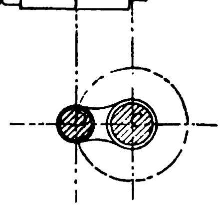


FIG. 55.

An oil cup, *e*, Fig. 54, is usually cast solid on one-half of the strap, for particular care must be given to the lubrication, the friction of eccentrics being much greater than that of cranks because they have larger sliding surfaces, between the sheave and the strap. It may also be noted that an eccentric requires more metal for its construction and space for its operation than a crank; it also absorbs more power than the crank, as a result of the greater leverage at which friction acts, and it is used in preference only where the throw is comparatively short.

A crank is used for converting the rectilinear motion of the piston into rotary or circular motion, while an eccentric is usually employed for converting the rotary or circular motion of the shaft back into the straight motion of the valve.

When the valve receives its motion from an eccentric, the periods of steam distribution are necessarily controlled by the position of the eccentric, and they occur earlier or later in the stroke as the eccentric is turned forward or backward on the shaft.

Now, referring to Fig. 54, we find that the point C is the center of the shaft, and the point D is the center of the eccentric sheave. As the sheave revolves with the shaft the point D will describe the small dotted circle on the shaft, whose center is the center of the shaft. As a result when half of a revolution is completed, the eccentric strap, *B B*, will be moved horizontally a distance equal to the diameter of the dotted circle, which equals the *throw* of the eccentric, or the travel of the valve. The distance between the center of the shaft C and the center of the sheave D equals the length of the crank arm, and is termed the *radius* or *eccentricity* of the eccentric, which is one-half of the *throw*. All movements of the eccentric are transmitted to the valve by means of an eccentric rod and valve stem, which connect the two.

In practice, the travel of a locomotive valve does not always equal the throw of the eccentric; the difference may be attributed to the influence of the link, and, in many cases, to the unequal length of the rocker arms.

With this brief description of the eccentric, and explanation of its use, we shall proceed to develop its relation to, and connection with, the movement of the valve.

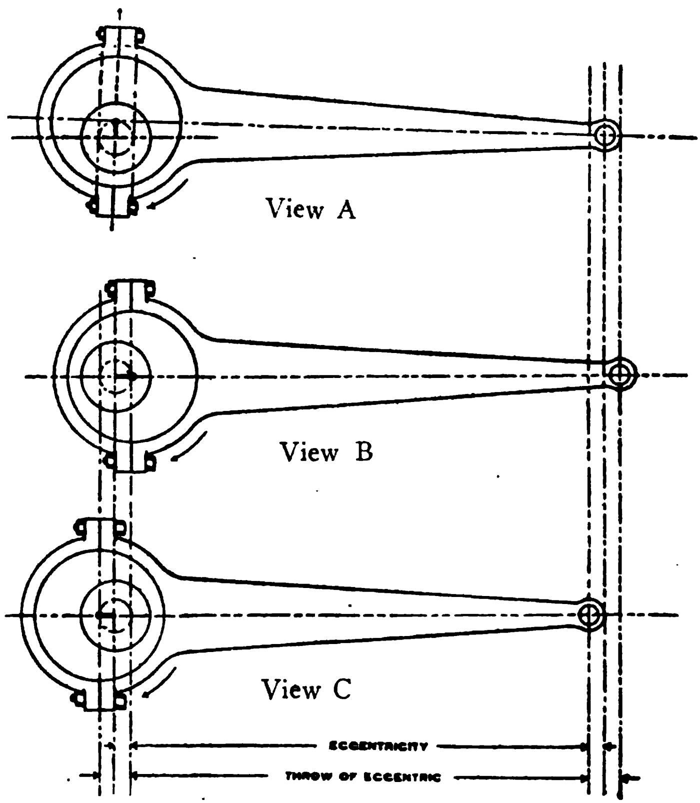


FIG. 56.

Eccentric Rod, or Blade.

The longitudinal motion of the eccentric is conveyed to the link, and valve gear, by means of rods, or blades. The rear end of the eccentric rod is generally fitted into the eccentric strap and made secure with bolts, while the forward end is forked and fastened to the link with a tapered pin.

The method of imparting the eccentricity of the eccentric through the eccentric rods is diagrammatically shown in Fig. 56.

Referring to view C, it will be seen that the center of the eccentric is to the left of the center of the axle. When this eccentric moves one-quarter, in the direction of the arrow, view A, the center of the eccentric will be at the top quarter and directly above the center of the axle, while another quarter movement in the same direction will bring the center of the eccentric to the right of the center of the axle, as shown in view B. It may also be observed that the quarter movement from view C to view A was sufficient to advance the forward end of the eccentric an amount equal to the eccentricity of the eccentric, while the two movements moved it an amount equal to the throw of the eccentric, or one-half the travel of the valve.

The Shifting Link.

A "shifting" link is moved up or down, through the medium of the reverse lever, and its intermediate connections, to reverse the motion of the engine, or secure a variable cut-off.

There are numerous forms of links in use, with considerable variations in detail, but the type shown in Fig. 57 is used to a considerable extent.

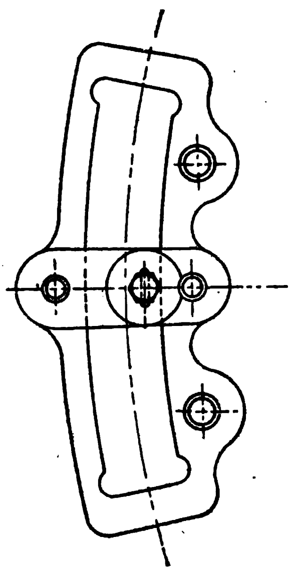


FIG. 57.

As previously stated, the forward motion eccentric rod is connected to the top of the link, with a tapered pin, and the backward motion eccentric rod is attached to the bottom of the link in the same manner; so the link block works in the top, or upper half of the link slot when the engine is operated in the forward motion, and in the bottom, or lower, half of the link slot when the engine is running backward.

Transmission Bar.

A transmission bar, Fig. 58, is simply a bar of steel or iron, supported by a hanger, with the forward end attached to the rocker arm, and the rear end fastened to the link block.

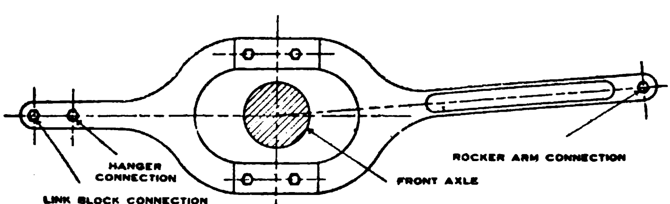


FIG. 58.

It is used as a connection between the link block and the rocker arm to transmit the motion of the eccentric to the valve, when the rocker is placed a considerable distance ahead of the link.

Another type of transmission bar, used in connection with the Stephenson gear, is illustrated in Fig. 52.

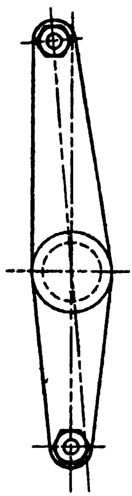


FIG. 59.

The Rocker.

The rocker, Fig. 59, is a lever composed of two crank arms oscillating on a rock-shaft, and its functions have already been described, page 56.

The rocker arms are rigidly set, but they may be of different lengths, and are generally bent, or set at an angle with each other, to partly neutralize the error introduced by the angularity of the main rod.

ALLEN LINK MOTION.

The Allen link-motion, Fig. 60, is a form of valve gear in which the link is raised, as in Stephenson's, and the valve rod lowered as in Gooch's, to reverse and produce cut-off; but, since the link and slider-block are both moved, each need only be moved half as far as in either of the other forms. The link is straight, and not a part of the circular arc, and is more easily manufactured.

The discovery of this motion was a natural sequence to the invention of the shifting and stationary links. By it a com-

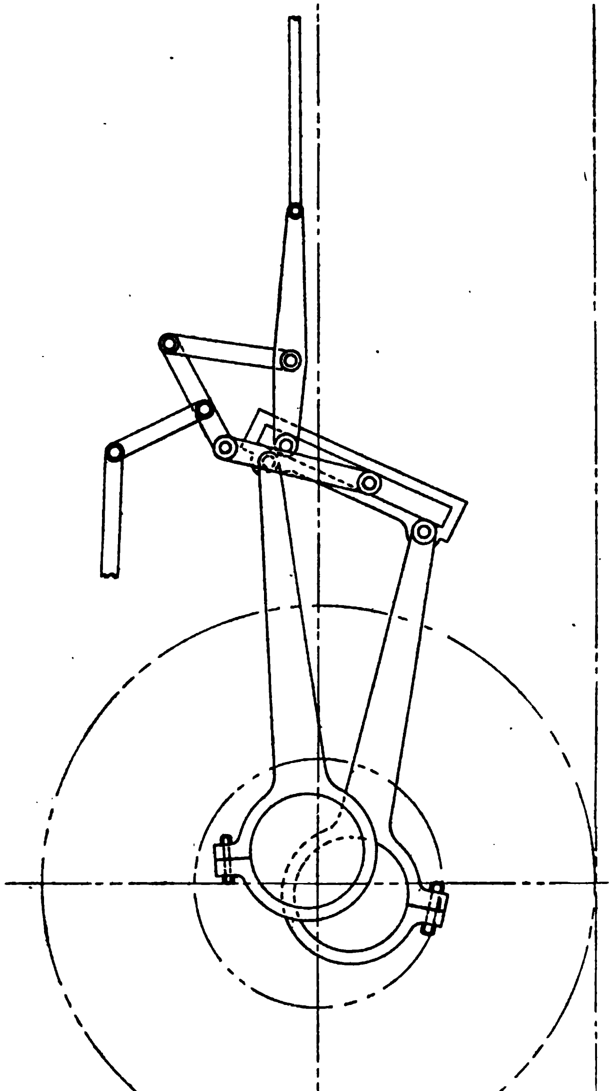


Fig. 60.

promise has been effected between the leading features of both motions, resulting in a more *direct* action and perfect balance of the parts with a reduced slip of the link block.

The location of the point of suspension, and attachments of the eccentric rod pins, upon, or back of, the link arc, are quite as variable for this, as for the shifting link motion; and the requirements of the other details generally indicate whether the reversing shaft should be placed above or below the central line of motion.

In proportioning the parts, the main object is to move the link and radius rod (when the crank stands at the zero or 180° locations) in such a manner that the link arcs peculiar to each motion shall always be *tangent* to each other. In this case all the locations of the link block will be found in one and the same *straight* line. This peculiarity has given rise to the title "Straight Link" motion, expressive of the form of the link.

The radius rod and main link are supported by rods from the reversing shaft arms, as shown in the illustration, Fig. 60, and the inequality in the lengths of the latter, which is essential to a proper suspension of the parts, incidentally tends to *equalize* the weights resulting on the opposite sides of the reversing shaft, thus greatly facilitating a change of the motion from one full gear to the other.

Well schemed motions of this type practically preserve the characteristic feature of the stationary link, viz.: a constant lead; yet from the nature of the case they possess at time slight inequalities in one or both of the full gears. These, however, are quite insignificant for a relatively long radius rod and short travel.

This form of link motion is particularly adapted for use upon locomotives with steam chests at the side of the cylinders, better known as "inside connected" engines.

We are unable to say whether or not this type of link motion is now used in America, but it is often referred to, and we assume a description of it will be of interest here.

ANDERSON VALVE GEAR.

The principal feature of this gear is a double eccentric crank arm, fastened to the end of the main crank pin. This arm has two pins, as shown in Fig. 61, which, in operation, take the place of eccentrics. These pins are placed in the manner given: Key way points are located on the axle, just as in the case with the eccentric motion, lines are drawn from the center of the axle, passing through these points, and distances equal to the radius of eccentricity, or half the valve travel, are spaced off. Then the crank arm is designed with arms, so constructed as to connect these points; the small arm extending from the outer end of the pin on the crank pin arms; by doing so the connecting rods will not interfere when the axle is revolving. The motion is then transmitted to a link similar to that used with the eccentric link motion, and continued to the valve by connection from link to valve rod, this being taken care of to suit the design of the locomotive. To reverse the motion, the link is raised and lowered, similar to the eccentric link motion. Of course, it is necessary to place the tumbling shaft arm outside the frame, rather than inside.

The Anderson gear is extremely simple, and is very accessible for inspection, oiling and maintenance, due to the fact that it eliminates the use of eccentrics and straps, and the heavy revolving parts found on eccentric motions. It also does away

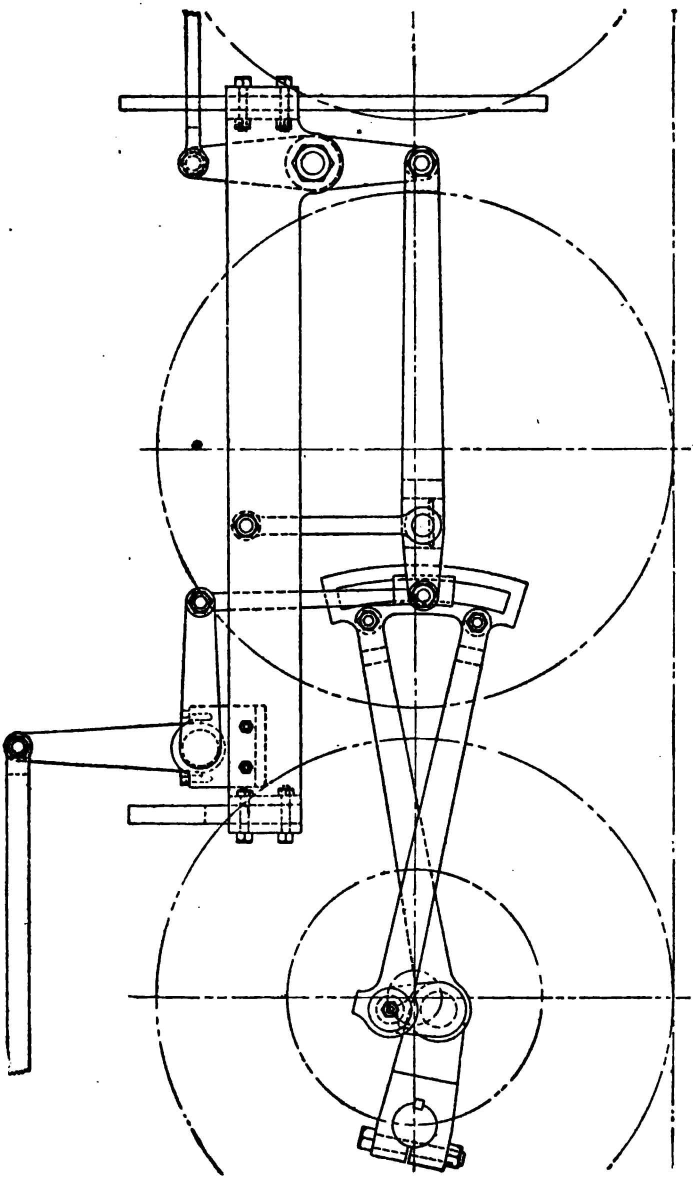


FIG. 61.

with the heavy rocker boxes, and the long transmission rods, which, in most cases, are curved and subject to considerable distortion. Frame breakage, too, is reduced, as compared to locomotives using the Stephenson type of gear, because of the fact that the frame may be better braced, as there are fewer moving parts between frames.

The crank arm is standard for each type of locomotive, and is interchangeable between locomotives of the same class and design, and the necessity of adjustment, after once applying the gear, is rare. The double crank arm enables the gear to take all of the motion transmitted to the valve from the axle, and to secure a variable lead—making for a quick starting locomotive, and an improved distribution of steam.

This valve gear was invented by Mr. J. A. Anderson, of Benwood, W. Va., and several locomotives of the Baltimore & Ohio Railroad Company have been equipped with the same.

JOY VALVE GEAR.

This form of valve gear, invented by the late David Joy, has been the subject of much discussion as to its practical value, when used on the locomotive. The concensus of opinion in this country is such that the gear has been employed on very few locomotives—due, as is generally believed, to the fact that while, in theory, the Joy valve gear gives almost perfect steam distribution, in actual practice its results are far from satisfactory.

Some years ago, however, this gear was extensively used in England, and on the Continent, and, for this reason, we have endeavored to obtain the opinion of those who have had experience, and are familiar, with the results obtained with this gear abroad.

Thus, we are able to quote Mr. A. R. Bell, Editor of "*The Locomotive Magazine*," London, England, who, in the course of our correspondence, in November, 1919, said:

The Joy valve gear may be regarded as obsolescent in British practice; only one line—the L. & N. W. R.—continuing to fit it to machines of new construction, and this company does not employ it in the case of their most powerful class of express engine, the Walschaert motion being used in this case. In Continental practice it is dead; and indeed, never achieved much favor in Continental Europe outside Russia save in a few instances. In England, following the example of the Continent, the Walschaert gear is rapidly coming into favor, for the same reasons as have led to its extensive employment in the United States.

The illustration, Fig. 62, is quite satisfactory as giving a good idea of the form of this motion commonly employed on main line locomotives. Another type, with a small return crank, was used principally for small locomotives and was relatively rare.

The only difference in the latest application of the Joy gear and the earlier type is an increase in the dimensions of the moving parts, and the modification necessary for inside admission valves. In some cases (rebuilt engines) this modification has been obtained by the interposition of a rocking lever between the valve spindle and radius rod.

The Joy gear is one of those distributions that, whilst giving an excellent effect in theory, is subject to several practical disadvantages. Its good points are: Abolition of eccentrics, uniform distribution, fairly light weight, possibility of locating the valve above and parallel with the axial line of the cylinder, and constant lead. It has the disadvantage of being somewhat diffi-

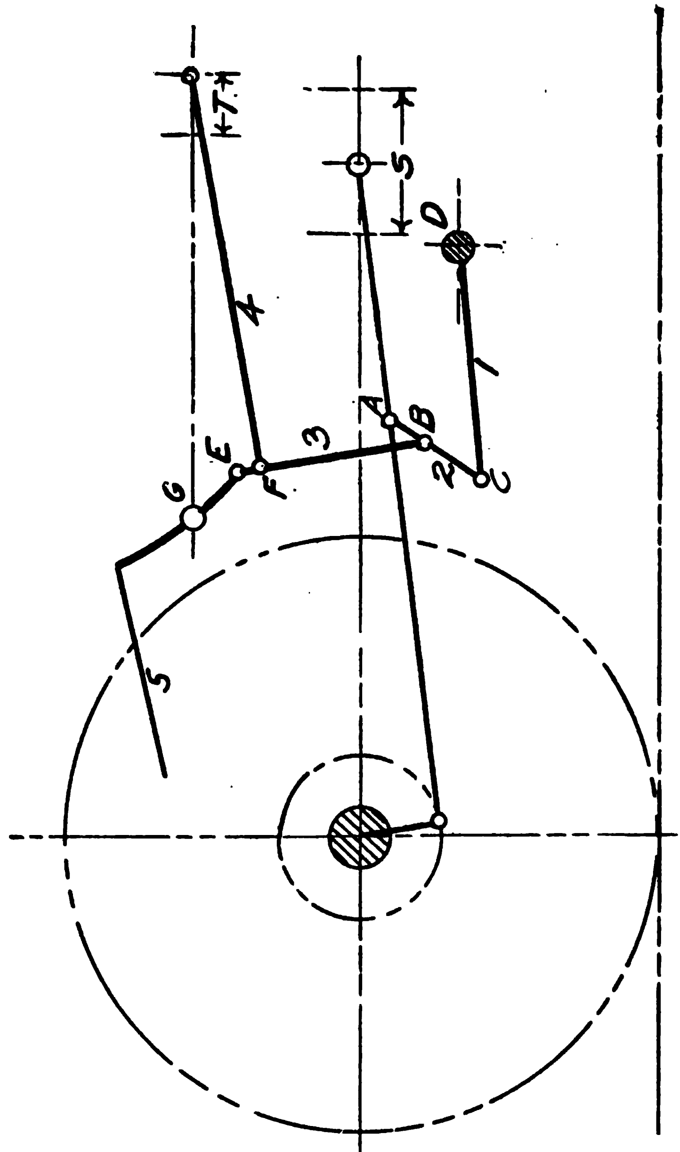


Fig. 62.

cult to set (owing to the position of the die-block in the curved slides being radically affected by the position of the driving axle in the pedestals). Vertical displacements when running are directly reflected upon the valve, particularly with machines having short connecting rods as inside-connected engines usually have; and a tendency to severe wear in the die-block and slide, especially due to the fact that the latter is in the position of maximum angularity when the valve is in full travel. That the vertical displacement of the axle seriously affects the function of the gear is demonstrated by the fact that the writer has known cases where the springs to the main axle have been set undesirably tight in an endeavor to compensate for this by reducing the vertical play of the axle whilst running; thus trying to cure one evil by the introduction of another.

The foregoing reasons are sufficient to show why this gear is going out of favor; especially as the increasing use of the outside cylinder permits the ready adoption of the Walschaert motion, which possesses all the good points of the Joy gear without the disadvantages to which attention is drawn above.

However, a brief explanation of the action of the gear, as shown in Fig. 62, should at least be of interest to the reader.

Operation.

A vibrating arm, or link, 2, derives its motion from the main rod of the engine, to which it is connected at the point A. The link, 2, is limited to move vertically by its connection, at C, with the radius rod, 1. This rod is pivoted at a stationary point on the engine, D. The rod, or lever, 3, is given a horizontal movement at its lower end by its junction, at B, with the link 2. The upper end of lever 3 imparts motion to the valve, through the rod 4, due to the connection at F. This point F, the fulcrum of the lever 3, moves vertically with the connecting rod, to the extent of the vibration of the connecting rod at the point A. Therefore the point F is supported in blocks, which slide in slots in the links. The links are curved, with a radius equal to the length of the arm 4. A shaft, corresponding to the ordinary lifting shaft, attaches to the links, and the position in which the point F of the lever 3 is shown corresponds to the center of this shaft. By means of an ordinary reversing rod, 5, the shaft, carrying the links, may be rotated, thus inclining, or tilting, the link guide slots to either side of a vertical position, and determining full forward and full reverse positions, and the points of cut-off in either motion. This is due to the point F being given a horizontal movement, because, when the links are tilted, the vertical movement of lever 3 causes the blocks in the links, and consequently the point F, to move in a path inclined to the vertical center line. The direction of this movement of F is dependent upon the position of the links, and the extent of the movement is relative to the amount the links are inclined to the vertical.

GOOCH LINK MOTION.

The Gooch link motion, is another form of valve gear, derived from Stephenson's, in which the link is stationary. This form of connection between the valve and the eccentrics, as shown in Fig. 63, is especially applicable to those circumstances in which the former requires no rocker. The mutual relation of the parts will be clearly perceived from an examination of the figure here shown, which illustrates one of the most successful methods of suspension. The main link is hung from a *fixed* point by a short bar called the "suspending link," and the link block connected with the valve stem through the "radius

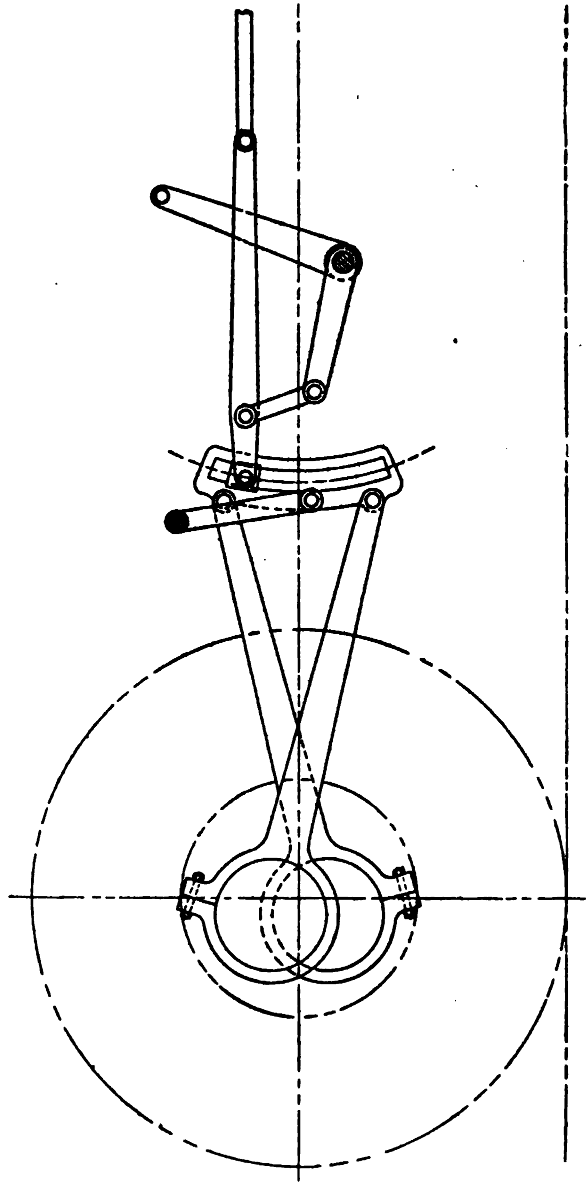


FIG. 63.

rod." By means of a reversing combination the block may be carried to any point from one end of the link to the other, that is, from full gear forward to full gear backward. But, since the link arc is always struck with a radius equal to the length of the radius rod, having its center in the central line of motion, when the crank occupies the zero, or 180° location, it must be evident that the block can be moved from one full gear to the other *without* altering the position of the valve, consequently, the lead opening will remain *constant* throughout the motion. Now it has been invariably the custom to simply define a stationary link motion as *"one in which the lead is constant,"* leaving it to be inferred that the angular withdrawal of the crank from its zero position at the moment of preadmission must also be a *constant* quality. In reality, this lead angle *increases* just as much for a stationary link motion as for a shifting one. The only difference between the two is that the lead opening of the stationary link motion is more ample and the angle slightly greater, for all except the mid-gear, than with the shifting link motion. Unlike the shifting link motion, however, the lead opening is *not* dependent on the arrangement of the eccentric rods, for these may either be crossed or open, without altering the result. But, for the purpose of meeting the other conditions of the motion, an arrangement like the figure shown should be adopted. As a general thing, more attention is paid to the equalization of the cut-off, and reduction of the slip, in the forward than in the back gear. For the accomplishment of this object, the center of the link should be dropped below the central line of motion, the angular advance of the backing eccentric slightly reduced, and the backing eccentric rod lengthened.

This type of gear is more complicated than the Stephenson, and requires nearly double the room between the shaft and the valve stem, on account of the radius rod.

This stationary link is seldom found in American practice, because of the fact that all modern locomotives are built with steam chests on the top of the cylinders, rather than at the side. On stationary engines, the link and governor are occasionally used cojointly, and, in such instances, the stationary link will be found best adapted to the requirements of the case, because its radius rod imposes a far lighter duty upon the balls of the governor than the shifting link with its rods, and additional friction of eccentric straps.

This type of link motion was used extensively on European locomotives in the early days of valve gears, but it has been largely replaced and it is now seldom, if ever, used.

Laying Out a Link Motion.

There are certain points in laying out a link motion which can be ascertained only in a technical manner, but in ordinary shop practice almost the entire motion work is laid out according to well established rules which, when understood, are much quicker and equally as good for all practical purposes.

Therefore, in order to show the reader wherein theory and practice differ, we shall first explain the practical methods, calling attention to these points that can be accurately determined only in a technical manner, and then we shall explain the correct technical method of laying out a link motion, and at the same time point out the errors of the link motion; and last, but not least in importance, show how to set a locomotive's valves.

SHOP PRACTICE REGARDING THE LINK MOTION.

How to Find the Travel of a Valve.

Let us assume that the width of the steam port is 1⅜", and the steam lap ¼". These figures, of course, and those following, may be supplemented by those pertaining to particular conditions. If the valve is to open the steam port fully for the admission of steam, with no overtravel, the travel of the valve can not be less than twice the sum of the steam port width and the lap. In this case it will be $2\times(1\frac{3}{8}+1\frac{1}{4})=5\frac{1}{4}''$, the travel of the valve. But, if the valve is to have overtravel, as modern locomotives all have, and is to move, for example, ⅜" beyond the edge of the steam port, we have $2\times(1\frac{3}{8}+1\frac{1}{4}+\frac{3}{8})=6''$. Also, the throw of the eccentric, when the rocker arms are of equal length, equals the travel of the valve. The valve travel may be approximately found by adding the width of one steam port, plus the steam lap, to one-half the width of one bridge, and doubling the sum.

When the eccentric throw is known, but the rocker arms are of different length, the valve travel may be ascertained as follows: Multiply the length of the top rocker arm by the eccentric throw, and divide the product by the length of the lower arm; the result is the travel of the valve.

To Find the Throw of an Eccentric.

If the eccentric is upon an engine, or is finished in the shop, you can easily determine its throw in this manner: Measure the greatest distance from the center of the axle (or the bore) to the outside face. By outside face we mean the bearing on which the eccentric strap fits. The difference between the greatest and the least distance will be its throw. If the eccentric is not yet laid off, and its correct throw is unknown, it may be determined as follows: If both arms of the rocker are of equal length, the throw of the eccentric should be the same as the travel of the valve.

If the rocker arms are of unequal length, the top arm being the longer, to find the throw multiply the length of the lower arm by the travel of the valve, and divide the result by the length of the top arm.

When the travel of the valve and the length of both rocker arms are known, perhaps the most simple and accurate way to determine the throw of the eccentric is to make a sketch of the rocker arm, similar to Fig. 64, using the center lines only.

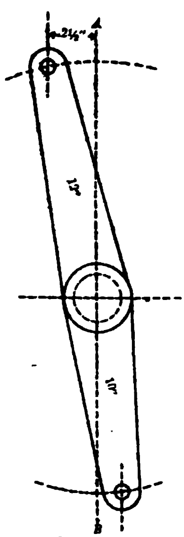


FIG. 64.

We shall assume the length of the top arm to be 12", the length of the bottom arm 10", and the travel of the valve 5".

First, erect the perpendicular line A B; now use a pair of dividers and from a point on the line A B describe two arcs, equal in length to the rocker arms. Now locate a point on the upper arc 2½" from the line A B (which equals one-half the travel of the valve). From this point draw a line, letting it pass through the center of the rocker shaft, and intersecting the lower arc; in our figure the entire rocker is shown in this position, so measure the distance from the center of the hole in the lower rocker arm to the line A B, which will equal one-half of the throw of the eccentric. Measurements should be made on each arc, but the circular movement is so slight that it has not been considered here.

How to Lay Off a New Eccentric.

If the eccentric be in two halves, fasten them securely together. Fit a wooden center, faced with a thin sheet of tin or copper, in the hole, as a shaft, keeping it parallel to the line R S, in Fig. 65. Keep the face of the wooden center flush with the planed side of the eccentric, if either side be planed. In place of wood, copper or tin may be used as a center. Find the exact center of the hole for the shaft, if it has been bored out. If it has not been bored out, draw A B, true with the planed faces of the two halves of the eccentric. Now the center may be located with a pair of hermaphrodites, as shown. Erect the perpendicular line R S through the center of the rib and at right angles to A B, and, with a pair of dividers, set equal to a distance of one-half the throw of the eccentric desired, lay off the point D, as indicated. This will be the center of the eccentric, and so, with D as a center, describe a circle to represent the circumference of the desired eccentric.

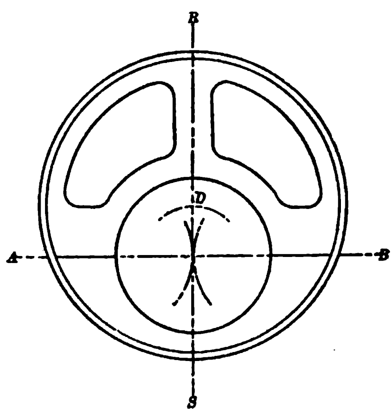


FIG. 65.

The hole for the shaft should be bored just large enough to turn freely on the shaft when the two halves of the eccentric are tightened securely.

To Find the Length of Eccentric Blades.

When speaking of the length of the eccentric blades upon a locomotive, it is generally understood that we mean the distance from the center of the eccentric strap to the center of the link pin hole. If the rocker has no backset, the correct length of an eccentric blade should equal the exact distance, on a horizontal line, between the center of the main driving shaft and the center of the rocker box, minus the distance from the link arc to the center of the link pin hole (the center of the link block travels in the link arc). If the lower rocker arm is backset, subtract the amount of backset from the length.

This length is only approximately correct, as the blades are invariably adjusted a little when setting the valves, but it is near enough for all practical purposes. The correct technical length would vary only slightly. It would be a trifle longer, as the eccentric blades are crossed when the crank pin is on its back center and open when on the forward center, which tends to shorten the blade. (See Technical Points, page 165.) The effect of long and short eccentric blades is clearly shown in the chapter on Errors of the Link Motion, page 172.

To find the length of an eccentric blade, place a straight edge across the two main shoes (see that the shoes are properly tightened), drop a plumb line through the center of the rocker-box, measure the distance from the straightedge to the line, and add to this one-half the thickness of the driving box, which should be bored out central. Subtract from this sum the distance from the center of the link arc to the center of the link pin hole, and also subtract the backset of the lower rocker arm, if it has any backset.

Now, to find the length of the eccentric blade alone, measure the distance from the center of the eccentric strap to its butt-end, or shoulder, and add to this about ½″ for liners, or clearance, as the case may be; subtract this length from the total length of the blade and strap, and the remainder will be the length of the blade alone.

To Find the Length of the Valve Stem and Yoke.

On most modern locomotives the cylinders and valve seat are parallel to the line of wheel centers. On these engines simply plumb the outside, or top, arm of the rocker shaft, by dropping a line through its center; the distance from this line to the center of the exhaust port will be the correct length for the valve stem from the center of the yoke to the center of the rocker pin hole.

Another way to find the length of the valve rod is to find the center between the port openings, plumb the rocker arm, and mark the valve rod with a tram. The distance between the center between the port marks and the tram is the amount the rod must be changed.

If the valve seat is inclined, set the top arm of the rocker at right angles to the valve seat; this may be done by placing a long straightedge on the valve seat, and, using a two-foot square at the rocker, by setting the center line of the rocker arm true with the square. Measure the distance from the center of the rocker arm to the center of the exhaust port; this distance will be the correct length.

To determine the length of the valve stem and yoke alone, without the valve rod, measure the distance from the center of the steam chest to the outside of the gland (when it is packed), and to this length add one-half of the travel of the valve, plus one inch for clearance (if space will permit add a few more inches so that the gland may be repacked with the stem in any position without pinching the engine). This will be the length from the center of the valve yoke to the shoulder. The remainder of the original length will be the length of the valve rod from the center of the pin hole to its shoulder.

Laying Off a Valve Yoke.

If the yoke is milled, or slotted, to fit the valve; finish fitting it by hand before laying off the stem. Keep the stem true with the face and sides of the valve, fit closely, but do not let the yoke bind or cramp the valve. When finished, the end of the stem should be permitted to raise and lower about ¼ to ⅜" without binding the valve. This is done on account of the circular motion of the rocker arm, and the yoke should be about 1-16" loose, endwise, between the yoke and the valve.

In laying out a valve yoke, be sure that the yoke is square with the stem before slotting it.

Now put the yoke on the valve, take it to a face plate, and locate two centers from which to swing and turn the stem. Keep these two centers central both ways, and see that you have sufficient stock for the stem to finish the required size.

If the entire yoke is still in the rough take it to a face plate and block it up edgewise, use a two-foot square and surface gauge and keep the stem at right angles to the yoke, divide the stock as nearly as possible every way, then locate two centers for turning up to the required size. Then, with the two-foot square, lay off the yoke for the valve, making it the exact thickness of the valve longitudinally, and allow from

1/32" to 1/16" for the valve to move endwise in the yoke. Drill small holes in the centers laid off for turning, in order to retain the original centers.

Finding the Length of the Link Hanger.

On a large face plate, or table, lay off, full size and with correct dimensions, the tumbling shaft and rocker arms, using their center points only.

Keep the top arms of both the tumbling shaft and the rocker perpendicular, or in their correct positions; and, from the center of the lower rocker arm and on a horizontal line with it, lay off the backset of the saddle stud. From this point find the distance to the center of the short arm of the tumbling shaft, which will be the correct length of the hanger.

If one link hanger is broken and the other one is all right, use the length of the good hanger.

Length of the Reach Rod.

Plumb the reverse lever and the top arm of the tumbling shaft (if they are in position), and find the exact distance between the center of the hole in the reverse lever and the center of the hole in the tumbling shaft arm, which will be the correct length from center to center. If the reverse lever and tumbling shaft are not yet in position, place a center in the foot casting for the reverse lever and another in the tumbling shaft stand; transfer a square line from each center onto the frame, and measure the distance between these two lines. If, for any reason, the reverse lever, or the tumbling shaft arm, cannot be set plumb in its central position, then set either in whatever position you can for its central position, and find the length as we first suggested.

To Find the Backset of a Standard Rocker Arm.

We will assume that the rocker box is already located, or that the reader knows the length of the eccentric rods and rocker arms. On a large face-plate, table, or finished board, draw the horizontal line A B parallel with the wheel centers (see Fig. 66), and locate the point H the correct distance from the main shaft and the correct distance above the line A B; this point in-

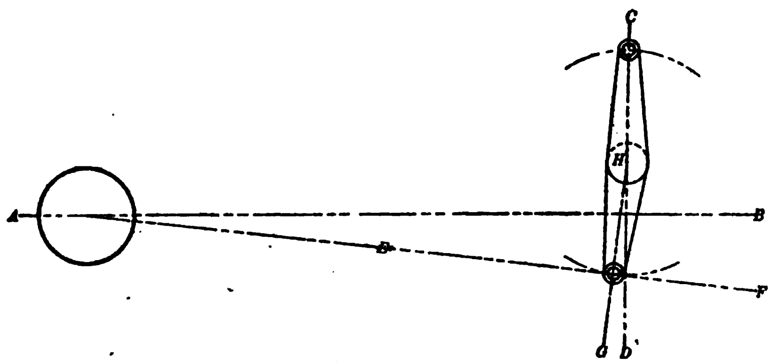


FIG. 66.

dicates the center of the rocker shaft. Now erect the perpendicular line C D at right angles to the valve seat, and with a pair of dividers, set to the length of each rocker arm, describe arcs from the point H equal to the length of each rocker arm. Then draw the line E F through the center of the shaft, intersecting the lower arc which represents the length of the lower rocker arm. Next, through the center H, erect the line G H at right angles to the line E F. The distance between the line C D and the line G H on the lower arc is the required backset; the line E F being the center line of motion.

If the top arm of the rocker stands plumb in its correct relation to the valve seat, and the line A B bisects the center of the lower rocker arm, the rocker will require no backset.

Remember that the top and bottom arms of the rocker are entirely independent of each other as regards the backset; either of them may require a backset, and the other not, or they may both be backset. The top arm should be at right angles to the valve seat, and the bottom arm at right angles to the center line of motion, then each will travel an equal distance each way from its central position.

To Find the Backset of a Direct Motion Rocker Arm.

The draftsman will usually determine the amount of backset for a rocker of this kind, since in doing so it is necessary to know the relative position of the center of the link block and the tumbling shaft when in their central positions. (See Technical Points, page 170.)

When these positions are known it is easy to determine the correct backset for the rocker arm. Make a full sized sketch as shown in Fig. 67. The line A B should pass through the wheel centers. The outside rocker arm should be set at right angles to the valve seat, and, with a length equal to the length of the inside rocker arm, describe the arc R S. Draw the line C D through the center of the link block pin hole when the block is in its central position, and let it intersect the arc R S. Then draw the line E F through the center of the rocker shaft and at right angles to the line C D. The point where the line E F intersects the line C D will be the center of the inside rocker arm, and the backset will equal the distance on the arc R S between the centers of the two arms, as shown in the illustration; the distance between the center of the link block pin hole and the center of the pin hole in the inside arm of the rocker will be the length of the extension rod. In the figure, we find that the link block is attached to the extension rod, while the back end of the extension rod is prevented from working up or down by the small hanger H. Excepting the slight circular movement imparted by the hanger H, the center of the link block remains in a horizontal line with the center of the main shaft. We therefore determine the offset by the center of the link block pin hole instead of by the center of the driving axle. The line C D is, therefore, the center line of motion.

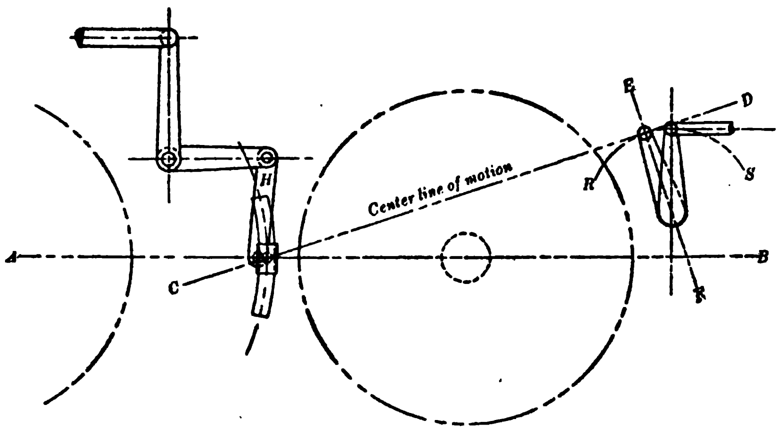


FIG. 67.

Laying Off a Rocker Arm.

Take the rocker to a face-plate, secure two V blocks of equal sizes, and line them up with parallel strips high enough to let each arm revolve. Next place the body of the shaft on the V blocks, keeping one V block near each end; then, with a surface gauge, set to any height, scribe a line on each end of the shaft, revolving the rocker to four different positions, and marking each end of the shaft for each position. From these four lines locate a center at each end of the shaft, place the arms in an upright position, and set each center true with the surface gauge; then try a two-foot square to each arm and see whether they are square with the shaft. Place the arms in a horizontal position, keeping the two main centers true, and carry a line through the boss on each arm. Now lay off the length of each arm, and see if the rocker will true up all around. If the arm requires any backset, then with the length of the arm describe an arc on the boss and lay off the center for that arm the required distance from the central line, being careful to lay it off in the right direction. When laying off the center of each arm, be careful to make all measurements at right angles to the center of the shaft. When you have the four centers correct, lay off the pin holes, making each 1/32″ smaller than the round end of the taper pin hole when finished; and have a small hole drilled in each end of the shaft to retain the main centers while in the lathe. If the rocker will not true up to the correct dimensions, return it to the blacksmith then, instead of doing so after you have it half finished in the lathe. If you have plenty of stock, divide it, as near as possible, in order to equalize the cut while in the lathe.

To Find the Length of Rocker Arms.

This is properly the work of the designer, but when the throw of the eccentric and the travel of the valve are known it is an easy matter to determine the length of each arm, if the position of the rocker box is known. Place a straightedge across the top of the frames, and measure the distance to the center of the rocker box, then measure the distance to the center of the stuffing box. If the rocker box is above the frame, the difference between these two measurements will indicate the length of the top arm, and if the rocker box is below the top of the frame, measure the distance it is below the frame, and add these two measurements together; in either case add 1/8″ more in order to divide the circular movement imparted to the valve stem (unless a Scotch yoke is used). We shall assume that the length of the top arm is 12″, the travel of the valve 5″, and the throw of the eccentric 4″. If both arms were of equal length, we know that it would require a 5″ throw to move the top arm 5″. It is therefore evident that the bottom arm must be less than 12″ in length. In this case, it is in the proportion of 5 to 4, and it may be determined as follows: Multiply the length of the top arm, 12″, by the throw, 4″, which equals 48, and divide this by the travel of the valve, 5″, which gives 9-3-5″—length of lower arm. If the throw, travel, and length of the bottom arm are known, find the length of the top arm as follows: Multiply the length of the lower arm by the travel of the valve, and divide the result by the throw of the eccentric.

Length of the Tumbling Shaft Arms, and Relative Position of the Tumbling Shaft and Rocker.

These are technical points and any attempt to determine them by ordinary measurement will result in a failure. (See Technical Points, page 170.)

The Quadrant.

The determination of the length and radius of the quadrant is properly the work of the draftsman, but they are easily determined in the shop, when the dimensions of the other parts are known. Make a sketch of the other parts, and locate the position of the reverse lever latch in each full gear. Find the distance from one extreme point to the other, and add to this the width of the reverse lever, plus a little more for clearance at each end, with sufficient stock for bolting both ends; this is the length of the quadrant.

To find the correct radius of the quadrant, draw the reverse lever latch up as far as it will go; then measure from the bottom of the latch to the center of the pin hole in the foot casting, and subtract 1/16″ from this distance for clearance for the latch; the remainder will be the correct radius for the top of the quadrant. When laying off the holes in a quadrant, for drilling, draw the latch up as far as possible and fasten it there. Then, if the boiler is cold, let the latch clear the quadrant about 3/32″ at the front end, and ¼″ at the back end, which will allow for the expansion of the boiler: when the boiler is warm the latch will have the same clearance at each end. There is nothing very important about a quadrant except the notches. How to lay these off will be explained later. Of course, the quadrant should be set properly and securely fastened.

Radius of the Link.

When the rocker arm has no backset, the correct radius of a link should equal the distance on a horizontal line from the center of the main shaft to the center of the rocker box. If the rocker has any backset, subtract the amount of backset from this length. The reason of this is clearly shown in our chapter entitled Errors of the Link Motion. Some manufacturers of locomotives make the link radius ¾″ per foot less than this length, but the distance given is the correct length.

To find the radius, place a straight edge across the front of the main jaws (providing the front jaw is square with the top of the frame). Then drop a plumb line through the center of the rocker box, and measure the distance on a horizontal line from the straightedge to the line. Add to this length the thickness of the main shoe, plus one-half the driving box. This will be the correct length of the radius. If the main jaws both taper, find the center of the jaw (see Shoes and Wedges) and drop a line through its center. Then find the distance horizontally from this line to the other line, dropped through the center of the rocker box; this length will be the correct radius. In either case, if the rocker has any backset subtract the amount of backset from its length. This rule will not apply to valve gears having an extension rod between the link and the rocker; on locomotives of this kind the radius will depend upon the relative positions of the tumbling shaft and link.

Laying Off a New Link.

Blue-prints are usually furnished for the purpose of laying off new links, but if they are not, proceed as follows: We will assume that the throw of the eccentric is 5″, and that the rocker arms are of correct length to secure the required travel for the valve, and that you intend to make the link block 5″ long and 3″ wide. First, draw the straight line A A, as shown in Fig. 68, through the center of the shaft and the center of the link; next, with a length equal to the correct radius of the link, describe the arc B B. Then add together the following amounts: 2½″, which is one-half the length of the link block; and 5″, which equals the throw of the eccentric; 1½″ for the slip of the link block (see Slip of the Link, page 152), and ¾″ for clearance between the link block and the end of the link. (In ordinary practice 1½″ is the amount usually allowed for the slip of the link and ¾″ is considered a safe margin for clearance, although, when a blue-print is furnished, you will find these amounts considerably reduced as the draftsman usually has the use of models with which he can try the maximum slip and thereby prove his work; in such a case ⅜″ is considered sufficient for clearance. The slip varies on different engines, but

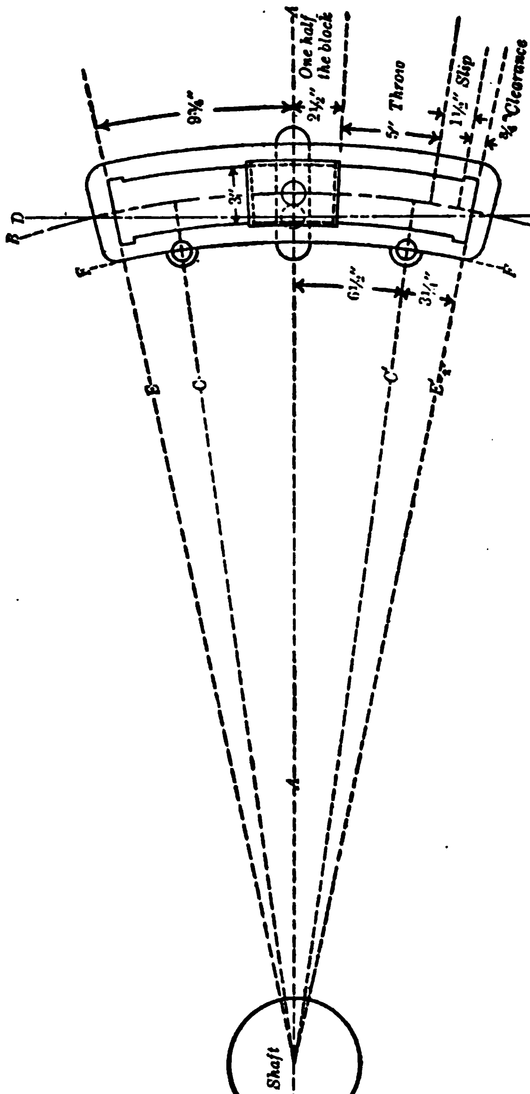

9⅜"
2½" One half the block
5" Throw
1½" Slip
¾" Clearance
3"
6½"
3¼"
A
B
D
F
E
C
C'
E'
Shaft

FIG. 68.

the figures we give herewith are considered safe.) Now, by adding the above amounts, we have a total of 9¾, which is one-half the inside length of the link. All measurements should be made on the arc B B, which is the link radius, with a flexible scale, if you have one; if not, cut a strip of tin 9¾″ long, and from the point where the line A A intersects the arc B B (which will be the center of the link block), lay off on the arc B B two additional points 9¾″ from the center; and from these two points draw the lines E and E′, passing through the center of the shaft. These lines indicate the inside ends of the link. Then add together 2½″, which equals one-half the length of the link block; and ¾″, which is the clearance; this amounts to 3¼″. So, in each full gear, the center of the link block should be 3¼″ from each end of the link. Therefore, from the two points where the lines E and E′ intersect the arc B B, lay off on the arc B B two additional points 3¼″ nearer the center of the link, and from these points draw the two lines marked C and C^1; the link pin holes will therefore be laid off on these two lines, which indicate full gear. Now, as the link block is to be 3″ wide, add 1½″ to the length of the radius, and describe another arc which will represent the front face of the link inside and the front face of the link block. Then subtract 1½″ from the radius, and describe another arc which will represent the back face of the link inside and the back face of the link block. Since the link block is to be 5″ long, lay off two additional points 2½″ from its center. From these points lay off the ends of the block, keeping them true with the center of the shaft. The link pin holes must be an equal distance from the link radius and should be as close to the radius as possible to avoid increased slip. Therefore, describe the arc F F (which is called the link pin arc), as close to the arc B B as will be consistent with a proper thickness of the link. The link pin holes should be laid off at the two points where the lines C and C[1] intersect the arc F F. Now, through the two points where the lines E and E[1] intersect the arc B B, draw the line D D. In ordinary shop practice, when the correct position of the saddle stud is not known, the point where the lines A A and D D cross each other is used as the point of suspension. This point is only approximately correct, but near enough for all practical purposes. To find the correct point of suspension, see Technical Points, page 167. The offset of the saddle pin is, therefore, indicated by the space between the line D D and the arc B B on the line A A.

Reason Why a Link Saddle is Offset.

The purpose of the offset in the link saddle is to obtain as nearly as possible an equal cut-off, and, at the same time, to permit the lead to be the same for each stroke, and to approach as nearly as possible a correct distribution of steam with a reciprocating engine and slide valve. It is done to overcome the inherent imperfection in the design of links, the angularity of the connecting rods, and the offset of the link pin holes.

Slip of the Link.

Some years ago the "slip" between the link block and the link was considered an important feature in the adjustment of the link and its connections, but it was found to be of small consequence, and, at present, is generally neglected.

In addition to the other two motions, the link block, being securely fastened to the bottom of the rocker arm, must move in the arc traversed by that arm, while the action of the eccentric rods on the link forces it to move in a sort of vertical motion during certain parts of the stroke; these two motions combined cause the link to slip in the block. This slip is caused partly by the circular movement of the lower rocker arm, thereby causing the block to slip also, but principally by the method of suspension, and the manner of attaching the eccentric blades to the link, the link pin holes being back of the link arc. The action of the link pins is similar to that of a knuckle joint between the eccentric center and the link arc through which the center of the link block must travel. The link slips most when in full gear and the slip diminishes as the block is moved toward the center of the link. By referring to "Technical Points," you will note the distortion introduced into the valve's motion by the angularity of the connecting rods and by this backset of the link pins from the link arc, and, while moving the link saddle pin back tends to equalize the motion, it also tends to increase the slip, which, if very great, would seriously impair the valve's motion. On marine engines, equality of steam is sometimes sacrificed to a reduction of the slip; but with the long connecting rods used upon locomotives, and the proximity of the link pin holes to the end of the link, little difficulty is found in keeping the slip within practical bounds. Raising the link saddle above the center of the link will also equalize the valve's motion, but in locomotive construction there are practical objections to doing this. Backsetting the link saddle pin has an effect equivalent to lengthening the eccentric rod during a portion of the stroke, and thereby equalizes the valve's travel. (See Rule 36 for Valve Setting.) Moving the eccentric rod pin holes farther from the radius of the link or closer together, tends to increase the slip. By referring to Fig. 68, you will notice that we allowed 1½" at each end of the link for the "Slip of the Link." The amount of slip varies on different locomotives, but 3" in the length of the link is considered a safe margin. This slip partakes of a sort of double movement during each revolution of the wheel; the block lowers and raises in the link twice during one complete revolution, as may be seen by anyone who will go under an engine and watch the link and block. The action is as follows: In full forward gear the block begins to slip down immediately after the crank pin has passed the top quarter, and continues to move downward until the pin has reached the forward center, at which point it begins to move upward until the pin has passed the bottom quarter. Then it again begins to move downward and continues to move downward until the crank pin has reached the back center, when it again moves upward until the pin has once more reached the top quarter, or starting point. In full gear backward, these operations are just the reverse. The amount of slip may be ascertained by measuring the distance from the end of the link to the link block, in either of these positions, with the reverse lever in full gear; the difference between the greatest and least the distances so obtained will indicate the amount of slip. An increase in the length of the lower rocker arm will decrease the slip, and an increase in the length of the link will decrease the slip, provided the distance between the link pin holes is also increased, but it will also increase the midgear lead.

To decrease the length of the link, and also the space between the link pin holes, will increase the slip and reduce the midgear lead.

TECHNICAL POINTS.

In this chapter we shall endeavor to explain how to locate certain points and determine certain lengths for a shifting link motion, which could not be found or explained by plain measurement. Strict theoretical rules prohibit the use of the link templet, but as it gives the same results when properly used as a conglomeration of link arcs and circles, which only confuse and mystify the ordinary mechanic or apprentice, we shall make use of the link templet to explain these subjects. Some of our drawings may appear a trifle complex, but we have made each as plain as possible, for a thorough comprehension of the subject, and in order that it may be more clearly understood by all our readers we have avoided the use of Algebra and Geometry. The subjects treated in this chapter will include:

First: The angularity of the main rod, showing the crank pin at full and half stroke on a center line engine.

Second: The position of the crank pin at full and half stroke on an engine whose cylinder axis is above the wheel center.

Third: The relative positions of the eccentrics to the center line of motion.

Fourth: Relative positions of crank pin and eccentrics at full and half stroke.

Fifth: How to find the correct length of eccentric blades.

Sixth: How to locate the point of suspension, which indicates the position of the center of the saddle pin.

Seventh: The relative positions of the tumbling (or lifting) shaft and the rocker.

Eighth: The length of the tumbling shaft arms.

Angularity of the Connecting Rods.

The eccentric is in effect a crank, and, being keyed onto the axle, must always remain an unvarying distance from the crank pin. It, therefore, follows that any irregularities imparted by the crank pin to the motion of the piston will also be imparted by by the eccentric rods to the motion of the valve. But, since the throw of the eccentric is much less than that of the crank pin, and since the eccentric rod is proportionately longer than the main rod, it follows that the distortions in the motion of the valve are necessarily much smaller than those in the motion of the piston, and they vanish entirely when the eccentric is on either center.

The point A in Fig. 69 represents the center of the cross head pin. Now, if the crank pin were on the forward dead center, it is evident that the center of the crosshead pin would be at the point B, which indicates the extreme forward travel of the

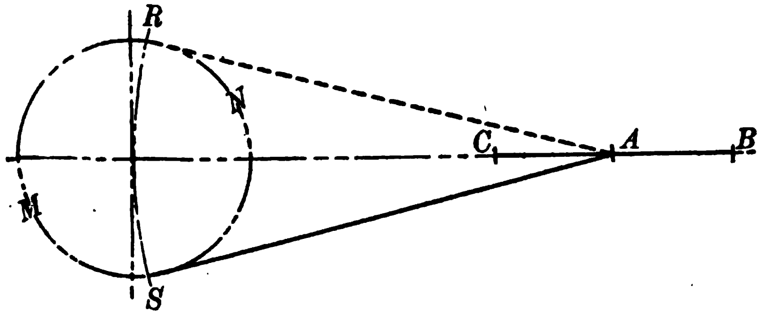


FIG. 69.

center of the crosshead pin; while if the crank pin were on the back dead center, the crosshead pin would be at the point C, which indicates its extreme back travel. It now becomes necessary to locate the positions of the crank pin at half stroke. It is evident from the above brief explanation that the length of the main rod must be equal to the distance from the center of the driving axle (or shaft) to a central point between the points B and C. This point indicates the center of the crosshead pin's travel, and, therefore, the half stroke of the piston is indicated by the letter A. Now, to determine the position of the crank pin at half stroke we must proceed as follows: From the point A, with a length equal to the length of the main rod, describe the arc R S. The two points where the arc R S intersects the crank pin circle must, therefore, indicate the positions of the crank pin at half stroke, for either stroke of the piston. We find these two points are farther forward (nearer the crosshead) than the top or bottom quarter. It is, therefore, evident that the crank pin must travel farther while the crosshead pin travels from A to C than it would travel while the crosshead pin traveled from A to B.

This irregularity will be scarcely perceptible when the piston is at the beginning of its stroke, and, therefore, the point of lead opening, but it would seriously affect the point of cut-off which occurs at intermediate points of the piston's stroke.

Partly to overcome this imperfection of the crank motion, and in order to obtain an equal cut-off, the link saddle pin is backset, but this only approximately corrects the inherent error of crank motion.

Experiments were carried out by making one steam port wider than the other to overcome this defect, but that caused one exhaust to be heavier than the other, and also proved injurious in other ways. To offset the effects of the angularity, valves are used, upon some stationary engines, which have more steam lap on one side of the valve than upon the other.

The early builders of steam engines used what is known as the slotted crosshead, or Scotch yoke, to overcome this defect, but it was found that setting the link saddle pin back answered the same purpose and was less expensive.

Cylinder Axis Above the Wheel Centers.

Many of the early locomotive builders set the center line of the cylinders a considerable distance above the driving wheel centers, under the erroneous theory that such position contributed to the adhesive quality of the engine, but the present practice, with rare exceptions, is to set the cylinders central to the main driving axle.

Upon an examination of Fig. 70 we discover another form of irregularity in the piston's motion introduced by the main rod. Here we find the line P Q represents a central line drawn through the wheel centers, and the line C B represents a line drawn through the center of the cylinders; this latter line also represents the path of the crosshead pin as explained in Fig. 69. Now, if from the point A, as previously explained, we

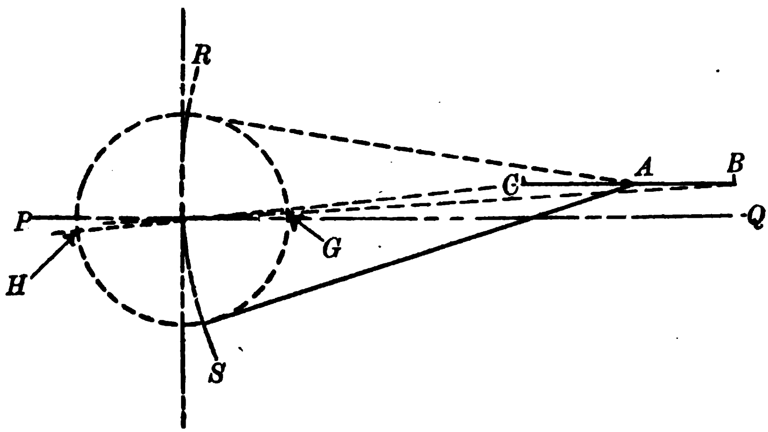


FIG. 70.

describe the arc R S, we find that the crank pin will be nearer the quarter in one stroke than in the other, and hence will cause a variation in the cut-off for the two strokes of the piston. Now, if we examine Fig. 70 a little closer, we shall find that the error is partly overcome by the position of the crank pin at both the forward and back dead centers. When the crosshead is at B, it follows that the position of the crank pin must be located on a line drawn from the center of the shaft to the point B; the crank pin center will therefore be at the point G when it is on the forward dead center. Likewise, a line drawn from the point C and passing through the center of the shaft, will bisect the crank pin circle at the point H, which is, therefore, the position of the crank pin when on the back dead center. It will be noticed that the point G is above the horizontal line P Q, while the point H is below it, therefore a line drawn at right angles to the line P Q and passing through the center of the shaft will not indicate the correct top and bottom quarter for this kind of an engine. It will be noticed, however, that the point G is closer than the point H to the line P Q; and a line drawn from the point G to the point H would not pass through the center of the shaft. Therefore, this irregularity is not entirely overcome. When finding the length of the main rod for this kind of an engine, if the crosshead is up, the length of the rod should be found by measurements on unequal surfaces.

Care should also be taken to get the crank pin on its correct center when putting up a main rod or finding the travel marks on these engines.

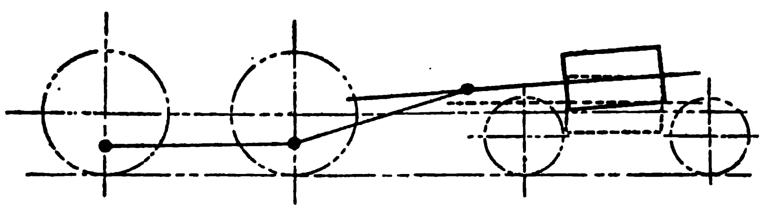


FIG. 71.

Cylinder Axis at Angle to Wheel Centers.

It has long been the practice to make the center line of the cylinder slightly above the center of the driving wheels, and incline the cylinder on an angle (usually 1″ in 48), on narrow gauge locomotives, with small wheels, to give sufficient room for the front engine truck under the cylinders.

The dotted lines in Fig. 71 show where the cylinder would be located if it were placed on a horizontal line with the center of the driving wheels, and the heavy lines indicate the position of the cylinder when placed at an angle.

It will readily be observed that this arrangement provides additional room for the engine truck.

Center Line of Motion and Angular Advance.

When no extension rod is used between the link block and the rocker arm, the center line of motion will be a line drawn through the center of the main shaft and the center of the pin hole in the lower rocker arm, as shown by Fig. 66, the upper arm of the rocker being set at right angles to the valve seat. When an extension rod is used between the link block and the rocker arm, as shown by Fig. 67, the center line of motion will extend from the center of the link block (in its central position) to the center of the pin hole in the arm of the rocker to which the extension rod is attached.

Now, we wish to call the reader's attention to a few points shown in Fig. 72. The horizontal line P Q bisects all the wheel centers, and we shall assume that in this case the line also passes through the center of the cylinder; therefore, the crank pin will be on "dead center" when its center is located on this line P Q. We have shown it on its forward center at the point G, at which point we find that the center of the lower rocker arm is below the line P Q, and that the lower rocker arm is backset. How to determine the amount of this backset has been previously explained. It will be noticed that the center line of motion does not pass through the center of the crank pin when the rocker has a backset. It will also be observed that the short line drawn through the two eccentric centers, F and B, is at right angles to the center line of motion, and not parallel to the perpendicular line L M, and that the eccentric centers are unequal distances from the crank pin. It follows, therefore, that the eccentrics are set by the center line of motion, and not by the crank pin.

Now, if the center of the lower rocker arm were located on the line P Q, the rocker would require no backset; and this line would then be the center line of motion; and the eccentrics would be an equal distance from the crank pin. But, again, if

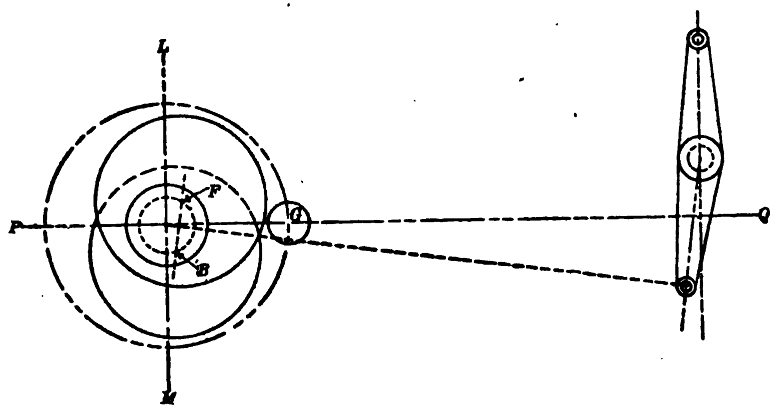


FIG. 72.

the engine had a straight rocker, and the line P Q was the center line of motion, it does not necessarily follow that the eccentrics be equidistant from the crank pin. If the cylinder center were above the wheel center, as shown by Fig. 70, we would again find that the eccentric centers were unequal distances from the crank pin. Therefore, the eccentrics are always set by the center line of motion and not by the crank pin. However, the builders of modern locomotives endeavor as far as possible to build straight line engines and to use straight rockers.

Another point to which we desire to call the reader's attention is the angular advance of the eccentrics. In Fig. 72 the angular advance is equal to the distance between the short line which bisects the eccentric centers and the center of the main shaft. In this case the advance is equal for each eccentric, but in some cases the angular advance of one eccentric is altered in order to improve the other motion, as will be explained later. In a case of this kind, where the angular advance of each eccentric is different, the distance between the center of the shaft and a short line drawn through the center of each eccentric, as shown by Fig. 72, would not indicate the correct angular advance of either eccentric, as such a line would not be at right angles to the center line of motion.

With rocker arms of equal length the amount of angular advance should be equal to the lap of the valve plus the lead.

Relative Positions of Crank Pin and Eccentrics at Full and Half Stroke.

We have made use of two former figures in order to call the attention of the reader to those points in the construction of an engine which must be considered when laying off a link motion. But in order to avoid the use of complex drawings, which only tend to confuse the reader, and in order to explain the following subjects in as clear and understandable a manner as possible, we shall assume that this is a "straight line" engine, whose wheel centers and cylinder center are on a horizontal line, which line is also the center line of motion. This engine, of course, has a straight rocker (by straight rocker is implied one with no backset). In Fig. 73 we shall use the lines P G and L M, the same as in the former figures, but in this case the line P G is also the center line of motion; the larger circle represents the path of the crank pin, and the small circle the path of the eccentrics. Therefore the points G and H snow the position of the crank pin when on forward and back centers, and T and W indicate the position of crank pin at half stroke (as explained by the angularity of the connecting rod). We shall, therefore, draw two straight lines from the points T and W to the wheel center. Next we shall locate the positions of

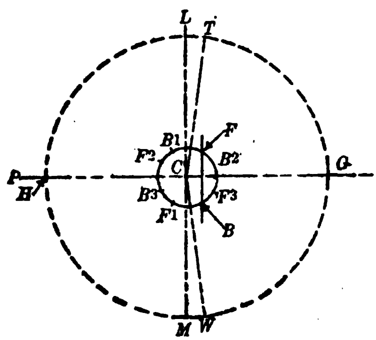


FIG. 73.

the two eccentrics, when the crank pin is at the forward center G. If the valve has no lap or lead, the eccentrics would be at right angles to the crank pin, and, therefore, on the line L M; but we shall assume that the valve has 7/8" steam lap and 1/16" lead. Therefore, we must advance both eccentrics that amount toward the pin, in order to give the valve the required amount of opening at the beginning of the stroke. Since the crank pin is at the forward center, G, the piston must be at the forward end of the cylinder, therefore the forward steam port must be opened to admit steam. By advancing both eccentrics toward the pin, the bottom arm of the rocker is forced forward and the top arm backward, thus opening the forward port. So we construct another short line parallel to the line L M, and 15/16" in front of it (which is the amount of lead and lap added together); the two points where this line intersects the eccentric circle will be the centers of the two eccentrics. We shall designate them by the letters F and B, as shown, and in this case they are equal distances from a line drawn from C to G. The distance the center of each eccentric is advanced from the perpendicular line L M, which must be perpendicular to the center line of motion, is called its "angular advance." We have, therefore, located the positions of the two eccentrics in the full stroke forward. We shall now locate them in full backward stroke when the crank pin is at H (the eccentrics are always an unvarying distance from the pin, being securely fastened to the axle), therefore they will be an equal distance from a line drawn from C to H, and they are indicated by the letters F^1 and B^1. We shall next locate them at half stroke. The pin being at T, they will be equal distances from the line C T and are indicated by the letters F^2 and B^2. We will now locate them at W. They will be equal distances from the line C W and are indicated by the letters F^3 and B^3. The points marked F, F^1, F^2 and F^3 indicate the different positions of the forward or go ahead eccentric, and those marked B, B^1, B^2 and B^3 indicate the positions of the backup eccentric.

Link Templet.

We shall now proceed to make a link templet, which we shall have occasion to use later on (a templet may be made of Russian iron or tin). Fig. 75 shows the outline of a link, the line R S being an arc of the correct radius of the link, and passes through its center; the line I J is the link pin arc, and is described from the same center as the arc R S. The lines P Q and X Y should be parallel to the center line M N, and all three lines should be marked on the templet as shown in Fig. 74. The center of the saddle pin will be located on the line

M N, at, or very near, the point where the line K L crosses it (as previously explained in Shop Practice).

Now cut out the templet, letting it cover all the space between the lines A B, C D, I J and R S, and we have a link templet as shown in Fig. 74. The points marked U and Z indicate the centers of the link pin holes.

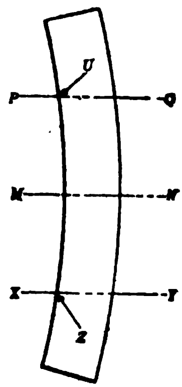


FIG. 74.

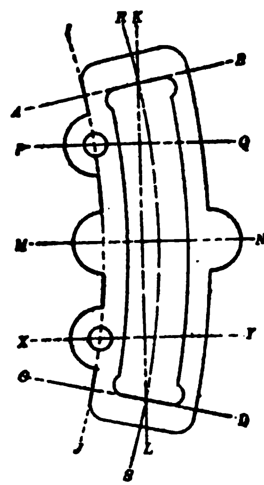


FIG. 75.

Correct Length of Eccentric Blades.

We shall now continue where we left off after locating the positions of the eccentrics for full and half stroke.

Draw the center line of motion, and the perpendicular line L M, the same as before. Locate the position of your eccentrics in full stroke when the crank pin is at G and H, as previously explained. Then locate your rocker arm the correct distance from the center of the axle, as shown in Fig. 76 by the letter N and parallel to the line L M. Find the exact distance between the points *c* and *p*, and from this subtract the distance from the link pin arc to the arc of the link proper, as shown by Fig. 75, and the remainder will be equal to the length from the center of the eccentric strap to the center of the link pin hole in the blade. Then, with a radius equal to this length, and from the point F, describe the arc F above the center line of motion. Now, if the link is raised or lowered, this arc F indicates the path through which the link pin travels, and what is true of this one arc is also true of all that may hereafter be made.

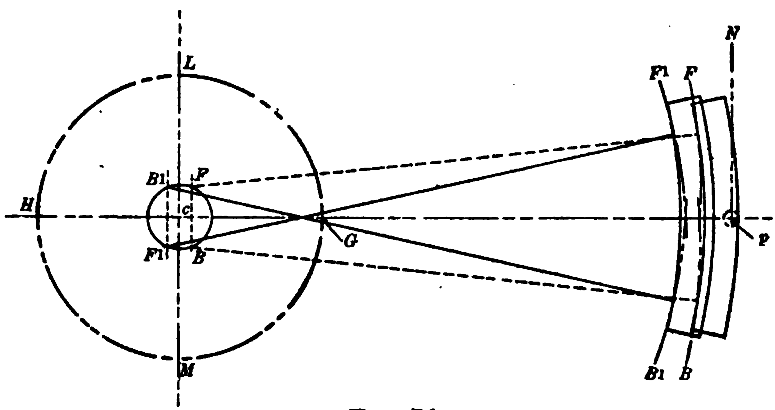


FIG. 76.

Next, from the point F¹ describe the arc F¹, and from B describe the arc B below the center line of motion, likewise from B¹ describe the arc B¹. Then place the templet on the drawing, keep the center M N on the center line of motion, and let the two points marked U and Z on the templet just touch the two link arcs F and B. Mark the shape of the templet on the drawing; this indicates the position of the link, when the crank pin is at the point marked G. Then slip the templet back until the two points marked U and Z just touch the link arcs marked F¹ and B¹, keeping the line M N on the center line of motion. Again mark the shape of the templet on the drawing; this indicates the position of the link when the crank pin is at H. The forward face of the link templet, marked R S in Fig. 75, indicates the center, or radius, of the link; therefore, the front faces of the templet should be equal distances from the point p, which is the center of the bottom rocker arm. Now, by describing from the point p, a small circle tangent to the face of the forward templet, we find that the back templet falls a little short of the circle. This is caused by the eccentric blades being crossed when the pin is on the back center, while they are not crossed when the pin is on forward center. We must, therefore, lengthen our blades one-half the amount of the distance from our small circle to the front face of the back templet, as shown in Fig. 76. This will be the correct length of the blades, and when connected up, the rocker arm will then travel an equal distance each way from its central position.

Correct Point of Suspension or Position of the Center of Saddle Pin.

The periods of admission and cut-off may be equalized by changing the point of suspension of the link, either up or down, or horizontally. A somewhat better distribution of steam can be secured by suspending the link above its center, but in locomotive construction there are practical objections to raising it. We have already explained that the center of the saddle pin would be located on the line M N, Fig. 74, and we shall now determine its exact distance from the front face of our templet, which face is the link arc. Having already found that the inequality in the motion of the piston is greatest when the crank pin is at half stroke (the crank pin and the eccentrics being rigid), it therefore becomes necessary to locate a position for the saddle pin, so that equal portions of the steam will be admitted alternately, and the cut-off be equal when the crank pin is at half stroke. Therefore, draw the center line of motion and locate the positions of the eccentrics when the crank pin is at half stroke, points T and W, as previously explained, and shown in Fig. 77. Next locate the rocker arm N O, and from the center of the rocker arm L describe the arc D E; now, with the correct length of the eccentric rods, and from their re-

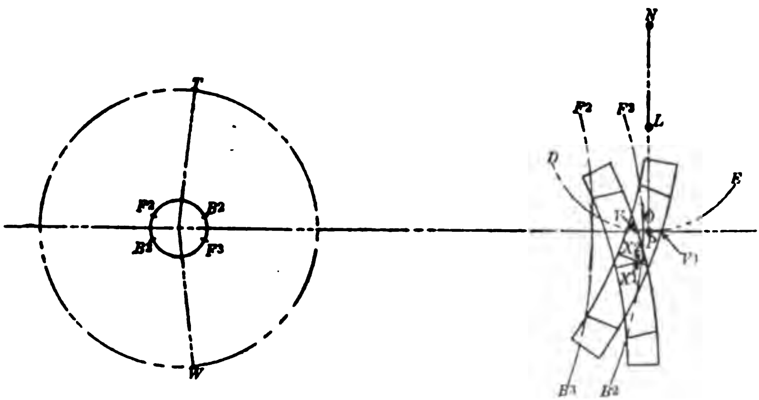


FIG. 77.

spective points, as explained in Fig. 76, describe the four link pin arcs are F, F¹, B and B¹. The forward arcs are above the center line of motion, and the back motion arcs below it. Now, when the rocker arm stands in a vertical position, the valve stands central on the valve seat, and we know that the valve has ⅞" outside lap (as previously explained); therefore the bottom of the rocker arm must move ⅞" either way from its central position to open the valve, or to reach either point of cut-off (assuming that the rocker arms are of equal length); therefore, from the point P, which indicates the center of the hole in the lower rocker arm, mark the two points V and V¹, on the arc D E, which represents the path through which the center of the lower rocker arm must travel. Now place the point U on the templet on the arc F^2, and the point Z on the arc B^2; then move the templet along those lines until its front face, R S, Fig. 75, touches the point marked V. The templet now represents the position of the link when the crank pin is at T, and we know that when the engine is moving forward the valve must have reached the point of cut-off at the back port when the crank pin has reached this position.

Now transfer the line M N from your templet onto the drawing, for future use. Again place the point U of the templet on the arc F^3, and the point Z on the arc B^3; then move the templet along these lines until its front face touches the point V^1. The templet now represents the position of the link when the crank pin is at W, and we know that, in the forward motion, the valve must at this point be cutting off steam at the forward port; to that once more we transfer the line M N from the templet onto the drawing. With a length equal to the distance between the lines R S and K L, and on the same line, M N, Fig. 75, and from the forward face of each templet, and on the line M N, lay off the two points X^1 and X^2. If a straight line drawn through these two points is parallel with the center line of motion, these points indicate the correct position of the center of the saddle pin. If not, by trial locate two points that are equally distant from the front faces of the templets, and that are also parallel to the line of motion; these points will be the correct position of the saddle pin. When the correct position is found, mark it on the templet for future use.

Relative Position of the Tumbling Shaft and Rocker, and Length of the Tumbling Shaft Arms.

The saddle pin is usually located in such a position as to obtain an equal cut-off at half stroke, where the irregularities introduced by the crank pin are greatest, and the tumbling (or lifting) shaft is located in a position to obtain an equal amount of lead in full stroke. Owing to the irregularities of crank motion, it is impossible to obtain an equal lead, and an equal cut-off, at all points: If one is equal then the other will not be; this is one of the imperfections of link motion. But the difference is so slight in full gear that equal cut-off is considered of less importance than an equal amount of lead at the beginning of each stroke, therefore the tumbling shaft is located and the length of its arms determined, in such a manner as to obtain the latter result. In making Fig. 78 we must combine all the foregoing problems. Draw the center line of motion, locate the eccentrics at full and half stroke, locate the rocker arm N P, and, from its center, L, describe the arc D E. Now, from the position of each eccentric, and with the exact length of the eccentric rods, describe each link pin arc, as shown in Fig. 76, making all the forward arcs above the center line of motion and all the back motion arcs below it. Locate the points X^1 and X^2, as explained in Fig. 77, which indicate the points of suspension when steam is cut off equal at half stroke. Mark these two points on your drawing, Fig. 78. Now, to determine the length of the tumbling shaft arms, and the position of the tumbling shaft, we must find the points of suspension in full forward, and full backward, strokes, at each end of the cylinder. The valve has ⅞" steam lap and 1/16" lead, therefore add these figures together and locate the points V and V^1 on the arc D E, 15/16″ from the point marked P. Place the templet on the drawing, with the line M N below the center line of motion, place the point of templet marked U on the arc F, and the point marked Z on the arc B, and move it along these lines until its front face touches the point V^1, and then mark the point X^3 on the drawing; this point will indicate the position of the saddle

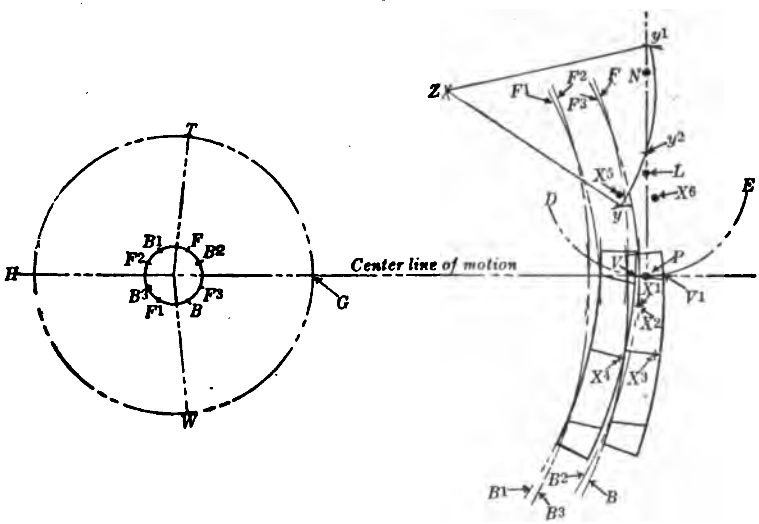


FIG. 78.

pin when the crank pin is at G, in full gear, forward motion. Again, place the templet on the drawing with the line M N below the center line of motion, place the point U on the arc F^1, and Z on the arc B^1, and move the templet until its front face touches the point V; now mark the point X^4 on the drawing. This point indicates the position of the saddle pin when the crank pin is at H, in full gear, forward motion. We must now find the points X^5 and X^6 in exactly the same manner, and using the same link arcs as before, except that the line M N of the templet must be above the center line of motion. We have not outlined the templet in these positions, as it would make the drawing appear more complex, but we shall continue the explanation. Place the point U on the arc F, and Z on the arc B, front face of the templet touching the point V^1. Now mark the point X^6. Place the point U on the arc F^1, and Z on the arc B^1, letting the front face of the templet touch the point V. Then mark the point X^5 on the drawing. These two points, X^6 and X^5, indicate the positions of the saddle pin when the piston is at each end of the cylinder, in full gear, backward motion. Now, from the two points X^3 and X^4, with a radius equal to the length of the link hanger, find the point marked Y, from the points X^5 and X^6, with the same radius, find the point Y^1. and from the points X^1 and X^2, with the same radius, find the point Y^2. Now, from these three points, Y, Y^1 and Y^2, find the point Z on the drawing, which indicates the position of the center of the tumbling shaft, and the distances from the point Z to the three points, Y, Y^1 and Y^2, equal the lengths of the tumbling shaft arms. The length of the top arm of the tumbling shaft is not so important, as it is made to suit other details of the engine.

ERRORS OF THE LINK MOTION.

The link motion, in actual practice, has three inherent sources of error, which tend to cause cut-off, release, and compression to occur at different points in the stroke of the piston, at the two ends of the cylinder. In order of importance, these sources of error are: the placing of the eccentric rod pins behind the link arc, the angular vibration of the eccentric rods, and the angular vibration of the main rod. This last source of error, however, has been shown to be a corrective element in the link motion, rather than a disturbing factor. Shortening the connecting rod reduces the error, and reduces the necessary offset of the hanger, or saddle, stud in the link, which offset has, for its object, the hanging of the link so as to give it a rising and falling motion on the stud. This will counteract the difference in motion between forward and backward strokes. If an adjustable stud be provided, its location is best determined by actual trial upon the engine.

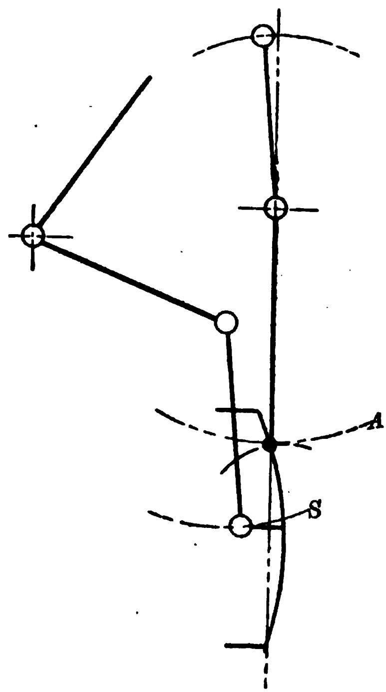


FIG. 79.

As we have said, the connecting rod has a corrective effect upon the other errors, and so reduces the necessary offset of the saddle stud. The remaining offset is to correct for the error due to the location of the eccentric rod pins. The final location of the saddle stud, in locomotive practice, is always behind the link arc, as shown in Fig. 79.

Errors Due to Angularity of Main Rod.

As just explained, the offset of the saddle stud is made to compensate errors, and to bring the cut-off at the same point in both strokes. Of these errors, we shall first study that due to the angularity of the connecting rod.

In Fig. 80 we have shown a skeleton representation of the crosshead pin, the connecting rod, and the crank pin, in their relative positions. Due to the angularity of the main rod, when

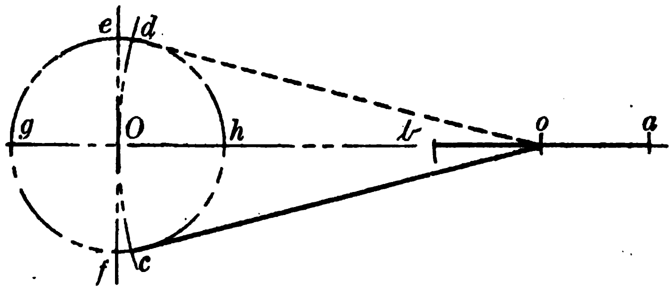


FIG. 80.

the rod is swung through an arc, passing through the center of the shaft *O*, with the crosshead pin at the center of the stroke *o*, the crank pin will cut the circle at the points *c* and *d*, rather than at the true quarter positions, *e* and *f*. The result of this error is that during the forward movement of the crosshead pin from *b* to *a*, the crank runs ahead of its correct position, while during the backward movement of the piston (when the crosshead pin travels from *a* to *b*), the crank lags behind its correct position. These errors exist throughout the stroke, being greatest at the quarter, when the variation is the amount *e d*, or *f c*, and least at dead center, when the crosshead pin is at *a* or *b*, and the crank pin at *h* or *g* (the error here is reduced to zero).

Of course, due to the fact that the eccentrics and links are driven by the motion of the crank pin, this error will be introduced into the movement of the valve, and during the backward movement of the piston, the events (cut-off, release and compression) at the front port, will occur late; assuming that the other two sources of error do not exist; and during the forward stroke, the events at the back port will be early.

To correct for this angularity of the main rod, set the saddle stud *outside,* or *ahead* of, the link arc.

Errors Due to Angularity of Eccentric Blades.

The action between eccentrics and links is much similar to that between crank and crosshead, with this important difference. With the crank and crosshead, the error is zero at dead center,

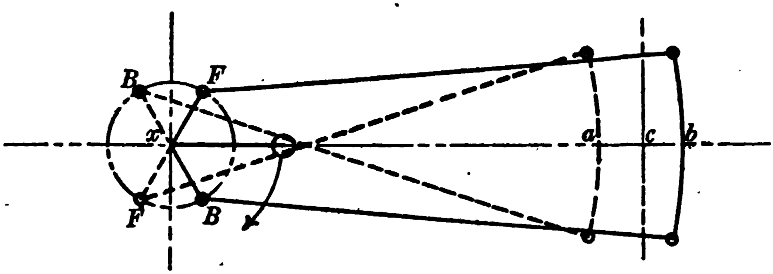


FIG. 81.

and greatest at the quarter, as shown in Fig. 80, but this is not true of the eccentrics and links. The maximum error, here, is at, or about, 180° rotation, where the error before was at its minimum.

This change in angle of the eccentric rod is shown in Figs. 81 to 84. In the former, the full lines represent the smallest angle between the eccentric rods and the center line of the axles, and in the latter, the full lines represent the greatest angle.

It may be seen, by referring to the figures, that from the position of the smallest, Fig. 81, to that of the greatest angle, Fig. 83, the crank pin and eccentrics have rotated through one-half of a revolution. Therefore, the error due to the angularity of the rods varies from the smallest to the largest amount during half a revolution, or while the crank pin passes from one dead center to the other. This changing angle of the eccentric rods causes the link movement to be variable, just as the angularity of the main rod caused a variable motion of the crosshead.

With the crank pin on forward dead center, the distortion of the link motion, due to the angular motion of eccentric blades, during one complete revolution, is shown in Figs. 81 to 84, which show the relative positions of the crank pin and eccentric rods, in quarter turns, for one revolution.

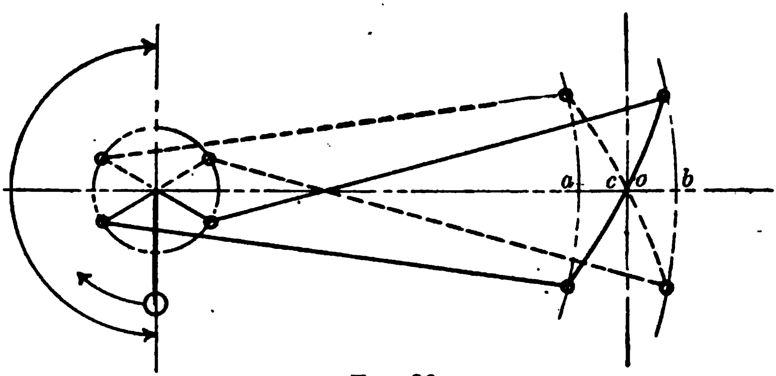


FIG. 82.

Omitting the main rod errors, Fig. 81 shows the pin on forward dead center. The radius of the arc of the link is *x c,* and the location of the link arc when its center is on the center of the axle, is shown at *c.* The center of the link, for this position, is shown at *b.* In Fig. 82, the relative positions are shown when the crank pin and eccentrics have rotated through one quarter turn. Here the link center has moved only a slight amount, represented by *b* and *o,* in the full line diagram. That is, the link has moved back from point *b* to point *o.* In the next figure, the parts are advanced a quarter turn, completing one half revolution, or a full stroke of the piston, and the crank pin is on back dead center. Here the link has moved a much larger amount, from *o* backward to point *a*, showing clearly that, during this second quarter, the movement of the link has been much faster than during the first quarter.

The position of the parts, on top quarter, is shown in Fig. 84. Here the link center has moved through the exact distance it moved while the crank pin turned through the second quarter of a revolution, and has returned to the point *o*. Referring once

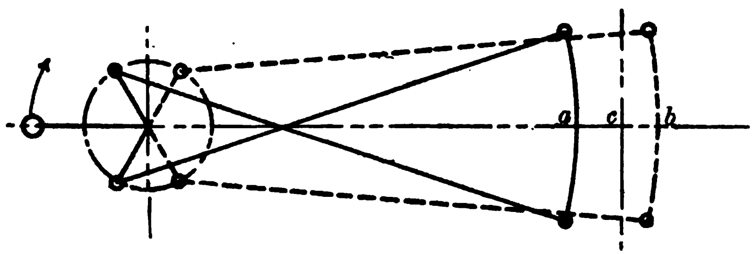


FIG. 83.

more to Fig. 81, when the crank pin has returned to forward dead center, and completed one revolution, the link has moved from *o* forward to *b*.

It is evident, now, that the slow travel of the link occurs while the crank pin moves from top quarter to bottom quarter (the fourth and first quarter turns, running forward), or while it is ahead of the vertical center line through the axle. This slow motion is the effect of a comparatively small change in the angle of the eccentric blades, while they are open. The faster link travel occurs during the movement of the crank pin from bottom to top quarter (second and third quarter turns, running forward), and while it is ahead of a line running vertically through the axle. This fast movement is produced by a comparatively large change in the angle of the eccentric blades, while in crossed position. An increase in the angle of the blades, therefore, causes a greater error due to angularity, as follows:

The movement of the link is slow while the crank pin turns through the front half of its path, and, while the piston is in the forward half of its stroke (either direction), and while the crank pin turns through the back half of its path, the movement of the link is fast. In othr words, the valve events occur *late in front,* and *early behind.*

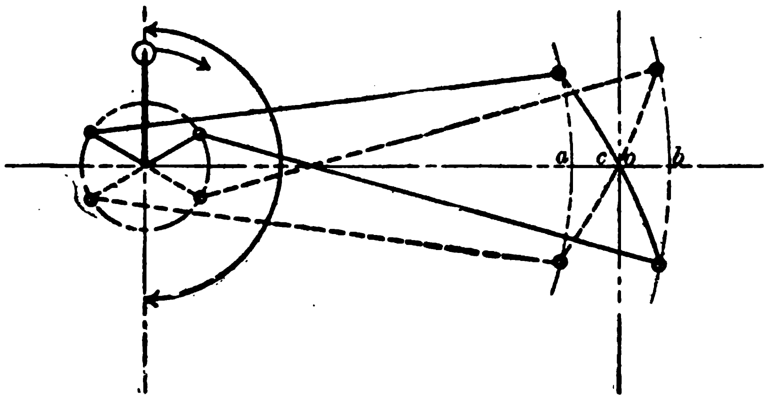


FIG. 84.

This error, also, is corrected by offsetting the saddle stud *ahead* of the link arc. This correction necessitates a greater amount of offset than did the error due to the angularity of the main rod. It is evident from the diagrams that the error is more pronounced with short blades than with long, for the changes in angle are larger with short blades.

The Error Due to the Location of the Eccentric Rod Pins Back of the Link Arc.

In the previous study of the movement of the link, the eccentric rod pins were assumed to be located in the link arc, and in previous discussions of the subject it has been tacitly assumed that the errors introduced by setting these pins back of the arc are so small as to be negligible. This, however, is by no means the case, this error being, in fact, by far the most important of

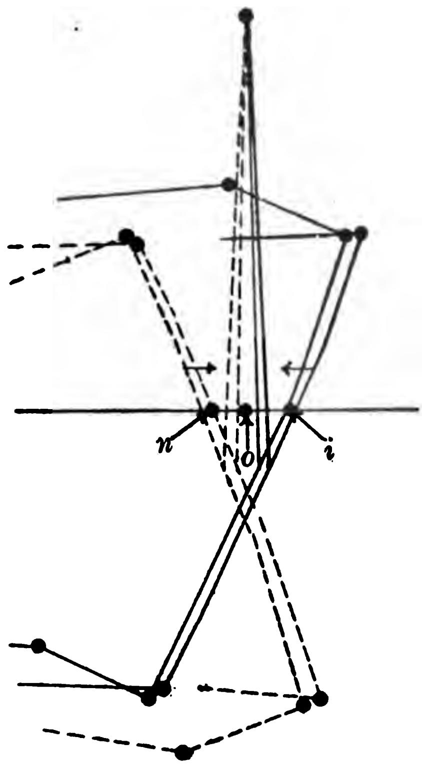


FIG. 85.

the three, over-correcting, as it does, both of the others, and resulting in finally locating the saddle stud inside of the link arc instead of outside, where the previous errors alone would place it. This error, like the others, may be best studied by insulating it so far as possible, although it is not possible to separate it from the eccentric rod error, as will be seen. It is, however, possible to separate it from the connecting rod error. The nature of the error may be seen from Fig. 85, which shows both forms of link, one having the eccentric rod pins located in the arc, and the other having these pins located three inches back

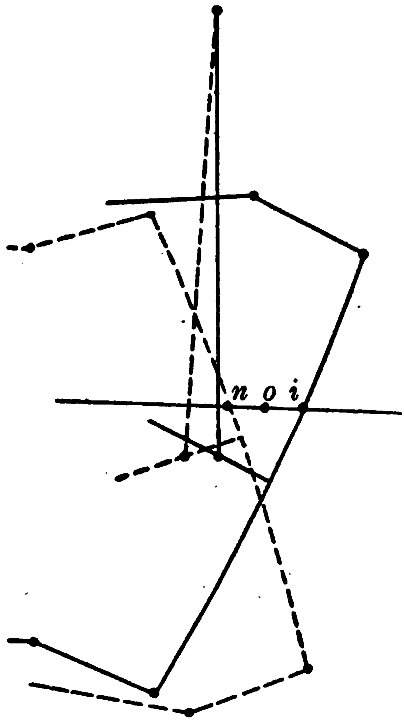


FIG. 86.

of the arc, as is customary. The eccentric rods for the former link are, of course, three inches longer than the latter. The saddle stud is located over the center of the link arc, as is shown in the diagram, and the links are shown approximately in the positions which they would occupy for a cut-off at one-third stroke, the full line links being in position for the rearward stroke of the piston, and the dotted line links being in position for the forward stroke. The movement of the crank is supposed to be by a Scotch yoke, so that no connecting rod errors are introduced. It will be seen at once that the setting of the eccentric rod pins back of the link arc makes the lines joining the extremities of the arc and the centers of the eccentrics crooked, whereas with this pin located on the arc, this line is, of course, straight; consequently the effect of placing these pins back of the arc is, for the positions shown, to draw the link having the offset pins nearer the shaft than the link which has the pins on the link arc. The action is that of a knuckle joint, any bending of which must draw the link toward the shaft. The rock shaft reverses this action on the valve, so that the drawing of the link toward the shaft pushes the valve away from the shaft. Pushing the valve away from the shaft quickens the cut-off for the front port, or rearward stroke, and delays it for the rear port, or forward stroke; that is, the effect of the offset pin is to make the cut-off too early in the rearward stroke, and too late in the forward.

This effect will be seen to be the direct opposite to those produced by the connecting, and eccentric, rods, and it obviously calls for an adjustment of the saddle stud in the opposite direction, as shown in Fig. 86, which shows the position of the saddle stud necessary to equalize the cut-off at one-third stroke, with the Scotch yoke connection, the stud being on the concave of the link, where it is in all cases located in actual engines.

The Final Offset.

It is obvious that the final offset of the stud is the resultant of all three. The offset of the eccentric rod pins varies within narrow limits only, but the length of the eccentric rod varies within wide limits. It is obvious that since the error of short rods alone would place the stud farther outside the link arc than long ones, they subtract more from the offset of the stud due to the eccentric rod pins than long ones, the result being that

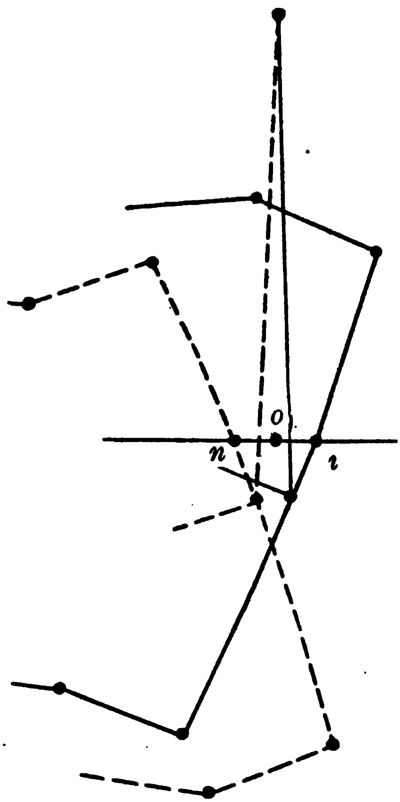


FIG. 87.

the final offset is less with short arms than with long ones; and this is found to be the case, the offset in actual engines ranging between about five-eighths of an inch and one and a quarter inches, depending upon the length of the eccentric rods. The connecting rod also varies in length, but its influence is so small that the variation in its length between usual limits has but little effect upon the final result. To illustrate this, Fig. 87 has been constructed, in which the connecting rod has been shortened by trial, until the offset of the saddle stud has disappeared, placing that stud over the link arc. With the other proportions unchanged, it has been found necessary to shorten the connecting rod to a remaining length of three feet before this result is accomplished. In the diagram in which the saddle stud is located, the following proportions have been used:

Stroke of piston, 24 inches.
Length of connecting rod, 91 inches.
Radius of link arc, 69 inches.
Length of link between pin centers, 12 inches.
Offset of eccentric rod pins, 3 inches.
Travel of the valve, 5½ inches.
Lap, ⅞ inch.

VALVE SETTING.

Introduction.

The operations involved in setting the steam valves of a locomotive have long been regarded by the average shop man as a subject of mystery, too complicated to be understood by any one except the expert valve setter, yet there is nothing mysterious about them. Any man of ordinary intelligence may, by reasonable application to the subject, become familiar with the necessary operation of valve setting. Of course, to master the principles thoroughly requires more diligent study, and experience in the work, than some might at first suppose. We shall endeavor, however, to explain the various operations generally employed so clearly and minutely as to enable the shop man to gain a good practical knowledge of the proper method of setting locomotive valves.

Object of Valve Setting.

If all the details of a valve gear could be made and assembled exactly as designed, the operation of "valve setting" would be unnecessary. In practice, however, such accuracy cannot be obtained; and after the gear has been assembled some adjustment is usually required.

The object to be accomplished is to adjust the valve gear parts so that each valve will be in such relation to its piston that, when steam is admitted to the steam chest, it will also be permitted to enter one end of at least one cylinder, and thereby start the engine in motion. The movement of the valves must be such as to cause the driving wheels to revolve continuously in one direction until the motion of the valve gear is reversed;

and when the valve's motion is reversed, the wheels must revolve continuously in the opposite direction. This is the elementary principle of valve setting; but it is not all that is accomplished.

Preparations for Valve Setting.

In practically all modern shops, the driving wheels are set on rollers, operated by compressed air, or of the ratchet type, which makes it an easy task to rotate the drivers; although the practice of "pinching" the entire engine forward and backward by means of crowbars, is sometimes used. In either case, the method of setting the valves is the same, the only difference being that, by the first method, the work may be accomplished much more rapidly and economically, than by the "pinching" process.

Before commencing the task of adjusting the valves, it is advisable to see that you are provided with a valve tram, a crosshead tram, a wheel tram, a large and a small pair of dividers, a prick punch, a two-foot rule, and such other tools as you may think necessary. As your experience increases, the number of tools required will decrease in proportion.

The next step should be to determine the type of valve you are to adjust, for, as previously explained, the outer edges of a slide valve are the working edges, which control the admission and cut-off, while the opposite is generally true of piston valves.

The practice herein described may differ in detail from the method followed in some shops, but it has been found to be satisfactory, and is one that is generally followed.

In setting valves, it is advisable to follow a certain definite program, adjusting each part in its logical order, and that plan will be followed here.

Regardless of the type of valve gear, and assuming that we are to work on a "cold engine," equipped with slide valves, the first thing to be done, in all cases, is to have the port openings properly located on the valve rod. An explanation of the method of marking the port openings, for both inside and outside admission valves, should, therefore, open our discussion of the subject.

How to Mark the Port Openings of Outside Admission Slide Valves.

Points of Admission and Cut-Off.

After ascertaining that the valve is connected to the valve gear, without lost motion, and without cramping or twisting of the

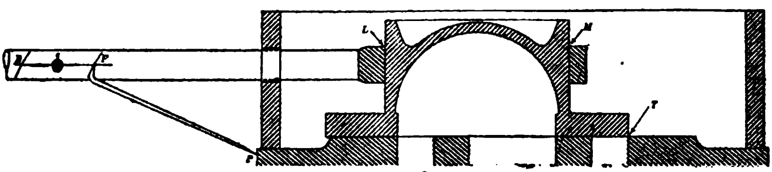


FIG. 88.

valve yoke, see that the valve is square on its seat, so that its steam edges are exactly parallel with the edges of the steam ports so as to cause admission and cut-off to occur precisely at the same time for the entire length of the port.

Now, with the steam chest cover removed, and the ports exposed, if there is any lost motion between the valve and the valve yoke, place liners between the back of the valve and the yoke, as indicated by the letter L, Fig. 88, and then move the valve until you can just insert a thin piece of tin between the forward edge of the valve and the front edge of the steam port, as indicated by the letter T. The valve is now cutting off the front port and is in position to mark the valve rod with a tram, one end of which is set in a prick punch hole at any convenient point, such as P, on the back wall of the steam chest, so scribe the point F on the valve rod. Then remove the liners and place them between the forward end of the valve and the valve yoke, at M. Now move the valve forward, until the piece of tin will just fit between the rear edge of the back port and the rear edge of the valve. With the tram set at the same point P in the cylinder, again mark the valve rod, at B. A horizontal line may now be drawn along the center of the valve stem, and its intersection with the arcs scribed by the tram will give the exact port marks B and F. The center F is the front port mark, and the center B is the back port mark, and the distance between the points F and B will be equal to twice the lap of the valve. A point midway between the two arcs, c, will indicate the central position of the valve, provided the front and back steam laps are equal, which is usually the case in locomotive practice. It is the usual practice to set the valve, however, after the steam-chest covers have been bolted into place.

Points of Release and Compression.

The exhaust port marks can be obtained in the same way, but, as the exhaust edges are invisible when the valve is on its seat, they are obtained by locating a center, c, exactly half way between the two port marks, and then scribing the exhaust port marks a distance from this center equal to the exhaust clearance, or exhaust lap, as the case may be. To do this, set a pair of dividers at the amount of exhaust clearance, or exhaust lap, and scribe a circle on the valve rod, from the center c, and its intersection with the horizontal line will mark the points of release and compression. If the valve has neither exhaust clearance nor exhaust lap, the point c will represent both release and compression.

How to Mark the Port Openings of Inside Admission Piston Valves.

The method of setting inside admission piston valves is generally similar to that previously described. When setting piston valves, the steam chest heads should be removed, for the sake of convenience. The piston valve is not in plain sight, however, so its line and line positions T T, Fig. 89, are generally determined by observation through peep holes, as H, provided for the purpose. By inserting a very thin strip of metal in this peep

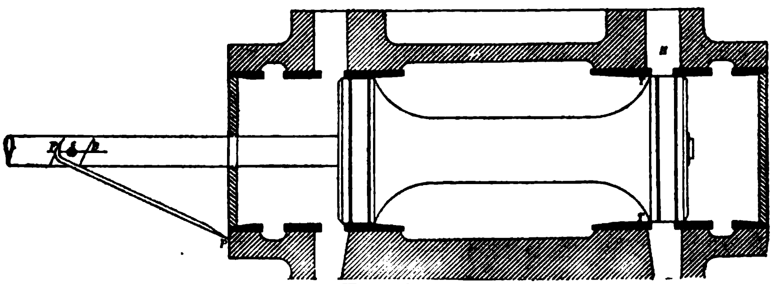


FIG. 89.

hole, and allowing it to come between the edge of the port and the edge of the valve ring, the valve may be gently tapped into position of exact port opening or closure. Thus, Fig. 89, with a feeler inserted at H, and between the valve and the port at T, the valve may be set so that it just closes the right hand port. In this way, the front and back port openings are located on the valve stem, by tramming from any convenient point, as P, on the back wall of the steam chest.

These openings may also be determined in another way. The distance between the face of the valve chamber and the edge of the port is known. As the head of the chamber is removed, a rod of iron or steel may be inserted longitudinally into the chamber, bearing against the outside ring of the valve. Then move the valve, and the metal rod with it, until the length of rod inside the chamber, measured from the face of the chamber, is equal to the distance between the face of the chamber and the edge of the inner port, minus the distance between the working edge of the inside ring of the valve to the outside edge of its valve ring.

It must be remembered, however, that to perform corresponding functions of the slide valve, this valve moves in a direction opposite to that of the slide valve. It will also be observed that, with the slide valve, the forward port mark F is the front and the rear port mark B is to the rear, while the locations of the port marks, and the letters indicating the same, are reversed with inside admission valves.

How to Mark Port Openings of Outside Admission Piston Valves.

With valves of this type, the arrangement is the same as that used with outside admission slide valves, and the method of setting is the same. The line and line positions of the valve, however, must be observed through peep holes, or by measurements, as in the case of inside admission piston valves.

Locating Dead Centers.

It is common practice to use the old markings on the drivers, when tramming the valve rod. But this should not be done, as the main and connecting rods have loosened, frames have dropped, due to relaxation of springs, and lost motion has developed, thereby introducing error. It is recommended that new dead center marks be found on the wheel and crosshead.

Place the crank on the dead center. The term implies no turning effort, because positive piston pressure can cause no rotative impulse, and is the position of the driving wheel when a straight line from the center of the wrist pin in a crosshead, through the center of a driving wheel, intersects the center of a main pin (crank pin), as shown in Fig. 90. In this position a crosshead is at one extremity of its stroke, therefore there must be two dead centers, as the illustration shows, one the "for-

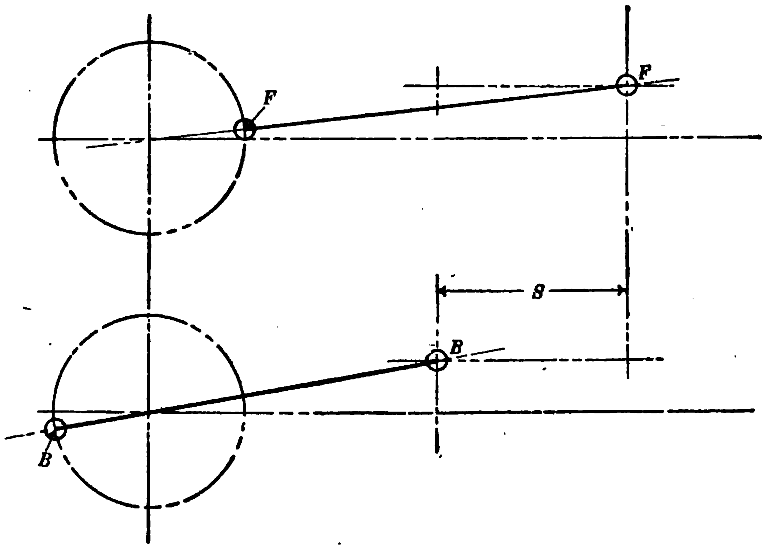


FIG. 90.

ward" center, and the other "back" center. But, as there are a "forward," and a "back," center on each side of the locomotive, there are, in all, four dead centers.

For the sake of convenience, we shall first find the right forward, or front, dead center. The method of finding this position is as follows: With the driving wheel connected to the crosshead, move the crank forward by revolving the driving wheel until the crosshead reaches a point about one inch from the end of its stroke, as shown in Fig. 91. With a crosshead tram, set on a point A, on the front guide block, scribe an arc B on the crosshead, to mark this position. At the same time, without changing the position of the crosshead, from some convenient stationary point on the wheel cover or frame, such as the point P, scribe the arc X on the face of the tire. Continue to revolve the wheel, as indicated by the arrow, until the crosshead passes the end of the stroke t and begins its backward movement, stopping it when the tram previously used reaches exactly from the point A on the guide to the point B on the crosshead, as shown in Fig. 92. Then re-tram from the point

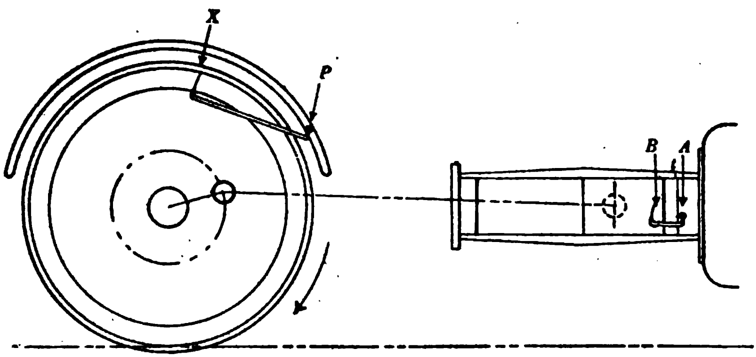


FIG. 91.

P to the tire, using the same tram as in Fig. 91, and, with the wheel tram set at the point P, scribe another arc Y on the face of the tire, as shown in the illustration.

Now, with a pair of hermaphrodite calipers set about one-half the thickness of the tire, draw a line r s parallel to the wearing face of the tire so as to cross both of the arcs made with the wheel tram at the points X and Y. With a pair of dividers, carefully bisect the line r s, between the points X and Y, and scribe and prick punch the exact center between X and Y, as shown at Z. This point is the dead center mark, and the engine crank is at right front dead center when the wheel is revolved until the tram reaches exactly from the point Z to point P. Both the front and back dead center positions must be determined in the same way, except that, when locating the back centers, the crosshead tram is used from the back guide block, or rear end of the guide.

The effect of measuring on the crosshead is to find a point where the crosshead travel (from the end of the stroke) is the same, both when approaching, and when leaving, the end of the stroke. By transferring these points to the driving wheel and dividing them to find the central point, the actual end of the stroke, or dead center, is determined absolutely.

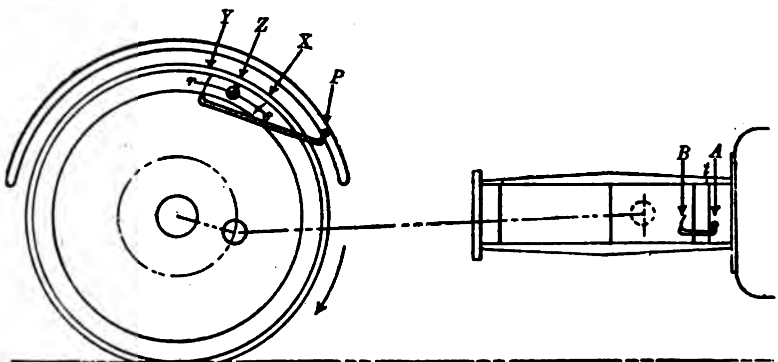


FIG. 92.

To obtain the quarter positions, set the crosshead on the forward dead center and measure back on the guides one-half the piston stroke. Then revolve the wheel forward until the crosshead reaches the half stroke position marked, and, with the wheel tram, obtain the quarter position by tramming from the point P to the tire. The opposite quarter stroke position may be obtained by the same method, preferably revolving the wheel in the same direction so that the influence of lost motion in the parts may be minimized.

The four important points on the stroke are now definitely established on the wheel, and are ready to be used for valve setting.

Setting Outside Admission Slide Valves.

The reader now understands what the port marks indicate, and how to locate the four dead centers, and will experience little difficulty in grasping the method of setting the valves. As the method of setting the valve varies with the type of valve gear employed, we believe it will simplify matters to assume that our valve motion is controlled by the Stephenson valve gear. The operations of adjusting the valve when controlled by other valve gears will be treated later on.

There are many points to be considered, and comprehended, throughout the operations of adjusting a valve controlled by a Stephenson valve gear, which could not well be explained here, so the reader will find it advisable to first make a perusal of them under the chapter on "Rules for Valve Setting," where they are grouped in systematic order; and then proceed on the subject of valve setting.

Trying the Lead.

First, place the engine on dead center, at whatever center may be most convenient. We shall assume the right front center. With the engine in this position, it will be observed that the crank pin is about six inches below its dead center. Since, as the crank pin approaches its dead center, the wheel must turn backward, the valve gear must be in full back motion—therefore, place the reverse lever in the back notch of the quadrant (refer to Rule 23). Move the wheel back, and, with the wheel tram, catch the dead center. Then, with the valve tram, beginning at or above the horizontal line on the valve stem, scribe an arc extending below the line considerably.

The engine is now on dead center on the right side, and the extreme travel of the crosshead is designated by a line scribed on the guides at the front end of the crosshead. Place the reverse lever in the forward notch, and move the wheel backward until the valve stem is seen to move; now all the lost mo-

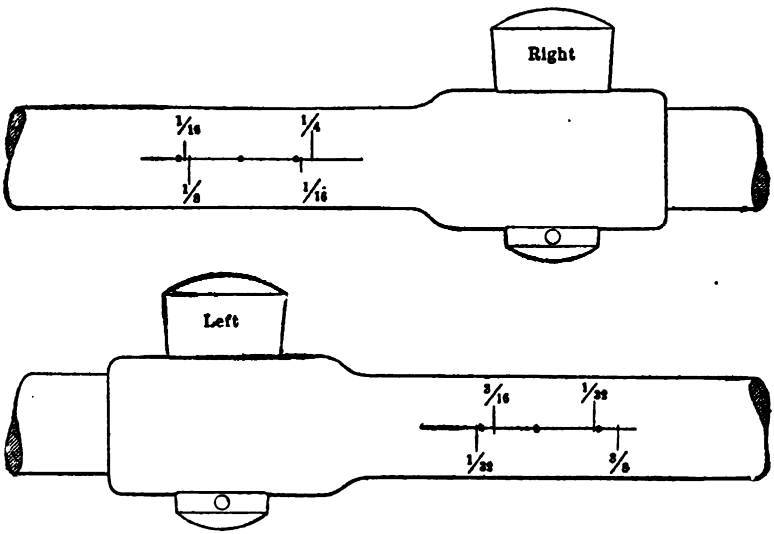


FIG. 93.

tion in the valve gear has been taken up. (Rule 23.) Turn the wheel forward, and once more catch the dead center with the wheel tram; then mark the valve stem with the valve tram, this time *above* the horizontal line of the valve stem.

These marks give the lead at the right front port, on right front dead center for the forward and the backward motions in full gear; go to the left side of the engine, turn the wheel and find the left front center, and repeat the operation, marking the valve stem for each motion, just as previously done. Do not fail to mark the travel. Return to the right side and obtain the back center, repeat the operation, and go to the left side once more; find the left back center and again mark the valve stem and crosshead. The engine has now been run over once, and it is now easy to see what changes must be made.

Making Alterations.

In explaining this subject, we shall assume marks and figures corresponding to the tram marks made above on the valve stems, as shown by Fig. 93. We shall also assume a desired lead in full gear, both forward and backward motion, of 1/32". Commence by examining the two forward motion tram marks on the right side (those above the horizontal line on the right valve stem); we find ¼" lead at the forward port mark, and 1/16" lap at the back port mark. (Reference to Rule 25 will aid in distinguishing between lap and lead.) Since both of these marks were made while in the forward motion, the length of the right forward motion eccentric blade must be altered, so as to equalize these tram marks at each end. To determine whether to lengthen or shorten the blade, and the amount of change, see Rule 26. It is evident that the blade must be shortened 5/32", or one-half the sum of the lap and lead, 5/16"; so mark the *R F* eccentric blade thus—(shorten 5/32")—*but make no changes until all of the tram marks have been examined, and all of the blades have been marked with their necessary alterations.* Before proceeding, it might be well to ascertain what effect the above change will make in our tram marks. When an eccentric blade is shortened, the rocker arm will, of course, force the valve stem forward. Assume that the valve tram be set to the forward tram mark on the right side, and held stationary while this change in the length of the eccentric blade is made. The result, it is very readily seen, is that the point of the tram is 5/32" closer to the forward port mark. And if, while the change is being made, the valve tram be set to the back port mark, the point of the tram will extend 3/32" back of the back port mark, since there is but 1/16" lap at this point, while the valve stem has moved 5/32". We shall now see how the port marks will come after this change. By subtracting, from the ¼" lead, 5/32", we have still 3/32" lead, at the forward port opening. Then, subtracting the 1/16" lap at the back port mark from the 5/32" change, we have 3/32" remaining—the lead at the back port opening; we therefore have 3/32" lead at each end.

However, we have 3/32" lead at each end, whereas we require but 1/32" lead of the valve in full gear, when the valves are set. The amount of reduction of lead must be, therefore, 1/16". Mark on the frame, or on the firebox, with chalk, "R. F. Ecc. 1/16" lead off." Next, examine both back motion tram marks on the right side (those below the horizontal line on the valve stem). We find here 1/16" lead in front, and ⅛" lap behind; by carefully observing Rules 25 and 26, it is evident that the right back eccentric blade must be shortened by one-half of the sum of the lap and lead, 3/16", or 3/32", which will give 1/32" lap at each end. But we require 1/32" lead at each end, and must therefore give the right back eccentric 1/16" more lead, in order to overcome the lap, and to have the required amount of lead. Therefore, mark "R. B. Ecc. 1/16" lead on."

Both left forward motion tram marks should now be examined. Here is found 3/16" lap in front, and 1/32" lap behind; from Rule 26) it is evident that the blade must be lengthened 5/64", or one-half the difference between the lap in front and the lap behind, 5/32", giving 7/64" lap at each end. To obtain 1/32" lead, rather than 7/64" lap, at each end, we must give the eccentric 9/64" lead; mark, therefore, "L. F. Ecc. 9/64" lead on."

Next try the two left back motion marks. Having 1/32″ lead in front, and 3/8″ lead behind, we must, according to Rule 26, lengthen the left back eccentric blade by one-half the difference between the front and back leads, 11/32″, or 11/64″, giving 13/64″ lead at each end. This is more than the required amount, as we want but 1/32″ lead, and should be marked, "L. B. Ecc. 11/64″ lead off."

By now altering all of the eccentric blades the amount indicated, as marked above, the valves are properly set with reference to lead (see Rules 2, 26 and 27), and we are all ready to try the cut-off.

Trying the Cut-Off.

Were the Stephenson link motion a perfect valve gear, this operation would be unnecessary, for if the valves were "square" in full gear (corner notch), they would be square always; that is, with the reverse lever in any other notch in the quadrant. But, due to the inherent errors of the link motion, previously explained, this is not the case, and equalizing the cut-off at one position only equalizes the events for that position, and the valve is out of square with the lever in any other notch. It is best, therefore, to have the steam equalized in the running cut-off—the position where the locomotive does most of its work, and this point naturally varies in different kinds of service.

Therefore, the cut-off should be equalized with the engine "hooked up," in her "working notch," rather than "square" in the corner notch, and lame when hooked up.

The working cut-off is usually taken at about twenty-five or thirty per cent of the piston stroke, so that the cut-off should be tried at about six or eight inches of the stroke, for passenger locomotives, and seven to ten inches for freight locomotives.

For purpose of explanation, we shall assume a passenger locomotive, 24" stroke, cutting off at 25%. We shall, therefore, try the cut-off at 6". Move the wheel *forward* until the right main crank pin has passed front dead center, and, as the crosshead recedes from the extreme forward travel mark, measure the distance from this mark to the front end of the crosshead. Set the engine so that this distance measures six inches; place the reverse lever in the front notch, draw it back slowly, until the valve tram corresponds to the right front port mark, and place the lever in the nearest notch of the quadrant. This is the position of the reverse lever for cut-off at 25%, or 6".

Now turn the wheel *backward* until the right valve stem begins to move, signifying that all the lost motion is taken up. Again move the wheel forward until the valve tram is true with the right forward port mark, and measure the distance between the port mark and the front end of the crosshead. Let us assume that it is now 5¾", mark this amount on the forward end of the right guide bar, thus: 5¾".

The reverse lever is left in the same notch throughout. On the left side of the engine, now, move the wheel forward until the left main crank pin has passed the forward dead center, and, as the crosshead moves back from the travel mark, catch the left front port mark with the valve tram—then measure the distance from the crosshead to the travel mark (front). Indicate this measurement on the front end of the left guide bar, as: 6¼" (assumed).

Returning to the right side, move the wheel forward until the right back dead center is passed, and catch the right back port with the valve tram. Measure the distance between the back travel mark and the rear end of the crosshead, recording the amount, say 5¼", on the back end of the right guide bar. Again, repeat this process, on the left side, recording the distance on the back end of the left guide bar, as 6½". We shall now ascertain what alterations are necessary to be made to equalize the cut-off at 6", or twenty-five per cent.

Changes to Be Made in the Cut-Off.

Although the lead, in full gear, was found to be perfect, we see, by trying the cut-off, that the steam is not perfectly equalized in the running cut-off. This is due, as has been explained, to the errors peculiar to the shifting link motion. However, any lost motion, or imperfect construction of the parts of the valve gear, will necessarily seriously affect the cut-off, and in order to thoroughly understand the effects of various changes that may be made to equalize the cut-off, the reader will be greatly aided in studying the "Rules for Valve Setting," more particularly numbers 28, 33 and 36, as well as 16, 17, 18, 34, 35, 37, 38 and 40.

By referring to our figures recorded on the guide bars, we find that, on the right side, steam is cut-off at 5¾" in front, and at 5¼" behind—a difference of only one-half inch. This amount is considered slight in ordinary practice, and, if no defects are located by reference to Rule 33, we must sacrifice perfect equalization of steam in full gear in order to equalize the cut-off. This is accomplished by shortening the right forward eccentric blade 1/32", as there is a difference in the points of cut-off of ½", as stated.

When the difference in the cut-off at both ends is slight, as in this case, you can tell whether to lengthen or shorten the blade by observing Rule 40. On the left side, we find the engine cuts off steam at 6¼" in front, and at 6½" behind; the difference we find to be only ¼"; according to Rule 40 we must lengthen the left forward eccentric blade 1/64". Now, compare the cut-off on both sides of the engine, to find out which side cuts off steam quickest; to do this, add the amount of cut-off at both ends on each side. On the right side we have 5¾"+5¼"=11", total amount of cut-off on the right side. On the left side we have 6¼"+6½"=12¾", total amount of cut-off on the left side. By this process, we find that the right side cuts off steam earlier in the stroke than does the left side; the difference between the total cut-off on each side being 1¾". In this case, the left side (or whichever side cuts off steam latest) is called "the heavy side." Now, in order to secure an equal distribution of steam, the cut-off on both sides of the engine should be the same. It is, therefore, evident that we must make some alteration in the valve gear in order to equalize both sides. See Rules 17, 33, 34 and 37. If, after reading these rules, you fail to locate any cause for this inequality of cut-off on the two sides, you must line the rocker boxes, or tumbling shaft, up or down. It is considered good practice to line the rocker box, rather than the tumbling shaft, when it can be done conveniently, for, unless you exercise due care when placing liners under the tumbling shaft stands, you may cramp the bearings; besides, if the engine be new, or has received a thorough overhauling, and the links are properly hung, it should not be done.

In the present case, we shall suggest four changes, any one of which will equalize the cut-off, and whichever one is most convenient may be employed. Line the left rocker box 7/64" down, or the right rocker box 7/64" up; or line the tumbling shaft stand on the left side 5/32" up, or the right tumbling shaft stand 5/32" down. See Rules 33, 34 and 37. When you have made this alteration the running cut-off will be equalized. In some cases the cut-off is tried with the reverse lever in several different notches of the quadrant, and the variation of cut-off between full and midgear, is averaged. The variation between this practice and the method we have explained is very slight, and, while the exhaust may not be so perfect in other notches, it is considered the better practice to have steam perfectly equalized in the running cut-off, where the engine performs most of its work.

Now, in order to try the exhaust opening, and closure, we must find additional points on the valve stems. If the valve is line and line at the exhaust edge, you know that the points of opening and closure will be indicated by the dead center between the port marks. But, if the valve has exhaust lap, it will be quite different, as the valve will cut off the exhaust before it has reached its central position, while it will not release until it has passed its central position. On the other hand, if the valve has exhaust clearance, these conditions will be reversed. This is true, also, of inside admission piston valves. So it will be well for the novice to distinguish these points by special marks or letters, when found.

To find these points, use a pair of small dividers, and, from the dead center between the port marks, with a radius equal to the exhaust lap, or exhaust clearance, as the case may be, describe a small circle on each valve stem. Make two centers where the circle crosses the horizontal line, and mark each whatever it represents, as previously explained. Now, remember that the opening, when the valve is moving forward, will be the closure when it moves backward, and *vice versa*.

When you have these marks, proceed to try each, by measuring the crosshead, as you did the cut-off, and record each point. Then compare these figures; alterations may be effected the same as in the cut-off. The process of equalizing the cut-off incidentally, has the same effect upon the exhaust closure, but remember that compression is of more importance than release, and compression, and lead opening, should be made as nearly perfect as possible. Indicator diagrams will expose these defects much more clearly than they could be explained here.

To find the maximum port opening, and maximum travel of the valve, place the reverse lever in full gear and turn the wheel one complete revolution, marking the extreme travel of the valve in each direction with your valve tram; the distance between each extreme point indicates the travel of the valve, while the distance from either extreme point thus found, to the port mark, indicates the maximum port opening. To determine the minimum port opening, and minimum valve travel, place the reverse lever in the center notch, and repeat the operation.

Laying Off a New Quadrant.

This task, due not to the importance of the quadrant itself, but to the fact that it has a great influence upon the motion of the valve, is a very important one. For it is by the correct use of the reverse lever in the quadrant that the locomotive is enabled to develop its utmost, in speed and power, under various working conditions. It is therefore quite necessary that one, to lay off a quadrant, should have a thorough knowledge and understanding of the valve motion.

Only the two extreme notches of the quadrant, in some shops, are located, and the entire face is then slotted with fine notches. Again, a templet, or the old quadrant, is used to lay off the new quadrant, and when the two extreme points are determined, stop plugs are placed in the quadrant. But we shall assume, for convenience of explanation, that the quadrant is to be laid off while setting the valves.

First to be located are the two extreme notches of the quadrant. This may be done by commencing on either side of the engine, and at either dead center, but, as the right side is within view of the man handling the reverse lever, let us begin on that side. Place the engine on right front dead center, and throw the reverse lever forward until the link block clears the top of the link by the amount allowed at each end of the link for clearance and slip, clamping the lever in this position temporarily. Now, by having the wheel turned one complete revolution, the two forward eccentric blades may be equalized, as has been previously explained. At the same time, note the full travel of the valve (errors in the design of the valve gear sometimes necessitate a shortening of the link block, or other alterations, to obtain the required travel of the valve).

Now place the crank pin on forward dead center and try the lead. Record the amount each eccentric must be moved, but do not change them as yet. Since the point of cut-off is of more importance than the travel of the valve, however, proceed now as follows: Turn the wheel forward, and measure the distance between the forward travel mark and the crosshead. When the crosshead has reached the full gear cut-off, which, for a piston stroke of 24", is usually 21", or seven-eighths of the stroke, try, with the valve tram, for the forward port mark on the right side (see Rule 24). Having ascertained, in this manner, the amount the right forward eccentric must be moved to secure the proper lead, it will be seen (Rule 28) that by changing the position of the eccentric on the shaft, the point of cut-off is also altered. So set the crosshead according to the amount that the right forward eccentric must be moved, and adjust the lever until the forward port mark corresponds to the valve tram. This will be the position of the reverse lever for full forward gear, and the quadrant may be scribed accordingly, for its forward notch.

Again place the crank pin on either dead center, and draw the lever back until the link block clears the bottom of the link as before (the amount of clearance and slip), and by turning the wheel backward, the two back motion blades may be equalized. Then, in exactly the same manner, try the travel, and locate the point of cut-off for full backward gear. Now, the eccentrics may all be altered the required amount, and the task of setting the valves may be completed. Try the full gear lead, and the full gear cut-off, in each motion, merely to prove the accuracy of the work.

The notches for intermediate points of cut-off for both motions may now be determined. Suppose, for instance, that a notch in the quadrant is to be located for a six-inch cut-off in forward gear. Place the reverse lever in the forward notch, and turn the wheel forward. As the crank pin passes its forward center, measure the distance between the front end of the crosshead and the foreward extreme travel mark. When this distance measures 6″ stop, and draw the reverse lever toward the center notch, until the valve tram corresponds to the foreword port mark. Mark the quadrant at this point, which will be the position of the lever for a 6″ cut-off. The other intermediate points are all located in the same manner, except that in the back gear the lever is placed in the back motion, and drawn toward the center, and the wheel is turned backward rather than foreward. When the full gear cut-off is the same for both motions, the center notch is located by determining the central point between the forward and backward full gear notches, using a pair of dividers. Always work on that face of the quadrant to which the reverse lever is attached.

Laying Off Eccentric Keys Before Setting the Valves.

Due to the increase in size of the modern locomotive, with a corresponding decrease in available space between the frames, it is becoming very difficult to adjust eccentrics and key them on

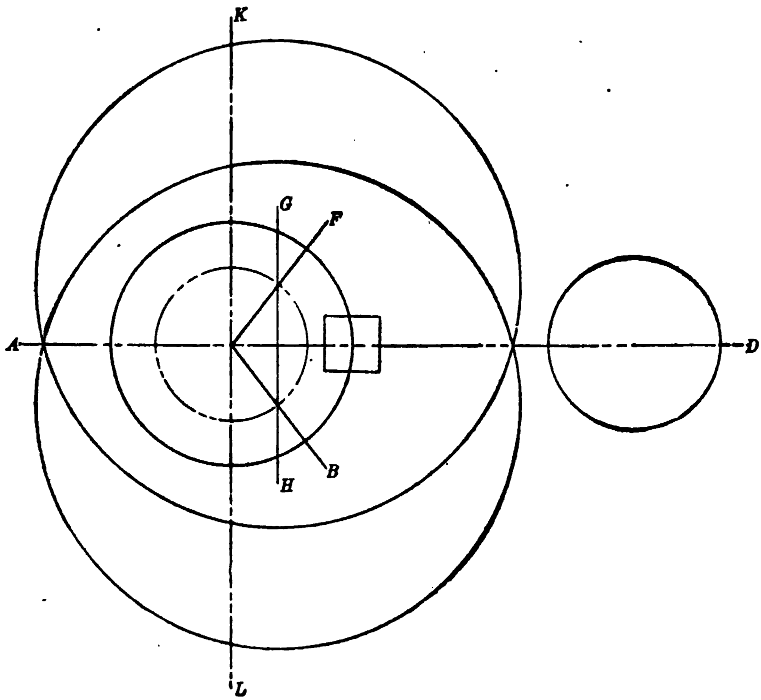


FIG. 94.

properly. The task is not only difficult, but is a cause of delay, and it is for this reason that it is becoming common practice to cut the keyways in the axle in advance. Even if one or more of the eccentrics should then require a slight offset, no serious damage results therefrom, and, if the man performing the work be proficient, the keyways may be laid off perfectly in this manner.

The following method may be used with any size of shaft, and for any throw of eccentric, with a straight rocker; if the rocker arm has backset, a little more work is necessary, because the center line of motion must be located. (See Fig. 94.) It is necessary to locate the center line of motion, as the eccentrics are always set by it, and not by the crank pin. These points may be laid off on the end of the shaft, and then transferred to their proper positions with a box square. Or, if the wheels are on the axle, a full size drawing may be made, and one or more pieces of tin of correct length be cut, to reach from the center line of motion to the center of the keyway. First, draw the center line of motion, A D, as shown in the figure; then, through the center of the shaft, erect the perpendicular line K L. If the two arms of the rocker are of equal length, the valve travel and the eccentric throw must be equal. To the steam lap of the valve, add the required lead, or subtract the negative lead, if used, and draw the line G H, parallel to the line K L, this distance from the line K L.

But, if the rocker arms are of unequal length, determine, by the use of a pair of dividers, how far the bottom arm moves while the top arm moves the amount of lap plus the lead. The line G H should be made this distance from the line K L, and, when correctly located, describe the broken line circle, using the center of the axle as center, and with a radius of one-half the throw of the eccentric. This circle, of course, represents the path of the eccentric centers, which are the points where the line G H cuts this circle. From these two points, and through the center of the axle, draw the lines F and B, allowing them to cut the circle representing the outer diameter of the shaft. These two lines, F and B, where they intersect this circle, indicate the correct positions for the center of the keyways, and may be transferred, by means of a box square, to the positions desired to locate the keyways.

Another successful method of accomplishing satisfactory results is that in which a board, 1½x10 inches, about six feet in length, is used. In the center of the board is cut a half circle the size of the largest shaft (perhaps 10 inches), and a V-clamp on the side of the board, to fasten it on the axle, is provided. This clamp may be raised or lowered so as to bring the top of the board to the center of any size of axle. A small bevel is inserted at one end of the board so that, if the line from the center of the driver to the center of the link is the same, the board may be set level. If the center line of motion is not horizontal, however, the board must be set at the correct angle by means of a plumb. The plumb bob frame is marked for the proper incline for the various classes of locomotive engines.

The board should be clamped on the axle near the eccentric; then, by plumbing the crank pin on the opposite side, the main crank pin is placed on dead center. Then place the eccentric plumb on the shaft, with the large part of the eccentric either up or down, according to whether it is the forward or backward motion. This will, of course, place the valve central on the seat, when the eccentric may be given the proper advance toward the crank pin, measured on the top of the board, which corresponds to the center of the axle, and the keyway be marked. The proper amount of advance to be given depends upon the outside lap of the valve, and the desired lead, as explained previously.

RULES FOR VALVE SETTING.

These rules for valve setting are applicable generally to slide valves, as they apply to the Stephenson link motion. Inside admission piston valves are more extensively used in connection with gears of other than the link motion type—the more modern forms of valve gears. The setting of these piston valves with the modern gears is thoroughly explained elsewhere in this volume.

Rule 1.

Relative Positions of Eccentrics to Crank-Pin.

On indirect motion engines, the forward motion, or go-ahead, eccentric should follow the crank pin when the engine moves forward, and the rearward motion, or backup, eccentric should lead the pin.

With direct motion, this arrangement is just reversed, that is, when moving forward the forward motion eccentric should lead, and the backup eccentric should follow, the pin. To distinguish direct from indirect motion, see Rule 39.

The rib of each eccentric is set at about the third spoke away from the pin, ahead of or in back of it, depending, naturally, upon whether the valve is of inside or outside admission type.

Rule 2.

To Change the Lead.

On engines with indirect motion, the lead is increased by moving the rib of the eccentric toward the crank pin, and is decreased by moving the rib away from the pin. The reverse is true of engines of the direct motion type.

Rule 3.

Method of Changing the Lead.

Mark the eccentric and the shaft (axle) with a tram, and move the eccentric on the shaft the exact amount indicated by the markings on the valve stem, providing the rocker arms are of equal length. If the arms are of different lengths, the distance the eccentric should be moved is proportional to the difference in the lengths of the rocker arms. If the top arm, therefore, be 12" long, and the bottom arm 9" long, a movement of 3/32" of the eccentric will produce a movement of ⅛" in the valve stem.

Rule 4.

To Set an Eccentric in Case of Emergency.

When setting an eccentric in an emergency, while out on the road, or setting temporarily in the shop, before setting the valves

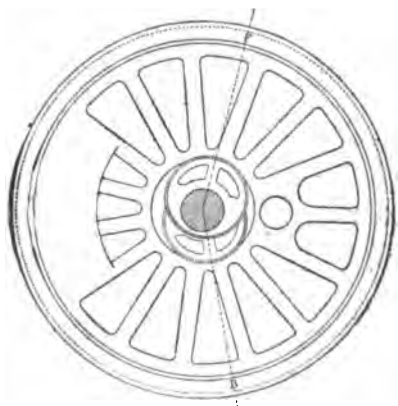


FIG. 95.

(assuming that there are 15 spokes in the wheel, that the valve has ⅞" steam lap, and that 1/32" lead is desired) place the rib (belly) of the eccentric slightly past the third spoke from the crank pin, as shown in Fig. 95. This position will be about right on most engines; of course the exact position depends upon the type of the valve, lap of the valve, lead required, number of spokes, center line of motion, etc. See Rule 15.

Rule 5.

Angular Advance.

The *angular advance* of an eccentric means the distance the center of the eccentric is advanced toward the crank pin, from a line drawn through the center of the axle, at right angles to the center line of motion. It is indicated by the space between the lines G H and K L, in Fig. 94. The line A D is the center line of motion. See Rule 15. When the rocker arms are of equal length, the *angular advance* is equal to the amount of lap of the valve plus the lead in full gear.

Rule 6.

Travel of Valve in Mid-Gear.

When the rocker arms are of equal lengths, the travel of the valve in mid-gear equals twice the angular advance, plus the increase of lead in mid-gear. The increase of lead in mid-gear varies from ¼" to ⅜", according to the radius of the link.

Rule 7.

To Find the Amount of Steam Lap.

The distance between the port marks always indicates the total steam lap; the valve has one-half that amount on each side.

Rule 8.

Worn Eccentrics.

When eccentrics are worn out of round very badly, you can reduce the size of one or all of them my truing them up in the lathe, without affecting the valve motion, providing that you do not change the throw. Of course, the eccentric blade will have to be lengthened, so as to divide equally.

Rule 9.

Throw of Eccentrics.

Keep all four eccentrics at exactly the same throw; a difference in their throws will affect the travel of the valve, and also the point of cut-off.

Rule 10.

Steam Ports.

Steam ports should all be kept of exactly the same size. A variation in their sizes will make one exhaust heavier than the other, and sometimes break the crank pins.

Rule 11.

Amount of Lead to Be Given.

It is impossible to lay down any definite rule as to the proper amount of lead that should be given, although for convenience in our explanation of valve setting, the engine is given 1/32" lead all around in full gear. The amount of lead which would give the best results on one engine might not do so on another, owing to the difference in the design of different locomotives; the best practical amount for each engine can be ascertained by the use of an Indicator. The volume of the clearance space, the cut-off, and other causes influence the amount of lead given, which varies from ¼" positive lead to 3/16" negative lead. Advocates of early admission claim that the cylinder and piston have become cooled during exhaust and should be reheated as early in the stroke as practicable, while the argument in favor of a late admission is, that while the crank is at or near the dead center, pressure against the piston will have no effect to turn the shaft, but rather prevent its turning by increasing the friction on the pins and main shaft. For past passenger service, the engine with the greater lead will be more satisfactory, while for freight service conditions, a lesser amount of lead is advantageous. But there is not much loss or gain by either early or late admissions, within practical limits, and the lead should be such as will produce the best results otherwise. See Rule 13. For an explanation of why an engine is given lead, see "The Slide Valve," page 37.

Rule 12.

Lead for Allen Valves.

A divergence of opinion exists as to the proper amount of lead to be given for Allen valves. The best practical amount to be given each engine can be determined by the use of an Indicator. Owing to the double opening (the supplementary port being equal to the length of the steam port), the volume of steam admitted to the cylinder by an Allen valve will be double the amount that a plain slide valve will admit, with an equal amount of lead. But, a decrease of lead will delay the cut-off and every other event of the valve. However, within practical limits, this variation will be so slight that it is considered good practice to give Allen valves only one-half the amount of lead given plain slide valves. The Allen valve will maintain the initial pressure better, at early cut-offs, than the plain slide valve, and the pressure will be more uniform during the period of admission, on account of the double opening.

Rule 13.

Lost Motion in the Valve Gear.

Lost motion in the valve gear decreases, and often nullifies, the amount of lead given the engine in full gear. Therefore it is important that all lost motion should be taken up when setting the valves, otherwise an old engine may be running blind in full gear; and yet, with a valve tram, may apparently have the required amount of lead. This evil would be readily apparent on an Indicator diagram.

Rule 14.

Effect of Increased Lead.

Lead increases as the reverse lever is drawn toward the center notch, in proportion to the radius of the link. This increased lead is sometimes injurious to slow, hard-pulling engines. To remedy this, the angular advance of the backup eccentric is decreased, in order to benefit the forward gear. Decreasing the angular advance of one eccentric gives the other motion almost a constant, unvarying amount of lead between full gear and mid-gear. The Allen design of valve gear, Fig. 60, which has a straight link, was designed to overcome this increase of lead in full gear.

Rule 15.

Relative Distance of Eccentrics From the Crank-Pin.

Very many believe that when the valve is given an equal amount of lead in the forward and backward motions, the eccentrics are an equal distance from the crank pin, yet they are not always an equal distance, even if the cut-off be equal in each extreme notch. Remember that the eccentrics are set from the center line of motion, and not from the crank pin, so that the bottom of the rocker will travel an equal distance away from its central position. (See Technical Points, 161.) As the eccentric blades are so long, and as the line of motion is usually so close to the crank pin when it is on center, for all practical purposes set the eccentrics by the pin, when setting them temporarily.

Rule 16.

Equalization of Admission and Cut-Off.

A perfect equalization of admission and cut-off for both gears is practically impossible, with link motion. If the lead be perfect the cut-off will not be, and vice versa. Equalization of the back gear is usually sacrificed to benefit the forward gear. This is due to the inherent imperfection of link motion, but it can be made almost perfect by carefully performing the operations required in setting the valves.

Rule 17.

Equalization of Steam.

When the exhaust is perfect, steam is not always perfectly equalized, and it is sometimes necessary to sacrifice a perfect exhaust to a correct equalization of steam. For example, if one cylinder were considerably larger than the other, on a single expansion engine, the larger cylinder should be given an earlier cut-off, or the small cylinder a later cut-off, in order to equalize the volume of steam admitted to each cylinder. Such an inequality is readily discovered by the use of an Indicator. An imperfect equalization will impart unequal stresses upon the crank pins, and, if the inequality be great, will cause the engine to jerk, and sometimes will break the crank pins.

Rule 18.

Perfect Equalization of Steam.

By referring to "Technical Points," we find that it is impossible to secure a perfect equalization of steam in both gears, with the shifting link motion. If the leads be equal, the cut-off will not be, and vice versa. This is due to the angularity of the connecting rods, and to the offset of the eccentric blade pins back of the link radius. By advancing the exhaust on the back edge of the valve in proportion to the clearance, the compression may be equalized in both ends of the cylinder, and the exhaust will sound perfect. Those familiar with the Indicator diagrams will readily perceive the advantage of this alteration.

Rule 19.

Full Gear and Mid-Gear.

Full gear means that the reverse lever is in either extreme notch in the quadrant. Mid-gear implies that the reverse lever is in the center notch of the quadrant.

Rule 20.

Dead Center, Quarter, and Eighth.

The engine is on "dead center" when the main pin, the center of the main driver, and the center of the cylinder are in line. By the "quarter" is meant that the pin is at right angles to, and above or below, its dead center position; and by "eighth" is meant the position of the main pin half way between dead center and quarter. See "Locating Dead Centers," page 189.

Rule 21.

Distinction Between Port Marks and Tram Marks.

The distinction between port marks and tram marks are these: While they are both made with the same tram, the port marks are the points obtained when first marking the valve stem, and they indicate the points of lead opening and of cut-off, and the "exhaust" port marks indicate points of release and compression.

The tram marks are those marks secured while setting the valves, and indicate the amount of lead or lap the engine has, at given points.

Rule 22.

To Equalize the Tram Marks and Change the Lead.

To equalize the tram marks at each end, change the length of the eccentric blade. To change the lead, move the eccentric on the axle. (See Rules 2 and 3.)

Rule 23.

Trying Forward and Back Motion Port Marks.

When trying forward motion port marks, always see that the reverse lever is in the forward notch, and have the engine moved back on dead center enough to take up all lost motion. Move forward and catch the dead center; then mark the valve stem. When trying the back motion port marks, place the lever in the back notch and move ahead enough to take up all the lost motion; then move back and catch the dead center, and mark the valve stem.

Rule 24.

Which Port Marks to Work From.

When the crank pin is on the forward center, always consider the valve tram marks from the front port mark; when on back center, from the back port mark.

Rule 25.

To Distinguish Lead From Lap.

If a tram mark comes between the port marks, it indicates so much steam lap (sometimes called blind); if outside, so much lead. But occasionally when blades and eccentrics are out badly, a tram mark may lap outside of the opposite port mark and appear as lead. Always consider which center the pin is on, and which port mark you are trying. (See Rule 24.)

Rule 26.

To Determine Whether to Lengthen or Shorten an Eccentric Blade.

Unless some positive rule is understood, it requires much time for deliberation, and often puzzles good mechanics, to determine whether to lengthen or shorten an eccentric blade. The reader who carefully follows this explanation will experience no difficulty from this source, and can tell at a glance whether to lengthen or shorten a blade, and thereby save much time and avoid mistakes. In order to more clearly explain this subject, an illustration, Fig. 96, is used.

The two forward motion tram marks are shown above the horizontal line, and, are indicated by the letters F and F1, while the two back motion marks are below the line, and are indicated by the letters B and B1. The point marked *o* indicates the exact center between the port marks (not the tram marks). All measurements should be made on the horizontal line. First examine the two forward motion tram marks. With a small pair of dividers, find the exact center between the marks F and F1; the center thus found, indicated by the letter *f*, is in front of the point marked *o*. Therefore, the eccentric blade should be shortened an amount equal to the exact distance between the points *o* and *f*. If the rocker arms are of different lengths, move the blade enough to cause the valve stem to move the amount thus indicated.

Now examing the two back motion tram marks; with the small dividers again find the exact center between the marks B and B1. The center thus found, indicated by the letter *b*, is in back of, or behind, the point *o*; therefore, the eccentric blade should be lengthened an amount equal to the distance between the points *b* and *o*.

The rule is, *always bring these two centers together.* If the center between the tram marks is in front of the center of the port marks, *shorten the blade;* if behind it, *lengthen the blade.*

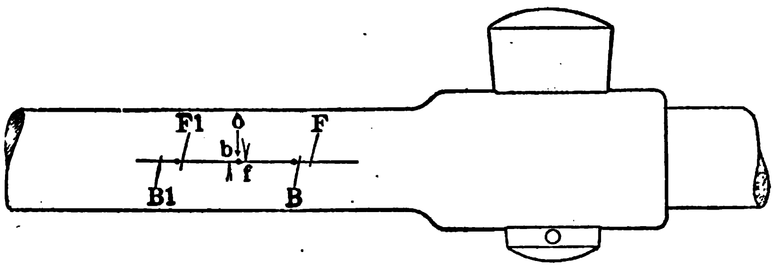


FIG. 96.

On a direct motion engine, the changes should be made in exactly the reverse manner. (See Rule 25.) Another method, largely practiced, is to add the lead and lap together and divide the amount, etc. This method is often puzzling, for even good mechanics sometimes confuse lead and lap. The rule here given is less liable to mistakes, for, as it does not matter whether the marks indicate lead or lap, you are safe when you work from their centers.

Rule 27.

Proving Alterations by Trial

In some shops the lengths of all eccentric blades that require alteration are changed, and then the engine is run over, to prove the work, and also run over again, after moving the eccentrics, before trying the cut-off. This is considered good practice, but it cannot always be done on "hurry-up jobs;" besides, if you are accurate in making the changes, it is unnecessary. (See Rule 28.)

Rule 28.

Effect of a Change of Lead Upon the Cut-Off.

If you have a "hurry-up job," change the lengths of the eccentric blades, but try the cut-off before moving any of the eccentrics on the axle. You can figure this way: If the lead be increased 1/8" by moving the eccentric on the shaft, the engine will cut off steam 1/2" earlier in each stroke. And, by adding together this 1/2" at both front and back points we have a total difference of 1" in the cut-off, on that side. Therefore, by increasing the lead 1/8", we reduce the total amount of cut-off on that side by 1". Remember that increased lead hastens every operation of the valve. (See "Lead will affect the point of cut-off," page 61.) It is likewise true that a decrease in lead always delays the point of cut-off an equal amount. Therefore, by decreasing the lead 1/8", the cut-off is delayed, and the total amount of cut-off is increased 1" on that side.

Rule 29.

Connecting Eccentric Blades to the Link.

Care should be taken when connecting eccentric blades to the link. Remember that, as a general rule, the forward motion blade should be attached to the top of the link, and the back motion blade to the bottom of the link. (See Rule 39.) The eccentrics are not in the same positions on all engines. On some, the back motion eccentric is next to the box, and on others the forward motion eccentric is next to the driving box. When coupled up incorrectly, the engine will move in the opposite direction to that indicated by the position of the reverse lever. Through negligence, good mechanics sometimes make this error.

Rule 30.

To Change the Valve Stem Instead of Both Blades.

If there is a right- and a left-hand thread nut on the valve stem (or rod), and if both eccentric blades on the same side require lengthening an equal amount, you can shorten the valve stem this same amount, and thereby avoid changing both blades: or, if the blades need to be shortened, lengthen the valve stem. Of course, if the rocker arms are of different lengths, the change of the valve stem should be of a proportioned amount. If the valve stem is of correct length, do not alter it. Very few of these valve stems are now in use. If it be necessary to change the valve stem length on a locomotive, the stem should be carefully marked for the guidance of the blacksmith.

Rule 31.

Altering the Lead.

The lead of an engine may be altered by changing the lengths of the reach rods, or link hangers, or by lining the rocker arms, or tumbling shafts, up or down, but what is added to the two go-ahead eccentrics is taken off the backup eccentrics, or vice versa.

Rule 32.

The Effect of One Blade on the Other's Motion.

If one eccentric blade be of incorrect length, it will affect the other's motion, and cause the valves to sound "out." The effect will be greatest when the reverse lever is hooked up in mid-gear, but it will be scarcely perceptible in full gear.

Rule 33.

Equalizing the Cut-Off.

After the lead has been equalized in full gear, it is customary to run the engine over again, to see if the steam is equalized at early points of cut-off, particular attention being given the running cut-off, which varies from about 4" to 6" for a passenger engine and from about 6" to 9" for a freight engine. If the engine is "out" very badly in the running cut-off, locate the cause at once, before making any alterations, to equalize the cut-off. Should one side of the locomotive cut-off steam three or four inches earlier at one end than at the other end, and the other side cut-off equally, you may be sure that there is something wrong. First, examine the link saddles carefully; occasionally a saddle is fastened to the link "up side down," which will change the off-set of the saddle pin. See that the saddles are not transposed, and that the saddle is securely fastened to the link, and not working "too and fro;" sometimes a saddle becomes loose, and, if the holes are oblong, it will move. Next examine the back-set of the rocker arms, and ascertain whether or not they are the same; one arm may be bent or sprung, and thereby cause the back-set of the rocker to be imperfect, but, unless the arm is sprung considerably, say ¼" or more, it will not cause this defect. See that the valve stem keys and eccentric blades are tight. If you cannot locate any other cause, the defect can be remedied by changing the off-set of the saddle pin. (See Rule 36.) If the difference in the cut-off, front and back, be very slight, it may be remedied by adjusting the eccentric blades, but it will be at the expense of a perfect equalization of steam in full gear. Should the running cut-off be imperfect on both sides, the cut-off taking place too early at the same end on each side, and each side at the same portion of the stroke, see if the top arm of the tumbling shaft is loose. Ascertain whether or not the reach rod is of correct length by trying the cut-off in full gear in each motion. (See Rule 38.). If the reach rod is of correct length, this defect may be overcome by lining the rocker boxes, or tumbling shaft stands, up or down, or by changing the length of the link hangers, or "lifters."

Should the engine show three or four inches "heavy" on one side; in other words, carry steam that much farther on one side than on the other, before cutting off, examine the arms of the tumbling shaft and see whether or not either is bent or sprung, either up or down. See that the two link-lifters are of an exact length, and that all of the eccentrics have the same throw, and that the two links are of the same radius, and that the off-sets of the link saddles are the same. If the lap or travel of the valves, or the sizes of the steam ports, are not the same on both sides, it will produce this defect in the cut-off. If you fail to locate the defect by the foregoing suggestions, place a straightedge across the frames, level it by the cylinders, and try the centers in the rocker boxes, and the tumbling shaft and its short arms—the centers on each side of the engine should correspond. Now you can remedy this defect by lining under the rocker box or tumbling shaft. Make any change that will raise the link or lower the link-block, on the "heavy" side of the engine, or lower the link or raise the link-block on the "light" side. (See Rules 16, 17, 18, 28, 34, 36 and 37.) If any part be correct do not alter it.

Rule 34.

Effect of Lining the Rocker Box and Tumbling Shaft.

Remember that when you place a liner under one tumbling shaft stand, you thereby raise both links, one about one-third as much as the other, and that when you line a rocker box up or down it will not affect the other side. Therefore, a 1/16" liner under a rocker box will effect the cut-off as much as a 3/32" liner placed under the tumbling shaft stand.

Rule 35.

Alteration of Main Rods.

If the main rods require alteration, figure on the alteration when calculating the changes for cut-off.

Rule 36.

To Correct Unequal Cut-Offs by Changing the Link Saddle.

When an engine does not cut-off equally, front and back, and you have located the cause in the link saddle, examine the saddle carefully and see that it has not been bolted to the link "up side

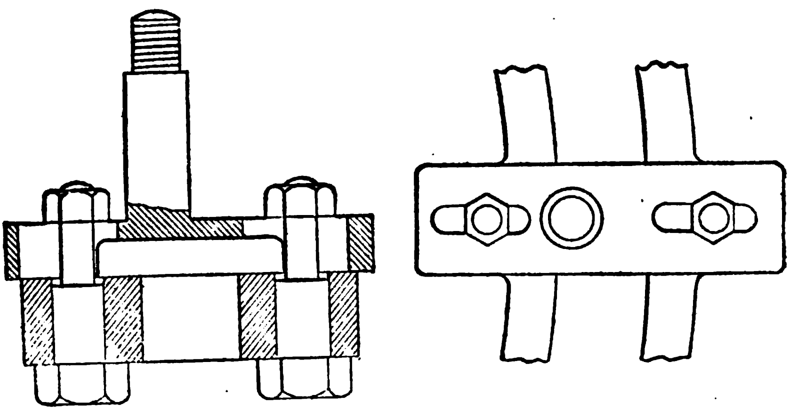
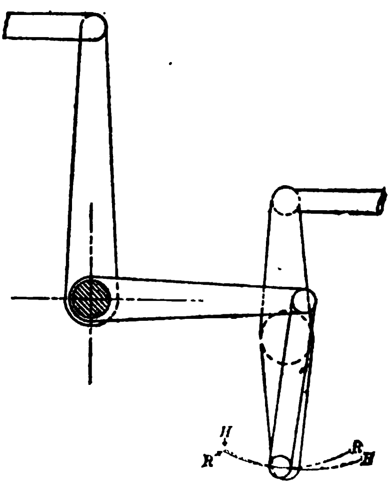


FIG. 97.

down," or saddle transposed; that is, the right saddle fastened to the left link, and vice versa. If the saddles are apparently correct, use a temporary link-saddle (see Fig. 97), if you have one; if not, use temporary bolts in the place of the link saddle bolts. Be sure to mark the original position of the link saddle before removing the bolts. We shall assume that the engine cuts-off steam equally front and back on one side, and on the other side it cuts off at 4" in front and 7" behind. Now we must determine which way to move the saddle, to increase or decrease the off-set of the saddle pin. Let us first ascertain just what effect this off-set of the saddle pin has on the cut-off in the forward motion.



Fi. 98.

Since the link and the link block are suspended from different points, the saddle pin being inside the link arc, it follows that the link block pin and the link saddle pin sweep through different arcs. By referring to Fig. 98, we find that the path of the saddle pin is represented by the arc H H, and that the path of the link block pin is represented by the arc R R. If we consider the positions of the eccentrics, we find that the link will be lowered when the crank pin is on the back center. Now we know that raising the link increases the lead, and thereby hastens the cut-off. It follows, therefore, that this off-set of the link saddle pin hastens the cut-off behind, and delays the cut-off in front. (It has an effect equivalent to an increase in the length of the eccentric blade, while the crank pin is on the forward center, and a decrease in its length when the crank pin is on the back center.) If the saddle pin had no off-set at all, the cut-off would occur earlier in the stroke in front, and later behind (this imperfection is caused by the angularity of the connecting rods, the main rod and the eccentric rod, and by the off-set of the eccentric rod pin holes from the link arc). Now, as the off-set of the link saddle pin tends to equalize the cut-off in both strokes, it must, therefore, delay the cut-off in front, and hasten the cut-off behind.

As our engine cuts off at 4" in front, and 7" behind, it is apparent that we must increase the off-set; therefore, move the saddle back 9/64" from its original position (3/32" *for every one inch difference in the cut-off.*)

The rule is, *if steam be cut off earlier in front than behind, increase the off-set. If earlier behind than in front, decrease the off-set.* After making this change, run that side of the engine over again and try the cut-off, and you will find it about right front and back. Mark the saddle when correct. If it is the original saddle that was on the link, rose-bit the holes, and countersink for the heads of the bolts. If the change made be very great, plug the holes in the saddle and drill new ones, or replace the saddle with a new one.

The exact amount the link saddle pin should be off-set depends entirely upon the design of the engine. Engines whose cylinder centers are above the wheel centers will require more off-set in the saddle than center line engines.

Rule 37.

Lining the Link-Block.

Lining a link, or link-block, 1/16" up or down, will make 1" difference in the cut-off on that side.

Rule 38.

Trying Length of Reach Rod.

If you have an old engine that carries her steam too far, or not far enough, in either gear, and you wish to try the length of the reach rod, by trying the cut-off in each extreme notch of the quadrant, proceed as follows: First try the cut-off in both gears; if it carries steam too far in the forward gear (for example, if it cuts off at 22" in the forward gear, and at 20" in the back gear, and should cut off at 21" in both gears, with a 24" stroke) place the reverse lever in the forward notch of the quadrant; now use a tram (from a running board bracket, wheel cover, or anything stationary), and scribe a line on the reach rod. Then turn the wheel forward, and, as the crank pin recedes from either center, use a rule between the end of the crosshead and the travel mark it is leaving, and, when the distance measures 21", stop. Now draw the reverse lever slowly toward the center notch, and catch the correct port opening with the valve stem tram; then stop moving the lever, and again mark the reach rod with the same tram. The difference between those two lines on the reach rod is the amount it should be shortened. In cases of this kind, one or more notches of the quadrant are sometimes plugged up.

Rule 39.

To Distinguish Between Direct and Indirect Valve Motions.

The rocker arm is usually the means by which the motion imparted to the eccentric is reversed. With the Stephenson link motion, if both arms extend in the same direction from the rocker shaft, the motion is direct. However, if they extend in opposite directions from the shaft, one up and one down, the motion is reversed, and the valve is indirect.

With the more modern valve gears, the motion is direct when the link block is working below the center of the link, when it has the same direction of motion from the eccentric to the valve. But when the link block is working above the center of the link, the motion is indirect. Therefore, reversing the engine, or placing the link block above the center of the link, changes the valve motion from direct to indirect.

Inside admission valves, as the modern piston valve, also influence the valve motion. When the valve admits steam from the inside, it must move in the opposite direction to the outside admission valve, that is, in a direction opposite to that of the piston at the commencement of the piston stroke. To accomplish this, the valve may be driven direct, without a rocker, leaving everything else the same. The rocker will give the desired motion, however, if both eccentrics are moved half way around the axle.

The following is a simple means of distinguishing between direct and indirect motion: If the crank pin and eccentric are on the same side of the driving shaft, and a rocker is provided to reverse the motion, the valve is indirect, outside admission. But if the crank pin and eccentric are together without a rocker arm, the valve has inside admission, direct.

If the crank pin and eccentric are on opposite sides of the driving shaft, with a rocker to reverse the motion, the valve has inside admission, and is indirect. With crank pin and eccentric opposite without a rocker, however, the valve has direct motion, and is outside admission.

The distinction may be stated briefly as follows: When the eccentric rod moves forward, and produces a forward movement of the valve, the motion is direct. But, if when the eccentric moves forward, the valve moves back, the motion is indirect.

Rule 40.

Equalizing the Running Cut-Off.

When the difference in the cut-off in both strokes on the same side of the locomotive is slight, and you can locate no other defect, you should sacrifice a perfect equalization of steam in full gear in order to equalize the steam in the running cut-off. If the cut-off shows "heavy" in front; that is, carries steam further in front than behind, shorten the eccentric blade; if "heavy" behind, lengthen the blade. A change of 1/32" in the length of the blade will make a difference of ½" in the cut-off. at each end.

Rule 41.

Giving Lead When Boiler is Cold.

If the quadrant is attached to the boiler, and the boiler is cold, give the two backup eccentrics 1/32" more lead than the two go-ahead eccentrics, as the expansion of the boiler when hot will raise the links slightly and thereby equalize the lead.

Rule 42.

Setting Valves on a New Engine.

After setting the valves upon a new engine, or upon one that has received a thorough overhauling, shorten all four eccentric blades 1/64". When the engine settles down, and the wedges are set up, the blades will be of correct length, and the steam distribution will be equalized.

	Valve				Per Cent Cut Off Full Str'ke	Running Cut Off	Exhaust Clearance		Cylinder	
	Travel	Lap	Lead Run. Cut Off	Lead Full Gear		Per Cent Str'ke	Piston Valves	Slide Valves	Diam.	Area
	Ins.	Ins.	Ins.				Ins.	Ins.	Ins.	Sq. Ins.
Fast Passenger	5 5½ 6 6½	15/16 1 1/16 1 1/8 1 1/8	1/4 1/4 1/4 5/16	0	85.0 84.2 85.0 84.0	25	1/8	1/16	17 -18 18½-20 20½-24 24½ up	227-254 269-314 330-452 471 up
Passenger	5 5½ 6 6½	7/8 1 1 1/16 1 3/16	1/4	0	85.6 86.0 85.0 85.7	25	1/8	1/16	17 -18 18½-20 20½-24 24½ up	227-254 269-314 330-452 471 up
Fast Freight	5 5½ 6 6½	7/8 15/16 1 1/16 1 1/8		0	86.6 87.5 86.5 87.0	33	0	0	17 -18 18½-20 20½-24 24½ up	227-254 269-314 330-452 471 up
Freight	5 5½ 6 6½	13/16 7/8 15/16 1		0	88.3 90.0 89.3 89.5	50	0	0	17 -18 18½-20 20½-24 24½ up	227-254 269-314 330-452 471 up
Switcher	5 5½ 6	13/16 7/8 15/16		0	88.3 90.0 89.3	66	0	0	17 -20 20½-22 22½ up	227-314 330-380 398 up
Light Locomotives	2½ 3 3½ 4 4½ 5	13/32 1/2 9/16 11/16 3/4 13/16		0	88.3 87.9 88.7 87.0 87.5 88.3	50	0	0	5 - 6 6¼- 8 8¼-10 10¼-13 13¼-16 16½ up	20- 26 31- 50 53- 79 87-133 143-201 214 up

The above table, recommended by the American Locomotive Company, will give the reader a knowledge of the latest practice in setting the valves for the Stephenson gear.

THE WALSCHAERT VALVE GEAR.

History.

Before proceeding into a discussion of this well-known form of valve gear, it seems to the writer that a brief history of the life and work of the inventor should be of interest to the reader, and with this idea in mind we present the following facts:

The inventor of this valve gear was born in Malines, Belgium, on January 21, 1820, and was named Egide Walschaerts. Some years later, for reasons unknown to the writer, he saw fit to omit the final letter *s* from his surname, so we shall do likewise in referring to him in this work.

About the year 1835, when the line from Brussels to Malines was opened, his native city was made the central point of the Belgium State Railways. The event proved to be the turning point in the career of young Walschaert, for the construction of the locomotive made a strong impression on his mind, which lasted until the end of his life.

In 1838, while a student in the College of Malines, he exhibited some models of a stationary and a locomotive engine of his own construction, at a local exposition. He continued his experiments with more or less success and secured a position as a mechanic in the Belgium State Railway shops in Malines in 1842, and became a foreman in the shops at Brussels two years later, when his ability as an engineer had clearly demonstrated his qualification to rule the motive department, yet he was never permitted to rise to a higher official position.

In those early days the locomotives used in Belgium were imported from England, and they had only been in service a few years when Walschaert was made a foreman. But it is now evident that they did not meet with his approval, for he soon began work upon a new system of valve motion, which invention he completed in the year of 1844.

Under the rules of the Belgium State Railways, in force at that time, a mechanic, or foreman, was not allowed to patent a discovery or device for his own benefit and profit, because the products of his brain, as well as of his hands, was considered the property of his employer. So to circumvent the rule Walschaert induced an engineer of the Belgium State Railways to seek a patent for him. The application for this Belgium patent was filed by his friend, Mr. Fisher, on October 5, 1844, and the patent was issued November 30, 1844, for a term of 15 years.

It may be stated here, to the credit of Mr. Fisher, that he never claimed credit, personal or otherwise, for the invention of the valve gear on which he secured a patent.

The rule referred to, however, did not apply to foreign countries, and Walschaert secured a patent on his valve gear in France on October 25, 1844, in his own name.

The invention immediately attracted attention in Bëlgium, but some time elapsed before it was adopted by the leading railways of Europe. Considerable effort was made to introduce the valve gear in America about 10 to 15 years after it was invented, but the construction of our locomotives and the existing conditions were not then favorable. About 1897 a few American locomotives were equipped with the gear and the results obtained during the following years were so satisfactory that the valve gear gradually worked its way into general favor.

The original design of Walschaert's valve gear, upon which a Belgium patent was issued to his friend Fisher, is shown in Fig. 99.

While there is considerable difference in construction of the mechanism patented in 1844, it is, in principle, similar to the

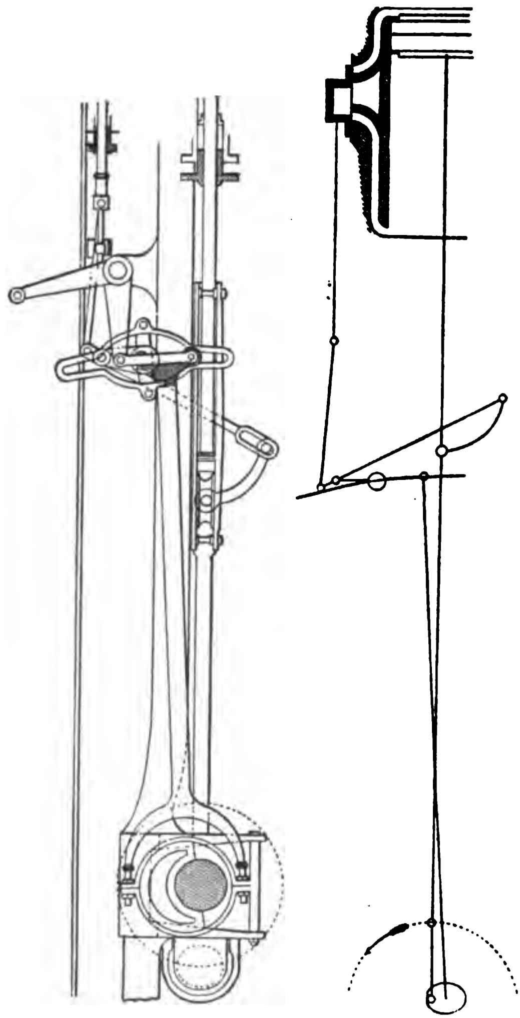


FIG. 99.

valve gear in use today, which was constructed by Walschaert in 1848, and first used on locomotive No. 98 of the Belgium State Railways, when it ran from Malines to Brussels on September 2, 1848.

While it is true that the link and combination lever are now usually placed in a different position to shorten the eccentric and valve stem, Fig. 100 of the original design shows, in a general way, the valve motion in use today, for quite a number of locomotives at the present time require an arrangement similar to that of the illustration.

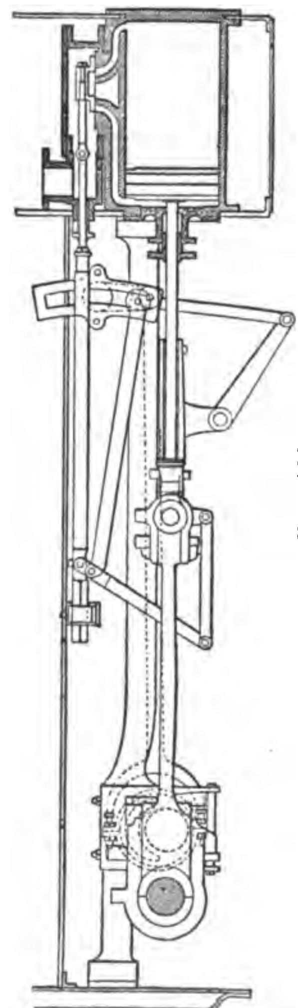


FIG. 100.

At the time Walschaert began his work, Sharp's valve-motion, which consisted of two eccentrics with forked rods, was the only system in extensive use, and it is very doubtful whether Walschaert ever saw the link motion credited to Stephenson and Howe, which was invented in 1843, before he completed his own system.

The inventor's activity during the following years was confined, to a great extent, to the duties imposed on him as foreman, but a number of other improvements and inventions conceived in his ingenious brain were used by practical men of that time.

It is now difficult to understand how Walschaert's inventive spirit was maintained in the face of the obstacles encountered, and the field to which his services were restricted. It certainly was not stimulated by any monetary reward, for he never received much, if anything (except a few medals), for the invention of his valve gear.

Then death deprived him of the pleasure of seeing the general adoption of his valve gear by the locomotives of the twentieth century, for he died on February 18, 1901, at Saint Gilles, near Brussels, in Belgium, at the age of 81 years and 29 days.

It has been well said that his work now meets requirements which did not exist during his lifetime and, for this reason, he is entitled to credit for solving a problem of, to him, a future generation.

Analysis.

The principal advantage of this gear lies in the accessibility of its parts, which are placed entirely outside of the driving wheels; this facilitates oiling, inspecting and cleaning operations which are frequently difficult in performing on locomotives equipped with the Stephenson link motion. Furthermore, in heavy engines equipped with the Stephenson gear, the eccentrics must be made of large diameter to secure the required throw. This increases the velocity of the rubbing surfaces and the tendency to heat, especially in the case of locomotives which have comparatively small wheels and are employed in high speed service.

In the Walschaert gear the various parts are pin-connected and are easily lubricated, hence troubles due to over-heating are reduced to a minimum. Furthermore, the Walschaert gear, as usually constructed, transmits the moving force of the valve in a practically straight line; consequently there is less springing and yielding of the parts than in the Stephenson link motion.

By removing the valve gear from between the frames, as in the case of the Walschaert valve gear, a better opportunity is afforded to introduce stronger frame bracing, and this reduces the possibility of the frame breaking. It is purposed here to explain the theory and action of the Walschaert valve gear, in as simple and plain a manner as possible, with the help of numerous illustrations.

In commencing the study, or entering into an analysis, of any particular form of locomotive valve gear, it must be assumed that the principle of the plain steam engine in its most primitive form is already understood; at least, of the valve itself. As to the valve, it is a study in itself, and we must become acquainted with it before starting into the subject of the mechanism that operates it.

So far as the distribution of steam in the cylinder is concerned, the constant lead, which is a feature of this motion, is not considered objectionable, and it has some distinct advantages. Under such conditions it is possible to determine the amount of lead the engine should have at the most economical point of cut-off. This point once determined, and so designed, it cannot be altered by anyone in the shop or roundhouse. Another advantage is that it prevents valve setters from attempting to produce results by moving the eccentrics into improper relations one to another.

The constant lead of the Walschaert valve motion prevents the sealing of the cylinders by the piston valve when the piston is at the end of its travel, or approaching it, whereas with the link motion, either by derangement or excessive wear, the valve laps the ports at the end of the stroke, thereby causing excessive compression and many other troubles.

Another feature of this motion, which appeals to the engineers, is the comparative ease of handling the reverse lever when the locomotive is running at a high rate of speed.

Lead is given in order that steam may be admitted between the piston and the cylinder head, toward the completion of the stroke, as a means of cushioning the piston and thus tempering the sudden reversion of its motion. There are many cases in which the variable lead is preferred; for instance, with a variable lead, the longest cut-off in starting can be obtained, combined with the proper amount of lead at the ordinary running cut-off.

In the case of passenger locomotives particularly, a steam distribution like this is often desirable. The Walschaert valve gear, as already stated, may be designed to give a variable lead. This practice has recently been followed in a number of instances. The favorable results for starting are, however, obtained at the expense of the distortion of the valve events in back motion by robbing one to favor the other.

For this reason the Walschaert valve gear, with variable lead, is suitable only for passenger and fast freight locomotives and not for slow freight or switching locomotives. With a variable lead, so arranged that the lead increases as the reverse lever is hooked up, the eccentric crank lags behind the correct position for a constant lead; in other words, it is so set that the link is not in its central position when the crank pin is on the center. The general arrangement of the Walschaert valve gear depends largely on the general design of the locomotive.

Contruction.*

The reliability of the Walschaert valve motion on locomotives, when properly applied, has practically put the former standard (the Stephenson motion) out of use. It is, therefore, considered that a complete elementary explanation of the theory of this gear, as applied in practice, is desirable.

This will involve considerable repetition of data which has been previously published, but the object is to show more intimately the fundamental co-relation between the various parts that make up this gear, in order to make as plain as possible its extreme simplicity and to enable even a layman to master the gear without further instruction.

Like any other device, the Walschaert valve motion is merely a development of some more simple form. A clear understanding of the underlying principles of this valve gear is, therefore, best obtained by starting with the original form and tracing the various steps in the development.

Fig. 101 represents a simple form of valve motion. This consists of a single crank eccentric, driving a plain valve without lap or lead, by means of an eccentric rod directly connected to the valve stem. Assume that the engine is to run forward; with the main pin on the back center, as shown, the eccentric crank pin must be on the top quarter. The valve will then be in a central position on its seat with all ports closed. The engine

*Courtesy of American Locomotive Co., New York, N. Y.

could not, therefore, start of itself when the throttle was opened. Connect another engine of the same kind to the wheel on the other side of the axle. If, then, the main pins are set at right angles to each other, as in the case of the locomotive, the valve of the left hand engine would be in a position to admit steam behind its piston and start the engine forward. The eccentric crank of the engine shown in the illustration would then move its valve forward. This would uncover the back steam port and admit steam behind the piston.

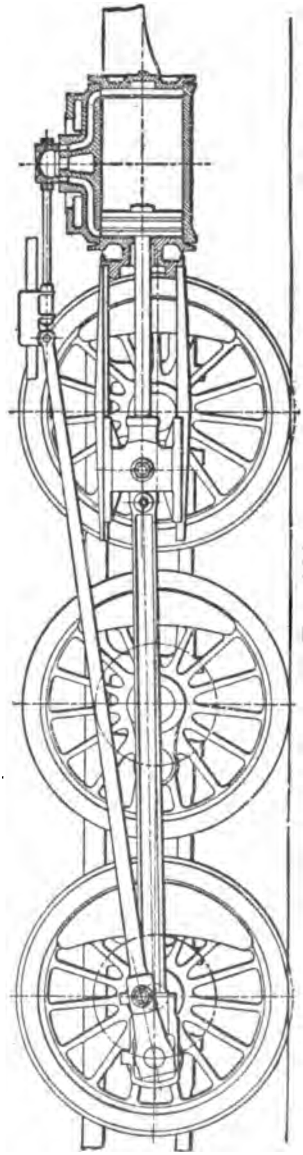


Fig. 101.

Such an engine, however, would run in only one direction. If, for example, the wheel was pinched backward to start the engine backward; with the eccentric crank in the position shown, the valve would be moved back. Steam would be admitted ahead of instead of behind the piston.

In order to make the engine run backward, the eccentric would have to be on the lower quarter. In such a case, if the wheel was turned to the left or backward, the eccentric would be a quarter of a revolution ahead of the main pin. It would thus move the valve forward, opening up the back port and admitting steam behind the piston.

The introduction of some means for reversing is the first step necessary in the development of this simple form of engine. This can be accomplished by introducing between the eccentric crank and the valve stem a beam pivoted at its center as shown in Fig. 102. Here the link is the beam.

Referring to diagram "A," Fig. 102, with the valve stem connected to the lower end of the link and the main pin on the upper quarter, the valve is in the position to start the engine forward. Assume, on the other hand, that the main pin and eccentric crank remain in the same positions and the valve stem is connected to the upper end of the link, as shown in diagram "B." The valve is then in its extreme position to the left. When the throttle is opened, the engine will run backward. It is evident, then, that a radius rod connected to a block which slides in a curved slot in the link, as shown in the diagram, and the necessary mechanism for raising and lowering the block, would give an engine which could be reversed.

This engine, however, is far from an efficient machine. The valve does not close the admission port until the main pin is on the center, at which time the valve is in its central position on the seat. Steam will thus be admitted to the cylinders throughout the full stroke of the piston, irrespective of the travel of the valve.

In order to govern the period for the admission of steam; or, in other words, to change the cut-off; a different design of valve must be used. The valve must be given lap. Also, the valve should have lead, at least at the ordinary running cut-off.

If lap and lead are given to the valve, the valve motion shown in Fig. 102 must be so modified that, when the crank pin is on

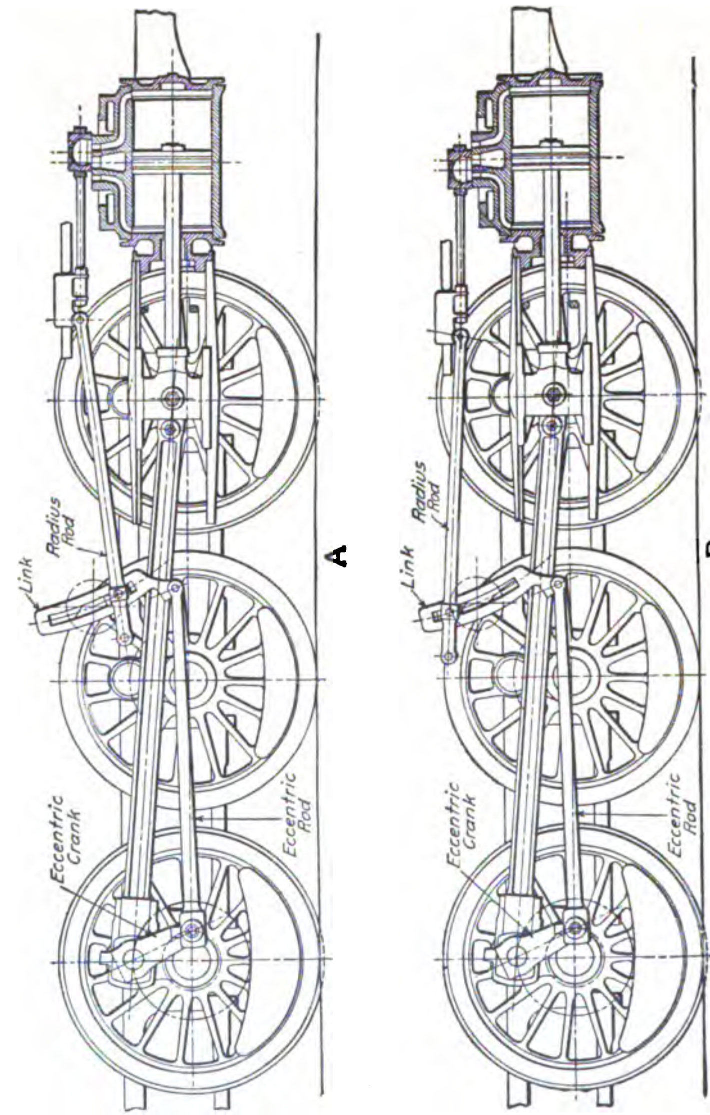

Link
Radius Rod
Eccentric Crank
Eccentric Rod
A
Link
Radius Rod
Eccentric Crank
Eccentric Rod
B

Fig. 102.

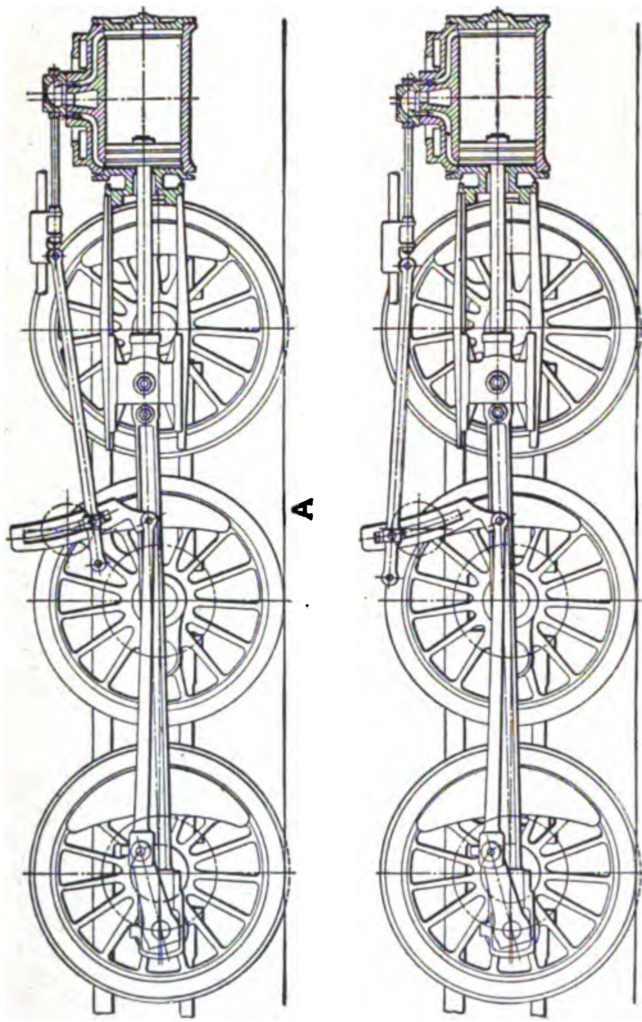


FIG. 103.

either of the centers, the valve will be advanced a distance equal to the lap plus the lead.

A study of Fig. 103 shows that this advance of the valve cannot be obtained by any change in the position of the eccentric crank relative to the main pin. With a single eccentric crank, as shown, the engine could not be reversed if the eccentric crank was not a quarter of a revolution from the main pin.

Fig. 103 shows the same valve motion as Fig. 102; except that the valve has one inch outside lap and the eccentric crank has been advanced to more than a quarter of a revolution ahead of the main pin. With the link block at the bottom of the link, as in diagram "A," the advance given to the eccentric will have moved the valve forward a distance equal to the lap plus the lead, and the engine will run forward. But if the link block were moved to the upper end of the link, as in diagram "B," the valve would be moved back. Steam would then be admitted to the front end of the cylinder when the piston was at the end of the back stroke. The engine would then stall and, therefore, could not be reversed.

In the Walschaert valve gear, the motion for providing lap and lead to the valve is derived from the main pin through the lap and lead lever, which is connected to the crossehad.

The introduction of the lap and lead lever is the next step in the development of the Walschaert valve gear.

Referring to Fig. 104, when the link block is in the center of the link, as in diagram "A," there will be no movement of the radius rod as the link is swung back and forth by the eccentric. Assume that the radius rod is connected with the lap and lead lever at the point "R." Also assume that the upper end of the lever is connected with the valve stem crosshead at "V," and the lower end to the crosshead arm by a short link, as shown.

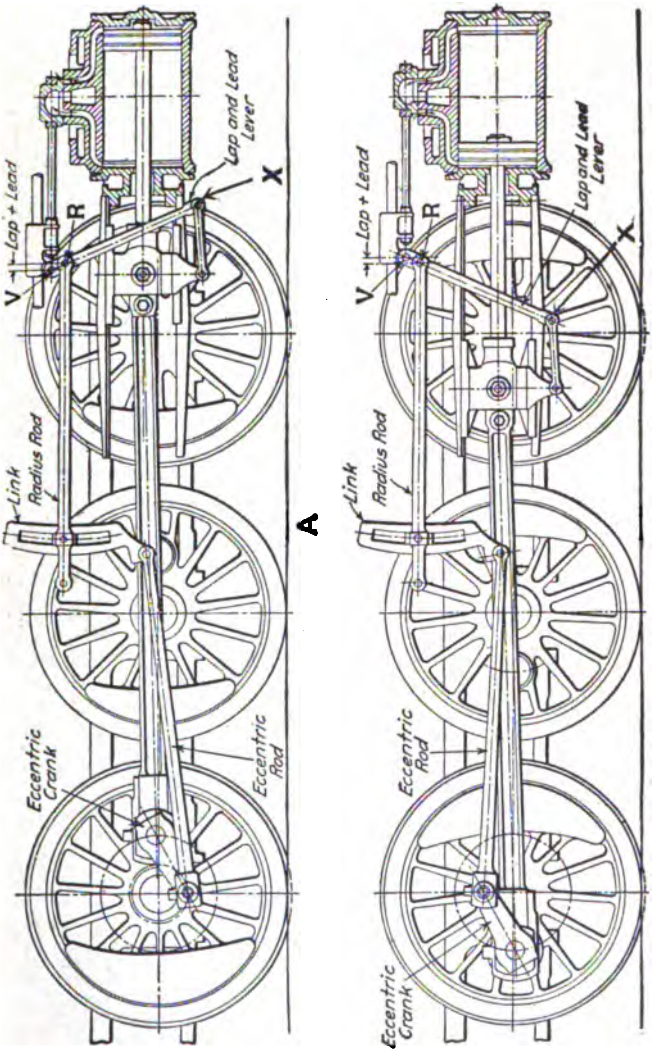

Lap + Lead
R
V
X
Lap and Lead Lever
Link
Radius Rod
Eccentric Crank
Eccentric Rod
A
Lap + Lead
R
V
X
Lap and Lead Lever
Link
Radius Rod
Eccentric Rod
Eccentric Crank
B

FIG. 104.

With such a construction, as the crosshead moves back and forth, point "R" being stationary, the point "V" will rotate about it. The valve will thus be moved back and forth.

With the main pin on the forward center, as in diagram "A," the angle assumed by the lap and lead lever has moved the valve back a sufficient distance to uncover the front port. When the main pin is on the back center, as in diagram "B," the inclination of this lever has opened the back port.

Therefore, the motion of the valve is derived from two distinct sources, viz.: the eccentric crank and the main crank. The former gives the travel to the valve minus the lap and lead. This motion is transmitted through a slotted link oscillating on a fulcrum, in which slot the link block is moved at will above or below the fulcrum. By this means the motion of the engine can be reversed so as to run either forward or backward and it may, therefore, be called the reversing motion of the gear. The lap and lead motion, imparted to the valve by the main crank, is not reversible, as it is symmetrical with relation to the dead center position of the crank in whichever direction the engine is running, and is always constant, regardless of the position of the link block.

The action of the Walschaert valve gear as a whole may be best learned by tracing the movement of the valve through a complete revolution of the wheel.

Figs. 105 to 108 show a series of diagrams representing different positions of the crank pin. For the sake of simplicity, the valve and cylinder are shown in section. The other parts of the gear are represented by their center lines and center points only. These diagrams are purposely drawn out of proportion. The valve and the eccentric throw have been enlarged in order to show more clearly the positions of the edges of the valve relative to the edges of the cylinder ports.

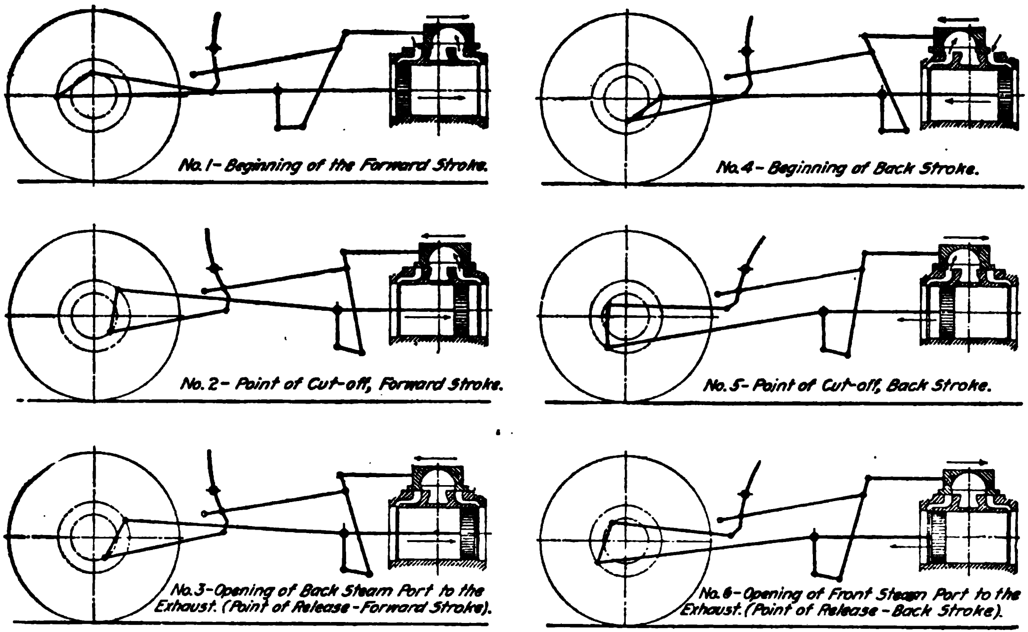

No. 1- Beginning of the Forward Stroke.
No. 4 - Beginning of Back Stroke.
No. 2 - Point of Cut-off, Forward Stroke.
No. 5- Point of Cut-off, Back Stroke.
No. 3-Opening of Back Steam Port to the Exhaust. (Point of Release - Forward Stroke).
No. 6-Opening of Front Steam Port to the Exhaust. (Point of Release - Back Stroke).

FIG. 105.

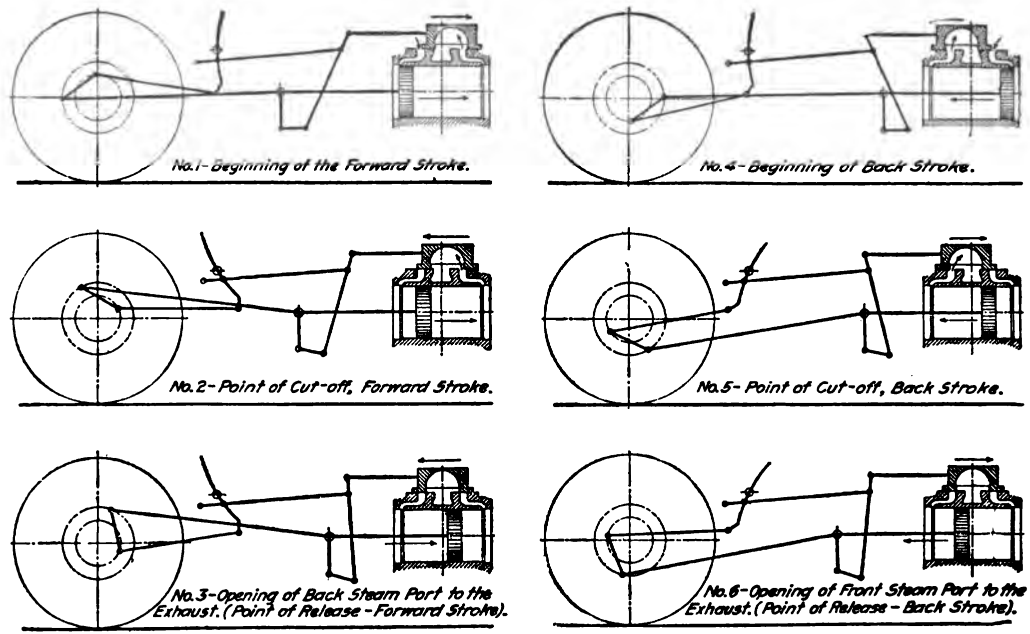

No. 1—Beginning of the Forward Stroke.
No. 4—Beginning of Back Stroke.
No. 2—Point of Cut-off, Forward Stroke.
No. 5—Point of Cut-off, Back Stroke.
No. 3—Opening of Back Steam Port to the Exhaust. (Point of Release—Forward Stroke).
No. 6—Opening of Front Steam Port to the Exhaust. (Point of Release—Back Stroke).

FIG. 106.

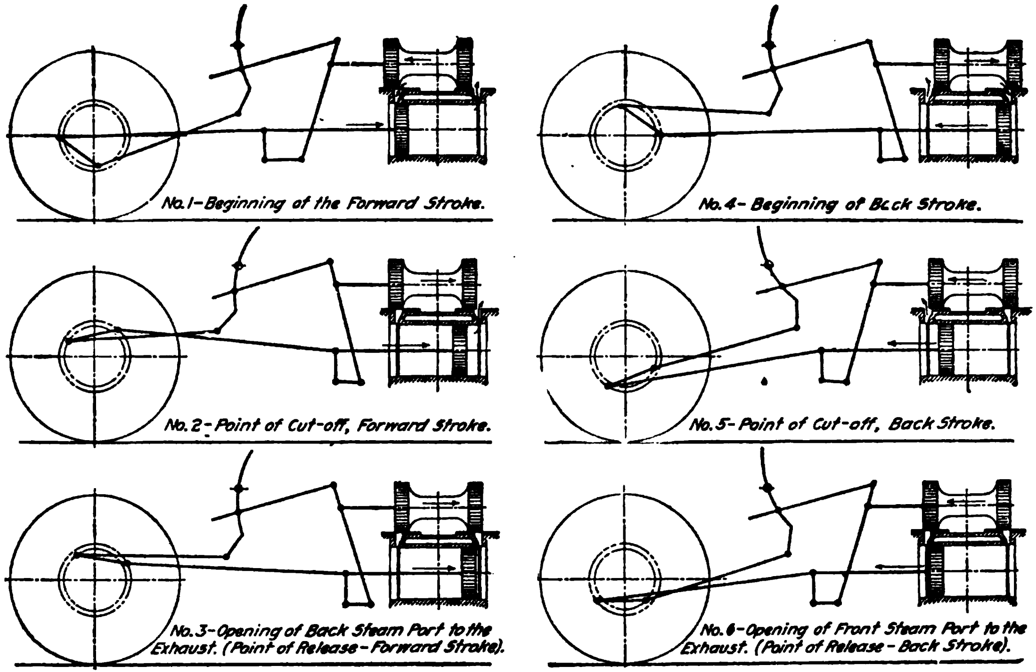

No.1-Beginning of the Forward Stroke.
No.2-Point of Cut-off, Forward Stroke.
No.3-Opening of Back Steam Port to the Exhaust. (Point of Release-Forward Stroke).
No.4-Beginning of Back Stroke.
No.5-Point of Cut-off, Back Stroke.
No.6-Opening of Front Steam Port to the Exhaust. (Point of Release-Back Stroke).

FIG. 107

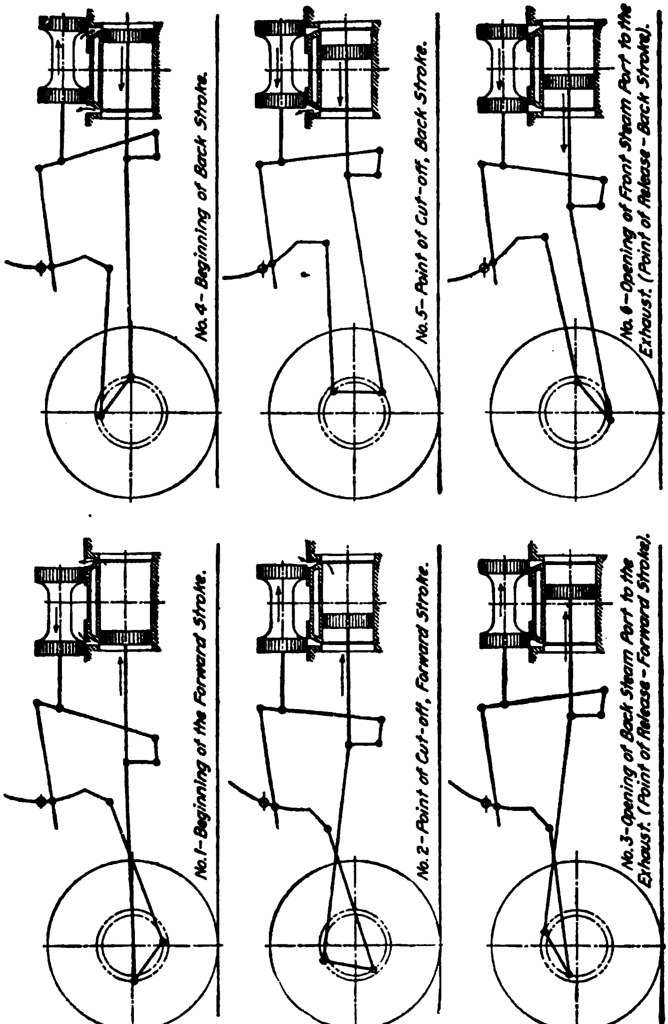

No. 1–Beginning of the Forward Stroke.
No. 2–Point of Cut-off, Forward Stroke.
No. 3–Opening of Back Steam Port to the Exhaust. (Point of Release–Forward Stroke).
No. 4–Beginning of Back Stroke.
No. 5–Point of Cut-off, Back Stroke.
No. 6–Opening of Front Steam Port to the Exhaust. (Point of Release–Back Stroke).

Fig. 108.

In Fig. 105 the valve has outside admission; and the motion is represented with the reverse lever in full gear forward. Fig. 106 represents the same arrangement of valve motion. In this latter case, however, the reverse lever is hooked up and the engine is cutting off at about 25 per cent of the stroke.

Figs. 107 and 108 represent the Walschaert valve gear as arranged for piston valves having inside admission.

In the Walschaert valve gear, the valve, as previously explained, receives motion from two distinct sources: First, from the eccentric crank. This gives the valve its full travel. Second, from the lap and lead lever. This gives the valve its lead. The valve would receive a travel from this source equal to twice the lap plus twice the lead, even if the eccentric rod were disconnected.

Considering diagram 1, Fig. 105, the valve has outside admission. Consequently, forward motion being taken from the bottom of the link, the eccentric crank leads the main pin. Also the radius rod is connected to the lap and lead lever below the valve stem.

The link is in its central position. Therefore, the valve would likewise be in its central position, if it were not for the motion given to it by the lap and lead lever. The travel of the crosshead to the back end of the stroke has caused this lever to rotate about its point of connection with the radius rod. This has moved the valve forward, as indicated by the arrow, a distance equal to the lap of the valve plus the lead.

Considering diagram 1, Fig. 106, the main pin, eccentric crank, link and crosshead are in the same positions as they are in the corresponding diagram in Fig. 105. The link block, however, is nearer the center of the link. This has not affected the lead for the following reasons: The length of the radius rod is the same as the radius of the link, or approximately so.

Thus, as the link is in its central position the raising of the link does not cause any movement of the front end of the radius bar. This point, which is the fulcrum of the lap and lead lever, is in the same position as in the diagram previously considered. Consequently, this lever has moved the valve to the same position as before; and the lead is the same as when the reverse lever was in full gear.

In Figs. 107 and 108 the valves have inside admission. The radius rod is thus connected to the lap and lead lever above the valve stem.

The diagrams in Figs. 105 to 108 represent the various valve events throughout a complete revolution of wheels. Comparison between corresponding diagrams in the four figures brings out very clearly the difference in the arrangement of the Walschaert valve gear for outside and inside admission valves. It also shows the effect on the various valve events of hooking up the reverse lever. For example, it will be noticed that in Figs. 106 and 108 the valve cuts off the steam from the cylinders and opens the steam ports to the exhaust at a much earlier period in the stroke than it does in Figs. 105 and 107. The latter figures represent the engine with the reverse lever in the corner notch; while Figs. 106 and 108 represent running with a short cut-off.

With the eccentric crank a quarter of a revolution from the main pin, and with the eccentric of such length that the link is in its central position at the end of the stroke, the lead in the Walschaert valve gear is the same for all cut-offs.

To change the lead of the Walschaert valve gear, under the conditions given in the preceding paragraph, it is necessary to change either the lap of the valve or the distance between the connecting points of the lap and lead lever. Reducing the lap of the valve increases the lead, and vice versa. Changing the lap also changes the points of cut-off. If the lap is reduced, the cut-off will occur at a later period in the stroke; while if it is increased the opposite result will occur.

The following rules govern the changes to be made in the lengths of the arms of the lap and lead lever to increase or decrease the lead:

To *increase* the lead, make the upper arm (or distance between the valve stem and radius bar connections) longer in proportion to the lower arm (or distance between the radius rod connection and bottom connection of the lever).

To *decrease* the lead, make the upper arm shorter in proportion to the lower arm of the lever.

Arrangement.

A general outline of the gear for an inside-admission valve is shown in Fig. 109, where the names are given of all essential parts.

The names define, as nearly as practicable, the special function each one has to perform in the combination of the gear. Starting at the crank pin "D" we have the eccentric crank, eccentric rod, link, radius rod, combination lever, combination link, crosshead arm and link block "g" (indicated only by letter). Fig. 110 is a similar arrangement of an outside-admission gear with no change in the names of any of the working parts.

It will be noted that in Fig. 109 the eccentric crank follows the main crank, while in Fig. 110 it leads. Also in Figs. 109 and 110 the radius bar is shown in the bottom portion of the link in forward motion, thereby reducing the stresses on the link trunnion bearings when running forward.

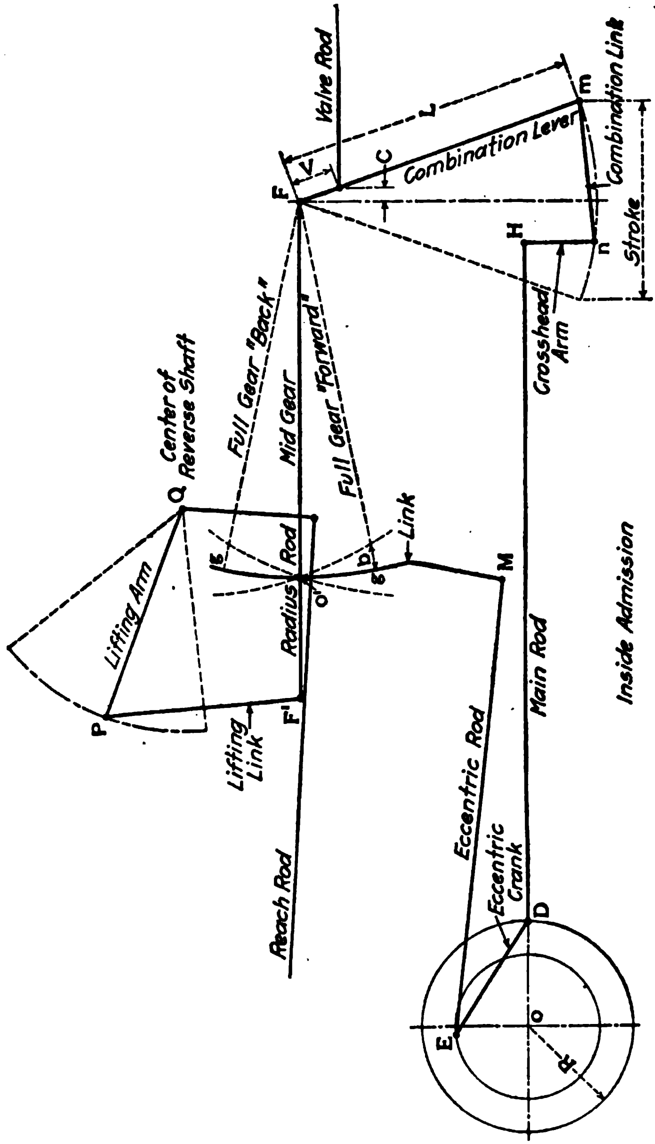

Valve Rod
L
V
C
Combination Lever
m
Combination Link
F
H
n
Stroke
Crosshead Arm
Full Gear "Back"
Mid Gear
Full Gear "Forward"
Center of Reverse Shaft
Q
Lifting Arm
Radius Rod
O'
B
D
B'
Link
M
Main Rod
Inside Admission
P
Lifting Link
F'
Reach Rod
Eccentric Rod
Eccentric Crank
D
E
O
R

FIG. 109.

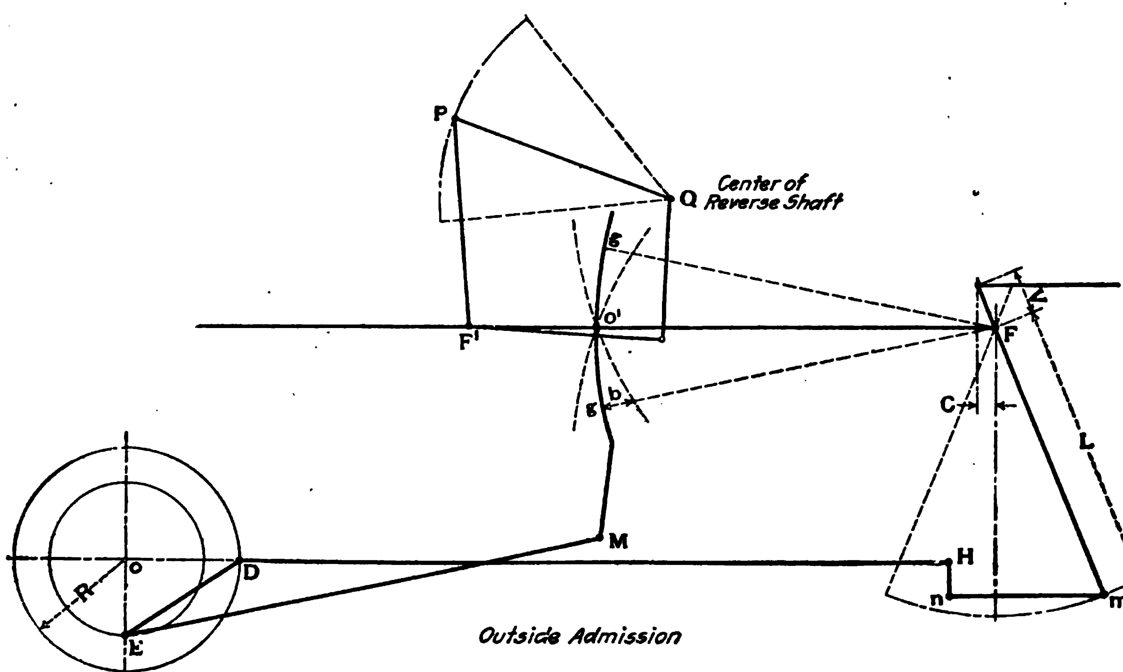

P
Q
Center of
Reverse Shaft
g
O'
F'
b
g
F
V
C
L
M
H
n
m
o
D
R
E
Outside Admission

FIG. 110.

Before proceeding with the details of the gear, the stroke of the piston, lap and lead, virtual eccentric throw, valve travel and the point of cut-off must be shown. Of these the stroke, lead, valve travel and cut-off are determined with respect to the size of engine. It is advisable to add 1/32-inch to the actually desired lead to provide for wear of pins and bushings.

Lap.

When lead, cut-off and valve travel are given, the lap is found by the Reuleaux diagram, Fig. 111.

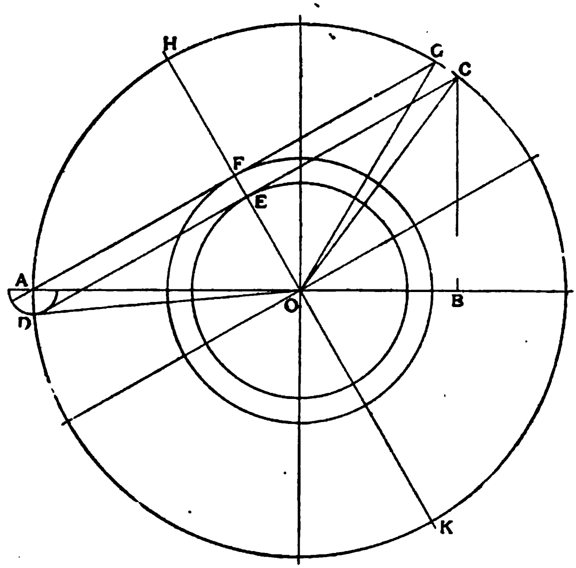


FIG. 111.

A circle with a diameter equal to the travel of the valve is drawn and on this diameter the same percentage as the desired cut-off percentage in the cylinder is measured from "A" to "B." From "B" is drawn a perpendicular line which intersects the circle at "C." O-C will be the cut-off position of the crank.

With the lead as radius and "A" as center, draw a semi-circle on the opposite side of the diameter to that of the location of point "C." From the cut-off point "C" draw a line tangent to the lead circle and extend same until it intersects the valve circle at "D." O-D will be the position of the crank when the valve starts to open. Draw a line through "O" perpendicular to the line C-D, which it will intersect at "E." The distance E-O will be the required lap. The given lead may now be added to the lap directly by drawing a line through "A" parallel to line C-D which will intersect the extension of line E-O at "F." O-F will be the lap and lead, or the amount of motion to be imparted to the valve by the combination lever.

It will be noted that the angularity of the main and eccentric rods has some effect as to the exactness of the cut-off points in front and rear ends of the cylinder. This layout will, therefore, give an approximate average of what may be obtained in practice.

This method is equally applicable to the ordinary double eccentric motion with shifting links when the centers of the eccentrics are located at "A" and "G" with the crank at "K" for direct motion and outside-admission valve, or for inside-admission when the motion is transmitted to same by means of a rocker. For inside-admission with direct motion, or outside-admission using a rocker, the crank is located at "H" for the same eccentric position; provided, in all cases, that the center line of the cylinder passes through the center of the axle, otherwise their relation to the crank must be changed to suit the angularity of the main rod when on the dead centers, and the irregularity introduced thereby divided between the two dead centers. In the shifting link motion the link block will follow a convex curve from "A" to "C" in reversing from one full gear to the other, with increased lead toward central position with open rods, and a concave curve with diminishing lead toward center with crossed rods.

Returning to the Walschaert gear, we find that the combined motion of the single eccentric and the crank give an exactly equivalent result in full gear, but in linking up, the effect on the motion of the valve is equivalent to a link block following the straight line A-G from full forward to full backward gear, thus giving a constant lead E-F at all cut-offs.

Virtual Eccentric Throw—With Lap and Lead Known.

The required throw of point "F" (Figs. 109 and 110), being the final throw imparted by the crank E through the link to the combination lever, may be termed the virtual eccentric throw. It can be found either mathematically or geographically by the following formulæ, and as per Figs. 112 and 113, respectively. In both figures the valve circle is indicated by No. 2 and the virtual eccentric circle by No. 3.

$$b = \frac{R\sqrt{a^2 - c^2}}{R - c}$$ for inside admission and

$$b = \frac{R\sqrt{a^2 - c^2}}{R + c}$$ for outside admission valves,

where b = the radius of a circle with a diameter equal to the throw of point "f" (Figs. 109 and 110), representing the eccentric throw which may therefore be called the virtual eccentric throw.

R = the length of the crank = ½ piston stroke.

a = the radius of a circle whose diameter is equal to the valve travel.

c = the lap plus lead = distance O-F in Fig. 111.

Combination Lever.

It is evident that the motion of any point on the combination lever can be represented by a circle, as the two connections to which the motion is imparted, namely, "m" and "F" in Figs. 109 and 110 emanate from two circular paths, the main crank "D" and the eccentric pin "E" which is set at an angle of 90 degrees with the former, providing that the link connection of

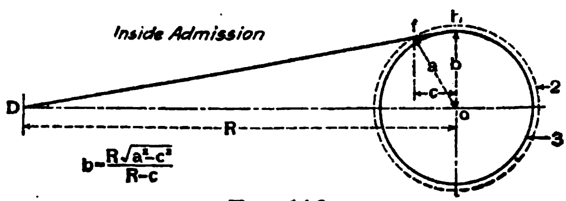


FIG. 112.

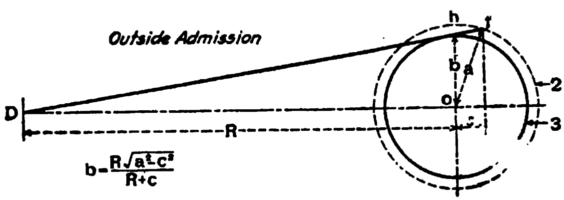


FIG. 113.

the eccentric rod is located on the center line of the engine in its central position. Even if the link connection is off the center line of the engine when the eccentrics will be more or less than 90 degrees in the lead or following of the crank, the effect of the movement of point "F" (Figs. 109 and 110) will be the same as if the eccentric crank had exactly 90 degrees angle to the main crank and consequently the line "b" in Figs. 112 and 113 will always be represented at right angles to the crank "D."

With "O" as center of axle, lay out the length "R" of crank "D." Through "O" draw a line perpendicular to length "R"; also with "O" as center draw the known valve circle.

For inside-admission valves, the lap and lead "C" is laid off from "O" toward the crank "P." From this point draw a line perpendicular to D-O until it intersects the valve circle (No. 2) at "f." From "D" draw a line through "f" until it intersects the vertical diameter of the circle at "h," when O-h or "b" will be the radius of the vertical eccentric circle No. 3.

For outside-admission valves, Fig. 113, the lap plus lead is laid off on the opposite side of "O" to that of the crank and a perpendicular line is drawn in the same manner as in Fig. 112 until it intersects the valve circle at "f." Where the line D-f intersects the vertical center line of axle at "h," the radius "b" of the virtual eccentric circle is found by the distance O-h.

For outside admission valve the eccentric circle will always fall inside of the valve circle; but for inside-admission the eccentric circle may fall inside of, on, or outside of the valve circle, depending on the relations between the stroke, valve travel and lap and lead.

The length of the combination lever will be made to suit the distance between the piston rod and valve stem, but its proportions will be $V : L = C : R$. The point of valve stem connection, measured from "m," will be L—V for inside, Fig. 109, and L+V for outside admission valves, Fig. 110, the valve stem connection falling below the radius bar connection in the former case, and above it in the latter case.

As a further illustration of how all the connecting points on the combination lever can be represented by circles, refer to Fig. 114 for inside, and Fig. 115 for outside-admission valves which makes the whole system clear and complete. In these figures the swing of the lower end of the combination lever is represented by the crank circle "1" from which it receives its motion. On the same center is drawn the valve circle "2," the virtual circle

"3," the lap-plus-lead circle "4," and the lap circle "5." Tangent to lap and lead circle "4" is drawn a line f-f perpendicular to the crank O-D until it intersects circle "2" at "f." By drawing the line D-f, extended to the transverse center line of the

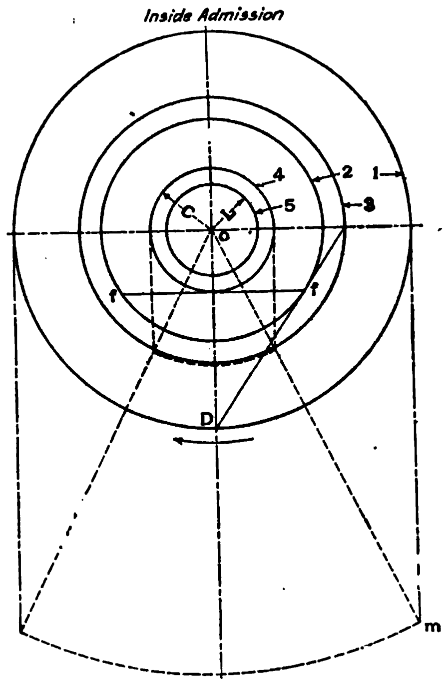


FIG. 114.

axle "O," it will intersect the eccentric circle at "h," which is in this respect a repetition of that found in Fig. 112.

Fig. 115 gives the same arrangement, but for outside-admission, except that the line f-f is drawn tangent to the lap and lead circle on the opposite side of the axle "O" to that of the crank and the line D-f is drawn intersecting the transverse center line at "h," as in Fig. 113.

Length of Radius Rod.

With due consideration of the length of the eccentric rod, the radius rod should be as long as circumstances will allow. It is

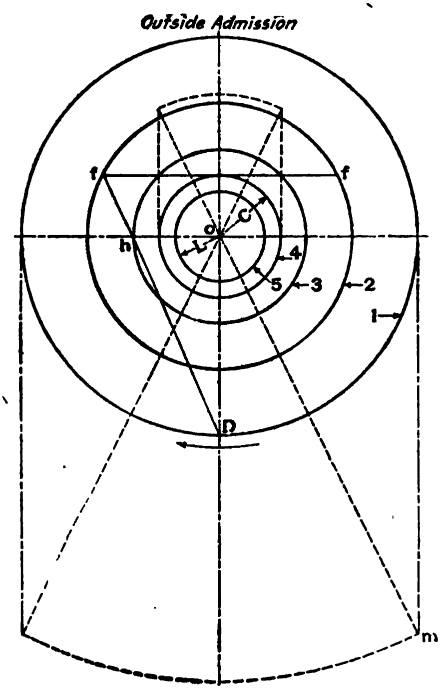


FIG. 115.

also subject to a minimum limitation in length, and a minimum of three times the travel of the link block "g" from full "forward" to full "back" gears is a good rule. See Figs. 109 and 110.

Link Slot Radius.

From the point of connection between the radius rod and the combination lever as center, strike off the link slot with a radius equal to the length of the radius rod.

Link Fulcrum.

At the same distance and on the same side above or below the extended center line of the valve stem as the average distance of the radius rod connection (point "F"), is the preferable location of the fulcrum of the link "O," Figs. 109 and 110. With a reasonably long radius rod a slight deviation from this will have little or no detrimental effect on the movement of the valve.

Link Swing.

The angular swing of the link should never exceed 45 degrees, preferably 40 degrees, from one extreme to the other.

Link Block Movement.

Full "forward" and full "back" positions of the link block "g," Figs. 109 and 110, will be at a point on the link where the swing is equal to the diameter of the virtual eccentric circle whose radius is "b," Figs 112 and 113.

Throw of Eccentric Pin.

Determine the actual throw of the eccentric pin "E" by the relation of the location of the link block "g" in full gear, Figs. 110 and 109, to that of the eccentric rod connecting pin "M" from the link fulcrum. Generally the eccentric pin "E" will have two or three times greater throw than the virtual eccentric motions of points "g" and "F."

Length of Eccentric Rod.

Make the eccentric rod E-M as long as practicable, using three times the actual eccentric throw of pin "E" as a minimum length.

Eccentric Rod Connection to Link.

Special attention must be given to the location of the connection of the eccentric rod with the link pin at point "M" in Figs. 109, 110 and 116. This location must be plotted out because of the irregularity due to this point "M" generally being some distance above the center line of the engine and also be-

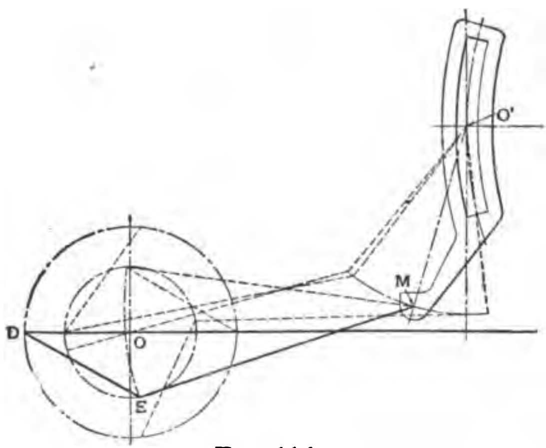


FIG. 116.

cause of the angularity of the eccentric rod. The normal angle of 90 degrees between the crank "D" and the eccentric pin "E" must be modified to suit the height which the link pin "M" is above the center line of the engine, and is determined in connection with the fore and aft locations of the pin "M." With limited lengths of eccentric rods, the rear half of the eccentric circle gives a shorter throw than the front half. To compensate for this, the pin "M" will have to be moved back of the tangent of the link, until it gives, by the upward rise in its rear throw, the same angular swing to the link as that of the front from its central position when the crank is at its dead centers.

Great care should be exercised in locating this point so that the natural irregularities, unavoidable in the transmission of circular into lineal or oscillating motion, are properly compensated for in these particular points to get a correct and satisfactory motion of the gear as a whole.

Lifting Shaft Location.

The location of the lifting shaft "Q," Figs. 109 and 110, has an important bearing on the proper movement of the link block

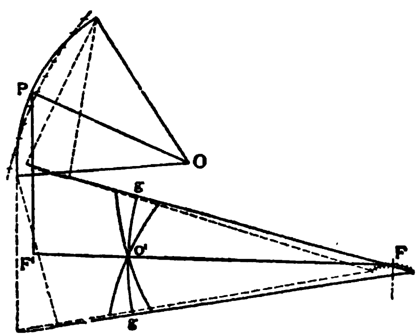


FIG. 117.

"g" and consequently on the movement of the valve at all points between full forward and full backward positions. The lifting arm connection "P," Figs. 109 and 117, at upper end of the lifting link, should follow as near as possible a circular path described by the upper end of the lifting link in a position perpendicular to the radius bar while the link block "g" is moved from one extreme position to the other, with the main crank "D" at one of its dead centers, or being the same as a circle drawn with "F" as center and a radius equal to $\sqrt{s^2+u^2}$, where "s" is the length F′-F of the radius bar and "u" the length of the lifting link F′-P. The arm P-Q must necessarily be shorter than the radius thus obtained, and the lift link shaft "Q" should then be so located that with a convenient length of this arm, the point "P" will describe a curve intersecting the former at about 50 per cent cut-off position of the arm in both front and back gears. If, however, it is desired to favor the forward gear, a somewhat earlier intersection may be selected in the back motion, when the forward half of the curve will coincide more closely to the theoretical circle.

A somewhat closer refinement in this respect, is obtained by the location of the lifting shaft on the center line of motion back of the link, Fig. 118.

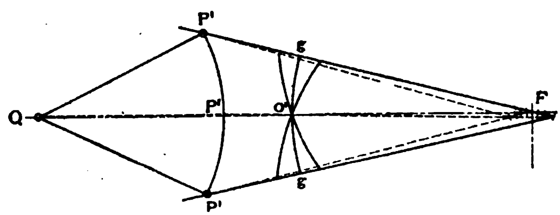


Fig. 118.

The back end of the radius rod being raised and lowered by means of a sliding bearing, causes the link block "g" practically to follow the chord of the circular path described by the point in the link determining the throw of the block in all positions.

In the usual arrangement of the Walschaert valve gear, the lead is constant and does not vary as the reverse lever is hooked up. The Walschaert valve gear can, however, be so designed as to give a variable lead.

There are many cases in which the variable lead is preferred. With a variable lead, the longest possible cut-off in starting can be obtained, combined with the proper amount of lead at the ordinary running cut-off. In the case of passenger locomotives particularly, a steam distribution like this is often most desirable.

The favorable results for starting are, however, obtained at the expense of the distortion of the valve events in back motion. For this reason, the Walschaert valve gear with variable lead is suitable only for passenger and fast freight locomotives; and not for slow freight and switching locomotives, and should therefore be used with discretion. This variation in lead is accom-

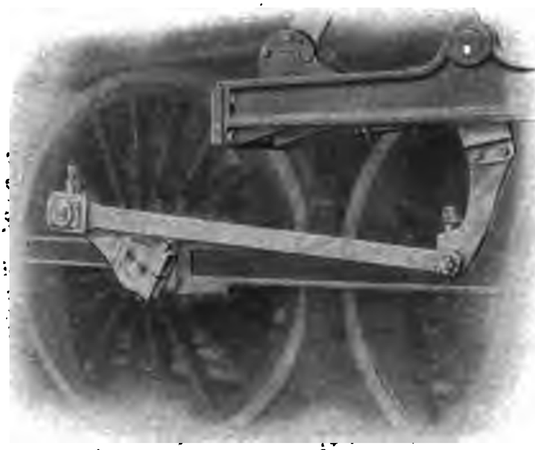


FIG. 119.

plished by designing the combination lever for the maximum lead in mid-gear and reducing it to a desired minimum at full gear by shifting the eccentric crank in the proper direction until this occurs.

The general arrangement of the Walschaert valve gear depends largely on the general design of the locomotive. Some of the ordinary forms of construction of the various parts of the gear are shown in the following illustrations.

Fig. 119 illustrates an eccentric crank and eccentric rod. The former is secured to an extension of the main crank pin by a binding bolt. The eccentric crank is split so that it can be drawn to a tight bearing by means of the bolt. This bolt, together with a key, keeps the eccentric crank fixed in its position on the main pin. It may be easily removed, if necessary, by driving out the binding bolt. This construction has the ad-

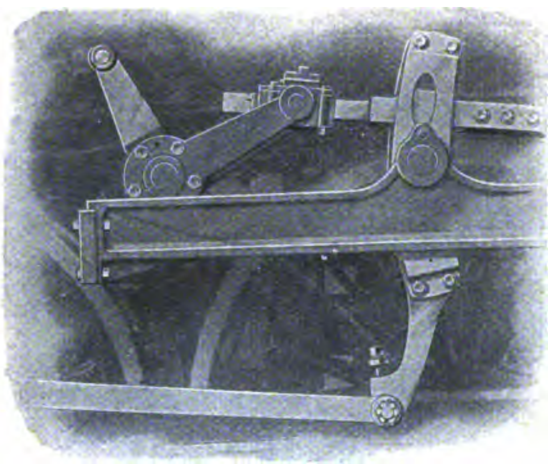


FIG. 120.

vantage that it permits of using a solid bushing on the side rod at the main crank pin.

Fig. 120 illustrates the link and reverse shaft. In the design shown, the radius rod is directly connected to the lift shaft arm by means of a slip block. In such a case, the reverse shaft arm is made in two parts, the outer part being easily removable. This permits of readily disconnecting the reverse shaft and radius rod and makes a very simple arrangement.

In another style of the reversing mechanism, also commonly used, the arm of the lift shaft is connected to the radius rod by means of a link. This type of construction is shown in Fig. 121.

This figure also shows the American Locomotive Company's latest patented arrangement of self-centering guide for the valve stem of the Walschaert valve gear, as shown later. It will be noticed that the guide is cast in one piece with the steam chest head. Consequently, the guide is always central with the piston

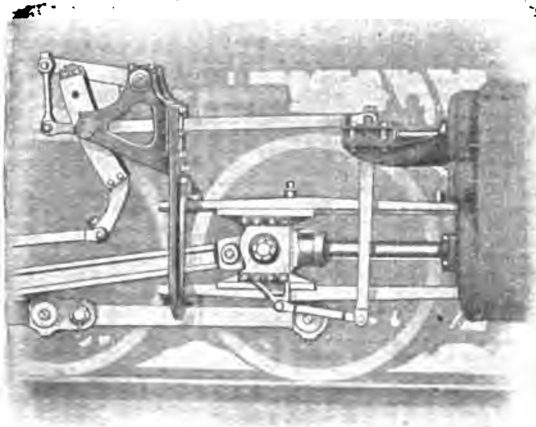


FIG. 121.

valve chamber. An arrangement of this kind has the advantage that it can be erected, taken down and replaced without any lining up. The guides are so constructed that they can be easily adjusted for wear by means of liners. This arrangement also permits the use of a straight design of lap and lead lever without forks. The lever is connected to the valve stem crosshead by a pin passing through its wings. In case it is necessary to take down the lap and lead lever, the upper portions of the guide may be removed by taking out four bolts. This gives access to the pin connecting the lap and lead lever to the crosshead.

Fig. 122 illustrates the latest arrangement of the combination link directly connected to the wrist pin. This arrangement has the advantage of eliminating the crosshead arm and bushing, thereby reducing the number of parts and also reducing the reciprocating weights. It also simplifies the construction of the guide yoke end.

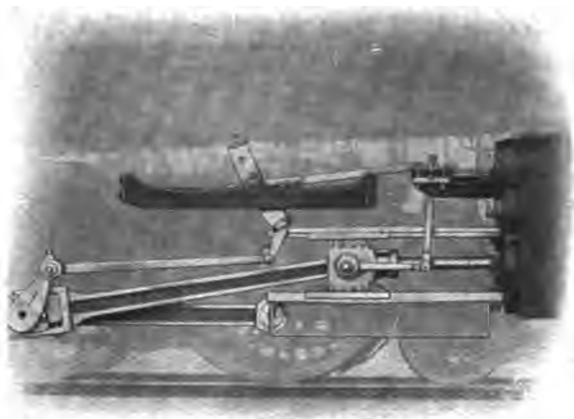


FIG. 122.

However, on long stroke engines, the pins are so close together that a good arrangement is not practical.

In Case of Accidents.

Having considered the principles and construction of the Walschaert valve gear, the next phase of the subject to take up is what should be done in case of accidents. This is of the utmost interest to all engineers. It would be impossible to lay down rules to cover every case that might arise. It is possible, however, to consider some of the more usual or more possible accidents; and to determine the best and quickest courses to follow in such cases.

The accidents to be considered may be divided into two general classes; those in which it is not necessary and those in which it is necessary to block the valve to cover the ports.

The first class of accidents includes only those cases in which it is necessary to take down the main rod. The valve must necessarily be blocked if the main rod is disconnected.

In the second class of accidents, the damage to the machinery may or may not make it necessary to take down the main rod.

Main Rod Up, Valve Not Blocked.

The following is considered a good rule to follow in regard to taking down the main rod. When it is possible to lubricate the cylinder and relieve compression, otherwise than by removing the cylinder cocks, the main rod may be left up if in a condition to run. For example: if there are relief or vacuum valves in the cylinder heads, these may be removed. This will prevent compression and also permit of lubricating the cylinder.

Considering the first of the above mentioned classes of accidents in which the valve does not have to be blocked; suppose, for example, an eccentric crank, eccentric rod or the foot of the link is broken.

Fig. 123 illustrates how the valve gear may be disconnected in such cases. Take down the eccentric rod, disconnect the radius rod from the lift shaft and secure the link block in the center of the link. The valve on the lame side then receives a motion from the lap and lead lever. Its travel will be equal to twice the total amount of the lap plus the lead. This gives a port opening equal to the amount of the lead. Consequently, the main rod may be left up, as the cylinders will be lubricated. Furthermore, though the cut-off on the disabled side will be very short, the steam that is admitted will do a certain amount of work, and the engine can be reversed.

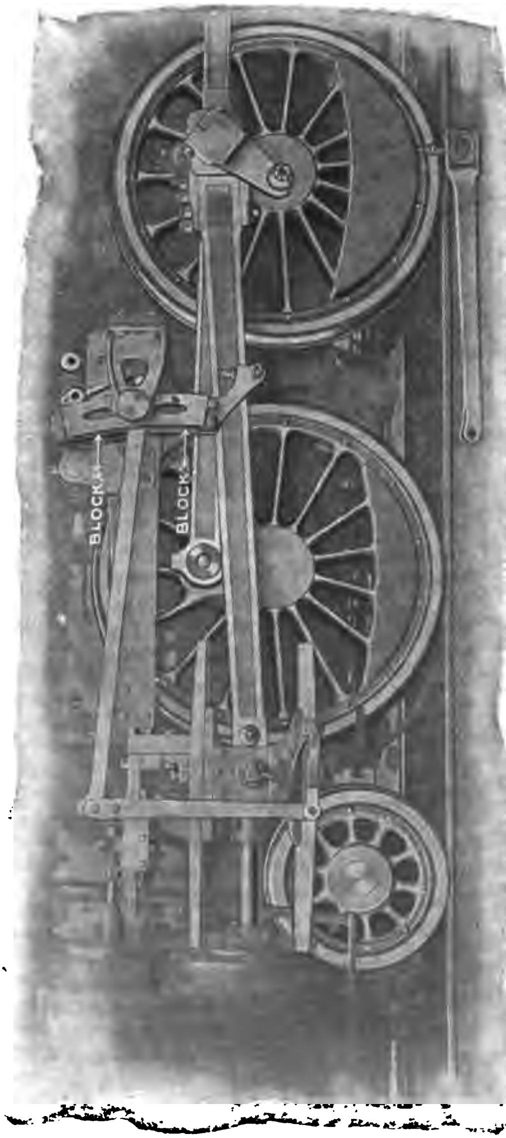


FIG. 123.

The link block may be secured in the center of the link by means of two blocks wedged in position, as shown in the illustration.

With the link block thus secured, care must be taken not to come to a stop with the main pin on the lame side on either quarter. In such an event, the lap and lead lever would be in a perpendicular position, or approximately so; and the valve would be practically central on its seat. As the crank pin on the other side would be on a dead center, it would, thus, be impossible to start the locomotive.

In Fig. 123 the radius rod is connected to the lift shaft arm by means of a link or hanger. It can thus be readily disconnected.

When the radius rod is directly connected to the lift shaft arm by means of a slip block, as shown in Fig. 124, the reverse shaft arm is made in two pieces. To disconnect the radius rod, it is necessary only to remove the outer section of the lift shaft arm and remove the slip block.

Considering the second general class of accidents: those in which the valve has to be blocked to cover the ports. This class, as already stated, may in turn be divided into two other classes: First, cases where it is necessary to take down the main rod; second, those in which the main rod is in condition to run. In the latter cases, the rules governing what is considered good practice as to leaving the main rod up or taking it down have been previously given.

Taking up these two kinds of accidents in their order:

Main Rod Down, Valve Blocked.

Assume that the main rod is broken or the piston rod bent. In the case of the inside-admission valves, the locomotive may be

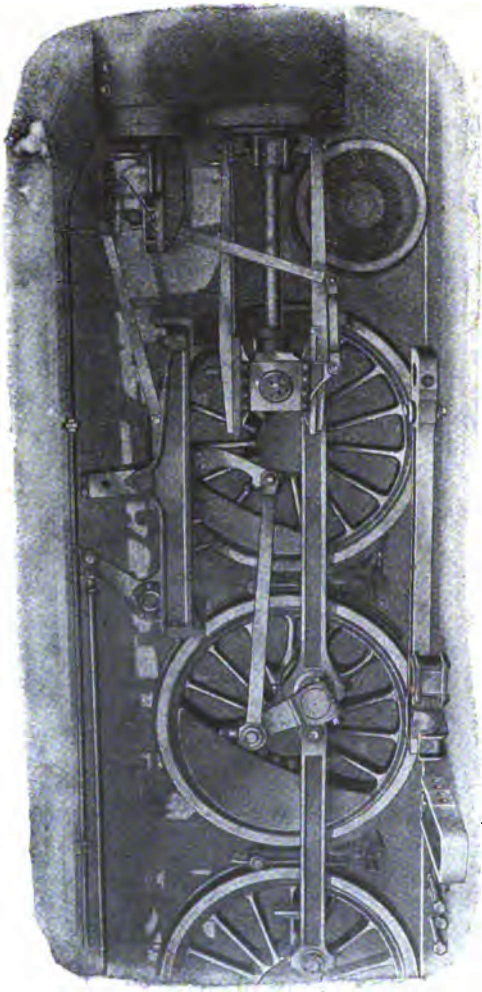


Fig. 124.

disconnected and blocked as shown in Fig. 124. Disconnect the radius rod from the lap and lead lever. Suspend it clear of the latter. Secure the valve to cover the ports. This can usually be done by means of a set screw provided for this purpose. In the design illustrated, the set screw will be noticed on the side of the valve stem guide. If no set screw is provided, a valve stem clamp is usually included in the tool equipment. Clamp or block the crosshead at the back end of the guides as shown.

With the valve motion disconnected in this way, the reverse lever is free to operate the other side; and the locomotive can be run in on one cylinder.

If the crosshead arm, lap and lead lever connector, or lap and lead lever is broken, the same method may be followed. Of course, such of the broken parts as would in any way interfere with running the locomotive would have to be removed.

In the case of *outside* admission valves, the same course may be followed as illustrated in Fig. 124, and described above; except that the lap and lead lever must be taken down. If this were left up, the radius rod would strike it as the latter moved back and forth to the motion of the link.

Main Rod Up, Valve Blocked.

In such a case, if the valve has *inside* admission, the locomotive might be blocked as shown in Fig. 125. For example: suppose the lap and lead lever, lap and lead lever connector, or crosshead arm were broken. Disconnect the radius rod from the lap and lead lever. Suspend it clear of the latter. Tie the lower end of the lap and lead lever ahead to clear the crosshead on the forward stroke. The locomotive can then be run in on one side.

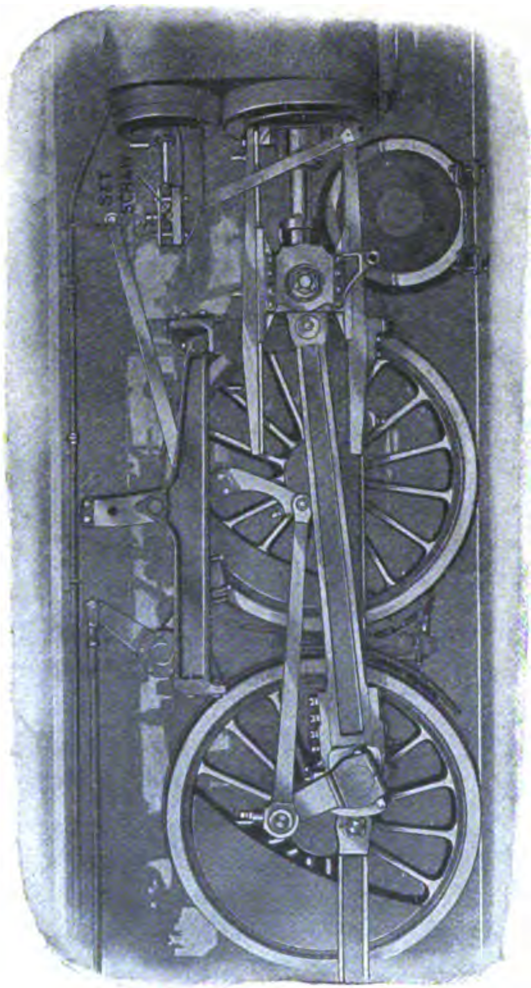


FIG. 125.

If the valves have *outside* admission; under conditions similar to those assumed in the case of Fig. 125, the lap and lead lever must be taken down. In other respects, the same method may be followed.

When the radius rod is suspended, a chain or wire should be used. A rope would be quickly cut through by the motion of the radius rod.

Setting the Valves.*

Assuming that the port marks and dead centers have been located in the manner previously explained (pp. 189-192), proceed as follows:

Setting Outside Admission Slide Valves.

As the distance between the front and back port marks is equal to twice the lap of the valve, when the valve is in position

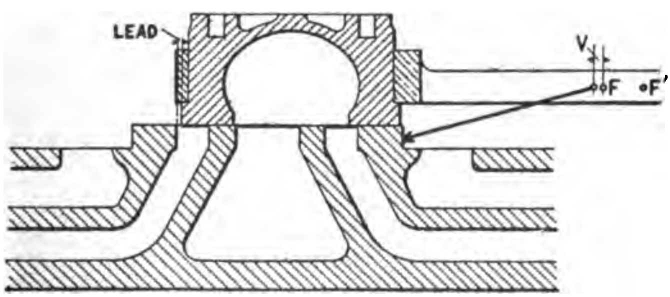


FIG. 126.

for lead opening (main crank on either dead center), the port should be open by an amount equal to the desired lead opening, as shown in Fig. 126.

If the tram is used with the valve on its lead, then the distance between the point so found and the point F on the valve stem will be equal to the lead (see dimension V on Fig. 126).

*Courtesy of The Baldwin Locomotive Works, Philadelphia, Pa.

With this statement in mind, proceed as follows:

Hook up the gear so that the link block is exactly central with the link. Place the main crank on the forward dead center, and tram to the valve stem. Revolve the wheel to the backward dead center, and again tram to the valve stem. Measure the distance between the two points so obtained, and compare the same with the specifications. The distance should be equal to twice the sum of the lap and lead. Variation from the specified

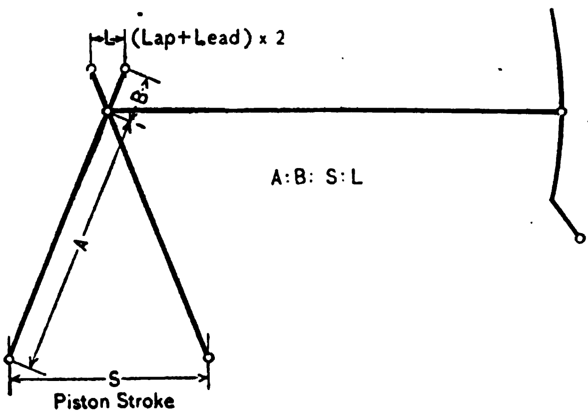


FIG. 127.

figures means that an error exists in the combining lever, the upper and lower arms of which are made respectively proportional in length to twice the lap and lead and to the stroke of the piston. See Fig. 127.

Assuming that the distance L, as trammed on the valve stem, is found correct, the procedure is now as follows:

Place the gear in forward motion, with the link block at a point in the link that will give the specified maximum valve travel when the wheels are revolved in a forward direction (this position of the link block is obtained by experiment).

Place the main crank on the forward dead center, and with the valve tram scribe on the stem, measuring the distance between the point so obtained and the punch mark F. This distance should be exactly equal to the specified lead.

Revolve the wheel in a forward direction until the main crank is on the back dead center, and similarly scribe a line on the valve stem, measuring the distance between the point so obtained and the punch mark F^1. This distance should also be exactly equal to the specified lead.

Place the gear in backward motion, as instructed above for forward motion, and examine for lead at the front and back, exactly as described in the case for forward motion, except that the wheel must be revolved in a backward direction.

If all of the points so found are exactly to specification, the valve setting is square. A check should now be made by placing the piston on the forward dead center, and moving the link block through its entire travel in the link. This should in no way disturb the position of the valve.

With the gear set for full stroke forward and full stroke backward, the maximum travel should be examined with the piston at half stroke. The travel so measured will not be exactly square at the front and back, as this position represents half stroke measured from the piston travel, and does not take into consideration the angularity of the main rod.

The Walschaert gear has inherent peculiarities that frequently cause slight irregularities in the travel of the valve. If, in full gear, this irregularity does not amount to more than one-quarter inch out of the square between the front and back positions of the valve, it may be ignored, and the squaring of the travel may be considered as nearly perfect as the design will allow.

In special cases it may be desirable to square the valve travel when the gear is hooked up to give the cut-off at which the engine is most frequently worked. Thus, on a passenger locomotive, the most satisfactory results may be secured when the travel is squared at one-third or one-half stroke cut-off. In such a case, a slight irregularity at full stroke will probably be unavoidable, but will not prove detrimental. In any event, the particular circumstances under which the engine is to work must determine at what point of cut-off the valve travel shall be squared.

Corrections.

If, on trial, the valve gear is found to be out of square on the lead points, the following examples will serve to explain the corrections that should be made.

For example, suppose the specification calls for the following:

Maximum valve travel, 5½".

Eccentric crank throw, 11".

Constant lead, ¼".

Outside lap of valve, 1".

Link block below link center in forward gear.

It is very important that the following dimensions check exactly with the drawings:

Length of combining lever between central fulcrum and upper and lower arm centers (see Fig. 127, dimensions B and A).

Eccentric crank throw and length of crank arm.

In the case under consideration the prick punch marks on the valve stem will be two inches from center to center (this is twice the valve lap).

A change in the length of the eccentric rod results in a change in the position of the valve, approximately in proportion to the eccentric throw and valve travel. In the present case, this is as eleven to five and one-half, or as two to one. In other words, a change of one-quarter inch in the length of the eccentric rod will move the valve approximately one-eighth inch when the link block is in full gear and the main crank is on the dead center.

The influence of eccentric rod changes on the direction (ahead or back) of the movement of the valve is explained by reference to Fig. 128. An examination of Fig. 128 will show that if the

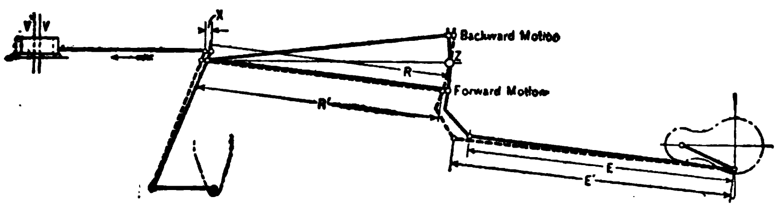


FIG. 128.

eccentric rod E is lengthened to E^1, then the radius rod R will be moved ahead to the position R^1, and the valve stem will be moved a distance X in the direction of the arrow, thus displacing the valve from position V to position V^1.

The following rules can thus be formulated:

If the link block is *below* the link center when running ahead, then——

In forward motion. If the eccentric rod is lengthened the valve is moved ahead.

If the eccentric rod is shortened, the valve is moved back.

In backward motion. If the eccentric rod is lengthened the valve is moved back.

If the eccentric rod is shortened, the valve is moved ahead.

If the link block is *above* the center when running ahead, then, in each case, the valve will be moved in the direction opposite to that stated above.

Corrections made to the link radius rod will have approximately full influence on the movement of the valve, viz.: any variation in the radius rod will produce approximately the same variation in the movement of the valve.

The link fulcrum Z (see Fig. 128) is a fixed point; therefore, the direction of movement due to changes in the radius rod will vary directly with such changes, and the following rules can be formulated.

In either forward motion or backward motion. To move the valve ahead, lengthen the radius rod the amount desired.

To move the valve back, shorten the radius rod the amount desired.

This is true whether the link block is above or below the link center in forward gear.

With these facts in mind, two examples will be considered.

Let it be assumed that, on tramming to the valve stem with the main crank on the dead centers, the following irregularities in the lead are noticed for the engine under consideration. The small circles on the diagrams represent the prick punch (port) marks F and F^1 (see Fig. 126) on the valve stem, while the crosses represent the irregularities in the lead when trammed to the valve stem (see Fig. 129).

The first procedure will be to divide the error between the forward and backward motions, as follows:

Error in forward motion—

$\left.\begin{array}{l}\text{Front, } \tfrac{3}{8}''-\tfrac{1}{4}'' \text{ lead} = \tfrac{1}{8}'' \text{ error} \\ \text{Back, } \tfrac{1}{4}'' \text{ lead}-\tfrac{1}{8}'' = \tfrac{1}{8}'' \text{ error}\end{array}\right\}$ To square the lead, the valve must be moved ⅛" ahead.

Error in backward motion—

Front, 7/16"—¼" lead=3/16" error
Back, ¼" lead—1/16"=3/16" error
To square the lead, the valve must be moved 3/16" ahead.

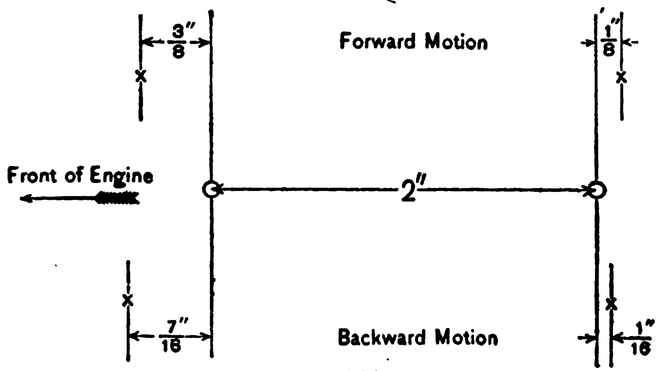


FIG. 129.

As the errors in the two motions occur in the same direction, it follows that the greater one partially neutralizes the effect of the lesser, and that the combined or average error will be the difference between the two, viz.: three-sixteenths of an inch minus one-eighth inch equals one-sixteenth inch average error.

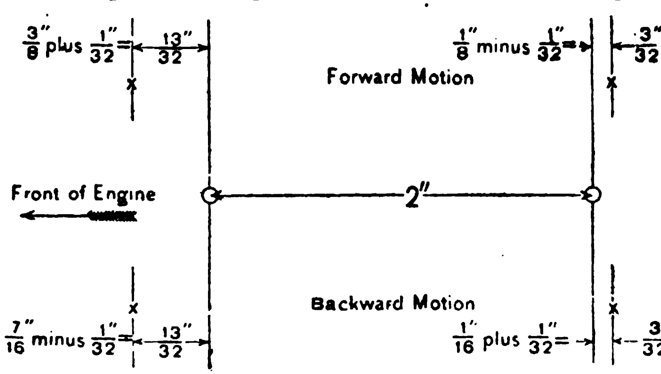


FIG. 130.

To divide an average error of one-sixteenth inch equally about a central point, it will be necessary to move the valve one-half this amount, or one-thirty-second inch (in this case one-thirty-second inch back in forward motion).

According to a preceding statement, the eccentric rod must be shortened one-sixteenth inch (in the proportion of two to one) to move the valve one-thirty-second inch. When this has been done the valve stem points will tram as shown in Fig. 130.

The errors in forward and backward motion have thus been equalized, and it remains only to square the lead front and back for both motions. The valve as now standing is five-thirty-seconds of an inch too far back to equalize the lead, viz.:

13/32"—¼" lead=5/32" error front.
¼" lead—3/32"=5/32" error back.

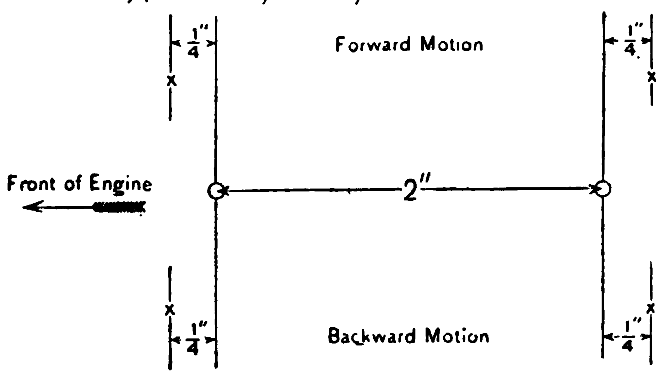


FIG. 131.

As the influence of the radius rod is direct, it follows that by lengthening this rod the amount required (five-thirty-seconds of an inch) the valve will be squared, and can be trammed to the dimensions shown by Fig. 131. These dimensions are the ones required by the specification.

The valve has thus been squared and the errors corrected in the first example, by the changes noted below:

Eccentric rod shortened 1/16".

Link radius rod lengthened 5/32".

A final trial of the valve and cut-off, etc., can now be made in the previously described manner.

Let it be assumed that on tramming for lead, results are obtained as represented by Fig. 132.

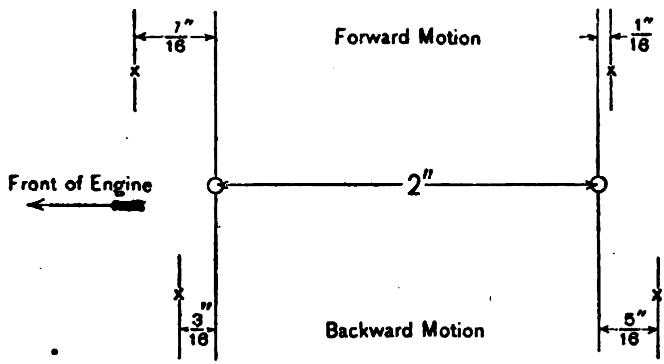


FIG. 132.

Divide the error between the forward and backward motions as follows:

Error in forward motion—

Front, 7/16"—¼" lead=3/16" error
Back, ¼" lead—1/16"=3/16" error
To square the lead, the valve must be moved 3/16" ahead.

Error in backward motion—

Front, ¼" lead—3/16"=1/16" error
Back, 5/16"—¼" lead=1/16" error
To square the lead, the valve must be moved 1/16" back.

As the errors in the two motions occur in opposite directions, it follows that they augment each other, and that the combined or average error will be the sum of the two, viz.: three-sixteenths of an inch plus one-sixteenth inch equals one-quarter inch average error.

To divide this error equally about a central point, it will be necessary to move the valve one-half the amount, or one-eighth inch (in this case one-eighth inch ahead in forward motion).

According to a preceding statement, the eccentric rod must be lengthened one-quarter inch (in the proportion of two to one) to move the valve one-eighth inch. When this has been done, the valve stem will tram as shown in Fig. 133.

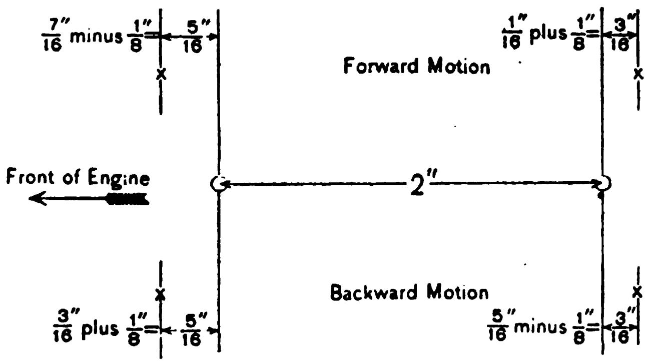


FIG. 133.

The errors in forward and backward motion have thus been equalized, and it remains only to square the lead front and back for both motions. The valve as now standing is one-sixteenth inch too far back to equalize the lead, viz.:

5/16"—¼" lead=1/16" error front.

¼" lead—3/16"=1/16" error back.

To move the valve ahead one-sixteenth inch the link radius rod must be lengthened one-sixteenth inch, and the lead will then be squared. When trammed for lead, the results will be as shown by Fig. 131. These dimensions are the ones required by the specification.

The lead has now been squared, and the errors in the second example have been corrected by the changes noted below:

Eccentric rod lengthened ¼".
Link radius rod lengthened 1/16".

Trial of the valve travel, cut-off etc., can now be made in the manner previously described.

From the above it is evident that the errors in forward and backward motion are equalized by changing the length of the eccentric rod; the lead is then squared by changing the length of the radius rod. Theoretically the radius rod should not be changed, but the amount necessary is so slight, that it makes practically no difference in the movement of the valve.

Setting Inside Admission Piston Valves.

The method of setting inside admission piston valves is generally similar to that previously described. It must be remembered, however, that to perform corresponding functions, this valve moves in a direction opposite to that of the slide valve. When setting piston valves, the steam chest heads should be removed, for the sake of convenience. The line and line positions of the valve, are determined by observation through peep holes provided for the purpose, or by measurement, as previously explained. The test for lead is made as described on page 276, the combining lever occupying positions as shown in Fig. 134.

As in the case of the slide valve, methods used for correcting errors can best be explained by two examples. First, suppose the specification of a locomotive having inside admission piston valves, calls for the following:

Maximum valve travel, 5¾".
Eccentric crank throw, 15½".
Constant lead, ¼".
Steam lap or valve, 1".
Link block below link center in forward gear.

The influence of eccentric rod changes on the direction (ahead or back) of the movement of the valve, is explained by reference to Fig. 135. An examination of this figure will show that if the eccentric rod E is lengthened to E^1. then the radius rod R will be moved ahead to the position R^1, and the valve stem will be moved a distance X in the direction of the arrow, thus displacing the valve from position V to position V^1. Therefore, the rules applying in the case of outside admission slide valves also apply to this style of valve, as follows:

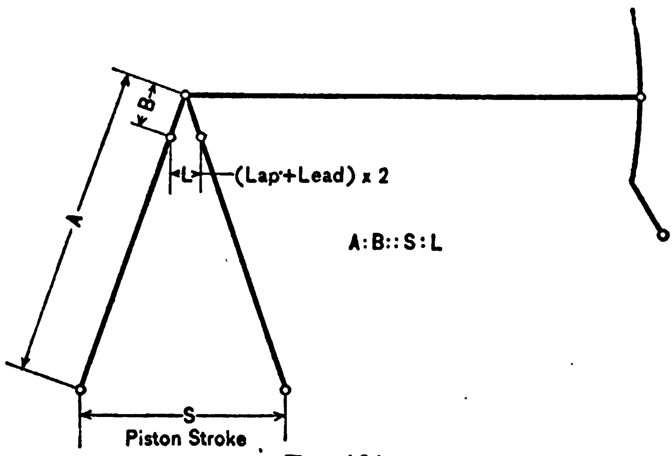


FIG. 134.

In forward motion—If the eccentric rod is lengthened, the valve is moved ahead. If the eccentric rod is shortened, the valve is moved back.

In backward motion—If the eccentric rod is lengthened, the valve is moved back. If the eccentric rod is shortened, the valve is moved ahead.

If the link block is above the center when running ahead, then, in each case, the valve will be moved in the direction opposite to that stated above.

In any case, regardless of whether the gear is in forward or backward motion—To move the valve ahead, lengthen the radius rod the amount desired.

To move the valve back, shorten the radius rod the amount desired.

It must be remembered that with an inside admission valve, the front port opening is increased if the valve is moved ahead, and the rear port opening is increased if the valve is moved back.

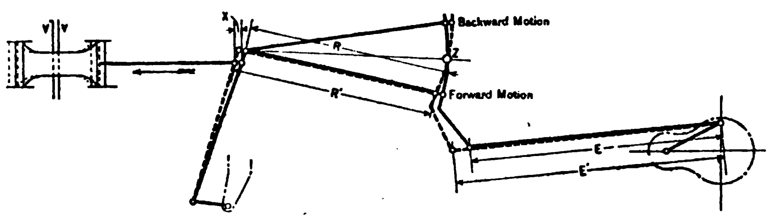


FIG. 135.

Bearing these facts in mind, two examples will now be considered.

Let is be assumed that, on tramming to the valve stem with the main crank on the dead centers, the following irregularities in the lead are noticed for the engine under consideration. The dots on the diagram represent the prick punch marks on the valve stem, while the crosses represent the irregularities in the lead when trammed to the valve stem (see Fig. 129). These irregularities are the same as those used in the corresponding case for slide valves, therefore the same diagrams are referred to. In the present case, however, as the valve is arranged for inside admission, the lead marks for the front steam port will appear on the back end of the valve stem, and those for the back steam port on the front end. In other words, when applying Figs. 129—133 to a locomotive with inside admission valves, the front of the engine should be considered on the right instead of on the left. The terms "front" and "back" in the text apply to the steam ports, and not to the positions of the marks on the valve stem.

The first procedure will be to divide the error between the forward and backward motions, as follows:

Error in forward motion—

Front, 3/8"—1/4" lead=1/8" error Back, 1/4" lead—1/8"=1/8" error	To square the lead, the valve must be moved 1/8" back.

Error in backward motion—

Front, 7/16"—1/4" lead=3/16" error Back, 1/4" lead—1/16"=3/16" error	To square the lead, the valve must be moved 3/16" back.

As the errors in the two motions occur in the same direction, it follows that the greater one partially neutralizes the effect of the lesser, and that the combined or average error will be the difference between the two, viz.: three-sixteenths of an inch minus one-eighth inch equals one-sixteenth inch average error.

To divide an average error of one-sixteenth inch equally about a central point, it will be necessary to move the valve one-half the amount or one-thirty-second inch (in this case, one-thirty-second inch ahead in forward motion).

In the engine now under consideration, the eccentric crank throw is fifteen and one-half inches and the valve travel five and three-quarters inches. Hence the ratio of eccentric throw to valve travel is approximately as two and seven-tenths to one. Therefore, according to a preceding statement, the eccentric rod must be lengthened two and seven-tenths times one-thirty-second, or approximately five-sixty-fourths of an inch to move the valve ahead one-thirty-second inch. When this has been done, the valve stem points will tram as shown in Fig. 130.

The errors in forward and backward motion have thus been equalized, and it remains only to square the lead front and back for both motions. The valve as now standing is five-thirty-seconds of an inch too far ahead to equalize the lead, viz.:

13/32"—¼" lead=5/32" error front.
¼" lead—3/32"=5/32" error back.

As the influence of the radius rod is direct, it follows that by shortening the rod five-thirty-seconds of an inch, the valve will be moved back that amount and the lead squared. The valve stem can then be trammed to the dimensions shown in Fig. 131. These dimensions are the ones required by the specification.

The valve has thus been squared and the errors corrected in our first example, by the changes noted below:

Eccentric rod lengthened 5/64".
Radius rod shortened 5/32".

A final trial of the valve, cut-off etc., can now be made in the previously described manner.

Let it be assumed that on tramming for lead, results are obtained as represented by Fig. 132.

Divide the error between the forward and backward motions, as follows:

Error in forward motion—

Front, 7/16"—¼" lead=3/16" error Back, ¼" lead—1/16"=3/16" error	To square the lead, the valve must be moved 3/16" back.

Error in backward motion—

Front, ¼" lead—3/16"=1/16" error Back, 5/16"—¼" lead=1/16" error	To square the lead, the valve must be moved 1/16" ahead.

As the errors in the two motions occur in opposite directions they augment each other, and the combined or average error will be the sum of the two, viz.: three-sixteenths of an inch plus one-sixteenth inch equals one-quarter inch average error.

To divide the error equally about a central point, it will be necessary to move the valve one-half the amount, or one-eighth inch (in this case one-eighth inch back in forward motion).

According to the rule previously stated, the eccentric rod must be shortened two and seven-tenths times one-eighth inch, or approximately eleven-thirty-seconds of an inch, to move the valve one-eighth inch. When this has been done the valve will tram as shown in Fig. 133.

The errors in forward and backward motion have thus been equalized, and it remains only to square the lead front and back for both motions. The valve as now standing is one-sixteenth inch too far front to equalize the lead, viz.:

5/16"—¼" lead=1/16" error front.
¼" lead—3/16"=1/16" error back.

To move the valve back one-sixteenth inch, the radius rod must be shortened one-sixteenth inch, and the lead will then be squared. When trammed for lead, the results will be as shown in Fig. 131. These dimensions are the ones required by the specification.

The lead has been squared and the errors in our second example have been corrected by the changes noted below:

Eccentric rod shortened 11/32".
Link radius rod shortened 1/16".

Trial of the valve travel, cut-off etc., can now be made in the manner previously described.

The accompanying table has been found by the American Locomotive Company to give goods results in setting the valves of the Walschaerts Valve Gear.

	Valve			Per Cent Cut Off Full Gear	Running Cut Off		Exhaust Clearance		Cylinder	
	Travel	Lap	Lead		Per Cent Stroke	Port Opening	Piston Valve	Slide Valve	Diam.	Area
	Ins.	Ins.	Ins.			Ins.	Ins.	Ins.	Ins.	Sq. Ins.
Fast Passenger	5	7/8	1/4	83 0	25	19/64	1/4	1/8	17 -18	227-254
	5½	15/16		84 2		5/16			18½-20	269-314
	6	1 1/16		83 3		21/64			20½-22	330-380
	6	1 1/16		83 3		21/64			22½-24	398-452
	6½	1 1/8		84 0		11/32			24½-26	471-531
	7	1 1/4		83 6		23/64			26½ up	552 up
Passenger	5	7/8	3/16	84 0	25	1/4	3/16	1/8	17 -18	227-254
	5½	15/16		85 0		17/64			18½-20	269-314
	6	1 1/16		84 0		9/32			20½-22	330-380
	6	1 1/16		84 0		9/32			22½-24	398-452
	6½	1 1/8		85 0		19/64			24½-26	471-531
	7	1 1/4		84 4		5/16			26½ up	552 up
Fast Freight	5	7/8	3/16	84 0	33	5/16	1/8	1/16	17 -18	227-254
	5½	15/16		85 0		21/64			18½-20	269-314
	6	1 1/16		84 0		23/64			20½-22	330-380
	6	1 1/16		84 0		23/64			22½-24	398-452
	6½	1 1/8		85 0		3/8			24½-26	471-531
	7	1 1/4		84.4		25/64			26½ up	552 up
Freight	5	13/16	1/8	86.9	50	27/64	0	0	17 -18	227-254
	5½	7/8		87 5		29/64			18½-20	269-314
	6	1		86.5		1/2			20½-22	330-380
	6	1		86.5		1/2			22½-24	398-452
	6½	1 1/16		87.1		17/32			24½-26	471-532
	7	1 3/16		86 5		37/64			26½ up	552 up
Switching	5	15/16	1/16	87.7	66	41/64	0	0	17 -20	227-314
	5½	7/8		88 2		11/16			20½-22	330-380
	6	1		87.2		25/32			22½ up	398 up
Light Locomotives	2½	13/32	1/16	86.9	50	7/32	0	0	5 - 6	20- 28
	3	1/2		86.5		1/4			6¼- 8	31- 50
	3½	19/32		86.5		9/32			8¼-10	53- 79
	4	5/8		87.0		5/16			10½-13	87-133
	4½	3/4		86.8		23/64			13½-16	143-201
	5	13/16		87.7		3/8			16½ up	214 up

THE BAKER VALVE GEAR.

The Baker valve gear is a development of the Baker valve gear which was originally designed for traction engines, and was patented in March, 1903.

In 1908 the valve gear, then known as the Baker-Pilliod, was first applied to a locomotive on the Toledo, St. Louis & Western Railroad, and, after passing through a period of experiment, it was discontinued in 1910, being superseded by the present Baker valve gear which was patented in 1911, and is now in use on about 6,000 locomotives, including 850 built for the United States Railroad Administration.

Description.

The Baker valve gear is an outside radial gear which receives its motion from two points: namely, the eccentric crank attached to the main crank pin, and the main crosshead. It does not use the link and block common to other gears, and, as a result, there is an absence of sliding friction. The reverse bars and reverse yoke, used in the Baker valve gear, take the place of the link used in the Walschaert and Stephenson valve gears.

The combination lever of both the Baker and Walschaert types of gear moves the valve the amount of lap and lead in each direction, while the balance of the valve travel is obtained from the eccentric crank. In the Baker gear, the eccentric crank transmits its motion through the eccentric rod to the gear connecting rod, radius bar, bell crank and valve rod, dispensing with the oscillating radial link, link block, and radius bar employed in the

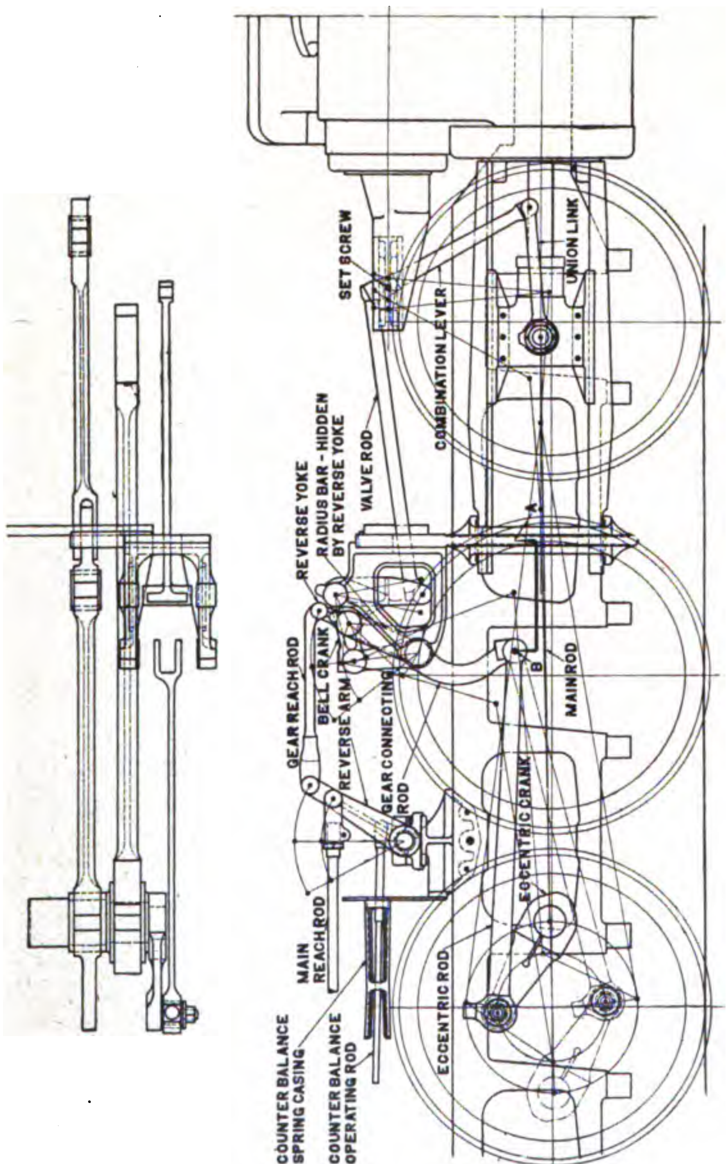

SET SCREW
UNION LINK
COMBINATION LEVER
REVERSE YOKE
RADIUS BAR - HIDDEN
BY REVERSE YOKE
VALVE ROD
A
B
GEAR REACH ROD
BELL CRANK
REVERSE ARM
GEAR CONNECTING
ROD
MAIN ROD
ECCENTRIC CRANK
MAIN
REACH ROD
ECCENTRIC ROD
COUNTER BALANCE
SPRING CASING
COUNTER BALANCE
OPERATING ROD

FIG. 136.

Walschaert gear, and thus avoids the usual variation occasioned by the slip of the link block.

As shown in Fig. 136, this gear consists of an eccentric crank, eccentric rod, gear frame (which contains the reverse yoke, radius bar, bell crank and gear connecting rod), valve rod,

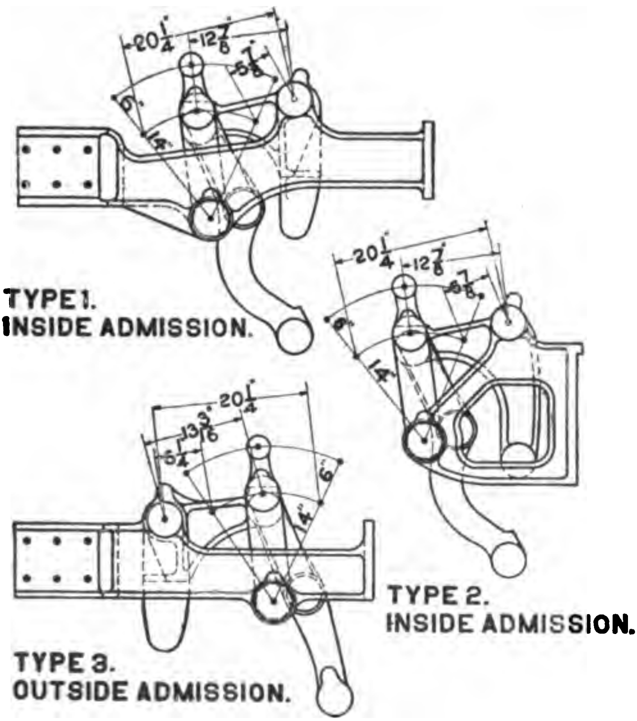


FIG. 137.

combination lever and union link, together with the customary reversing arrangement of reach rods, reverse shaft, reverse lever, etc.

The movement of the valve is controlled by the reverse yoke, shown in detail in Fig. 137 which is pivoted in the gear frame and at its upper end is attached to the gear reach rod. In the reverse yoke are pivoted the radius bars, to the lower ends of which the gear connecting rod is attached. This gear connecting rod extends downward, connecting with the eccentric rod, and upward, connecting with the bell crank which is pivoted in the gear frame. This may be seen in Fig. 138, showing a section through the gear (inside admission). The motion derived from the eccentric crank (the reversing movement) is carried through the valve gear to the bell crank. It is here combined with the motion of the combination lever (lap and lead movement) and transmitted to the valve by means of the valve rod and valve stem. The combination lever can be attached to the bell crank or to the valve stem crosshead. The principle is the same in either case.

The minimum vertical movement of the gear connecting rod is obtained with the reverse yoke in mid-gear, and, as the reverse yoke is moved toward either extremity, the valve travel is increased through the increased oscillation of the bell crank, by means of a greater vertical movement of the three gear connecting rod bearings.

A movement of the lower end of the gear connecting rod to the left, with the reverse yoke in forward motion, produces a lifting action of the gear connecting rod, which moves the valve back by revolving the bell crank clock-wise. This same movement of the lower end of the gear connecting rod will produce a falling action of the gear connecting rod with the reverse yoke in the back motion, which reverses the movement of the bell crank and valve. The crosshead connection moves the valve the amount of lap plus lead in each direction. This makes the lead constant, and independent of the cut-off.

The Baker valve gear is direct in the forward motion and indirect in the back motion for inside admission, and the opposite for outside admission.

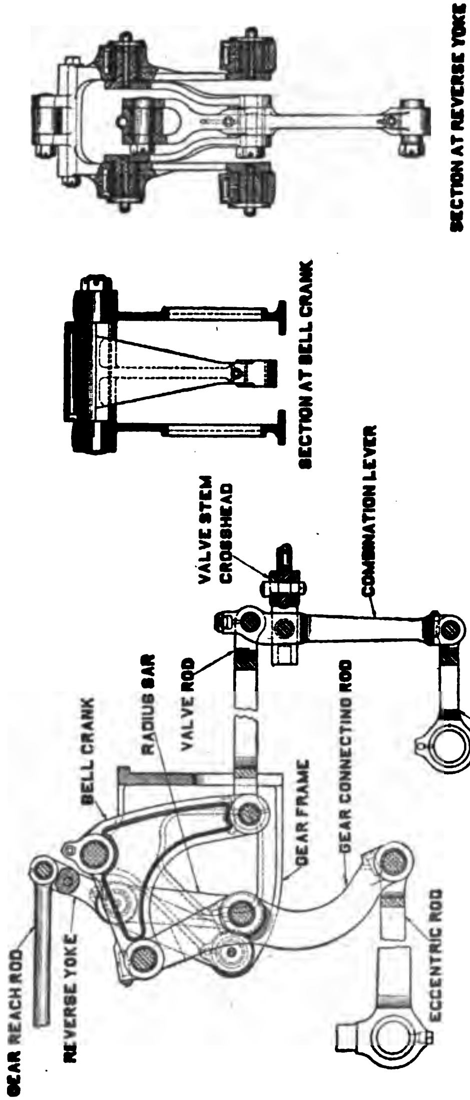

SECTION AT REVERSE YOKE
SECTION AT BELL CRANK
VALVE STEM CROSSHEAD
COMBINATION LEVER
VALVE ROD
RADIUS BAR
BELL CRANK
GEAR CONNECTING ROD
GEAR FRAME
UNION LINK
GEAR REACH ROD
REVERSE YOKE
ECCENTRIC ROD

FIG. 138.

Principle of Operation.

Inside Admission. The action of the Baker valve gear may be best learned by tracing the movement of the valve through a complete revolution of the wheels. Figs. 139 to 142 show a series of diagrams representing the different positions of the crank pin. For the sake of simplicity, the valve and cylinder are shown in section. The other parts of the gear are represented by their center points only. These diagrams are purposely drawn out of proportion. The valve has been enlarged in order to show more clearly the positions of the edges of the valve, relative to the edges of the cylinder ports.

Figs. 141 and 142 represent the Baker valve gear as arranged for piston valves, inside admission.

Forward Motion, Full Stroke. Fig 139, diagram 1, illustrates the engine with the reverse lever in full forward motion. The engine is represented as being on the back dead center. As the wheel is rotated in the direction of the arrow, the eccentric rod moves away from the cylinder, carrying with it the gear connecting rod to which it is coupled. This produces a movement of the radius bars also away from the cylinder, which in turn sets up a lifting action on the horizontal arm of the bell crank, thus moving the valve backward. This operation continues until the crank pin practically reaches the top quarter, when the eccentric rod movement reverses (because of the action of the eccentric crank), with the result that the valve starts to close.

Fig. 139, diagram 2, shows the valve in the act of closing the rear cylinder port, or in the position of crank end cut-off. As the wheel continues to rotate in the direction of the arrow, beyond the point shown in diagram 2, the eccentric rod continues to move forward, producing, by means of the gear connecting rod

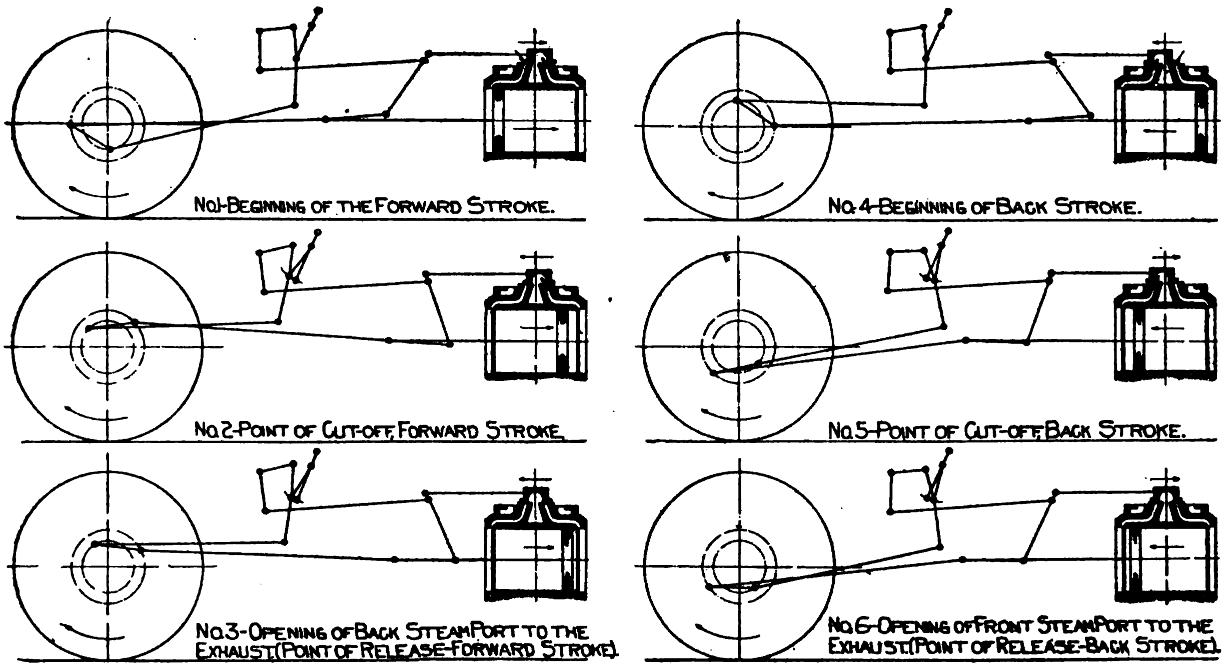

No.1-Beginning of the Forward Stroke.
No.2-Point of Cut-off, Forward Stroke.
No.3-Opening of Back Steam Port to the Exhaust. (Point of Release-Forward Stroke).
No.4-Beginning of Back Stroke.
No.5-Point of Cut-off, Back Stroke.
No.6-Opening of Front Steam Port to the Exhaust. (Point of Release-Back Stroke).

Fig. 139.

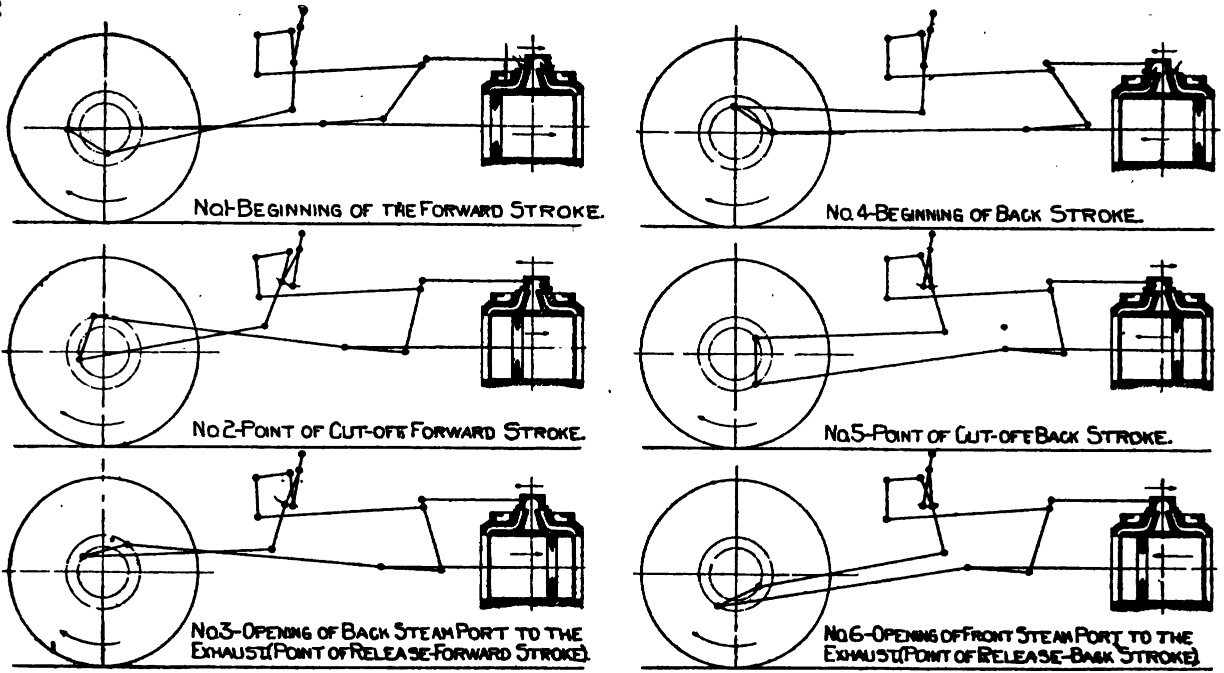

No 1-Beginning of the Forward Stroke.
No 4-Beginning of Back Stroke.
No 2-Point of Cut-off Forward Stroke.
No 5-Point of Cut-off Back Stroke.
No 3-Opening of Back Steam Port to the Exhaust (Point of Release-Forward Stroke)
No 6-Opening of Front Steam Port to the Exhaust (Point of Release-Back Stroke)

FIG. 140.

and radius bars, a downward movement on the horizontal arm of the bell crank, thus continuing to move the valve towards the front, until the engine reaches practically the bottom quarter, when the direction of movement of the valve is again reversed in the manner described above. From the description it will be noticed the Baker valve gear, inside admission is direct in forward motion; that is, the valve moves in the same direction as the eccentric rod.

Forward Motion Running Positions. The position of the reverse yoke governs the amount of valve travel, and also the percentage of cut-off obtained. As the reverse yoke is brought nearer its mid-position the arc of the radius bars becomes more nearly horizontal, and produces a less upward and downward movement of the horizontal arm of the bell crank, thus decreasing the valve travel.

Fig. 140 shows the engine in the same position as Fig. 139, with the exception that the reverse yoke has been brought nearer to the central position, and shows the effect on the various valve events of "hooking up" the reverse yoke. For example, it will be noticed that (in Fig. 140) the valve cuts off steam from the cylinders and opens the ports to the exhaust at a much earlier period in the piston stroke than it does in Fig. 139. The latter figure represents the engine with the reverse yoke in the full stroke position, while Fig. 140 represents the engine in a shorter *cut-off* position.

Backward Motion. The reversing of the Baker valve gear is accomplished by bringing the radius bar bearings in the reverse yoke past the central position, away from the bell crank. The movement of the eccentric rod away from the cylinder then produces a downward action on the upper arm of the bell crank, causing the valve to move forward. This is just the opposite

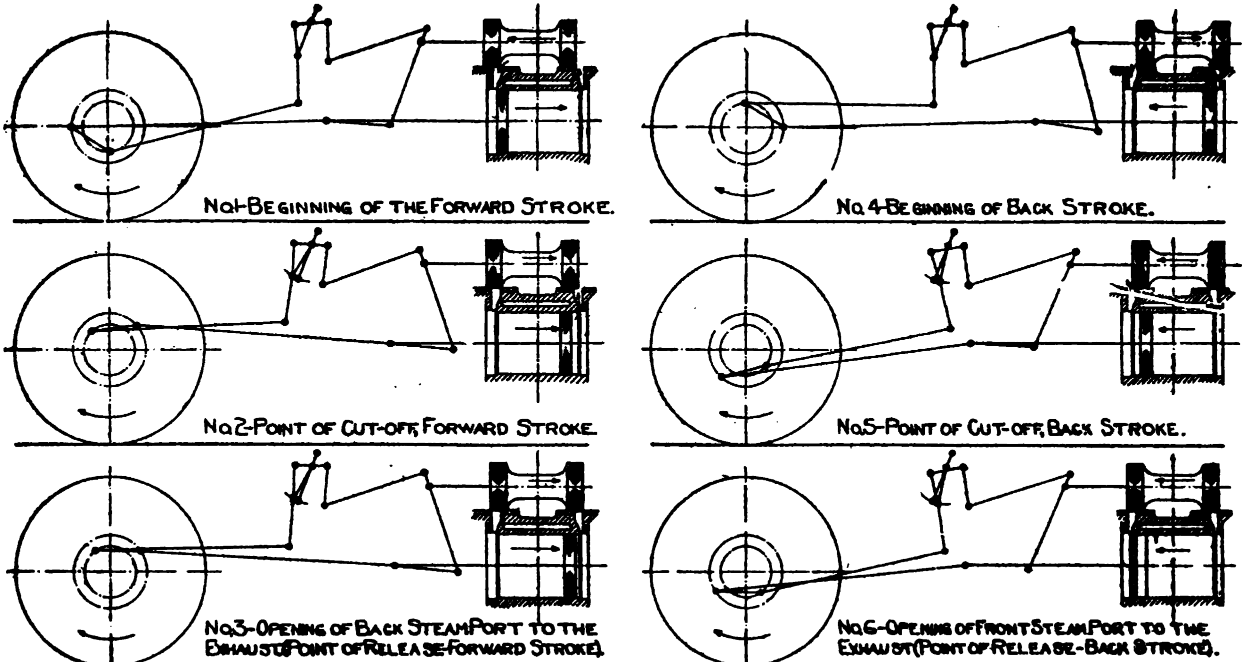

No 1-Beginning of the Forward Stroke.
No 2-Point of Cut-off, Forward Stroke.
No 3-Opening of Back Steam Port to the Exhaust (Point of Release-Forward Stroke).
No 4-Beginning of Back Stroke.
No 5-Point of Cut-off, Back Stroke.
No 6-Opening of Front Steam Port to the Exhaust (Point of Release-Back Stroke).

Fig. 141.

to the effect of this eccentric rod movement with the reverse yoke in forward motion. The inside admission gear is therefore indirect in back motion; that is, the valve moves in a direction opposite to the movement of the eccentric rod. The "hooking up" is accomplished exactly the same in backward as in forward motion.

Mid-Gear Position. The Baker valve gear is in mid-position when the radius bar bearings in the reverse yoke are exactly in line with the reverse yoke bearing in the frame, and the bell crank gear connecting rod bearing.

Outside Admission. The essential difference between the two gears is that in the outside admission gear the reverse yoke is in front of the bell crank, and moves away from the bell crank for forward motion, while in the inside admission gears this movement is just the opposite. The eccentric crank location is the same. Figs. 139 and 140 represent the Baker valve gear, as arranged for slide valve, outside admission. A comparison of these figures with Figs. 142 and 141 brings out clearly the difference in arrangement.

Lap and Lead Movement. The lap and lead movement is obtained from the combination lever by simply proportioning this lever so that when its long arm receives a movement equal to the piston stroke, its short arm will deliver to the valve a movement equal to two times the sum of the lap plus the lead. This movement is separate from and entirely independent of the rest of the valve gear action and, therefore, remains constant, regardless of the reverse yoke or the reverse lever positions.

To Change the Lead. To change the lead of the Baker valve gear, it is necessary to change either the lap of the valve or the distance between the connecting points of the combination lever Reducing the lap of the valve increases the lead, and vice versa

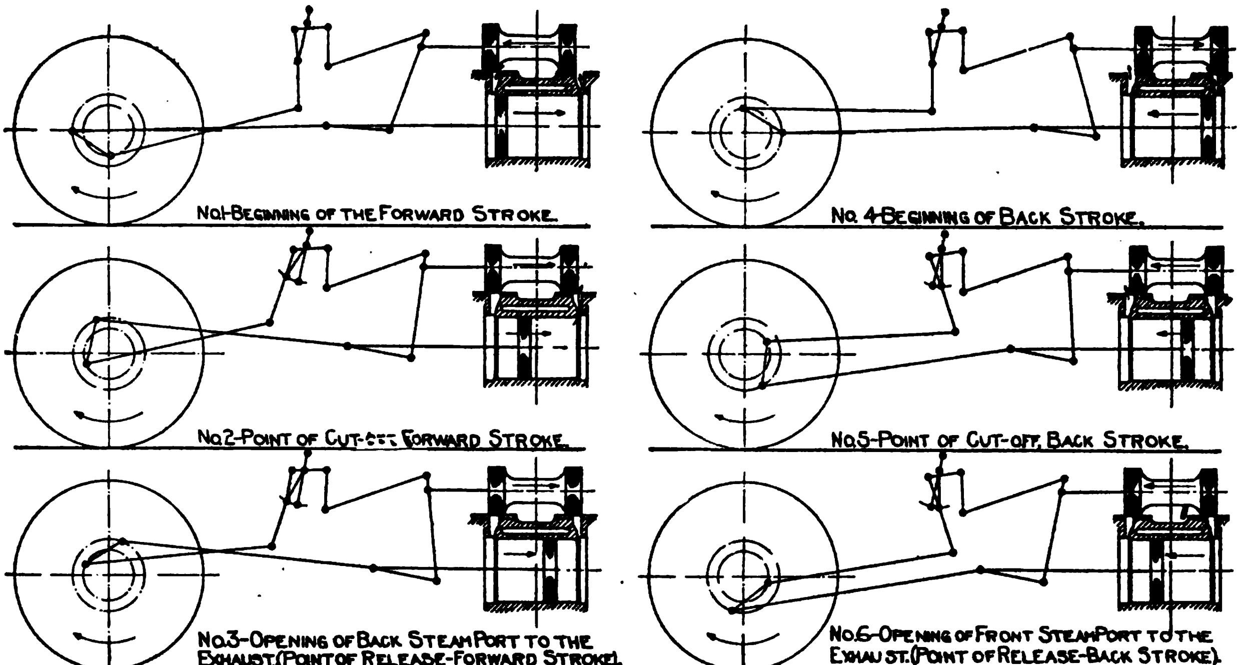

No.1-Beginning of the Forward Stroke.
No. 4-Beginning of Back Stroke.
No.2-Point of Cut-off Forward Stroke.
No.5-Point of Cut-off, Back Stroke.
No.3-Opening of Back Steam Port to the Exhaust. (Point of Release-Forward Stroke).
No.6-Opening of Front Steam Port to the Exhaust. (Point of Release-Back Stroke).

Fig. 142.

Changing the lap also changes the points of cut-off. The effect of reducing the lap is therefore not only to increase the lead, but also to increase the point of cut-off. The effect of increasing the lap will be just the opposite. To increase the lead by means of a change in the combination lever, it is necessary to make the upper arm longer, or to make the lower arm shorter. To decrease the lead these changes are just the opposite.

Rules. The reverse yoke moves forward in the forward motion.

The eccentric crank follows the main pin in the forward motion, and leads it in the backward motion.

Combination levers for inside admisson valves have the valve stem attached below the valve rod connection.

Combination levers for outside admission have the valve stem attached above the valve rod connection.

Eccentric Crank Setting.

Locate the eccentric crank on the main crank pin as nearly as possible to specified throw, and clamp it temporarily. Catch each of the four dead centers, in the manner previously explained, and, with a tram of convenient length, tram from the center of the eccentric rod pin to any convenient stationary point, such as the guide or guide yoke. (See Fig. 136.) The scribing is done with the "A" end of the tram, and this should be as nearly on a horizontal line with the front end of the eccentric rod as circumstances will permit. The lines obtained on the front and back dead centers should come exactly together. If they do not, the crank should be moved in the required direction until the lines do come together.

The object to be accomplished is to so locate the eccentric crank that it will (by means of the eccentric rod) place the gear connection rod in exactly the same point on each dead center. This is essential, for otherwise it is impossible to obtain the same lead condition on both front and back dead centers, and in both forward and backward motions. Having set the eccentric crank as explained above, no further change should be required, and it may be permanently secured.

Valve Rod and Eccentric Rod Adjustments.

The errors in these rods manifest themselves by distorting the entire steam distribution. The most convenient method, however,

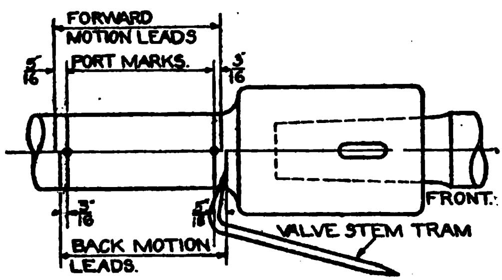


FIG. 143.

of calculating these errors is through their effect on the lead. The methods herein described have been found generally satisfactory, although there are several other good ways.

Place the reverse lever in full forward position. Rotate the wheels forward and catch successively each of the four dead centers, and scribe the valve stem to show the lead.

Pace the reverse lever in full backward position. Rotate the wheels backward, and scribe the valve stem to show the lead at each dead center. (Figs. 143 and 144 illustrate two convenient valve stem trams.)

Scribe all forward motion lines above the center line of the valve stem, and all back motion lines below the center line, to avoid mistakes.

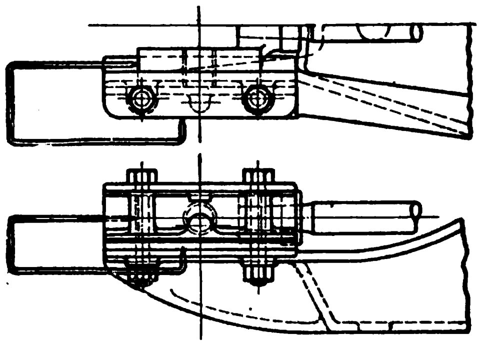


FIG 144.

Example 1. Assuming the condition that has been obtained to be as represented by Fig. 143, the valve rod error is determined first as follows: Add the leads on one end to the leads on the other end; subtract the smaller from the greater and divide

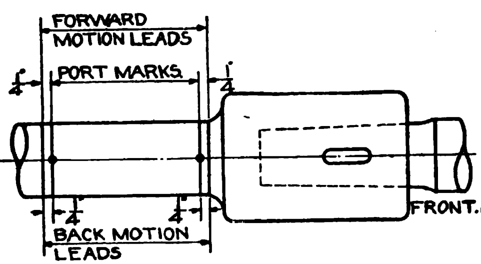


FIG. 145.

the result by four. The result of this calculation is the amount of change necessary to correct the valve rod and is always made in the direction necessary to correct the greater or controlling error.

In the above example the valve rod is found to be correct, because the result of the calculation is zero. The error, therefore, must be in the eccentric rod.

As the Forward motion leads 5/16"+3/16"=8/16", or 1/2"
And the Backward motion leads 3/16"+5/16"=8/16", or 1/2"
Then 1/2"—1/2"=0

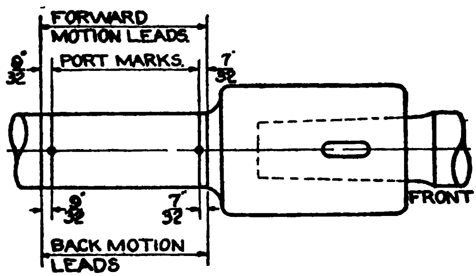


FIG. 146.

Example 2. Eccentric rod error is determined as follows: Subtract the lead on one end in one motion from the lead on the opposite end in the same motion, and divide the result by two. This gives the amount the valve must be moved to correct the reading. The ratio of the Baker valve gear is four to one; that is, the eccentric rod must be moved four times the amount of desired valve movement, or in this case—

As the Forward motion leads 5/16"—3/16"=2/16", or 1/8"
And the Backward motion leads 5/16"—3/16"=2/16", or 1/8"
1/8"÷2=1/16" Valve movement.
Then 1/16"×4=1/4" Change required in eccentric rod.

Therefore, the eccentric rod must be altered 1/4" to produce a movement of 1/16" in the valve. As has been previously explained, the Baker valve gear is direct in one motion and indirect in the opposite motion; consequently, the change in the eccentric rod will move the valve in one direction in forward motion and in the opposite direction in backward motion. This change will, therefore, correct the example, so that it will read 1/4" all around. (See Fig. 145.)

In inside admission gears the eccentric rod should be changed in the same direction the valve should move to correct the forward motion. In outside admission gears this is just the opposite.

Example 3. Assume the condition obtained to be as represented by Fig. 146. By calculating for the valve rod change in the same manner it is found that an alteration of 1/32" is necessary, and that this will correct the entire reading, and therefore no change on the eccentric rod is required. (See Fig. 146.)

Note: These three examples are purposely made up in this manner to make clear the result of an error in either the valve rod or eccentric rod when one rod is correct. When the valve rod alone requires adjustment it will be noted that the valve calls for a movement bodily in one direction. When the valve rod is correct and the eccentric rod is incorrect, the valve will require a movement in one direction to correct one motion, and in the opposite direction to correct the other motion.

Customarily, in valve setting, it will be found that the errors consist of a combination of valve rod and eccentric rod errors, but with the principle of the changes clearly understood no difficulty will be experienced in arriving at the correct result.

Example 4. Shows a combination of valve rod and eccentric rod errors and the method of calculating for the adjustment.

As the Forward motion leads 3/8"+1/16"=7/16"
And the Backward motion leads 1/8"+7/16"=9/16"
Then 9/16"—7/16"=1/8"

1/8"÷4=1/32" Change on valve rod to be made in the direction necessary to correct greatest, or controlling error. The effect of the change to the valve rod 1/32" would be to make the example read:

Forward motion leads $13/32'' + 3/32'' = 1/2''$
Backward motion leads $3/32'' + 13/32'' = 1/2''$
Therefore, $1/2'' - 1/2'' = 0$

This change, it will be noticed, has still further distorted the forward motion, although it has improved the backward motion, but—the error is now of the same amount in both motions, and in opposite directions.

Eccentric Rod Adjustment.

This is determined as follows: Subtract the smaller lead in one motion from the larger in the same motion and divide the result by two. This will be the amount the valve must be moved to correct the error.

$13/32'' - 3/32'' = 10/32'' \div 2 = 5/32''$, or
$3/32'' + 13/32'' = 10/32'' \div 2 = 5/32''$ valve movement.
$5/32'' \times 4 = 5/8''$ Change required in eccentric rod.

Explanation. The above figures show that in forward motion the valve should be moved ahead $5/32''$, and in backward motion the valve should be moved back $5/32''$ to square the lead.

Because this gear is direct in one motion and indirect in the other, the effect of changing the length of the eccentric rod is to move the valve in one direction in forward motion and in the opposite direction in backward motion. Consequently lengthening the eccentric rod a sufficient amount to move the valve $5/32''$ will correct the error in both motions, and will make the lead $1/4''$ on all four points. The eccentric rod should be altered four times the required valve movement, or $5/32'' \times 4 = 5/8''$.

Rules: *Valve Rod: The valve rod change is always in the direction to correct the greatest error. Eccentric Rod: On inside admission gears the change in the eccentric rod is always in* *the direction the valve should move to correct the forward motion. On outside admission gears this change is just the opposite.*

The valve setter, by experience, will be able to calculate and make the necessary changes in the valve rod and eccentric rod at the same time. The valve rod adjustment should always be determined first, as previously stated.

When the changes in the valve rod and eccentric rod, as outlined above, have been made, the lead should be measured on the four dead centers in each direction, to check the work.

At the time of checking the leads the extreme travel points of the main crosshead should be marked on the guide with a scriber, as these lines represent the extreme travel of the crosshead and are for use later when measuring the *Cut-off, Release* and *Compression.*

Adjusting Full Gear Valve Travel.

The full gear valve travels can be corrected and equalized between the two sides of the engine while checking the lead, as above. The quadrant, or the power reverse gears, should be laid out for extreme travel points. When lugs to limit the travel of the reverse yoke are provided on the valve gear frames, these should be chipped to correct height. The reverse yokes should touch these lugs when the maximum specified valve travel is being produced but—the power reverse gear should be adjusted so that the yokes will clear the lugs ¼". The lugs are for the purpose of providing a positive stop against creeping. In the case of engines equipped with a reverse lever or screw reverse, the valve travel can be limited by the reversing mechanism, and the lugs disregarded.

The full gear positions of the reverse yoke for inside admission should not be less than 5⅞" from the bell crank pivot in the forward motion, and not greater than 20¼" in the back motion, as shown in Fig. 137. With outside admission gear, Fig. 137, this dimension for the forward motion positions of reverse yoke should not exceed 20¼", and 6¼" in the back motion.

These figures should be checked after the boiler has been fired up, to correct errors caused by boiler expansion.

Checking the Cut-Off.

After the leads have been equalized the *Cut-off* should be measured in at least one position of the reverse lever, usually running position in forward motion. On passenger engines it is customary to measure it at 25% cut-off position, and on freight engines at 33%. If the cut-off does not vary more than ¾" the valve setting may be considered correct. Occasionally slight additional adjustment may be required to keep within this limit.

Breakdowns.

MAIN ROD UP, VALVE NOT BLOCKED.

If eccentric crank, eccentric rod, connecting rod, radius bar, horizontal arm of bell crank or reverse yoke below reach rod connection break.

(See Fig. 147.) Take down eccentric rod, block bell crank (if it is of the two-arm type plumb it in a vertical position, if of single arm type, make lower bearing 1¼ inches ahead of upper bearing); when bell crank is thus secured, care must be taken not to come to a stop with the main pin on the disabled side on either quarter. In such an event, the combination lever will be in a perpendicular position, or approximately so; as the main pin on the opposite side would be on dead center, it would thus be impossible to start the locomotive. When any of the above accidents occur, the valve on the disabled side then re-

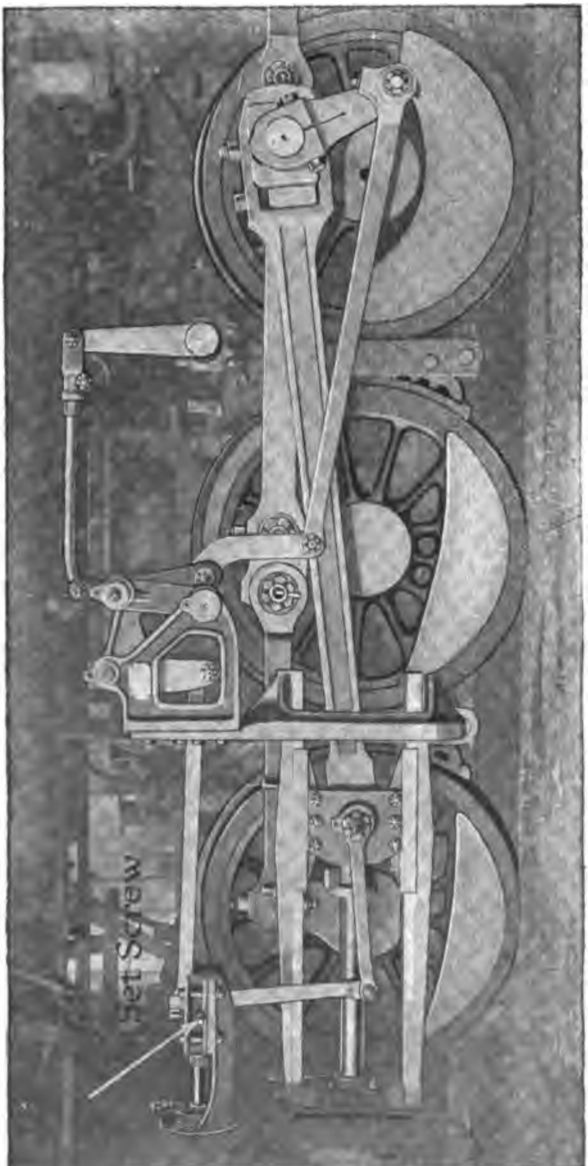


FIG. 147.

ceives a motion from the combination lever. Its travel will be equal to twice the total amount of the lap plus the lead. This gives a port opening equal to the amount of lead, consequently, the main rod may be left up, as the cylinder will be lubricated. Furthermore, though the cut-off on the disabled side will be very short, the steam that is admitted will do a certain amount of work; and the engine can be reversed.

If gear reach rod, reverse arms or reverse yoke at reach rod connection break.

Remove broken parts, block disabled side in desired position and proceed with train (it being understood that engine should not be reversed without first changing the blocking on the disabled side).

If main reach rod, reverse shaft bearing or reverse arm to which main reach rod is connected fail.

Block both reverse yokes at desired cut-off controlling speed with throttle.

The second class of accidents; those in which it is necessary to cover the ports. This class is subdivided into two other classes; first, cases where it is necessary to take down the main rod; second, those in which the main rod is in condition to run. Taking up these two kinds of accidents in their order.

MAIN ROD DOWN, VALVE BLOCKED.

If the main rod, main crosshead, guides or piston is damaged. (See Fig. 148.) Take down main rod and valve rod, leaving rest of gear intact. (If main rod is of the solid rear end type, making it necessary to remove eccentric rod and crank, valve rod may be left up.) Place valve over ports. It can readily be determined if valve is properly placed by applying brakes to engine, opening cylinder cocks and admitting a small amount of steam. If steam does not blow through, secure valve by means of set screw provided for this purpose, which will be found on

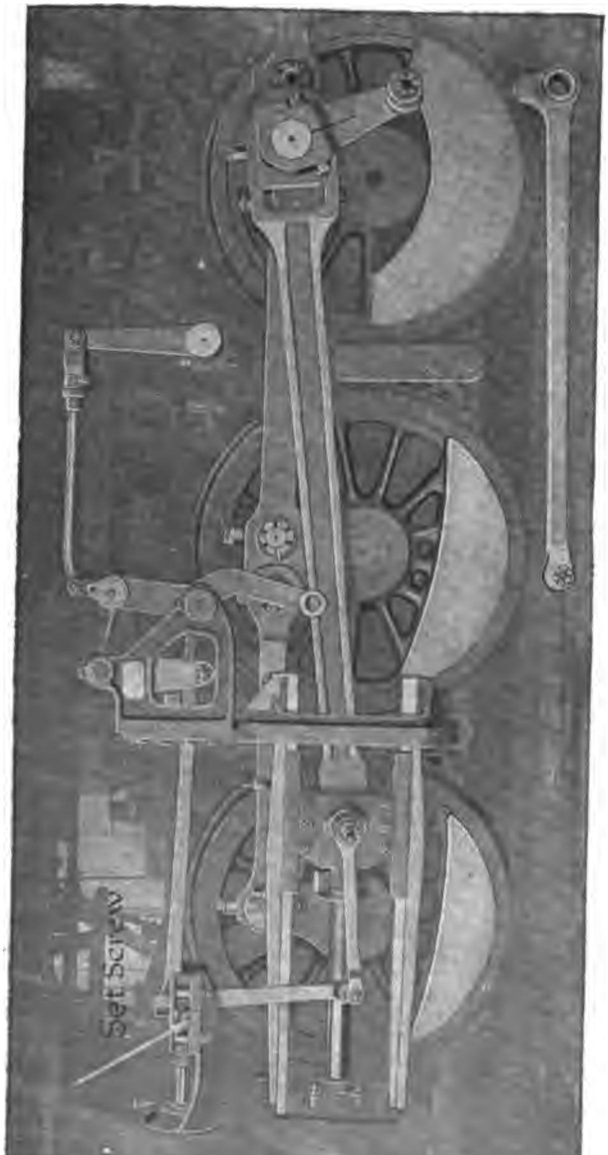


Fig. 148.

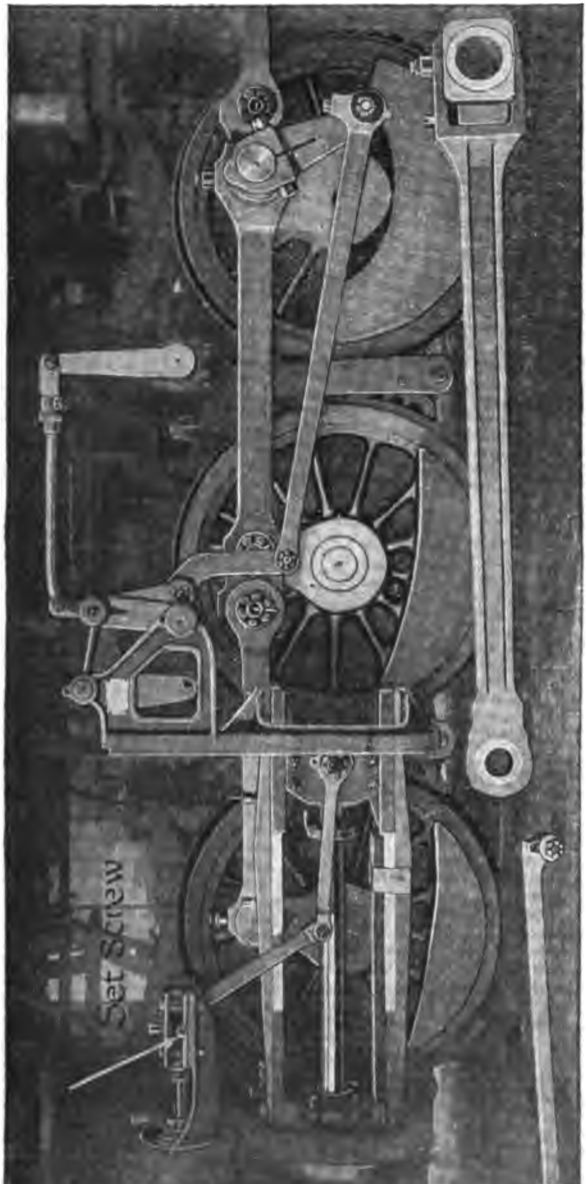


FIG. 149.

valve stem guide, valve stem crosshead or rocker boxes. If no set screw is provided, a valve stem clamp is usually included in the tool equipment. Clamp or block main crosshead at front or back end of guides as per railroad company's instructions. With the valve motion disconnected in this way, the reverse lever is free to operate the other side; and locomotive can be run in on one cylinder, by working engine at long cut-off.

If crosshead wrist pin should break and combination lever is driven from it.

Do the same as when main rod, main crosshead, guides or piston is damaged, except take down union link and wire bottom end of combination lever to some convenient place to keep it from swinging.

MAIN ROD UP, VALVE BLOCKED.

(See Fig. 149.) When main rod, main crosshead, guides and piston on the damaged side are in condition to run, the main rod may be left up, provided there are relief or vacuum valves in the cylinder heads, these may be removed so that cylinder may be lubricated and excessive compression avoided.

If vertical arm of bell crank should break and combination lever is suspended from it.

Take down combination lever, union link and valve rod.

If vertical arm of bell crank should break with combination lever suspended from valve stem crosshead or if cylinder head be broken or blown out.

(See Fig. 150.) Take down valve rod.

If union link, crosshead arm or combination lever fail.

Take down valve rod, fasten lower end of combination lever ahead to clear main crosshead on forward stroke. (See Fig. 151.) Of course such of the broken parts as would in any way interfere with the running of the locomotive would have to be reversed.

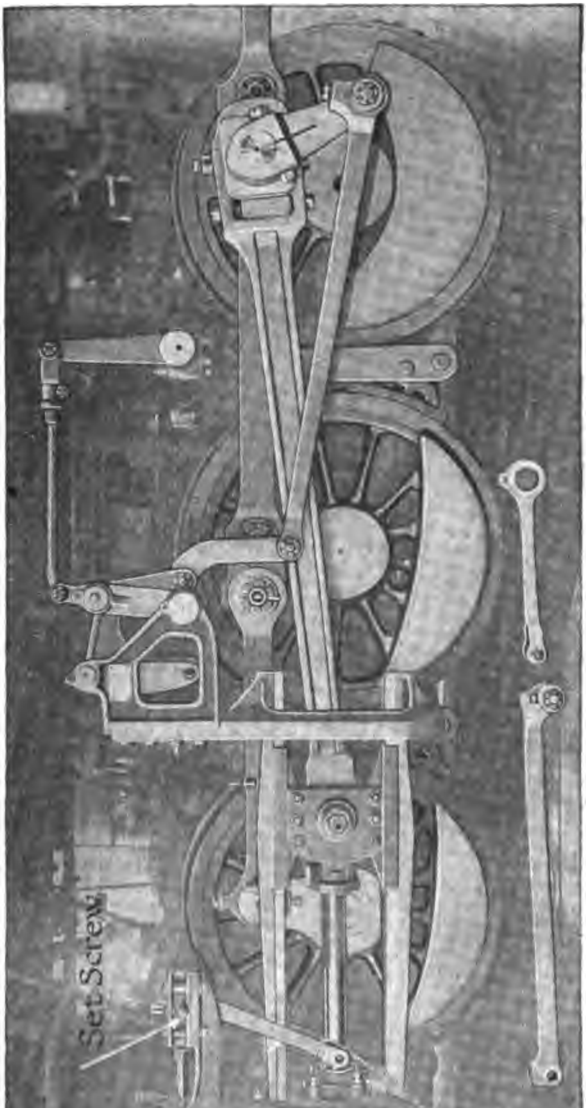


Fig. 150.

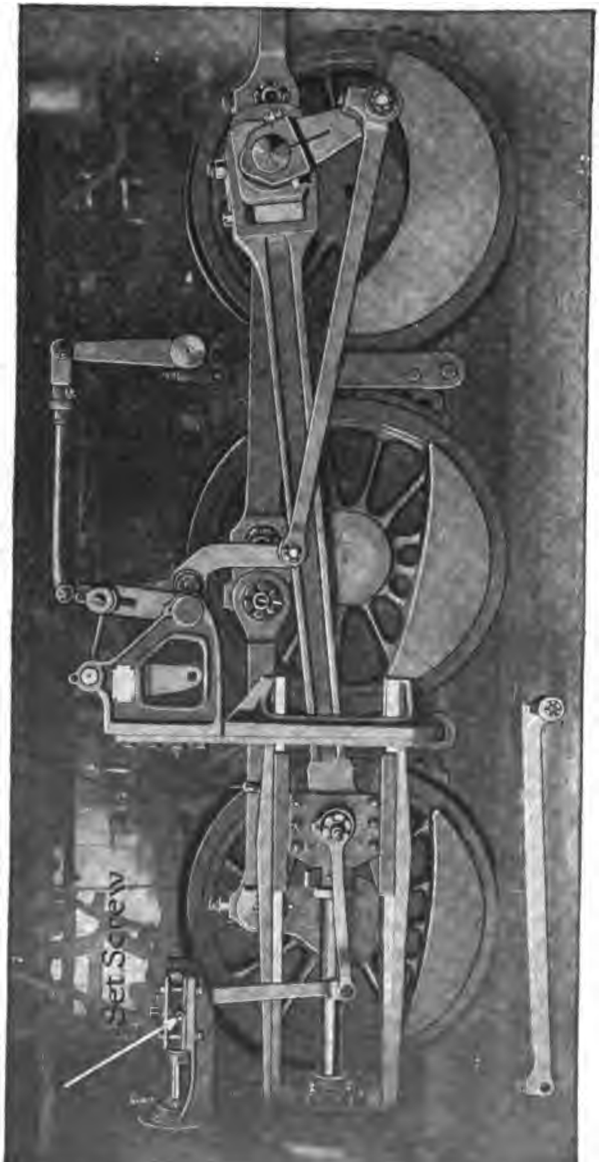


Fig. 151.

THE SOUTHERN VALVE GEAR.

This form of valve gear was invented by Mr. William Sherman Brown, of Knoxville, Tenn., who made application for a patent on the gear July 11, 1911, which was granted him July 23, 1912.

The inventor was employed as a locomotive engineer on the Southern Railway, and the first gear of this type was used on the Southern Railway Company's locomotive, No. 586, a 22x30-inch consolidated engine, employed in heavy freight service, which was turned out of the shop at Knoxville, Tenn., in February, 1913.

We understand that the new gear proved a success from the start, and that the first engine on which it was applied, with the same gear and set of valves, was still in service after traveling 200,000 miles, with practically no expense for work done to the valve parts, which were replaced identically as they were removed when the engine was shopped for general repairs.

The fact that the Southern Railway Company has equipped a large number of their locomotives with this gear, after giving it a thorough trial in all branches of the service, is strong evidence of its merit.

Construction.

While this gear is a radical departure from all previous outside valve gears, it is a gear that can be adapted to any class of locomotive, either inside or outside admission, and was designed with a view to reducing round-house repairs, and delays to power incident thereto.

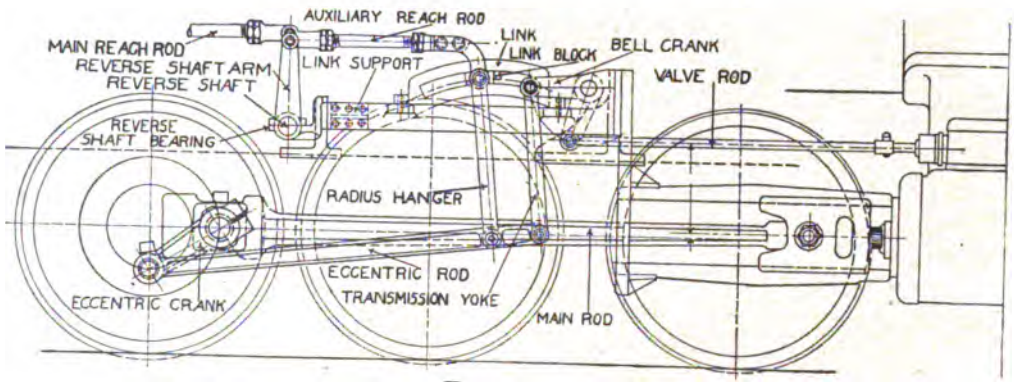

AUXILIARY REACH ROD
MAIN REACH ROD
REVERSE SHAFT ARM
REVERSE SHAFT
LINK SUPPORT
LINK
LINK BLOCK
BELL CRANK
VALVE ROD
REVERSE SHAFT BEARING
RADIUS HANGER
ECCENTRIC ROD
TRANSMISSION YOKE
MAIN ROD
ECCENTRIC CRANK

Fig. 152.

In designing the Southern valve gear, the inventor has eliminated many objectional features that have been found in other outside valve gears. This gear is very simple and compact, and contains but few wearing points. The crosshead connections have been dispensed with, and the gear has also been correspondingly reduced in weight.

In Fig. 152 we present the Southern valve gear, showing clearly the construction of the gear and the names of the parts. This, however, is an earlier type; the more modern form of the Southern gear is illustrated in Fig. 153, but the names of the parts are the same as those shown in Fig. 152.

If the valves are properly adjusted at the time the engine receives general repairs, the gear is so designed as to eliminate the necessity of any adjustment in the blacksmith shop, while the engine is in service. Simplicity of design has long been recognized as one of the greatest factors in the reduction of maintenance cost and the elimination of delays and repairs.

Transference from a rotary to a reciprocating motion is accomplished by direct movements and on straight lines, thereby doing away with strains and distortions. The links are horizontal and stationary, which does away entirely with the wear at this point, as the block only moves in the link when the reverse lever is moved to adjust the cut-off, or reverse the gear. The links being stationary also does away with what is known as the slip in the link block, found in some outside gears. There are but eight possible points of wear to the side, or a total of sixteen wearing points per locomotive, this being less than half contained in some gears.

Fig. 153 shows a view of the modern gear applied to a 26x32 Santa Fe type locomotive. It may be seen that this gear differs in some respects from that illustrated in Fig. 152.

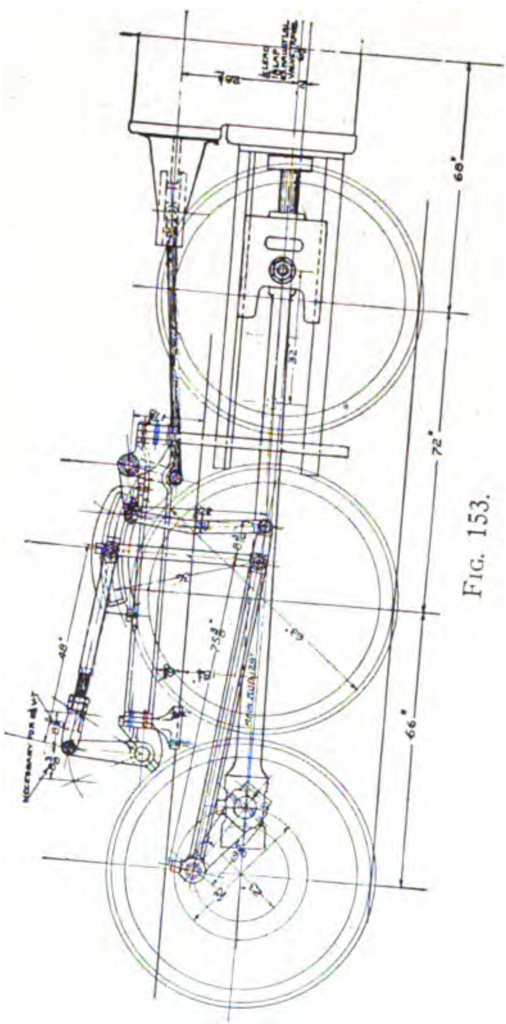


FIG. 153.

This gear will practically do away with engine failures due to breakage of valve gear parts. The different parts are so balanced as to reduce the wear on the pins and bushings to a minimum. A 22x30-inch consolidated engine, equipped with the Southern valve gear in heavy freight service, after making over 100,000 miles, did not show any appreciable wear on the pins and bushings; neither had there been one cent spent on repairs to the valve gear parts. All bearings, pins and bushings of the Southern valve gear are of such size as to insure long usage with very little wear. The forward end of the eccentric rod is supported by a bell crank hanger, which has at its top two bearings spaced widely apart, thus absolutely preventing any side slap on the eccentric rod.

Fig. 154 is a kinematic diagram, showing the varying positions of the different parts of the gear in forward motion. Starting on forward dead center, or on position Nos. 1—1 and C, moving to position Nos. 2—2 and C, bell crank then returning back over position Nos. 1—1 to position Nos. 3—3, thence to position Nos. 4—4, and then to position Nos. 1—1, we follow the diagram of the gear through the entire stroke.

The Southern gear is designed to eliminate all stress and strains on reverse lever and reach rod connections, and the reverse lever is easily handled while working a full head of steam. This feature appeals strongly to the engineers, enabling them to adjust their cut-off without fear of the lever getting away from them, and will induce them to work at as short cut-off as possible, resulting in a saving in fuel.

Fig. 155 is a kinematic diagram showing varying positions in back motion. Starting on forward dead center, or position Nos. 1—1 and C, moving to position Nos. 2—2 and C, then bell crank

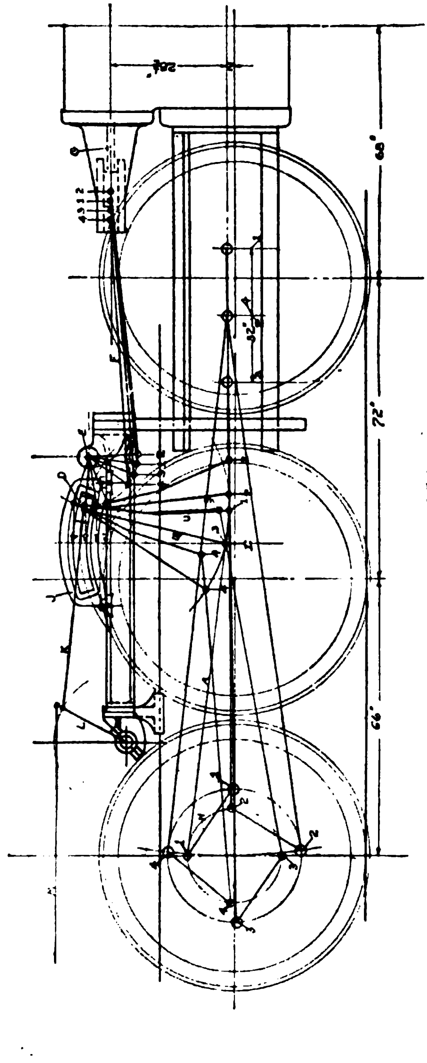


FIG. 154.

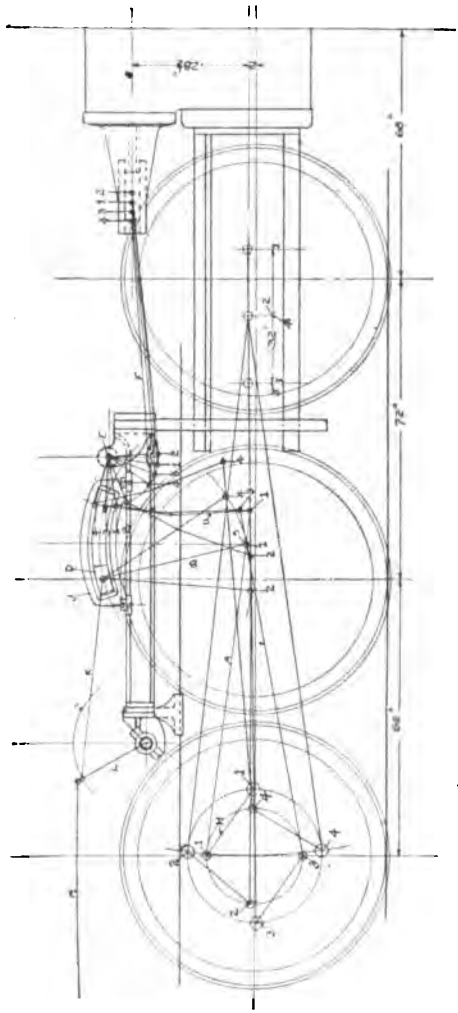


FIG. 155.

returning back over position Nos. 1—1 and C to position Nos. 3—3 and C, thence to position Nos. 4—4 and C, from which it moves to position Nos. 1—1 and C, the stroke is completed.

Directions for Setting and Adjusting Southern Valve Gear.

The method of setting and adjusting the Southern gear is clearly explained by the following rules, which have reference to the Southern valve gear layout illustrated in Fig. 156.

First. Set the link support so that dimensions conform to figures on erecting card. Then connect the gear as shown on the erecting card.

Second. Set the *links* so that dimension (M) conforms to erecting card.

Third. Set reverse lever in center of quadrant, adjust the main *reach* rod, so that *reverse shaft arms* will stand in vertical position.

Fourth. Adjust *auxiliary* reach rods so that *link block* will be in *center* of link when the reverse lever is in center of quadrant.

Fifth. Set *eccentric crank* for outside admission to *lead main crank pin,* and for inside admission to *follow main crank pin,* at a distance from center of main axle to center of *eccentric* crank pin to conform to erecting card, and proceed as shown by rule 6.

Sixth. Find dead centers in usual way. (Page 189.) With the engine on *front* dead center (F), tram from center of radius hanger pin (P) to any stationary point on cylinder casing or guide yoke, and scribe arc (a) as shown. Then place engine on *back* dead center (B) and again scribe arc (b) from center (P). If arcs (a and b) fall line and line, the eccentric crank setting is correct. If they do not, then knock eccentric crank to or from center until they do.

Seventh. Then revolve wheel or move engine one full turn, and tram in same center (P) and scribe arcs (c and d). If full travel (N) is within 1/16 inch of eccentric crank circle (O) shown on erecting card they are correct. If there is a difference of as much as ⅛ inch full travel of eccentric crank pin, lengthen the eccentric crank having the least *full* travel *one-fourth* of the *difference*, and reset cranks as per rules 5 and 6.

Eighth. Place engine on *front* and *back* dead centers (F) and (B) and move reverse lever entire sweep of quadrant. If valves move in same direction as link block, move link ahead, if in opposite direction move the link back, until still valve is found.

Ninth. To get the length of the valve rod, place engine on front *dead* center, allow ⅛ inch for lead, or amount shown on erecting card and adjust valve rod to lead. (In setting engines on jacks, have the wheels placed ½ inch under actual running height; for instance, if the blue print running height is 18 inches when the engine is on the rail, set the main wheels at 17½ inches running height on jacks.) Then try for full valve travel in forward position on each side of engine. If the valve travel is found to be unequal make necessary corrections by lengthening or shortening the auxiliary reach rods. For example, say 6 inches is the desired valve travel, lengthen the auxiliary reach rod on left side until 6-inch travel is obtained. Then proceed to run cut-off in full gear, and if necessary lengthen or shorten valve rod to equalize cut-off. Then run cut-off at about 9-inch piston travel and equalize same.

Tenth. Raising or lowering the links by means of liners has the same effect as shortening or lengthening valve rods, there-

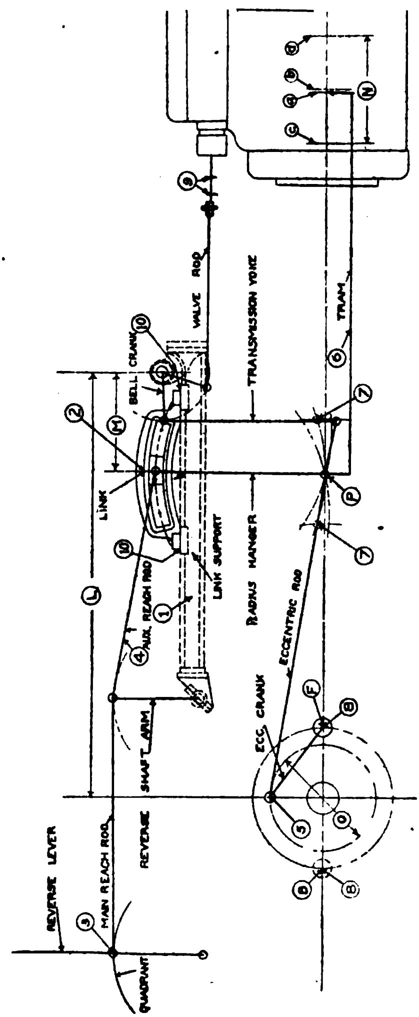

REVERSE LEVER
MAIN REACH ROD
QUADRANT
REVERSE SHAFT ARM
AUX. REACH ROD
LINK
LINK SUPPORT
RADIUS HANGER
ECCENTRIC ROD
ECC. CRANK
BELL CRANK
VALVE ROD
TRANSMISSION YOKE
TRAM

FIG. 156.

fore any derangement which may occur can be corrected without disconnecting any part of the gear.

Breakdowns.

First. If the eccentric crank or eccentric rods fail, disconnect the eccentric rod from the crank, radius hanger and transmission yoke, tie up the hanger and the yoke, clamp the valve in its central position, and proceed.

Second. If the radius hanger fails, disconnect the hanger from the rod and take down the eccentric rod. Then clamp the valve in its central position, and the locomotive is ready to proceed.

Third. In the event of the transmission yoke failing, disconnect it from the eccentric rod. Then proceed, first clamping the valve in its central position.

Fourth. When the horizontal arm of the bell crank fails, disconnect the yoke from the eccentric rod, tie up to clear, clamp the valve centrally, and proceed.

Fifth. If the vertical arm to the bell crank fails, or breaks, clamp the valve in its central position and proceed. It will sometimes be found necessary, however, to take down the broken vertical arm.

Sixth. In case of the failure of one auxiliary reach rod, or the reverse shaft arm, block both link blocks in the same position in the links, and in such a position as to give sufficient port opening to start the train and to control its speed by the throttle.

Seventh. If the main reach rod, or the middle arm to the reverse shaft, fails, or if both auxiliary reach rods fail, block the link blocks same as in case of one broken auxiliary reach rod, or reverse shaft arm. The power and speed of the locomotive may then be controlled by the throttle.

Laying Out the Southern Valve Gear.

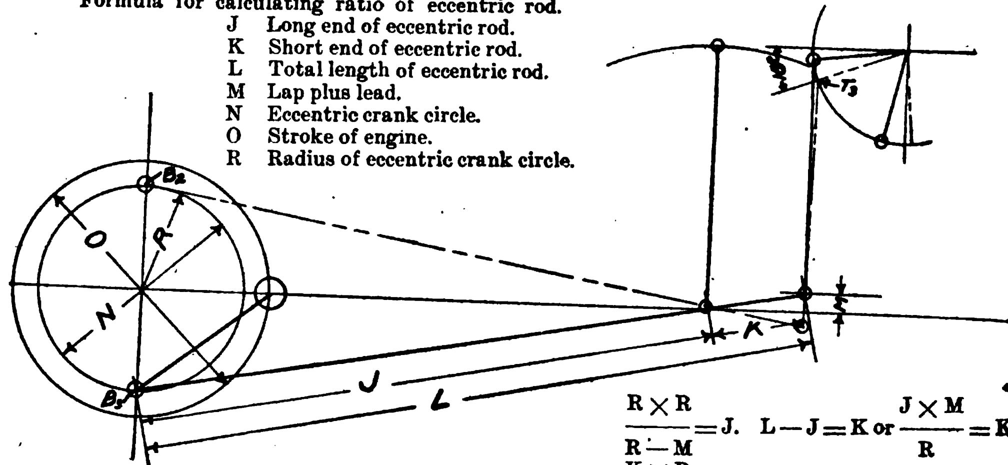
By J. B. Gwin, Chief Draftsman of the Southern Locomotive Valve Gear Co., Knoxville, Tenn.

Lay off the diagram on scale of half size, as follows: Lay off the center line through the center of the drivers, Fig. 158, and cut this line with a vertical through the center of the main axle at A. From center A, lay off both main pin circle and eccentric pin circle, d—d and e—e respectively. Now locate the center of cylinder; then, with tram set to the length of the main rod, locate the center of the crosshead pin on the center line of the cylinder at 1, from forward dead center. Now, with a straightedge through centers A and 1, draw the center line of the main rod. This is done in order to get the exact position of the main pin center above the center line of the drivers, which is very necessary for obtaining the length of the eccentric crank. Next, locate the bell crank center T to suit the available horizontal distance from the center of the main driver, and the vertical height above the center line of the drivers, if possible, so that the pin in the vertical bell crank arm will stand ½ inch below the center line of the valve stem, with the pin in its lowest position. Then, with radius equal to the length of the bell crank arms, from bell crank center T scribe arc T_1. Next drop a vertical from a point T_3, in Fig. 157. Then the length L measured on diagram, Fig. 157, will be the approximate length of the eccentric rod.

Now calculate the ratio of the long and short end of the eccentric rod, using diagram formula, Fig. 157. Set the tram to the equal length of the long end of the eccentric rod, and lay off arcs b—b, from centers B and B_1, intersecting at E, the

Formula for calculating ratio of eccentric rod.

J Long end of eccentric rod.
K Short end of eccentric rod.
L Total length of eccentric rod.
M Lap plus lead.
N Eccentric crank circle.
O Stroke of engine.
R Radius of eccentric crank circle.



$$\frac{R \times R}{R - M} = J. \quad L - J = K \text{ or } \frac{J \times M}{R} = K$$

$$\frac{K \times R}{J} = M.$$

FIG. 157.

center of the lower radius hanger pin when the engine is standing on either dead center.

Now, with the same tram, lay off arcs b_2—b_3, from centers C and D; then set the tram to the total length L of the eccentric rod, and lay off the arcs f and f_1, and the arcs C_1 and C_2, from the centers B and B_1, and C and D, to the extreme forward centers of the lower transmission yoke pin centers F, F_1, F_2, F_3, F_4 and F_5. Now set the tram to the length equaling the height of bell crank center T above center line of drivers, plus 1⅜ inches; this will be the suitable length of the radius hanger. For example, say that the height is 33⅛ inches, plus 1⅜ inches, which would equal 34½ inches, the length of the radius hanger. Now strike the arc G, from the center E; arc G will be the radial center, and the intersection of the arc G and the heavy line J_1 will be the vertical center of link and radius hanger. Now lay off the centers H and I. This distance will have to be found by trial, in order to obtain the proper valve travel. I is the link block center position, forward motion, and H is the same, for back motion. Take the tram with length equal to the radius hanger, and, from center I strike the arc J, also from center H strike arc K. Now lay off the eccentric rod center lines as follows: Lay off the center a, cutting centers B and E, and arc f at F. This for inside admission valves, and a_1 for outside admission valves, cutting centers B_1 and E, and arc f_1 at F_1. Lay off a_2, cutting center A and arcs b_2 and C_1, and J at L and F_2. Lay off a_3, cutting center A and arcs b_2 and o_1, and K at O and F_3. Lay off a_4, cutting center A and arcs b_3 and C_2, and K at N and F_4. Lay off center a_5, cutting center A and arcs b_3 and C_2, and J at M and F_5.

Now set tram to a trial length for the transmission yoke, say about 3¼ inches shorter than the radius hanger. From center

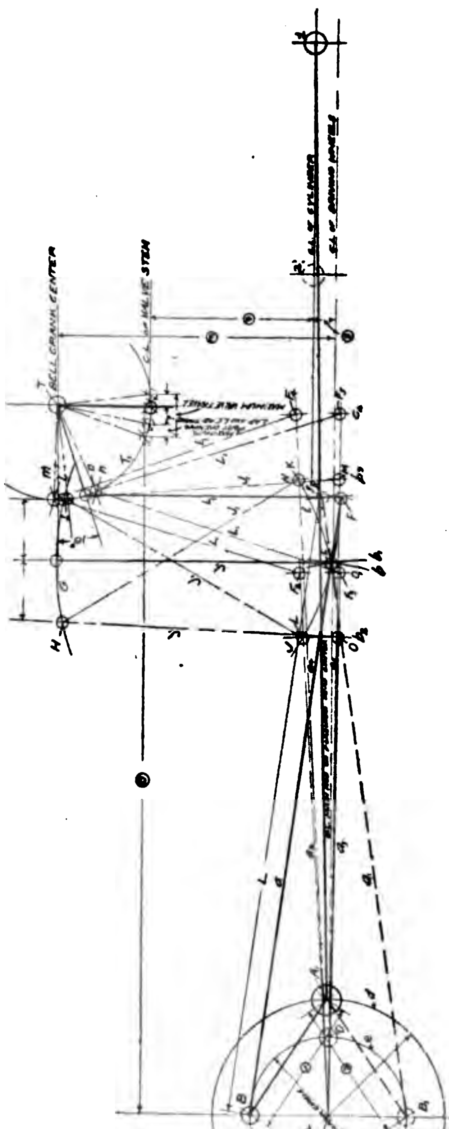


Fig. 158.

F_2, scribe the arc l, cutting the arc T_1. And from the center F_4, scribe the arc m cutting the arc T_1.

Now from the centers F_3 and F_5, scribe the arcs n and o, cutting the arc T_1.

Now, if the arcs l, m, n, and o intersect exactly on the arc T_1, this shows that the eccentric rod and transmission yoke are of proper length. But if the arcs l and m intersect on the arc T_1, and the arcs n and o intersect on the outside of arc T_1, the transmission yoke needs to be lengthened.

If the arcs n and o intersect on the arc T_1, and l and m on the outside, the transmission yoke needs to be shortened. But, if all four arcs l, m, n, and o intersect an equal amount outside of the arc T_1, the eccentric rod needs to be lengthened, while, if all four arcs intersect inside of the arc T_1, the eccentric rod must be shortened.

It is needless to say that any change in the length of the eccentric rod will call for a revised calculation for the lap and lead ratio, by the diagram formula, Fig. 157.

For inside admission, the main pin leads the eccentric crank pin, and with outside admission, the main pin follows the eccentric crank pin. The full black lines of the eccentric crank and rod are for inside admission, and the broken lines of the eccentric crank and rod are for outside admission.

THE YOUNG VALVE GEAR.

The Young Valve Gear patented by Mr. W. O. Young, of Chicago, Ill., September 2, 1913, and April 10, 1917, is a modification of the Walschaert gear, but is actuated wholly by piston reciprocation without intermediary crank connections to the driving wheels. In this respect it somewhat resembles the Lewis valve gear which had a limited use on locomotives between 1892 and 1895.

The Young gear is specifically designed for large cylinder locomotives. It is therefore arranged to produce much greater valve travel than has heretofore been found practicable with other gears without involving excessive angles to individual members. With this gear valve travels of 8½" to 9½" are obtained without distortion or serious angular thrusts.

The extremely long travel is a means to an end. It is for the purpose of permitting decidedly wider steam lap and still retaining late maximum cut-offs necessary for reliable starting power.

The advantage of unusually wide lap is greatly increased width of openings, particularly in early cut-offs, and consequently unobstructed steam flow for both inlet and outlet. The time of valve events, that is, the period of piston stroke at which pre-admission, release, and closure occur for given cut-offs, are practically the same as are caused by the Walschaert gear, but the increased velocity produced by the Young gear results in more decisive events and ample port areas.

The equipment is simple and durable in construction, positive in its action, and well standardized.

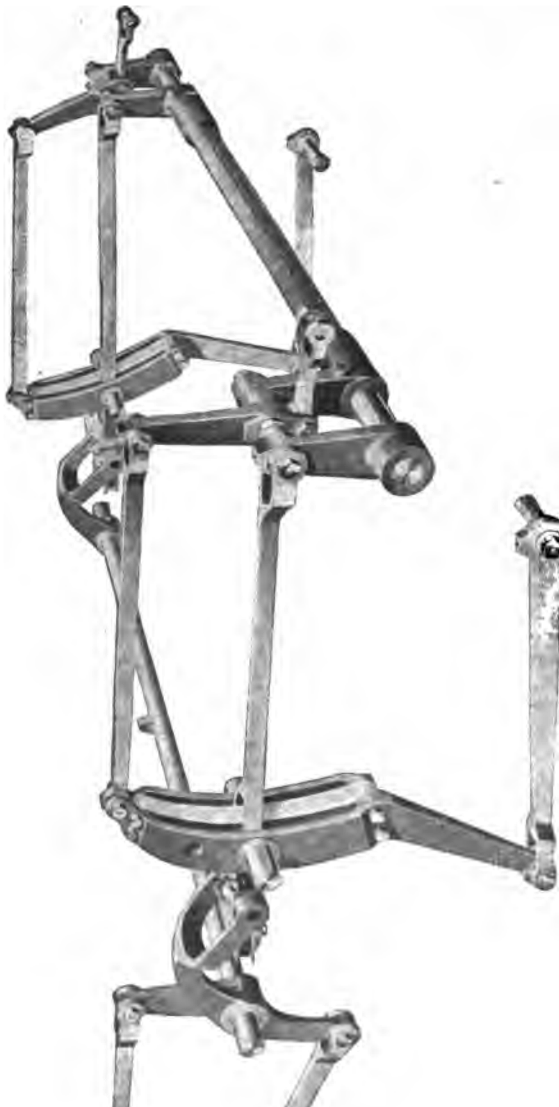


FIG. 159.

Design and Construction.

The Young design is an entirely new form of radial type of outside valve gear. Its simplicity of design and construction is due to the embodiment of the fewest possible working parts, as may be seen in Fig. 159.

The motion of this valve gear is derived entirely from the piston and crosshead connections, thereby eliminating distortions resulting from the slip of the driving boxes, wear of journal brasses, and settling of the equipment upon its springs. The 90-degree movement of valve travel is obtained by a pair of concentrically mounted rockers. These rockers transmit the variable component of the motion from the crosshead and link on one side to the lap and lead lever fulcrum on the other side.

Each piston, in operation, causes valve movement equal to the lap and lead on its own side and travel on the opposite side. The valve is moved a distance equal to its lap and lead each way from its central position by the movement of the piston through an entire stroke, when the radius bars are both central in the link. In this valve gear the source of motion is the piston alone, and the right engine is 90 degrees ahead of the left; while the left engine is 270 degrees ahead of the right, therefore the valve travel producing members must be direct on one side and indirect on the other. The various cut-offs are obtained with the radius bar in intervening positions between its central and extreme locations.

In Fig. 160 are the gear positions when arranged for 50 per cent cut-off, forward motion. They show the angles of the various members at the beginning of a stroke, at cut-off, and at release. The right gear is represented by heavy lines, and the left gear by light lines.

Valve travel in addition to lap and lead is produced through the oscillation of the link which causes the radius bar to oscillate the rock shafts. This motion is effective on the valves on the side of the locomotive opposite to the link imparting motion. As the front end of the radius bar acts on the lap and lead lever

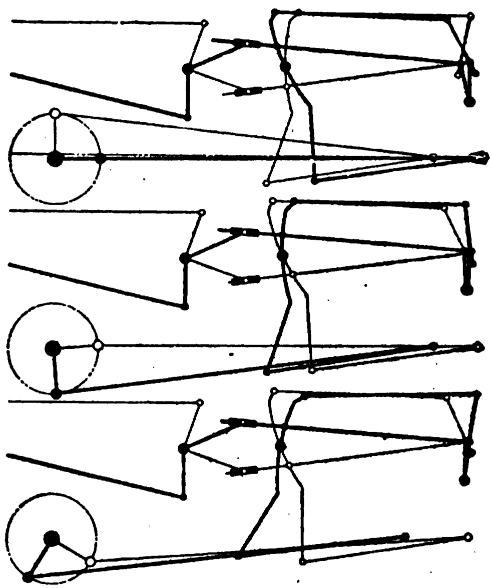


FIG. 160.

between its two extreme connections, unusually long travel is imparted to the valve without excessive link angularities. As the movement of the links is constant under all operating conditions, this is desirable.

Valve Setting.

In Fig. 161 is shown a diagram in heavy full lines, of the position of the right gear when the wheel is at its forward dead center and the radius bars are in mid-gear. The position at the back wheel center is shown in broken lines.

With the radius bars (VII) central, the rockers (IV) are in mid-position, and are unaffected by swinging the link (II) from one extreme to the other. As the combination links (V) are fulcrumed on the rockers and are attached to the links (II) by combination links (VI), the constant component of motion (lap and lead) is accomplished by each piston stroke.

Light full lines indicate the relative position of the left gear when the right is at its forward center.

When the right is at its back center, the left gear reassumes its former position, but the left main rod is then on its bottom quarter (shown by dotted lines) instead of on its top quarter. The relative position of the right gear to the left when the latter is at its wheel centers is the same as is the left to the right at its centers, but the right main rod is on the bottom quarter for the left forward center and the top quarter for the left back center.

The extreme positions of the radius bars (VII) are shown for full gear forward, the right bar by broken lines and the left by dotted lines. For full gear backward, these positions are reversed. The right bar is then below the center of oscillation of the link, and the left above. If measured on the link, when in mid-position, the total radius bar lift is approximately 16 inches (8 inches above and 8 inches below the link center), except for switching locomotives, on which the lift is slightly greater. With the radius bars in extreme or intermediate positions, the link swing causes oscillation of the rockers and produces the variable component of motion. In the rare cases where the left main pin leads, the radius bar lifts are reversed.

In locomotives using valve gears that derive a portion of their movement from a connection to the driving wheels, as

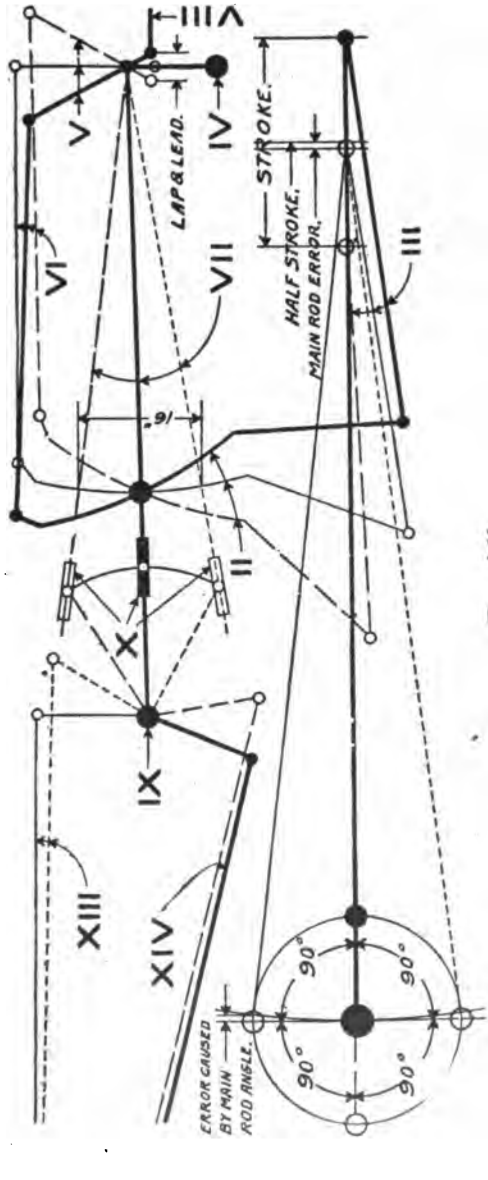

VIII
V
IV
LAP & LEAD.
STROKE.
HALF STROKE.
MAIN ROD ERROR.
III
VI
VII
.91
II
X
IX
XIII
XIV
ERROR CAUSED BY MAIN ROD ANGLE.
90°
90°
90°
90°

FIG. 161.

for instance, the Walschaert, it is difficult to make proper adjustments without revolving the wheels and taking centers.

With the Young gear, which has no connection to the driving wheels, the method of determining correct wheel centers may be simplified.

As one main pin is 90 degrees in advance of the other, reference to the diagram shows that with one pin at either dead center, the opposite pin is midway between; and the four pin positions divide the crank circle into four equal arcs. While the back end of the main rod, which is attached to the mid-positioned

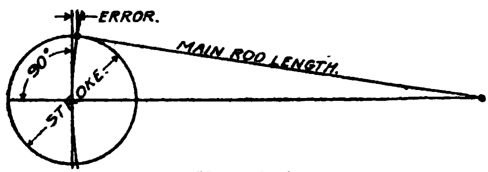


FIG. 162.

pin, has completed one-half of its stroke (measured by the pin circle), its front end has completed more than half of its stroke horizontally, due to its angular position. Therefore, at positions of dead wheel centers on one side, with its crosshead at either end of the stroke, the opposite crosshead is somewhat back of mid-position.

This error, caused by the main rod angle, varies with the length of stroke and length of main rod, and is usually between ⅝ inch and 1 inch at the crosshead.

All erecting cards of the Young gear show the amount of this error, and when known, correct wheel centers on either side are obtained by placing the opposite crosshead the specified amount back of mid-stroke.

Fig. 162 shows the method of determining the main rod error.

To Set the Valves Without Revolving the Wheels.

The locomotive need not be wheeled. The main rods should not be connected to the crossheads.

With the crossheads connected to pistons, mark on the guides the piston striking points. Show a mark midway between striking points and then place both crossheads back of mid-position the amount of the main rod error as shown on the erecting card. The above are the true positions the crosshead will occupy when the wheels on the opposite side of the locomotive are on either front or back centers.

With the union links (III) disconnected, place the reverse lever in position to center the radius bars. They are central when both links (II) may be swung by hand, back and forth, without movement to the rockers (IV). If one is central and the other is not, lengthen or shorten either reach rod (XIII or XIV), whichever is most convenient, until both links may be swung without affecting the rockers. This is the first adjustment to make. Its object is to equalize the radius bar lift for both gears, in order to insure equal cut-offs throughout their entire range on both sides of the locomotive.

With both crossheads in the positions previously placed, connect the union links (III). There should then be no movement to the rockers (IV) while throwing the reverse lever from full gear forward to full gear backward. If there is, take one side at a time, and lengthen or shorten the union link (III) to its required length. This is the second adjustment, and its object is to obtain constant lead for all cut-offs.

With radius bars central, and without changing the position of one crosshead, move the opposite one to one end of its travel, measure the lead, then move to the opposite end and measure the lead. If not alike, equalize by changing the thickness of the valve stem washer, or by lengthening or shortening the valve rod (VIII). Place that crosshead in its former position, and repeat the operation on the opposite side. This is the final adjustment for the purpose of securing equal lead for both ends of the cylinders.

The main rods may then be connected, and equal piston clearance at both ends of the cylinders arranged for. Clearance should be checked when the wheels are first revolved. Stop pins should be applied to limit full gear positions to the specified travel when the wheels are first revolved, or when the locomotive is first moved on its wheels.

To Set the Valves by Revolving the Wheels.

If desired, or if the main rod error is unknown, wheel centers may be taken in the usual manner.

Equalize the radius bar lift by the method described in the foregoing method of setting the valves.

With main rods and union links connected up, roll the wheels to a dead center, throw the reverse lever from one extreme to the other, and measure any movement to the rocker arm to which the radius bar on the opposite side of the locomotive is connected. Lengthen or shorten the union link on that side, if necessary, to insure a dead rocker for all reverse lever positions.

With the radius bars central, revolve the wheels and measure the lead at the four ports. Change the valve stem washers, if necessary, to secure equal lead for all four wheel centers.

Cut-offs may then be taken.

Refinements in Valve Setting.

While the valve setting instructions state that the reach rods should be of proper length to simultaneously center both radius bars, and while this is theoretically correct, it is in practice correct only so long as the gear is free from lost motion.

As one radius bar is operated by the upper portion of the link and the other by the lower portion, the effect of wear in the radius bar lifters and reverse shaft bearings causes the back ends of the radius bars to occupy slightly lower positions than initially arranged for, one at less and the other at greater distance from the link centers. Progressive wear tends to increase valve travel and delay cut-offs on the right side of the locomotive and to decrease the travel and hasten cut-offs on the left side, when operating in forward motion—in backward motion vice versa. This characteristic develops slowly, but it should be considered and allowance made when adjusting the reach rods. It is better, therefore, with the left radius bar exactly central, to arrange for the right radius bar to be sufficiently above its central position to cause between ⅛ and 3/16 inch reciprocation while swinging the link from one extreme to the other.

While the gear is new, the cut-offs in forward motion will then show approximately 6¾ inches to 7 inches of steam in the right cylinder when the left cylinder shows 7½ inches. These differences will gradually equalize and, after the gear is badly worn, they will become reversed. In backward motion the difference in cut-offs, one cylinder to the other, is opposite to that in forward motion.

In all valve gears considerable irregularity in lead, one port with another, is permissible, as its effect on distribution is negligible, provided approximately equal widths of port opening and points of cut-off and release are obtained. These are the three essential features. Regularity in closure, pre-admission and lead are of minor importance and they are relatively important in their above mentioned sequence.

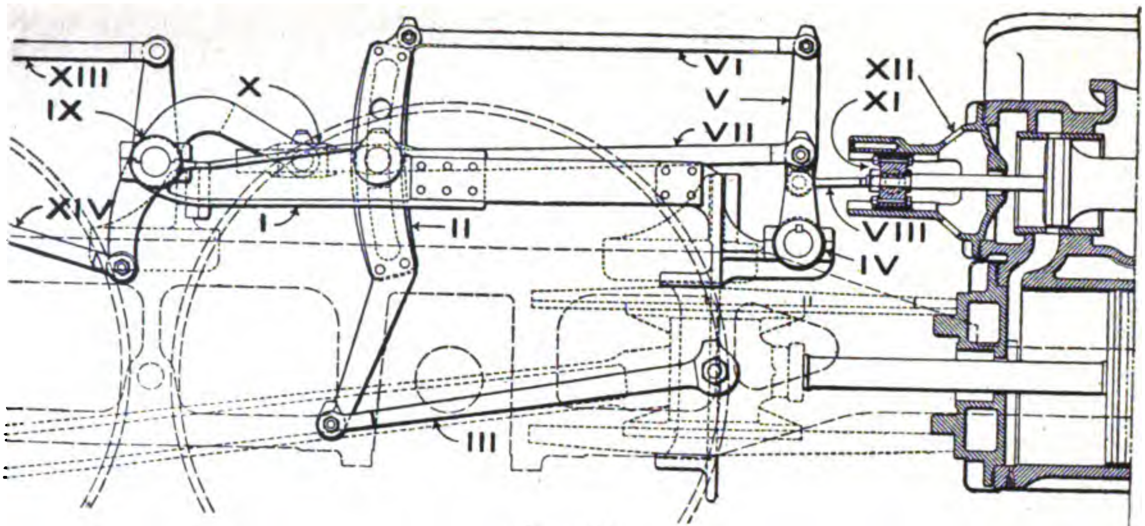


Fig. 163.

With lately designed Young gears, equalization of lead causes close equalization of all events. If found to be slightly irregular, the following is considered good setting for a new gear:

Forward Motion.

	Right Side.		Left Side.	
	Front.	Back.	Front.	Back.
Port opening	3/8	13/32	13/32	7/16
Cut-offs	7	6¾	7½	7¼

Backward Motion.

	Front.	Back.	Front.	Back.
Port opening	13/32	7/16	3/8	13/32
Cut-off	7½	7¼	7	6¾

This is assuming 5/16-inch lead and 28-inch stroke. If the lead is less than 5/16 inch and the stroke more than 28 inches the port opening will be somewhat less for the above cut-offs.

Breakdowns.

The structural simplicity of the Young gear insures unusual immunity from service failures; but the use of defective material, lack of proper inspection, or negligent care are not always preventable; therefore, failures may sometimes occur.

Provided both main rods and crossheads are unaffected, the result of failures from the Young gear are no more serious than from other gears. Either valve may be detached by disconnecting the valve link (VIII, Fig. 163) and the other valve continued in operation. The locomotive can then proceed, with one valve centrally secured and the other working steam.

This valve gear is manufactured by The Pyle-National Company of Chicago, Ill.

ACCIDENTS TO LOCOMOTIVES.

It requires less time to properly inspect and prevent an accident to the locomotive than it does to make repairs after the damage has occurred, and the prudent engineer, who inspects his engine regularly and replaces the loose bolts, tightens nuts and keys, looks for and carefully examines any cracks, flaws or other defects upon his engine, is seldom troubled with those annoying and sometimes dangerous break-downs while on the road. In fact, an observant engineer who becomes familiar with the regular and equal exhausts of his engine can often detect a lame exhaust in time to stop, and prevent a serious accident to his engine, as any defect in the valve motion, such as a loose eccentric, strap bolt, loose blade bolt, a loose valve stem key, broken valve yoke or slipped eccentric, will cause an imperfect exhaust. But breakdowns sometimes occur after a man has done everything that human foresight can suggest to aid in preventing an accident. These pages were written to assist those who are so unfortunate as to meet with a mishap on the road, who may through excitement, or lack of knowledge, forget something of great importance which would prevent an aggravation of the original injury.

In case of an accident, we presume that the engineer will first fully protect his train and comply with his book of rules regarding signals, flagman, etc., and will not neglect his boiler while working on the disabled engine. If the engine is in the ditch, or if the crown sheet is not fully protected, it is always advisable to kill the fire without delay.

All properly equipped engines should be supplied with a sufficient number of hand tools, wrenches, clamps, blocks, etc., for use in case of an accident, and a careful engineer will see that his engine is supplied with the same before starting out on the road.

It would be impossible to lay down rules to cover every case that might arise. It is possible, however, to consider some of the more usual, or more probable accidents; and to determine the best and quickest course to follow in such cases.

The accidents to be considered may be divided into two general classes; those in which it is not necessary, and those in which it is necessary, to block the valve to cover the ports.

The first class of accidents includes only those cases in which it is necessary to take down the main rod. The valve must necessarily be blocked if the main rod is disconnected.

In the second class of accidents, the damage to the machinery may or may not make it necessary to take down the main rod.

In a carefully prepared report presented to the Twenty-sixth Annual Convention of *The Travelling Engineers' Association* at Chicago, Ill., on September 13, 1918, locomotive failures were classified as follows:

Definition of What Constitutes an Engine Failure.

1. All delays waiting for an engine at an initial terminal, except in cases where an engine must be turned and does not arrive in time to be dispatched and cared for before leaving time.

2. All delays on account of engines breaking down, running hot, not steaming well, or having to reduce tonnage on account of defective engine, making a delay at a terminal, a meeting point, a junction connection, or delaying other traffic.

And the result of the committee's investigation, extending over a period of two years, was submitted in the following table:

Failures in Two Years Per 1,000 Miles of Road.

Air Brake and Piping	15
Ash Pan and Rigging	5
Axle, Broken Driving	2
Brake Rigging	3
Boiler Studs	1
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Disconnecting.

The question of whether or not it is necessary to disconnect one or both sides of a locomotive, presents itself in connection with so many breakdowns that the general principles should be thoroughly understood by every man having the control of a locomotive. When an engine failure occurs on the road, you alone must decide the question, without delay, and a great deal more than your personal safety may depend upon the wisdom of your decision.

For this reason, and on account of the fact that some of our directions might not otherwise be clear, we respectfully suggest that the reader should thoroughly understand the general rules governing the subject, which we shall hereafter attempt to make clear.

In order to avoid unnecessary repetition, and to assist in condensing the subject, we shall first explain what should be done in order to disconnect one side of a locomotive; and next what should be done, and what precautions should be taken, when it is necessary to disconnect both sides. Afterwards, throughout the chapter, we shall simply say "disconnect one side," or "disconnect both sides," as may be necessary.

Disconnect a compound locomotive just as you would a simple engine, keeping the separate exhaust valve open, and then work the locomotive simple instead of compound.

The following is considered a good rule to follow in regard to taking down the main rod. When it is possible to lubricate the cylinder and relieve compression, otherwise than by removing the cylinder cocks, the main rod may be left up if in a condition to run. For example: if there are relief or vacuum valves in the heads, these may be removed. This will prevent compression, and will also permit of lubricating the cylinder.

Disconnecting One Side.

When we speak of disconnecting one side, it will be implied that the engine is to continue its trip. Remove the main rod on the disabled side, and replace the liners and brasses in the straps, just as you found them. If a collar is not used between the main rod and the side rod, the latter can be held in position by pieces of wood, fitted and lashed lengthwise around the main pin with rope or wire. Secure the crosshead near the back end of the guides with a crosshead clamp, if you have one; if not, with hard wooden blocks, securing the blocks with rope so that they cannot work out. Remove the valve rod and secure the valve stem with a valve stem clamp, if you have one; if not, set the valve central upon its seat and cramp the valve stem by tightening the gland on one side. The valve can easily be set to cover the ports by opening the cylinder cocks and giving the engine a little steam; and then by adjusting the valve stem until steam is entirely shut off from both cylinder cocks. Do not remove the eccentric straps, or side rods, unless it is necessary. Whenever the eccentric straps are removed on one side, the top of the link should be tied to the short arm of the tumbling shaft, to keep it from tipping over, which would prevent reversing the engine. If it is necessary to take down one side rod, remove the rod directly opposite it. We advise this because the remaining side rod can exert no beneficial influence on the wheels, formerly controlled by the disconnected rod, when the engine cranks are passing dead center, but it imposes an enormous strain on the rod when starting or stopping, and is liable to break the rod, especially if the drivers should slip. If the opposite rod cannot be taken down remove all rods. The counter-balancing of the wheels, however, is affected by the removal of any of the rods. Do not remove the eccentric blades and leave the straps on the eccentrics, unless they will whirl and clear everything in all positions; otherwise they might punch holes in the fire box.

If the side rods have been removed from a ten-wheel engine, or pony engine, see that the forward crank pin will clear the crosshead in all positions; if not, *take no chances,* but disconnect both sides, blocking both crossheads clear forward, or wherever they will clear the crank pins, and have the engine towed in.

The method of blocking a valve is not controlled by the type of valve gear with which it is connected.

Disconnecting Both Sides.

This implies that the engine is dead and must be towed in. Therefore, place the reverse lever about half way down the quadrant, in the direction in which the engine is to be moved, remove both main rods and both valve rods, but do not block either valve if the crank pins clear the crosshead, nor remove the side rods, nor eccentric straps, unless it is necessary, and, when it is considered necessary, be sure to take the precautions previously explained.

In freezing weather, if the fire is down, all water should be drained out of the injectors, pumps, and feed and branch pipes. If there are no frost plugs, slack the joints and allow the water to escape. If there is danger of the water freezing in the boiler run it out of both the boiler and the tank. See that all oil cups are well filled before starting. Most roads are strict regarding the speed of dead, or disconnected, engines which are not counterbalanced perfectly, in order to avoid serious injury to the track, and require them to be moved at a low rate of speed.

Engine Derailed.

The first thing to do, after protecting yourself from approaching trains, is to ascertain whether or not the water is high enough at either end to leave the crown sheet, or the front end of the flues, unprotected by water, for a hot fire may melt or burn these parts if they are not well covered with water.

If the engine is in such a position that it leaves the crown sheet or flues unprotected by water, draw the fire; or, if you are not able to draw the fire, smother it with earth, sod, snow, or green coal.

Most engines, if they are not off badly, or too far away from the track, will help themselves on, without the aid of another engine, by using blocks under the wheels. A re-railing frog will be handy here, but if one is not at hand, jacks will, for light engines, aid materially by setting them to push the engine. An engine can usually be put back on the track more easily by moving it in the direction opposite to that in which it ran off; that is, if it left the rails when running ahead, it should be moved backward to replace it on the track. A careful inspection should be made for broken or sprung parts as soon as the engine is back on the rails.

Flue Burst.

Reduce the steam pressure and plug the flue with a wooden or iron plug. If you can get at the flue from the fire box door (using a long iron rod), you can plug the flue without drawing the fire, but a sharpened pole, or stick of wood, long enough to reach the flue, may be driven into it, and answer the purpose. The wood will burn off up to the water line, but it will not burn inside of the flue sheet. It is advisable to plug the front end of the flue first, and then the back end. If a lower flue bursts, or leaks badly, bank that part of the fire affected, and, if the pressure can be maintained, proceed. If the flue cannot be plugged, or its effect limited, cover the fire dead, open the blower enough to carry off the smoke in the fire box, lay a board on top of the coal, and go into the fire box and calk or plug the flue. This, however, cannot be done if there is a brick arch in the fire box.

The practice of plugging the flues is held to be inadvisable, and is discouraged by some roads.

Valve Yoke Broken.

A valve yoke usually breaks off at the neck of the valve stem, and can readily be discovered in the exhaust by a tremendous blow, or by the loss of two exhausts, and by a rocking motion of the engine. If the valve is pushed far enough ahead it will blow, but if it does not it is often mistaken for a slipped eccentric, and therefore you should examine the eccentrics first. A broken valve yoke will leave the valve at the front end of the steam chest, with the back admission port open. This may be discovered by placing the crank pin on the top or bottom quarter, opening the cylinder cocks, and reversing the engine; if the steam continues to come out of the back cylinder cock it indicates a broken valve yoke on that side. Again, stop one engine on dead center, open cylinder cocks, admit a little steam and move the reverse lever from one corner of the quadrant to the other. If the escape of steam is changed from back to forward cylinder cock, or vice versa, on the side where your engine is on the quarter, it must be the valve on the opposite side that is disabled. The oldest, and most accurate, method is to raise the steam chest cover and block the valve centrally, replace the cover, remove cylinder cocks or indicator plugs, remove valve rod and main rod, block the crosshead at the back end, and provide for lubrication; but this remedy requires considerable time and labor, and as *time* is a very important consideration on the road, and no serious objections are raised to other methods, providing the crosshead is securely blocked, we will state the other remedies.

Disconnect the valve rod and move the valve all the way ahead, remove the stem, if it is liable to blow out, and use a gasket back of the gland, or hold the valve stem intact with a valve stem clamp, and take out the relief valve. Block the crosshead at the front end, provide for lubrication, and remove the cylinder cocks or indicator plugs, to provide for a free circulation of air in the cylinders. The pressure will hold the valve forward, but if it should move it can do no harm, providing the crosshead is securely blocked. Another method is to remove the relief valve, push the valve clear back in the steam chest, replace the relief valve, open the cylinder cocks, admit a little steam, and push the valve forward slowly until steam appears at the rear cylinder cock. Then clamp the valve, remove the relief valve, and fit a block into the relief valve long enough to hold the valve back when the relief valve is replaced. Then block the crosshead at the back end. Still another way is to push the valve stem forward, and clamp it by cocking the gland, then block the crosshead at the front end. Leave the main rod up, and lubricate the piston and cylinder with the steam passing out of the rear cylinder cock opening.

Valve Yoke Cracked.

A cracked valve yoke is indicated by a slight exhaust when the engine is passing the back center. Stop the engine on the top quarter on the suspected side, place the reverse lever in the forward notch, set the brake, open cylinder cocks, then open the throttle to its limit. Next move the reverse lever backward slowly, and note its position in relation to the center of the quadrant when the steam stops flowing from the rear cylinder cock. If the reverse lever is back of the quadrant center, it indicates that the yoke is simply cracked or defective. The act of pulling the valve, when reversing, may cause the crack to open, and, in such a case, steam will show at the forward cylinder cock before the rear port is closed. When the yoke pushes the valve forward, the valve will sound all right, but when it pulls the valve back the engine will be lame. With careful handling, you may finish your trip without breaking the other side of the yoke. Work the engine in full stroke with a light throttle, and you will produce power sufficient to handle the train.

Steam Chest, or Cover, Broken or Cracked.

This is a very troublesome mishap which is often caused by reversing an engine while running, with closed throttle and cylinder cocks, at a fast rate of speed. By reversing the engine, under such conditions, the cylinders are converted into air compressors, and the air compressed is confined until the steam chest or steam pipes burst. If you believe that the chest is only cracked, disconnect the oil pipe, remove the casing, and, if it is only cracked on one side, by wedging between the chest and the studs opposite the crack, you may be able to close the crack enough to get in. Connect up the oil pipe and proceed. A brake shoe key may be used to advantage for this purpose. But if it is a serious crack, or if a piece be broken out of it, or a cover, and you have no way of clamping the broken parts together, disconnect that side, and then the quickest remedy is to use a blind gasket at one end of the steam pipe. But this is considered impracticable, owing to the corrosion of the bolts and nuts, the netting, and a very hot front end. Therefore take up the steam chest cover, remove the broken parts, disconnect the valve rod, and plug up the supply ports—not the steam ports—with wood, and clamp the plugs with steam chest studs. If steam enters from the side of the chest use a gasket there. If the chest is completely knocked off, clamp the wooden plugs with old bolts and fish plates, or whatever you are able to find, or sling a chain around the steam chest and cylinder, allowing it to pass outside of the guide bars at the rear end of the cylinder, and in close to the engine frame at the forward end of the cylinder, using a block or jack to make the chain tight.

If you are unable to make the ports tight, you can leave the train, run under light pressure to the nearest telegraph office, and report conditions of the engine to headquarters. Many steam chests may be saved, when you have no relief valves, by opening the throttle slightly as soon as you reverse the engine, for a reversed valve gear is virtually an air pump, and if the air cannot enter the boiler it must escape somewhere when compressed.

Piston Valve Broken.

Remove the steam chest head, and, if the valve is able to keep steam from the exhaust, clamp or block the valve centrally from the head, then disconnect the valve rod, etc., the same as for a slide valve. Open the cylinder cocks, provide for lubrication, and proceed. If the steam edge of the valve is broken enough to allow steam to reach the cylinder, remove the cylinder cock at the end where the piston head rests, and block the cylinder cock open at the end of the cylinder where the steam is present, to allow condensation to pass out and protect the cylinder in freezing weather. If the condition of the valve is such that it will not keep steam from the exhaust, prepare to be towed in. If the piston valve on a cross-compound is broken, remove the valve and reduce the pressure to about 100 pounds, or about one-half the usual amount carried, and proceed.

Piston Valve Body or Its Heads Broken.

In such a case insert a blind gasket in the joint of the live steam port. Remove the valve and all broken parts. Secure a cylindrical stick of wood, large enough to fill the valve chamber, and enclose same in a piece of sheet iron in the valve chamber; then disconnect the valve rod, provide for lubrication and circulation of air in the cylinder, and proceed.

Crosshead Broken.

If it be a slight break, such as a gib or a plate, it is sometimes possible to clamp the crosshead so that you can proceed, but care must be taken to see that the clamp does not strike the guide block at the extreme travel of the crosshead. However, if it be a bad break, take down the main rod, disconnect the valve stem, and clamp the valve in the center of its seat. If the piston is not broken, push it against the forward cylinder head, and then block the crosshead at the back end of the guides, if it is possible to do so. If the crosshead is broken so that it cannot be blocked, the safest method is to remove the piston from the cylinder. If the crosshead cannot be removed, set the valve so as to admit steam to the back end of the cylinder only, and then clamp the valve stem securely in that position.

Crosshead Pin Lost or Broken.

Take down the main rod, block the crosshead, and clamp the valve so that it will cover both ports.

Blocking the Crosshead.

It is considered better practice to block the crosshead securely at the back end of the guides where the construction of the engine is most suitable, but it may be blocked at the front end when necessary. Blocking the crosshead at the back end is preferable, however, because in case the crosshead blocking should give way and allow the crosshead to move, the cost of replacing, or repairing, a front head, is less than that of a rear head. Of course, where the guides are opposite the forward drivers, the crosshead must be blocked in the center, or front end, of the guides, to prevent the forward crank pin from coming in contact with the wrist pin or crosshead.

For the "Locomotive," or "Underhung," type of crosshead, block at the travel mark at the rear end of the guide bars, to prevent the cylinder packing rings from working down into the counterbore. Then secure the crosshead by inserting and lashing a block of hard wood between the crosshead and the guide block to hold it at the travel mark.

With the "Alligator" type of crosshead, to block it at the front end loosen one of the guide bolts with the plow bolt head, which holds the lower guide bar to the yoke, drive the bolt up so that the head will be about three-quarters of an inch above the bar; then cut a block about one-half of an inch longer than the distance from the bolt head to the crosshead. Sink the bolt head into the end of the blocking and lash the blocking down on the guide bar, so that it will prevent the bolt from working out, and will hold the crosshead secure.

When the crosshead is blocked, remove both of the cylinder cocks, or fasten them open if the steam edge of the valve, or the valve seat, is broken, take out the cylinder cock at the end where the piston rests, and block the cylinder cock open at the opposite end, to allow the condensed steam to escape from the cylinder.

Eccentric Slipped.

A slipped eccentric is generally indicated by an uneven exhaust, and when you are not certain which eccentric is out of alignment, it may be located in various ways. First, move the reverse lever to the forward notch, and if the valve squares up the go-ahead eccentric will be found all right; then pull the reverse lever up toward the center, and, if the engine becomes more lame as you approach the center, you may rely upon the fact that the back-up eccentric has slipped. When a go-ahead eccentric slips it will cause a later admission of steam, and when a back-up eccentric slips it will cause too early an admission. Now, having located the defective side, place the reverse lever in the center of the quadrant and you can ascertain whether the blade has slipped too long or too short, because, with an engine in this position, when the eccentrics are properly adjusted, the link will stand perpendicular; so by observing the incline of the link you can tell whether it is necessary to lengthen or shorten the blade. Of course, if the eccentric is keyed on the shaft, it is a simple matter to turn it on the shaft until the grooves of the eccentric and the shaft coincide, and then replace the key and tighten the set screws.

On engines with indirect motion, when the crank pin is on the forward center, the go-ahead eccentric will be above the pin, and the back-up eccentric below the pin, and when the pin is on the back center their positions are reversed. The rib of each eccentric is usually set at about the third spoke away from the pin; and ahead or back of the pin, depending, of course, upon whether it is an inside or an outside admission valve. The spokes in different wheels may vary, but so does the lead and lap. Remember, however, that the eccentrics should never be at right angles to the pin, but each should incline slightly toward the pin. Perhaps the quickest way to set a slipped eccentric approximately correct is by the old method of marking the valve stem, which may be explained as follows: Place the crank pin on either dead center on the disabled side, the forward center being the most convenient. If the forward eccentric has slipped (the forward motion eccentric is usually attached to the top of the link), place the reverse lever in the extreme back notch of the quadrant, and, with a knife or some other sharp instrument, scratch a mark on the valve stem as close as possible to the gland. Now place the reverse lever in the extreme forward notch of the quadrant, then go under the engine and move the slipped eccentric around on the axle until the same mark on the valve stem returns to its original position, and, after you make sure that the throw of the eccentric is on the opposite side of the main pin from the one that is solid on the axle, set the eccentric in that position.

If the crank pin is on the forward center, the center of the eccentric should be above the pin, and if on the back center, below the pin. If the back-up eccentric slips, go through the same performance in exactly the reverse manner, by placing the reverse lever in the forward notch and marking the stem, and then by putting it in the back notch and setting the eccentric as explained above. If both of the eccentrics on the same side slip, set each eccentric as near as possible to the positions we have previously explained. Then place the pin on the forward center, block the wheels, open the cylinder cocks, place reverse lever in the back notch, and give the engine a little steam. Now move the back-up eccentric until steam appears at the front cylinder cock, and then fasten that eccentric. Next, place the reverse lever in the forward notch, and move the go-ahead eccentric until steam appears at the same place—the forward cylinder cock; then fasten that eccentric and you are ready to proceed.

Another method is to get the engine on its dead center, as nearly as you are able by eye, and, if the forward eccentric has slipped, hook the reverse lever clear back; then clamp the valve stem so that the valve cannot move, and take out the bolt that connects the forward motion eccentric rod to the link. Then throw the reverse lever all the way ahead, being careful that the valves do not move, and, by moving the slipped eccentric until you can put the jaw bolt in, the eccentric will be set near enough to its correct position to run in. You must be careful, however, to see that the eccentrics are not in the same position on the axle, or you will have both of them set to run backwards; one eccentric must point up, and the other down. A back motion eccentric can be set in the same manner, except that the reverse lever must be thrown ahead. Clamp the valve stem, then take the jaw bolt out of the back motion rod and move the back motion eccentric until the bolt will go in without moving the valve or the rocker arm. When an engine is on dead center, the valve should be in exactly the same position, when the reverse lever is in the extreme back notch, as it is when the lever is in the extreme forward notch, so that if the valve rod does not move while the pin is out, and the reverse lever is being moved, the eccentric will be nearly correct. After the pin is put in, the valve rod will move while the lever is being moved, but, in the extreme notches, will show that the rod is in exactly the same place.

Eccentric Rod, or Strap, Broken.

If the forward motion eccentric rod breaks, it should be taken down, together with its strap. It is sometimes possible to substitute the back motion strap and rod on the forward motion eccentric, operating the engine in full forward gear. If the substitution of parts can be made, it will be well to chain loosely around the bottom of the link, to allow for the throw of the eccentric, and to prevent the link from being thrown too far at the bottom. If the change of parts cannot be made, the eccentric rods and straps should be taken down, and the engine disconnected on that side. If the eccentric strap is broken through the lug bolt, it can be patched well enough to get in by using a washer somewhat larger than the lug. Bend the washer over at the top, to hold the lug in place, passing the bolt through the washer, and tightening it and the opposite bolt securely.

If the eccentric strap, but not the rod, breaks, it is possible to place the back-up eccentric strap on the forward motion eccentric, and clamp the rod to the strap. But, in such a case, the two rods will always be actuated by the forward motion eccentric in full gear, and the engine can not be reversed.

If the main rod is not bent, or damaged, and is too heavy to be conveniently handled, it should be left up. In such a case, remove the indicator plug and lubricate the cylinder, or ease off the cylinder head stud a trifle, drive a few wooden wedges between the ends of the cylinder and its head, and lubricate the piston with oil through the opening. Also remove the rear cylinder cock, to provide for air circulation, to relieve the vacuum, and to allow for the escape of compression.

With a broken back motion eccentric rod or strap, remove the broken parts and secure the lower end of the link so that it will not turn over, as explained above. Run with reverse lever near full forward stroke, and proceed with full train.

Eccentric Blade Slipped.

A slipped eccentric blade is indicated by an irregularity in the exhausts. To reset the blade, when it slips, place the crank pin on either center, and the reverse lever in the forward notch, set the brakes, and block the wheels; then open the cylinder cocks and adjust the blade until steam appears at the front or back of the cylinder, depending upon the position of the crank pin. That is, if the crank pin is on the forward center, and the go-ahead eccentric blade has slipped, steam should appear at the front end of the cylinder, and vice versa. Secure the eccentric blade to the strap by tightening the bolts, and proceed.

Piston or Valve Stem Glands Broken.

If the gland breaks in two, try to wrap it with bell cord or wire. If a lug breaks off, make a wooden clamp; there are two nuts on each stud, so remove one of the nuts and tighten the other up against the temporary clamp. If you loosen one stud, wrap wire or rope around the steam chest and try to hold it secure, or, remove part of the packing and shove the gland in further, and try to hold it with one stud. If all remedies fail, disconnect one side.

Valve Rod Broken.

Remove the broken parts and block the valve to cover the ports.

Valve Rod Bolt Broken.

In case the bolt attaching the valve rod to the top rocker arm is lost or broken, use another bolt, or substitute the knuckle pin out of the front drawbar and secure it in place, but do not disconnect.

Crank Pin Broken.

If it is a main rod crank pin, disconnect one side and remove all rods on the disabled side. If a back pin on an eight-wheel engine, simply remove both side rods; if a back pin on a ten-wheel engine, remove the back pin and side rods only. If a front pin on a ten-wheel engine, remove the forward pair of side rods only.

Main Rod or Strap Broken.

Disconnect on the broken side.

Rod Set Screws Broken.

When it is necessary to remove a key from a rod, if the set screw is broken and cannot be backed with a chisel, if in the back end of a main rod, take the strap bolt out of that end of the rod, and block the crosshead. Then, with a pinch bar, move the engine until the key is loose. If the set screw is broken in a parallel rod. take the bolts out of the strap where the screw is broken, block the other drivers, and, with pinch bars, slip the wheels until the key is loose.

Saddle Pin or Link Broken.

Raise the link to the height necessary to give the desired cut-off to handle the train; block between the link block and the top of the link, remove all broken parts, and proceed, keeping the disabled link well oiled. Do not reverse. It is also practical to insert a block in the lower end of the link, but it must be cut one inch shorter than the distance from the link block to the end of the link, to allow for the slip of the link. If you wish to back up, the longer block must be placed in the top end of the link, to raise the link high enough to place the back-up eccentric in control of the valve, and to thus secure proper steam distribution.

Side Rod or Strap Broken.

Remove the broken rod and the parallel rod on the other side. If it is a ten-wheel engine, and this cannot be done, remove all of the side rods. If a front or back rod, or strap is broken, on a twelve-wheel, or consolidated engine, remove the broken rod and the one directly opposite it, if this can be done, and leave the others up.

Intermediate Side Rod Broken.

In this case all the rods should be taken down, and the engine must be towed in. Of course, it is possible for an engine, with all side rods removed, to move under its own steam, but such practice is not considered safe. For, if the main rod remains up on the disabled side, the wheels are almost sure to slip, and to injure the rods on the opposite side. Most roads forbid the practice of operating engines with all rods down on one side.

Guides, Blocks, or Bolts Broken.

If any of the bolts break, try to replace them. See that all of the nuts are tight, or they may be the cause of springing the piston. If a guide bar is broken badly, disconnect one side.

Guide Yoke Bent.

If a yoke is bent and will not hold the guides secure, disconnect one side.

Cylinders Broken.

Back Head.

Disconnect the valve stem on the broken side and place the valve central to cover the ports, provide for lubrication, and proceed. In warm weather cover the front end of the cylinder head with canvas or boards, to keep out the dust. If it be necessary to remove the guides and broken head, remove the piston also. If the cylinder of a compound be disabled, disconnect and block. Then operate as a simple engine at a very low rate of speed—4 or 5 miles per hour.

Forward Head.

Same as for the back head. Another method advocated, but seldom practiced, by which three-fourths of the power of the engine may be retained, is to remove the steam chest cover and plug up the forward steam port with wood and proceed, working both sides. This method is impracticable, however, owing to the shape of the steam port cavity on most engines, and the time it would require, and to the improbability of the block remaining intact.

Both Heads.

This is a "catch" question propounded to test a man's knowledge, for it is seldom that such an accident could occur, but in case it did happen the great majority of engineers would telegraph for new cylinder heads, or ask to have their engines towed in. However, if both pistons were all right and both steam ports were properly blocked, the engine could handle itself in such a condition. But such practice is not practical on account of the shape of the ports, and the difficulty, if not impossibility, of getting the ports securely blocked without interfering with the movement of the valve. Right here it may be well to state that it is the *steam port* which should be blocked, and not the supply ports.

Cylinder Key Lost or Broken.

If you are unable to substitute a spike or rod key, to replace it, run the engine light, or disconnect the disabled cylinder and proceed on one side, to avoid further damage.

Smoke Box, Front End, or Stack Broken.

Board up the opening to make it as nearly closed as possible, using the front end bolts to hold the boards. A cab curtain, or strips of sod, can also be used to advantage. When the stack is broken, a barrel, or petticoat pipe, placed over the smoke arch, will be sufficient to bring the engine in light.

Rocker Box Broken.

If it cannot be clamped securely, disconnect that side. If you can remove the rocker arm it will save taking down both eccentric straps, but if you cannot remove the rocker without difficulty, remove both eccentric straps and tie the top of the link to the short arm of the tumbling shaft, to prevent the link from tipping over.

Rocker Arm Broken.

Top Arm.

When either rocker arm is broken, it will deprive the engine of control on the disabled side. In such a case, it is advisable to clamp the valve with the back port slightly open, to remove the cylinder cocks, or indicator plugs, so as to provide for lubrication and circulation of air, and then to proceed on one side. If it is thought advisable to take down the main rod, disconnect the valve rod, clamp the valve central on its seat, remove cylinder cocks, clamp crosshead securely at the back end of the guides, or at the front end when necessary, place a collar on the main pin, and proceed. Of course, it is not necessary to take down the main rod in order to provide for lubrication, or circulation of air in the cylinder, for sufficient lubrication can be obtained when the piston is left connected to the main rod, by leaving the back admission port slightly open, with the cylinder cock removed, to allow steam to pass through the port and out of the cylinder cock; or oil may be poured into the indicator plug opening, when the valve is blocked centrally; also, the cylinder head may be wedged open a little, so that oil may be poured into the opening.

Lower Arm.

Follow one of the above methods, but make sure that the link on the disabled side will clear everything in both full gears. The rocker arm can be secured parallel with the engine frame, or, by securely fastening the link hanger on the disabled side to the one on the opposite side, the good link will hold the disabled link away from the rocker arm. Of course, when the link hangers are fastened together, it will not be necessary to disconnect the valve rod from the top rocker arm. If absolute safety is desired, and you have time to spare, take down both eccentrics, and tie the top of the link to the short arm of the tumbling shaft—to keep it from tipping over, which might prevent reversal of the engine.

Rocker Pins Broken.

If the top rocker pin should break, replace it with an old one, or disconnect that side. If the bottom pin breaks, it is sometimes possible to remove the top pin and turn the bottom arm up high enough to clear the link; in such a case tie it up to the guide yoke, but make certain that it will clear the link when in full gear in each motion. If you are in doubt, remove both eccentric straps, tie the rocker forward or back to clear the link, and then tie the top of the link to the short arm of the tumbling shaft, to keep it from tipping over. You can use bell cord or wire for this purpose. Then disconnect on the broken side.

Throttle Rod Disconnected or Broken.

If you are unable to shut off steam, make sure that it is the throttle rod at fault, and not leaking tallow cups, if tallow cups are used. If the throttle is open, do not attempt to proceed with your train until you are sure that the air brakes are in good working condition. Keep the boiler pressure low and the engine can be controlled by using the air brake and the reverse lever for stopping. When you find that you cannot close the throttle, or prevent steam from entering the cylinders, if you have a high pressure of steam, the engine will be apt to slip its drivers. In such a case do not use sand, but control the engine with the reverse lever until the pressure is reduced so that the engine will not slip. If much sand be used, there is danger of damaging the machinery. If the throttle valve will not open, it is probable that the rod has become disconnected inside of the boiler. If the engine has steam chest tallow pipes leading from the cab, you may be able to admit sufficient steam into the cylinders through these pipes, or, by removing the indicator choke plugs and steam chest valves from the oil pipe, sufficient steam may be obtained to run a small engine light, but you cannot obtain enough power in this manner to run a modern heavy engine. If you are unable to do this, the only remedy is to kill the fire, raise the dome cap, and connect the throttle rod; but this is not expected of a roadman, so prepare to be towed in.

Tumbling Shaft, Arms, or Stands Broken.

If either tumbling shaft stand becomes broken or bent, so that it cannot be clamped and used, it is sometimes possible to use a wooden block in its place. If the top arm of the tumbling shaft be broken or bent, so that it cannot be used, a pinch bar, or other iron rod, may be used, in some cases, across the engine frame, to hold the short arm in its working position; in such a case care must be taken to see that the rod is securely fastened, and that it does not interfere with the eccentric rods. If a rod cannot be used in such a position, place the reverse lever in a notch of the quadrant such that you are certain you can start the train, and then raise the disconnected link about level with the opposite one. Fit and fasten a block of wood in the slot, between the top of the link and the link block, to hold it in that position. It is not generally necessary to block the bottom of the disabled link, but, as a matter of safety, it will be to your advantage to do so. In fitting the blocks, space should be allowed for the slip of the link. In such a case, it is considered good practice to fasten the reverse lever securely at the quadrant, to hold the link in place, and to prevent reversing. Under no consideration should the reverse lever be dropped down enough to allow the lifting arm to pass under the blocked link, nor raised high enough to permit the lifting arm to pass behind the disabled link when the tumbling shaft is below the engine frame. If you wish to reverse the engine, you must also cut another block long enough to hold the link all the way up, and substitute it for the short block, before you reverse. If the short arm of the tumbling shaft be bent or broken, remove the link lifter on that side and block the link, as previously explained. In case both of the short arms be broken, remove both of the link lifters and block both of the links.

Link Hanger, Saddle, or Either Pin, Broken.

Remove all broken parts to avoid further damage, and then block the link the same as for a broken lower tumbling shaft arm.

Valve or Bridges Broken.

Remove the steam chest cover and ascertain the extent of the breakage. If it be serious, remove the broken parts and disconnect the engine on the disabled side. If the valve can be used to cover both steam ports, set it centrally upon its seat, and hold it in this position with a valve stem clamp, or you may use blocks, to keep the steam out of the cylinder. If you can block the steam ports, do so; then fill the valve chest with pieces of wood, and screw down the steam chest cover, to secure the blocks in place. Take out the cylinder cocks, make provision for lubrication, and proceed. The practice of using a thin board under the valve cannot be adopted with balance valves.

Valve Stem, or Rod, Broken.

If the valve stem or rod be broken inside of the steam chest, disconnect the valve stem from the rocker arm, on the disabled side, take out the front cylinder cock, or block it open, remove the steam chest cover and block the valve, leaving the front steam port open about 1/16 of an inch.

Link Broken.

Remove the eccentric straps, and take out the broken link; or fasten the top of it to the tumbling shaft arm with bell cord;

take out the cylinder cocks, or indicator plugs, and block the valve with back port slightly open. If one of the bolts connecting a built-up link be broken, or lost, substitute another bolt if possible, or remove the bolt from the bottom, and insert it in the top, of the link, binding the bottom of the link with wire.

Link Block Pin Broken.

If a bolt cannot be substituted, disconnect the top rocker arm from the valve rod, clamp the valve stem, block the valve with the back port slightly open, take out cylinder cocks, or indicator plugs, and lash the rocker arms parallel with the frame rail of the engine. If the arms will not clear the rods in this position, lash the link hanger on the disabled side to the one on the opposite side, to hold the link clear of the lower end of the rocker arm—or remove the eccentrics and blades and lash the link to the hanger.

Reach Rod Broken.

Use the same remedy as given for broken top arm of tumbling shaft.

Reverse Lever Broken.

If the break occurs in the reach rod hole, or below it, apply the same remedy as for the top arm of the tumbling shaft, or the reach rod. If, however, the break occurs above the reach rod connection, you can usually hold the reverse lever intact by fitting blocks inside of the quadrant, or, if it be a solid quadrant, by blocking it in any manner whereby you can secure a brace.

Extension Rod Broken.

The engine truck sometimes bends or breaks the long extension rod which connects the link block and the rocker arm on a ten-wheel engine. In such a case, disconnect one side and remove the broken rod.

Grates Broken or Burned.

When grates become broken, or are burned out, on the road, replace the same with fish plates, or any short pieces of iron at hand.

Pumps Fail to Work.

When the pumps fail to work, ascertain if there is plenty of water in the tank, and see that the tank valve is connected; open the heater cock for a few seconds, open the pet cock, close the heater, and try the pump. If it will not work now, slack down the lower pump joint. Now, if the water flows freely, move the engine about a dozen revolutions, and then tighten up the joints. If the water does not flow freely, the feed pipe, strainer, or hose is probably choked (or the inside lining of the hose may be torn loose), and must be cleaned out. If the pumps fail to work when the water flows through the joints, take the lower valves out and see that they are free. If you are still unable to find anything wrong, the pump will have to be repaired in the shop.

But little more need be said regarding pump failures. If the remedies suggested do not cause the pumps to operate, it is probable that you will not have sufficient tools with you, nor the time necessary, to make the repairs, such as changing the lift of, and facing and grinding, the valves.

Safety Valve, or Whistle, Blown Out.

If a safety valve or whistle blows out, start both injectors immediately, in order to secure an ample supply of water before the pressure is reduced and the injectors stop working. Then kill or smother the fire, and drive a soft wooden plug about 12 inches down into the hole. Dry pine is preferable as a plug, for the steam will cause it to swell and fill the hole. Now place a plank over the plug as a lever, to hold it in place, and bind the plank to the hand railing. If you have retained sufficient water in the boiler, fire up, but keep the steam pressure low, and proceed. In case of a safety valve blowing out, take care to see that the remaining valve will relieve the pressure properly. If a safety valve spring breaks, screw down the adjusting screw so that the safety valve will not operate. If this cannot be done, substitute a small nut for the spring, and screw down the adjusting screw until the safety valve is out of action.

Broken Water Glass.

Shut off both cocks and use the gauge cocks. The gauge cocks should be used frequently, even when the water glass is in good condition.

Blow-Off Cock, or Washout Plug, Broken.

If a blow-off cock, or washout plug, will not close, kill the fire immediately, and, if near a side track, allow train to drift on to same, so that it will clear the main track. If you cannot repair the blow-off cock, and the temperature is freezing, drain all water from the boiler, pumps, tank, etc., and prepare to be towed in. If the blow-off cock or washout plug blows out, drive in a soft wooden plug tightly, refill the boiler, if near a tank, fire up, and proceed under low steam pressure. If you are unable to refill the boiler, prepare to be towed in.

Injectors Fail to Work.

If both injectors fail to work, kill the fire and prepare to be towed in; you should, however, make every effort to keep the injectors working.

Lubricator Fails to Work.

Shut off the steam, disconnect the oil pipes from the lubricator, and oil through the pipes frequently.

Driving Springs or Hangers Broken.

With the heavy engines now in use, road men are not expected to jack up the engine, but even with a small engine the quickest way is to use wedges on the rail, when possible to do so, as time is usually an important consideration; but it should be done carefully or you may break other springs or hangers, or the engine may leave the rail. If you have an eight-wheel engine with overhung spring rigging, and a forward spring or hanger should break, place a fish plate or other piece of iron between the top of the back driving box and the frame on the broken side, which will save raising the wheel that much higher, and will permit of the use of a thinner wedge; now place a wedge on the rail, and run the back wheel up on it, which will take the weight off the forward box. Now block solidly with wood between the top of the forward box and the frame, remove the broken spring or hanger, and also the spring saddle, if necessary. Run the engine down off the wedge, remove the fish plate from the back box, and run the main wheel up on the wedge, which will take the weight off the back box and relieve the equalizer. Now pry up the front end of the equalizer, and block it securely. Then let the engine down, remove all loose parts, and the task is completed. If the back spring or hanger be broken, go through the same performance in the reversed manner, by running the main wheel up on the wedge first, etc. If the engine be a mogul, or ten-wheeler; to raise the weight off the main wheel run the forward wheel up on the wedge, and then to raise the weight off the forward pair, run the main wheel up on the wedge. If it be a mogul, and a forward spring or hanger be broken, you may have to remove both forward springs, and block on top of both forward boxes, but if it be only a hanger, a chain may sometimes be used to replace it. When you block both forward boxes, also block the intermediate equalizer to the truck. When the springs and equalizers are placed below the frame, proceed in the same manner; then block or chain up the equalizer until level, and remove, or secure, the broken springs and hangers. When the spring hangers straddle the frame, it is sometimes possible to block between the hanger and the frame. Should the large spring below the frame, and between the drivers, break, block the top of both boxes, or block between both long hangers and the bottom of the frame, and remove, or secure, the broken spring and equalizers.

Should the small coil spring hanger, back of the rear drivers, break, it may be possible to remove one of the small equalizers which ride the back box. If so, block on top of the box; if not, and you cannot hold the spring hanger in any other manner, you may be able to chain the back end of the small equalizers to the frame. If not, you must let the frame ride the box, but run very slowly.

Equalizers Broken.

Raise the engine the same as for a broken spring or hanger, when it is possible to do so. If an equalizer on an eight-wheeled engine, block on top of one box and block up the loose end of the equalizer, when possible, the same as for a broken spring or hanger. If it cannot be used, remove the equalizer and block on top of both boxes. If an equalizer below the frame, do likewise, or chain it up. If a forward equalizer on a ten-wheeled engine, block on top of the forward and main boxes, and block up forward end of the back equalizer. If it be a cross equalizer on a mogul, block on top of both forward boxes, and block on top of the back end of the long intermediate equalizer that goes to the truck. If the intermediate equalizer breaks, block between the boiler and the cross equalizer. If it be the cross equalizer on a four-wheeled pony, block on top of both forward boxes. When this equalizer is below or between the frames, it is sometimes possible to block between the hangers and the frame.

If a small equalizer that rides the back box, block on top of the back box and chain up the back end of the bottom equalizer. If it be a truck equalizer, block on top of truck boxes between the box and truck frame. Always remove or secure all loose parts.

Center Pin Broken.

If the center pin of a two-wheeled, or pony truck breaks, jack up the front end of the long equalizer high enough to chain it to the forward deck braces, or castings, or insert a fulcrum under the forward end of the equalizer and back the engine until the equalizer is raised higher than its ordinary position. Then run a tie or rail across, and on top of, the main frame, and chain the front end of the equalizer to the tie or rail. Another method: place a truck brass on top of the axle and allow the lower end of the equalizer to ride upon it.

Equalizer Stands Broken.

If the equalizer stand breaks, use the same remedy as for a broken equalizer. But if only the bolts break, you may be able to find some other bolts to replace them, or take bolts off some other part of the engine that will fit, and the loss of which will not impair the working of the other parts.

Center Casting Broken.

If an engine center casting on a four-wheeled truck should break, jack up the front end of the engine. If you can find two short rails, run these across the top of the truck equalizers and under the center casting; if you cannot find any rails, block up both sides between the truck frame and the cylinder saddle. But in this case, run very slowly when rounding curves, as the engine will not track very well. If the top, or male casting breaks, you should block up in the same manner. If the pony truck center casting breaks, block between the truck frame and the engine frame on each side, and go around curves very slowly.

Pedestal Jaws Broken.

If the break be in the main jaws, or between the main jaws and the cylinder, disconnect the valve on the disabled side, and proceed light. Do not attempt to haul any part of a train when the main jaws are broken, or you will place additional stress on the side rod on the disabled side, and on the frame on the opposite side; which may bend the latter. If the break occurs back of the main jaws, remove the loose parts and proceed with full tonnage.

Pedestal Bolt or Binder Broken.

If the pedestal bolt, or binder, of the main jaw be broken, and the shoe and wedge remain in place, remove a bolt from the rear jaw, insert it in place of the broken bolt in the main jaw, and substitute some other bolt in the back jaw. If this cannot be done, reduce your tonnage about one-half and proceed. If the bolt, or binder, be lost, it is probable that the shoe and wedge will be lost also. In such a case, use wooden blocks for the shoe and wedge, and support them on a substituted wooden binder; then proceed on one side with full train.

Setting Up Wedges on the Road.

Run the engine to a piece of straight, level track, and work with steam up, so that the parts of the frame near the firebox are expanded to the running length, as they will shorten when cold.

Place the engine on the top quarter on the side under consideration, place a block on the rail behind the driver on which the wedge is to be set up, and move the engine back until the driver bears against the block. This will throw the driving box solidly against the shoe at the front of the box, leaving all lost motion at the back of the box, so that the wedge will move freely. Now set the wedges moderately tight by means of the jam nut on the wedge, prying the wedge up with a small bar and then tightening the nut on the bolt. It is good practice to set the wedge up as far as possible, and then to pull it down, about 1/16" for main drivers, and ⅛" for all other drivers.

When it is necessary to set up a wedge which has a broken wedge bolt, this can frequently be accomplished by placing a nut on the top and the bottom of the wedge, securing them in place with wire. A broken wedge bolt often causes the wedge to stick to the box, resulting in the driving box running hot, but the above remedy will eliminate this trouble.

Engine Truck Spring or Hanger Broken.

First raise the front end of the engine with jacks. If it be a four-wheeled truck, pry up the frame on the broken side and block between the equalizers and the truck frame, close to the spring band, keeping it up level with the other side. If it be a two-wheeled, or mogul truck, block between the top of the truck box and the truck frame.

Raising a Defective Wheel.

If jacks are not available, or if the engine be too heavy to be raised with ordinary jacks, this may be done by running the wheel up on a re-railing frog, or wedges, or by lifting the engine frame by means of a fulcrum and levers. Now remove the oil cellar, and insert a block in its place; also insert blocks in the space between the pedestal brace and the bottom of the driving box, and see that the driving box is well oiled. If wedges are used, they should be of good hardwood material, for if they are crushed, or if they give way, and the wheel slips when elevated, the frame, or the driving boxes, are likely to be damaged, or the wheels derailed. Iron is preferable for blocking, with heavy engines.

Driving Axle, Wheel or Tire Broken.

An accident of this kind is very serious, as it may strip one side of the engine, or perhaps disable it. But, in most cases of this kind, the engine can be blocked up so as to reach the nearest siding, if not the terminal. If the injury be slight, by running slowly and by using care, the nearest siding may be reached, where you can notify headquarters, and where the engine should be properly blocked to run to the shop without assistance; however, this requires considerable good judgment and work on the part of the engineer. We shall endeavor to treat these various mishaps separately.

Main Wheel Broken Off the Axle.

Disconnect on the broken side, and remove all side rods and the broken wheel. The eccentrics will prevent the other main wheel from leaving the rail, even though it has a blind-tire. Use a jack and raise the axle on the broken side, remove the oil cellar, and fit a hardwood block between the driving axle and the pedestal jaw, or binder; then use old rod keys, or any kind of iron, and block between the spring saddle and the top of the frame, to keep part of the weight off the axle at this point; then remove the jack. Now raise the engine slightly on the broken side, and block between the top of the driving box, or boxes, nearest to the main wheel on the broken side, using hard wood or iron. If it be an eight-wheeled engine, also block between the engine truck equalizers and the truck frame on both sides, as additional weight will be imposed upon the truck. If it be a consolidated engine, it will only be necessary to block on top of the two boxes nearest to the main box. Then let the engine down and proceed slowly.

Main Wheel Tire Broken.

If it be merely a bad crack, or even if it be broken through, if the tire is still on the wheel, let the fireman stand on the steam chest and watch the tire while you run slowly and try to reach the nearest siding. If the tire be very loose, take off the rods and remove the tire. Disconnect on the broken side, and remove all rods. Then place a jack under the axle and raise the broken side, remove the oil cellar, and fit a hard wood block between the axle and the pedestal jaw, or binder, and also block under the spring saddle in order to keep the weight off the box. Remove the jack from beneath the axle, raise the engine on the broken side, and block the top of the driving box and the frame on the box, or boxes, next to the main wheel on that side. If it be an eight-wheeler, or consolidated engine, block the boxes the same as for a main wheel broken off of the axle. Now let the engine down, and, if the wheel clears the rail, proceed slowly. If the wheel will not clear the rail, block the engine a little higher on the broken side. If you cannot remove the broken tire, and it will not permit the wheel to revolve, disconnect on both sides and prepare to be towed in. Block up the main pair of wheels high enough to clear the rail on both sides, then block on top of all other driving boxes, and, if it be an eight-wheel engine, block on top of the truck equalizers.

Main Wheel Cracked.

If the wheel is not too badly broken, watch it closely and run slowly to the first siding, Disconnect on the broken side and remove all side rods, which will take the strain off the crank pin on the broken side; you may then be able to proceed. If the break is of such a nature that you consider it unsafe to proceed with the engine alone, or light, then block the wheel up the same as for a broken main tire, and see that the tire clears the rail, then proceed slowly.

Forward Wheel Broken Off the Axle.

Remove both side rods between the forward and the main pairs of wheels; if this cannot be done remove all side rods and remove the broken wheel, or chain it up to the frame. Now use a jack and raise the axle on the broken side; remove the oil cellar, and fit a hard wood block between the axle and the pedestal jaw, or binder. Block under the spring saddle on the broken side. If the forward pair of wheels have blind tires, unless there are collars on the axle, the opposite wheel must also be raised and blocked to clear the rail; in this case raise the front end of the engine and block on top of each main box and on both truck equalizers, and, if using a pony truck, block between the truck frame and the engine on both sides. If it be a six-wheel connected engine without an engine truck, block on top of the main boxes and under the back boxes, which will tend to counterbalance the weight. If it be necessary to disconnect the forward pair of springs on a mogul, an old truck brass may be used between the truck axle and the long equalizer. But if the forward pair of wheels have flanges, or collars on the axle, let the good wheel run on the rail and block on top of its driving box and on top of the main box on the broken side, and on the truck equalizer, as previously explained. See that the forward crank pin on the good wheel will clear the crosshead in all positions; if it will not, both sides of the engine must be disconnected and the engine towed in; but, assuming that the crank pin will clear, proceed slowly, especially when rounding curves.

Forward Wheel Tire Broken.

If the tire remains on the wheel, try to keep it there until the nearest siding is reached; then remove both side rods between the forward and main wheels. If this cannot be done, remove all side rods, and then, if the tire can be removed, take it off and block up the wheel. If you are unable to remove the tire, block up the wheel on that side, so that the tire will clear the rail. But if the tire be bent, or twisted, in such a manner as to prevent the wheel from turning, both wheels must be blocked up to clear the rails; block them the same as for the forward wheel broken off the axle.

Some engineers claim that the side rods may be left up by simply slacking both rod keys, but this is not considered good practice. It will be much safer to remove the two forward parallel rods, if it can be done conveniently; then to proceed carefully.

Forward Wheel Cracked.

If the break occurs at, or near, the crank pin hub, remove both side rods between the main and forward pairs of wheels, and all side rods, if necessary. Now, if the wheel be not too badly broken, it is probable that you can go ahead, but maintain a close watch on the broken wheel, and run slowly. If it is not safe to run this way, you must block up one or both wheels the same as for a broken tire.

Back Wheel Broken Off the Axle.

Remove both side rods between the main and the back pairs of wheels, and all of the side rods, if it is necessary. Place a jack under the axle and raise the broken end; remove the oil cellar and fit a hard wood block between the axle and the pedestal brace, or binder, on that side. Also block under the spring saddle, and let the good wheel remain on the rail, and block on top of its box; drive wedges between the drawbar and the chafing iron, and block on top of both main boxes. Leave your train and proceed cautiously.

If the axle should be broken between the two driving boxes, try to remove both wheels. If this cannot be done, they must both be blocked up high enough to clear the rail, and, if it is a heavy engine, a part of the weight of the engine must be transferred to the tender. Block between the equalizers and the engine truck frame on both sides; raise the back end of the engine and block on top of both main boxes, and wedge between the drawbar and the chafing iron. If this can be done, you may still run the engine light, but if it cannot be done run a short rail into fire box door and chain it to the drawbar, and block up under the back end of the rail on the tender. If the latter method is followed the engine must be towed in, so disconnect both sides.

Back Wheel Tire Broken.

Try to keep the tire on the wheel until you reach the nearest siding, running very slowly. Then remove both side rods between the main and rear pairs of wheels, and all side rods; if necessary, also remove the tire if you are able, and block the wheel up to clear the rail, leaving the opposite wheel on the rail. If you cannot remove the broken tire, and it will not allow the wheel to revolve, both wheels must be blocked up to clear the rail. Block them up in the same manner as for a back wheel broken off the axle.

Back Wheel Cracked.

Run slowly to the nearest siding, and then remove both parallel rods between the main and rear driving wheels; if this cannot be done remove all of the side rods. By removing the rod, the strain is taken off of the crank pin, and, unless the wheel be broken very badly, you may proceed slowly. If you consider it unsafe to run in this way, block the wheel up to clear the rail, the same as for a broken tire on the back wheel.

Cause of Tire Cutting.

One side of an engine being higher than the other will cause the tires to cut on the low side. If the driving axles are not square, or at right angles with the cylinders and center casting, or if they are not an equal distance apart, the wheels that are too far back will cut their flanges. When the engine truck wheel flanges are cut, it is an indication that the engine is not in the center of the truck. The front of the engine should be moved toward the cutting side. An engine not being central upon its truck may also cause the forward driving wheel flange to cut on the side opposite that on which the truck flanges are cutting.

Engine Truck Wheel or Axle Broken.

If a piece of the flange be broken off a truck wheel, run slowly, especially over frogs, switches and crossings. If a piece be broken out of a truck wheel, it is sometimes possible to chain the wheel, or to place a timber across the rail in front of it, so that it will slide to the nearest siding, where that pair of wheels must be removed, or blocked up to clear the rail. If it be a four-wheeled truck, chain the broken end of the truck frame to the engine frame, and then raise the engine and block solidly between the truck box and truck frame, and between the truck frame and engine frame above the good pair of wheels. Also block on top of the forward driving boxes, leave the train and run slowly, especially when rounding curves. If it be a two-wheeled truck, the truck frame should be chained to the engine frame, and the front end of the engine should be raised high enough for the truck wheels to clear the rail. Then block solidly between on top of the two forward driving boxes and under the two back driving boxes, but do not break the back driving box cellars. This will help to hold up the front end of the engine. Place fish plates on top of the back driving boxes before you raise the front end of the engine, which will prevent breaking the driving springs, or hangers. If a truck wheel be broken off, or if the axle be badly bent, block up in the same manner. It is sometimes easier to remove the truck entirely.

Engine Truck Spring or Equalizer Broken.

When a truck spring is broken, raise the front end of the engine and then block on top of the equalizer and under the engine truck frame. Also raise and block the opposite corner of the truck on that side, in the same manner. For a broken equalizer, place blocks on top of the engine truck boxes and under the engine truck frame.

Engine Truck Frame Broken.

Raise the weight of the engine and place pieces of heavy iron between the equalizers and the truck frame, or chain a piece of rail to it for a splice.

Tender Wheel or Axle Broken.

If you can find a piece of timber, a rail, or a cross tie, of the proper length, place it across the top of the tank directly over the broken pair of wheels; then block up under the timber, rail or tie, to protect the flange on top of the tender, and jack up the broken pair of wheels to clear the rail. While they are elevated, chain the truck to the timber, rail or tie, above the tank, on both sides so that it will carry the disabled truck. It is sometimes possible to slide the wheel, or truck, to a siding, by placing a tie across the rail in a position to prevent the wheel from turning.

Engine Frame Broken.

If a main frame breaks between the main driving axle and the cylinder, and the break opens very much, disconnect the disabled side and run in light. Do not attempt to work the engine to its full capacity, nor allow another engine to pull the disabled engine while it is attached to the train. If the upper rail of the frame be broken back of the main drivers, do not disconnect, but reduce your tonnage to a point of safety and proceed. If the lower rail of the frame be broken, and the break does not open enough to endanger the upper rail, do not give up the train, but proceed, and avoid all rough usage of the engine.

Draw Bar Broken.

If the engine be supplied with safety chains, and the train is not too heavy, the chains will hold the tank. If the engine is not equipped with safety chains, secure a chain from the tank box or caboose, and chain the tank to the deck. Safety chains should not have more than four inches of slack.

Driving Brasses Broken.

If a driving brass breaks, and is cutting badly, run that wheel up on a thin wedge; then use an iron block between the top of the frame and the spring saddle, which will take the weight off of that box; or, as a matter of safety, take out the driving spring.

Wedge Bolt Broken.

It is sometimes possible to screw the nut half way onto each part of the broken bolt, and thereby hold it up in place. If this cannot be done, then, with a wire, try to fasten a nut under the wedge to hold it up.

Driving Box Broken.

Raise the wheel to which the box is attached, and block it up in the same manner as for a broken tire.

Transmission Bar Broken.

Remove all broken parts, block the valve to cover the ports on that side, provide for lubrication of the cylinders, and clamp the disconnected valve securely.

Transmission Bar Hanger Broken.

Disconnect the transmission bar, and, if necessary, remove it; block valve and link the same as for broken tumbling shaft, etc.

Refilling a Boiler Without Steam.

To refill a boiler while the engine is being towed, close all openings through which air could enter the boiler; such as lubricator, cylinder and gauge cocks, relief valves, air pump steam valve and whistle valve, and screw down the heater valve; then open the main throttle on the injector and the tank valve. Place the reverse lever in full gear in the direction in which the engine is being towed; if the engine is moved fast enough the movement of the pistons in the cylinders will pump the air out of the boiler and create a vacuum therein, which will draw the water from the tank into the boiler.

Foaming or Priming Water in the Boiler.

When we say that a boiler is foaming, it is meant that the water is full of small bubbles, about 1/6 of an inch in diameter, and when we speak of it as priming, it is meant that water is carried with the steam away from the boiler and into the cylinders. Both foaming and priming are very dangerous, for the former may leave the crown sheet and flues, or tubes, exposed so that they will become overheated, while the latter will wash the lubrication off the valves and cylinders, and may cause damage to the cylinders.

Foaming is generally caused by mineral salts in the water, and a change in the temperature will increase the tendency to foam; but it may result from grease, alkali, animal, or certain vegetable matter, which is heated with the water, and its presence is generally indicated by the water in the gauge glass becoming disturbed, muddy, or foul.

Priming results from water foaming in the boiler, and may be detected by the sound of the exhaust, and by a spray at the funnel.

When the water foams open the cylinder cocks, close the throttle gently, until the water settles solidly, and then ascertain how much water is in the boiler. Put the pumps and injectors to work if necessary, and, if possible, blow some of the bad water out of the boiler. Open the throttle gently and work the foul water through the cylinders; then close the throttle and ascertain the height of the water. Be very careful in admitting steam into the cylinders, or you may knock the packing down, or perhaps knock a cylinder head out, or cause other serious damages. If the cause of foaming is found to be grease in the tank, flow the tank over the next time you take water, and, if you can get about one-quarter of a peck of unslacked lime, put it in the tank. A piece of blue-stone, about the size of a hickory nut, which may be obtained at any telegraph office, when placed in the hose back of the screen, will prevent foaming.

Priming may be stopped by injecting about one-quarter ounce of castor oil in the water gauge glass or in the tender, but this remedy is not recommended, for it will cause a film of oil on the crown sheet and flues, or tubes, and render them more liable to overheating.

BLOWS.

An engineer who is unable to locate a serious blow in his engine, which always causes a waste of fuel, and is generally a constant source of annoyance to himself, is placed in an embarrassing position. In addition, his inability to locate the blow and accurately report the same in the work book, may cause him to lose a trip, for misleading reports result in the loss of valuable time, and expense. The defect should be described so clearly that the shop man can make the repairs with a minimum amount of labor, whether or not there is steam pressure in the boiler when the repairs are being made. In connection with his report, the engineer should also describe the characteristics, and the location, of the defects observed, to assist the shop man in locating and repairing the trouble in case the engineer is mistaken in his deductions.

However, a locomotive is susceptible to so many different kinds of blows, some of which are very deceptive, that it is often difficult, and sometimes impossible, to determine the exact location of the source of trouble. But the engineer is expected to locate the blow and correctly report the same if it is possible to do so.

How to Locate a Blow.

We shall first call atention to a few of the parts of the engine where blows occur most frequently, and describe the various sounds, and the action, of blows under different circumstances, which may assist in determining the location of the blow before a test is made; we shall then explain the correct method of testing the engine and determining the loçation of the blow. A blow may be in the cylinder packing rings; the valve seat, the gibs, rings or rider of a balanced valve; it may be in the steam pipes or nigger-head, or it may be a crack, or a hole, in the steam ports.

When on right dead center, if the engine blows badly, and is unable to start the train, it is probable that the blow is on the left side. However, if the side which stands on quarter cannot start the train, the trouble will generally be found there.

There is a difference in the sound of valve and piston blows with which experienced men are familiar. But it may happen that both valve and piston blow at the same time, which complicates the matter of locating the trouble accurately.

If it is an intermittent or recurring blow, a round roaring, rumbling sound, like whor-r-r-r, you may depend upon its being in the cylinders, and you can usually locate in which cylinder it is by watching the crank pins on a slow pull, as the blow will usually be most noticeable as the piston is in the center of its stroke. If it is a continuous sharp, shrill sound, as whis-s-s-s, it is usually in the valve seat, but a valve sometimes blows intermittently when the valve cocks at one end. If it is a strong continuous blow and you have balanced valves it is possible that one of your valve strips, valve springs, rings or rider is broken. But if your engine has a plain slide valve, reverse the engine two or three times very quickly, as it may be only a cocked valve. If it will not reseat in this way, remove the oil plug from the steam chest cover and drive the valve down. Remember, however, that balanced valves do not cock. It may, however, prove to be a sand hole in the valve or between the steam ports. A steam chest blow is easily distinguished from a steam pipe blow because it will blow straight up the stack and make a clear, singing sound, while a steam pipe blow expends its force in the front end and makes no noise when going out of the stack.

A very serious effect on the draft, as well as a waste of steam, results from a steam pipe leak. In fact, a bad leak in the rear of the bottom joint will blow into the tubes and cause the engine to smoke at the door, when the engine is standing still with wide open throttle. As a test, the front end may be opened, and the joint covered with fine cinders. The cinders will blow away from the leak when the engine is given steam. However, the practical shop test is made with high pressure water.

If you have lost one exhaust it may be a slipped eccentric, for a slipped eccentric will usually cause the valves to sound out of square. A valve yoke cracked or broken on one side only will cause one exhaust to sound out of square while the other three are normal. When the valve stem breaks off it will usually cause a tremendous blow which will continue as long as the throttle remains open. But if you have a tremendous blow at one point only, and have lost one exhaust, and the three remaining exhausts are normal, it may be a broken bridge, or a crack, or a sand hole in the bridge. Notice the cylinder cocks before you stop, and see if steam appears at only one cylinder cock when the piston is at one end of the cylinder, and at both cocks when at the other end. If so, it is a very good indication of a broken bridge, but examine your eccentrics as soon as you stop. When an engine has a bad blow when in full gear which disappears when hooked up a few notches, it indicates that the valve travels too far and opens the exhaust port to direct steam chest pressure. This is sometimes caused by the top arm of the tumbling shaft working loose; perhaps the key is lost. When the exhaust nozzle is gummed up it produces a sort of asthmatic wheeze, or whistle, which is sometimes mistaken for a blow.

When two exhausts are heavy and two very light you may have blown out a nozzle tip, providing you have nozzles. When the dry pipe leaks, the engine will work water through the cylinders, and when standing in the roundhouse it may be discovered by a constant leak at the cylinder cocks. A leaky throttle generally leaks at all times, but is distinguished from a dry pipe leak, because the latter leaks both steam and water, and will show a stream of water at the cylinder cocks when the boiler water level is above the leak in the dry pipe. A leak at the bottom of the exhaust pipe will not cause a blow, but will affect the exhaust.

A blow in a compound is located by the use of the same tests as are employed with a simple engine, except that any blow on the high-pressure side will not be heard when the separate exhaust valve is closed. The pressure on the low-pressure side is increased by a blow on the high-pressure side, resulting in a relief valve pop on the low-pressure side when the compound is worked with full throttle.

Method of Testing.

From the descriptions we have given of different blows, you can usually determine about where the blow may be found and proceed to test that particular part, without giving the engine such a severe test as we have outlined, as this chapter necessarily covers all kinds of blows. We will first test the steam chest, and afterwards the cylinders. It is an easy matter to determine which cylinder contains the blow, but it is sometimes very difficult to locate the particular steam chest it is in, so follow the instructions closely.

Place each rocker arm alternately in a vertical position, block the wheels, open the cylinder cocks and give the engine a little steam. If steam does not appear at either cylinder cock you may depend upon the fact that the valve seats are tight. If your engine has balanced valves test the valve strips, rings or riders. A blow of this kind is sometimes very difficult to locate, but it can be done, viz.: If your engine has drain cocks screwed into the exhaust port, go under the cylinders and open the cocks and have the fireman give her a little steam; if steam appears at either cock that is the side your blow is on. Another method is to open the front end if you have a double nozzle, and you can see which side blows; if a single nozzle, climb up on the boiler and feel the draft on each side of the stack with a broom, or a lighted torch if at night. You can usually notice a difference in the draft. On whichever side the draft is the strongest the blow will be in the opposite chest. Or, you may put a little fresh coal in the fire box and watch its action on the smoke. This kind of a blow can sometimes be located by the increased friction, which will cause the valve stem to jerk when in motion, or it may be discovered by placing the crank pins on center alternately and handling the reverse lever under steam pressure; the blow will be on the side that handles the hardest while the pin is on the quarter (not the center). By placing the engine on the quarter you will only move the valve which is to be tested; the opposite side will then be on the dead center and its valve will not move with the Walschaert gear, and but very little with the Stephenson gear. Now let us return to the valve seat.

If steam appears at both cylinder cocks on one side while the steam ports are covered, it is evident that the valve seat on that side leaks, providing the opposite side is tight; the leak may be in the valve seat or beneath the false seat, or, if the valve has exhaust clearance, it may be a flaw in the valve itself.

If, however, steam appears at only one cylinder cock, on only one side of the engine, while the ports are covered, it may be a sand hole between the supply port and the steam port, but it is more probably a false seat loose on one side. If steam appears at the forward cylinder cock, the forward end of the false seat is loose, and if at the back cylinder cock, the back end is loose. If steam appears at both cylinder cocks on both sides it is evident that the valves on both sides blow.

A blow, with the engine in a certain position, is often noticed, and is usually due to a defective, or a stuck ring. The ring, however, is readily located. So as to obtain full valve travel, place the engine on right top quarter. The blow will not be noticed when the reverse lever is in the extreme corner, if the defective ring be on that side. But when the lever reaches about half the quadrant, the blow will occur, if the defective ring is on the back of the right valve. Test the ring on the front end of the valve in the same manner; place the reverse lever in the back corner, and gradually pull it toward the center until the admission ring is over the port. As the port is much wider than the ring, the defective exhaust ring allows an escape of steam into the exhaust passage.

We will now proceed to test the cylinder packing, placing each main pin on either quarter alternately, and, with the reverse lever in the forward notch, giving the engine a little steam, first setting the brakes. If steam escapes at only one cylinder cock, the cylinder packing on that side is all right, but before leaving it place the reverse lever in the back notch and try it there. If the above movements are executed with cylinder cocks closed, a defective cylinder packing will be manifested by a strong blow through the stack, which will cease as the reverse lever is pulled toward the center of the rack, and the valve moved to the center of its seat. Now, if steam appears at both cylinder cocks when one port is open, and at only one cock when the other port is open, it indicates a broken bridge (although a broken valve strip or ring might cause this, as well as a sand hole in the bridge below the valve seat). The particular bridge broken may be determined by noticing which port is open when steam shows at both cylinder cocks; if the forward port is open then it is the forward bridge, and vice versa. A broken bridge can usually be distinguished from a crack, or a sand hole, by the increased force of the blow. If steam appears in great volume at both cylinder cocks when the lever is in both motions, it is then impossible to say whether it is a broken valve seat or broken cylinder packing rings, so have the cylinder head removed first. If it is found to be all right, you know that it is the valve seat.

With reverse lever in the corner, when the engine is on the right top quarter, the throttle open and brake set, a blow at both cylinder cocks on the left side is no indication of a blowing valve or cylinder packing on that side. A blow, with engine on right top quarter, may be located as being on that side by pulling the reverse lever to the center of the quadrant, with a slightly opened throttle. For, when the lever reaches the center of the quadrant, if the blow stops, the cylinder packing on that side is probably at fault. However, if the blow does not stop, the valve should be tried by lifting the cylinder cocks with a stick, and thus ascertaining how much steam is escaping, with the valve central. In this manner one should be able to tell just where the blow is, almost as readily as if the defective rings could be seen, and much time is saved in the roundhouse.

Now let us assume that the right cylinder packing is to be tested. The left engine, in this position, is taking steam in the back end of the cylinder. If the rings are down, steam will blow past the defective ring and out the exhaust. Steam will back through the right exhaust passage, and in through the open exhaust port to the front end of the right cylinder, due to the fact that the nozzle is smaller than the exhaust passages to the right side. This will show at the cylinder cock as though the packing on that side were blowing.

Under the above conditions, with the reverse lever on the center, a blow at both cylinder cocks on that side is noticed. Steam backing through the exhaust passages causes this, and, with the valve central, there is an opening into both ends of the right cylinder, due to the exhaust clearance.

If a valve admission ring be broken, or stuck, there will be a blow at the stack, and also at the cylinder cock on the end having the defective ring. A broken cylinder packing ring can usually be distinguished from one that simply leaks by the volume of the blow. A packing ring that leaks will also show steam at both cylinder cocks when in both motions, but it will not be such a heavy blow as a broken ring will produce. Almost every engineer has had some experience with packing rings which simply blow, and can be distinguished from anything extraordinary, such as a broken packing ring or bridge. When you raise the steam chest cover, first examine the rings, or valve strips and springs. See that the valve strips do not fit too tightly on the ends, for the long strips expand 1/32" more than the valve and often cramp the short strips. Next examine the valve seat and face, then the bottom joint of a false seat, and the pressure plate, and face off all joints that need trueing up. Now examine the valve carefully for sand holes. If you cannot locate the blow elsewhere, fill the supply ports with water (one at a time), open the cylinder cocks, and ascertain if water leaks;

if it does there is certainly a sand hole or crack; if it does not leak, fill the cylinder and steam ports and see if the water leaks into the exhaust cavity. Open the drain cock at the bottom of the exhaust cavity.

With piston valves there is this difference: An engine may not blow in a standing test, but will when working, if the blow is due to defective exhaust rings or bushing; and the blow will take place at the point where engine sounds lame, which will help locate the defective ring.

The test for a broken packing ring in an inside admission piston valve is very much similar to that of a slide valve. However, because of the fact that in a piston valve the rings form the steam and exhaust edges of the valve, it is possible to ascertain which ring is defective by placing the engine on quarter on the side to be tested, with the reverse lever in the center of the quadrant, opening the cylinder cocks, and admitting steam. A blow through the exhaust, at the end where steam will show at the cylinder cock, shows that both rings are defective. If you get no blow, with the valve central, move the lever so as to allow the steam ring on the valve to open slightly the admission port—a blow through to the exhaust now will, of course, indicate a defective exhaust ring at the end of the valve where the steam port is open, and where steam is showing at the cylinder cock. But if you get no blow, by means of the reverse lever move the valve so that the exhaust rings will uncover the exhaust edge of the admission port—a blow through to the exhaust now indicates a defective steam ring at that end of the valve. By testing both ends of the valve in this manner, the defective rings are readily located. Usually, the defective steam rings are located when the valve is placed central on its seat, and steam is admitted to the steam chest, for, with open cylinder cocks, steam will show at the end of the cylinder when the ring is defective, and the test given is merely a proof.

Blows in Compound Locomotives.

The following method is used in testing for blows in high and low-pressure cylinder packing, for each type of compound.

If a cross-compound engine, simple it, and then make the same test as employed for a simple engine. With the Vauclain four-cylinder compounds, however, test the low-pressure first. A blow past the low-pressure piston will show the same as on a simple engine. However, if she blows past the high-pressure piston, it will make the engine stronger on that side, when working a full throttle, and the low-pressure exhaust will be much heavier. In order to test the valve on either side, cover the ports. If packing rings in the steam valve be broken, they will blow in one position and be tight in another.

To test the high-pressure packing of a tandem-compound, (containing four cylinders, the high-pressure being ahead of the low-pressure on each side, and both pistons connected to a single piston rod and crosshead), place the engine on the top quarter, lever in back gear, with the starting valve closed and the wheels blocked. Remove the rear indicator plug, or open the back cylinder cock of the high-pressure cylinder, so that steam coming from the back cylinder cock must pass the by-pass or starting valve, or blow past the piston packing. Then try the other indicator plug, or cylinder cock, by placing the reverse lever ahead. No steam will come through if the trouble is a leaky front end by-pass valve.

To test the low-pressure piston packing, the engine should be placed in the same position, with the lever in such a position as to admit steam into the front end of the high-pressure cylinder.

Open the starting valve, remove low-pressure cylinder back indicator plug, and give the engine steam. If steam comes from the open back cylinder cock, or the indicator plug opening, either the packing leaks or there is a leak in the by-pass valve. In order to determine which of the two leaks, close the back indicator plug and open the forward one. Now, if the blow still continues it is either leaking packing rings, or a leak past *both* by-pass valves. To make sure, it is then necessary to inspect the by-pass valves.

In order to locate a blow through the sleeve packing between the high and low-pressure cylinders of a tandem-compound, place the engine on top quarter, as in the previous test. Put the reverse lever in forward gear, with starting valve closed, block the drivers (or set the brakes) and open the throttle. No steam should get into the front of the low-pressure cylinder now until the engine moves. If steam does get into the front side of the cylinder, it indicates that there is a leak.

In carrying out this test, the indicator plug in the front end of the low-pressure cylinder should be removed.

In the case of a Baldwin balanced-compound, the test of the high-pressure piston packing is accomplished as follows:

The engine is placed with the outside main pin on the side to be tested on bottom quarter, the starting valve closed, the reverse lever in forward gear, and the brakes set (or drivers blocked). The indicator plug in the front end of either the high or low-pressure cylinder should be removed. Now, when the throttle is opened, steam will be admitted to the back end of the high-pressure cylinder, and any steam coming out of the plug opening will indicate a leak past the piston or the high-pressure valve. Test the high-pressure valve by moving the reverse lever to the center notch. With both ports covered, if the valve is tight the blow will stop, which should indicate that the blow was past the piston. To test the low-pressure piston, place the engine in the same position, that is, with wheels blocked, starting valve open, and back indicator plug out. Open the throttle, and leaky packing will be shown by steam issuing from the plug opening. If still uncertain, it is a simple matter to test the valve. Bring the reverse lever to the center of the quadrant, and the blow will stop if the valve is tight, for the port is covered.

As an aid in locating blows in a compound, a blow past the high-pressure cylinder packing in any compound always tends to increase the pressure in the low-pressure cylinders. And a blow past the low-pressure packing is invariably heard at the exhaust, usually on both forward and back stroke. Also, a blow past the by-pass valves, or valve bushings, occurs only at a certain place in a complete revolution.

By-Pass Valve.

In order to locate a blow in a by-pass valve on a Mallet, place the engine on either top or bottom quarter on the side to be tested. Place the reverse lever in full gear either forward or backward, and admit steam to one end of the low pressure cylinder. To do this, with Baldwin type locomotive, open the starting valve; with American type, open the main valve. If the by-pass valve at that end be either stuck or broken, steam will be allowed to pass through the by-pass valve into the other end of the cylinder. This will cause a blow through the exhaust port and out the stack, similar to that caused by defective cylinder packing or a broken valve seat.

This blow may also be located, while running, by occurring between exhausts. That is, if but one by-pass valve be defective, there will be three normal exhausts at the stack, and one blow.

But if both by-pass valves on one side be stuck, or broken, there will be a continuous blow at the stack.

In this case the defective valve may be located by the standing test just given. That is, place the engine in the position at which the blow occurs, and move the reverse lever in the opposite corner. If the blow ceases, the defective valve is at that end of the cylinder which was in communication with steam chest pressure through the ports, when the valve was in its first position. For instance, the test is made for the by-pass valve at the right low pressure cylinder, and the engine is standing with main crank pin on upper quarter. If the blow occurs with the reverse lever in the forward corner, it is evident that the defective by-pass valve is on that side, in back.

After locating the defective valve, take off the cap and remove the valve. If it be simply stuck, oil it thoroughly (with head-light oil) and replace it. If it be broken in such a manner that it will still seat if blocked down, block it in place, holding it with the valve chamber cap. However, if it be broken so that it will not seat, place a blind gasket between the by-pass valve chamber and the port communicating with the steam port. Or, if certain types of by-pass valve be used, a blind gasket may be slipped into the connecting steam pipe.

Cause of Valves Sounding "Out of Square."

Sometimes an engine's valves will sound "out of square" when the valve gear is perfectly adjusted. The following are some of the causes which may produce such an effect:

Driving wheels improperly quartered.

A main driving axle bent.

A patch inside of the steam ports, or ports of different size.

Cylinders of different size (not compound engines).

Eccentrics of different throw.

Links of different radius.

A hole in the petticoat pipe, or stack.

A leak at the exhaust pipe joint.

A valve-yoke cracked on one side.

Cylinders working loose on the frames.

Repairing Cylinder Packing.

Engineers are not expected to do this kind of work and should only do it in case of emergency when out on the road. If your packing blows, report it without delay. When it becomes necessary to overhaul the cylinder packing, and if steam is up, too much attention cannot be given the throttle while the cylinder head is off; otherwise you may receive a severe scalding.

First close the throttle and tighten the thumb screw, then open the cylinder cocks. Now move the engine until the piston head is within one or two inches of its extreme forward travel. If you cannot move the engine the main rod must be disconnected, but before doing this scribe a line on the guides, true with the back end of the crosshead, and when recoupled again the crosshead should be true with the same line (the liners sometimes become misplaced). Also, place a wooden block between the guides in front of the crosshead; this is merely for safety—to prevent knocking out a front cylinder head. Now, when you have the rod disconnected, move the crosshead forward to within an inch or two of its extreme forward travel. By moving the reverse lever you can then cover both steam ports on that side, to keep the steam out of the cylinder. Take off the cylinder head and follower, making a mark on the cylinder head, also on the follower, before taking them down, so that you can put them up to the same places. (Sometimes the holes only fit one way.) Be careful to place the cylinder head nuts and follower bolts in such a manner, when you remove them, that you will be able to put them back as you found them. Now, if solid ring packing, peen each packing ring inside until it fits the cylinder nicely; if dunbar packing with springs, spread the springs until they will hold the packing securely against the cylinder. If it be a solid piston head, the piston and head must be removed before you can remove the packing rings. When replacing the packing use a pair of inside calipers; hermaphrodites, or a pointed stick of the correct length will do; and set the piston tightly above the center of the cylinder.

Be sure that the packing bears against the cylinder all around; if it does not the packing should be taken out and the cause ascertained, and the defect remedied, if possible. After the packing is set out, clean off the follower and put it on. See that the heads of the follower-bolts press the follower, and that their ends do not touch bottom, then clean the cylinder head joints, and put the head on. In screwing up follower-bolts and cylinder head nuts, be sure to get them solid. Use proper care, especially in screwing the cylinder head nuts, or you will break the studs. When the head is put on, see that the joints lay close together all around; place the top nut in position, and run it on until it just comes against the head. Then put the bottom nut on in the same manner, and a nut on each side, and draw them, watching that the joints are together all around; then put on all the other nuts and draw them equally all around the head. You will sometimes find, after removing the follower, that the packing is not slack, although it seemed so before the follower was taken off. This shows that the packing was clamped and held by the follower, and that the packing is too long—that it was follower-bound. This may be remedied by placing a piece of wrapping paper between the follower and spider. Packing does not need setting if it blows only a little in starting—you rob the engine of its power by having it too tight. It should not be snug enough to prevent an engine's drifting freely. Packing should not be allowed to run longer than two months without being examined.

It is difficult to distinguish a by-pass valve blow from a valve packing blow on a piston valve, because: if you put the valve on the center, so as to block both ports, and if the valve packing is defective, there will be a blow up the stack. If the by-pass leaks so that steam gets down into one end of the cylinder, there will be a blow from the cylinder cock at that end. If steam were able to leak from one end of the cylinder to the other, there would be a blow from both cylinder cocks, like that from defective piston packing, but if the valve had any inside clearance steam would blow up the stack from one end, or both, of the cylinders, as the case might be, and the blow up the stack would be caused by a defective valve. In that case it would not be easy to decide which it was.

To distinguish between a by-pass valve blow and a broken admission ring, place the valve in the center of its seat, open cylinder cocks and admit steam. If steam appears at the cylinder cock it will indicate that the by-pass valve or admission ring is broken. Now move the main valve enough to cut off the admission of steam to the cylinder. If the steam is not cut off until the exhaust ring of the valve covers the admission bridge, it is an indication that the admission ring is broken, but if steam still appears at the cylinder cock and there is a heavy blow at the stack, it is the by-pass valve which is broken. This test applies to inside admission piston valves. The valve should be moved in the opposite direction when testing outside admission piston valves.

POUNDS.

A pound, as the term is used in connection with the locomotive, is not easily defined, but all engineers recognize it as a very disagreeable jerk, combined with an annoying sound, quite common to the locomotive on the road.

What are generally called pounds are really both knocks and pounds, and the distinction is somewhat indefinite. However, as an aid in distinguishing, a knock is heard, while a pound is both heard and felt. The neglect of a pound or knock, especially if serious, may be the cause of much trouble, even to the disabling of the engine. For this reason, it should be located and reported without delay, so as to relieve the engineer of responsibility, and to reduce the loss of time in the shop.

An experienced engineer can very often locate a pound, or a knock, by its own peculiar sound, but even to an experienced man, these are often misleading. One of the most deceiving knocks is that caused by a loose piston head (or spindle). This knock is apt to start suddenly, and is of such a nature as to lead one to believe that there must be an inch of lost motion somewhere, while in reality the amount may be of the thickness of tissue paper. This knock is often mistaken for that resulting from a loose wedge, driving brass, crosshead, or main rod brass; the noise is that similar to a crosshead being loose on the piston-rod, and occurs when passing both centers.

Causes of Pounding.

There are numerous causes of pounds and knocks in a locomotive, among the more common of which are improperly adjusted wedges, lost motion between the crosshead and guides, and loose or worn driving box brasses.

Other troubles often resulting in a knock or pound are loose pedestal binders, crosshead loose on piston rod (or a piston loose on rod) improper keying of the rod brasses, broken frame, broken or hot driving boxes, and improper length of main rods. A pound with steam shut off might indicate a broken follower bolt, loose spider, flat spot on tire, or compression trouble in the cylinder (this will cause a knock, rather than a pound).

Insufficient or faulty lubrication, also, will often cause a pound.

Location and Remedy of Pounds and Knocks.

With the use of a little common sense, pounds and knocks resulting from the foregoing, or any other, causes may be readily located. Some of the usual methods observed in locating pounds are as follows:

To test for pounds in the driving boxes, wedges or rod brasses, place the main pin on the suspected side on the top quarter. Open the throttle slightly, to give the cylinder a little steam, and reverse the lever under pressure. The pound, if it is on that side, will be readily observed if in the driving boxes, wedges or brasses. If it is not found, do likewise on the other side of the engine. Of course, in this test, the wheels are blocked, and it is necessary to have a man in the cab to operate the lever while another stands alongside the locomotive in order to observe the action of the various parts of the engine, and locate the pound.

If it is found that the axle has too much play in the boxes, or that the trouble is in the driving box or brasses, endeavor to adjust the faulty part at once, and report the repairs which may be needed at the terminal.

In the same manner excessive lost motion between the cross-head and guides may be located. This pertains to the guide at the sides, as well as the upper and lower guides; and the same is true of a piston rod not central between the guides, and these parts should be reported to be lined. The driving box wedges, too, should be reported when they have been set up as far as possible and the boxes are still loose between wedges and shoe.

To locate a pound in the main rod brasses, place the engine on top or bottom quarter on side to be tested, so as to have the pin between two rigid points, where any lost motion will show before the box moves or the wheel slips, set the brakes, admit steam to the cylinder, and work the reverse lever back and forth. Then, by watching the brasses, a pound is readily located.

Rod brasses should be reported to be closed or refitted when they are keyed solidly brass to brass, and are pounding on the pin. They should be reported to be lined when the key has been drawn, or driven to its full length, and the brasses do not close together or are too loose in the strap, lengthwise of the rod. That is, when the key is driven as far as possible and the brasses are working in the strap.

To key the back end of the main rod, place the engine on dead center, so as to key the brasses against the largest part of the crank pin, but to key the front end place engine generally on bottom quarter, so as to key against the largest part of the pin, and because it is easier to get at the screw which holds the front end key.

In locating pounds, the following facts are helpful:

Too long or too short a main rod pounds most when drifting with throttle closed, because the weight of the piston will take up all the slack in the main rod and its connections, and cause the piston to strike the head of the cylinder (front or back, according to whether the rod is too long or too short). To protect the cylinder heads, open throttle, so that steam admitted due to lead will cushion the stroke of the piston and take up its lost motion.

A loose piston on rod, or rod in crosshead, pounds hardest when working steam, as the pins pass the centers. It might be said with regard to the standing test given for locating a loose piston head, that as the lever is worked back and forth in the quadrant, the pound will be most noticeable when the lever reaches the front quarter, and will be lighter as it reaches the back quarter. When running, under steam, the opposite is true.

A loose cylinder bushing may be located only while engine is in motion and working steam, and the pound occurs at each end of the stroke, just before the pin passes the center, and generally before it reaches the eighth. This is because the packing rings are expanded by the steam against the walls of the bushing, and the friction created moves the bushing until it strikes the cylinder head.

Pounds When Drifting.

A pound which is noticeable when drifting may result from a loose follower bolt or head, the main rod keyed too long, or a flat spot on a wheel. Also, too much or too little compression will cause a knock, while running with steam shut off, but it is more likely that this cause is too great, rather than too little, compression, since all the air drawn in through the relief valves can not escape through the exhaust, and is compressed. This will cause a very high degree of compression, for air is much less compressible than in steam. However, on modern locomotives, the by-pass valve will relieve this excess pressure, and should obviate pounds due to it.

The loose follower bolts may be located by shutting off steam and allowing the engine to drift. The pound in the cylinder will occur when the loose bolt strikes the forward cylinder head as the engine passes the forward center on that side. Give the engine steam, and if the pound stops you have probably located a loose follower bolt on that side. For, when working steam, the compression, or preadmission, takes up the lost motion in the rod and connections, so that the loose bolt does not strike the cylinder head, but when steam is shut off, the piston travels the extra distance of this lost motion and the head is struck by the bolt, causing a pound.

LOCOMOTIVE POWER REVERSE GEARS.

Now that a power reverse gear of some sort has become a recognized necessity, not only on heavy locomotives, but on lighter power, including switching engines, it will perhaps be of interest to inquire into the reasons for its general adoption, while tracing the history of its development.

Several times in the past twenty years, sporadic attempts have been made to lighten the labor of the engineer by providing power operated means for handling the valve gear.

One of the earliest of these was developed by the Pennsylvania Railroad, and applied to a class of eight-wheel passenger locomotives. This apparatus consisted of a steam operating cylinder, and an oil cylinder which acted as a damper, and was also supposed to provide a positive lock. The device was not a complete failure, although much trouble was experienced in retaining the oil, and preventing movement of the gear through leakage past the locking piston.

Since no actual necessity for a power reverse could be shown at that time, this device was not developed further, and was soon abandoned.

John Player, of the Brooks Locomotive Works, saw, about this time, that some mechanical means would soon be required for reversing and positioning the valve gear, owing to the rapid increase in the size and power of locomotives. Foreseeing the difficulty of inducing the railroads, at the moment, to abandon the old "Johnson Bar," he attached a small four-way cock to the handle of the ordinary lever. This admitted air or steam to either end of a "booster" cylinder attached to the reverse shaft. By operating the latch and four-way cock in conjunction, the operation of reversing and "hooking-up" could be accomplished with very little manual labor.

However, the era of the power reverse had not yet arrived, and this device also went into the scrap heap, to be dug up periodically as a brand new idea by some one or other.

With the advent of the Mallet, "something had to be done." Not enough men could find standing room around a lever to reverse a heavy Mallet with stiff new valve gear.

This necessity brought out the original, "Mellin" reverse, which followed the Pennsylvania design previously mentioned, in that it had an air or steam cylinder in conjunction with an oil cylinder—in this case the oil cylinder functioning as a dash pot only. A novel arrangement of the reverse lever is the principal feature of this arrangement, which was the first successful power reverse used in this country. It is, however, a positive locked gear, and, for that reason, was gradually replaced by gears of the "cushioning" type, for reasons which will be made clear later on.

The original "Ragonnet" reverse gear, the invention of Mr. E. L. Ragonnet, was the answer of the Baldwin Locomotive Works to the demand for a power reverse for Mallet locomotives. Mr. Ragonnet adapted the old Brown floating lever principle, long used in connection with marine reverse gears, to locomotive service conditions.

The first Ragonnet gear was provided with crossed ports and a piston valve, details which for various reasons proved unsuccessful. It has, however, survived its "childhood diseases" due to the correctness of its main principle, that of the elastic cushion, as opposed to the positive lock.

The power reverse, having been developed to a successful state for Mallets, soon found a broader field due to the continued and rapid increase in size of simple locomotives. The point had been passed when an engineer could safely venture to adjust cut-off to suit road conditions while running. A vast amount of fuel was being wasted for this reason, as the only recourse was to throttling.

The screw reverse, widely hailed some years ago, proved abortive for the reason that it formed a positive and direct lock to the valve gear without the cushioning effect due to the springing of the forked end reverse lever, which had been in use for some time, necessitated by the wide fire box. Screw threads and nuts wore out rapidly, even when made of excessive size. Ball or roller thrust bearings did not remedy this trouble. Studs in the bolting-pads leaked and screw gears were not infrequently torn from their fastenings. However, the American Locomotive Company is now experimenting with a screw reverse employing a spring cushioning device, which, if successful, might serve to make this type popular once more.

In the early days of the Ragonnet reverse, considerable missionary work was necessary to overcome the almost universal prejudice against a device without a positive lock. The earliest forms, having small cylinders and a rather large cushioning range, were also subject to "creeping," due to leakage past valves or piston packing. These defects were due partly to improper design of these details and partly to lack of maintenance.

But now, both the Ragonnet, and the Alco gears, those most extensively used, are of the "cushioning" type.

The Ragonnet Gear.

The Ragonnet reverse gear, type "B," as illustrated in Fig. 164, is of the "floating lever" type, in which the valve is actuated by a rocker, on the outer arm of which hangs the floating lever. The upper end of the floating lever is connected to the reverse lever, while the lower end is connected to and moves with the crosshead.

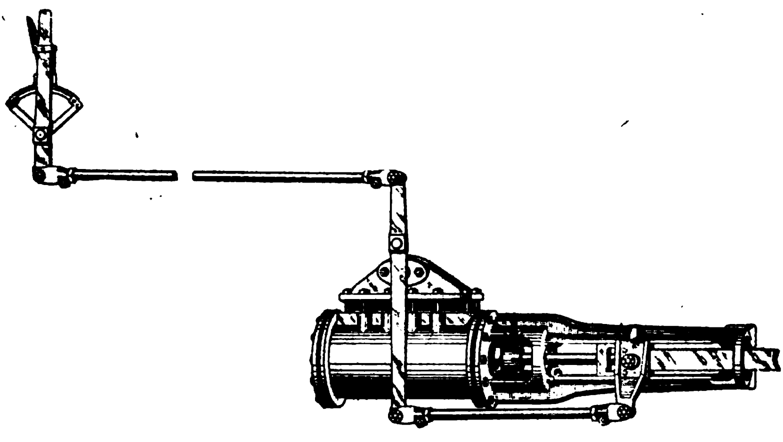


FIG. 164.

When the handle of the reverse lever is moved forward or to the right, for instance, the lower pin of the floating lever acts as a fulcrum and the valve is displaced toward the right. The back port being thus opened to pressure and the front port to exhaust, the piston commences its forward movement. If the lever is then latched, the top pin of the floating lever is thus anchored, and in turn becomes a fulcrum point. The continued movement of the piston then causes the valve to move to the left and thus again closes the ports.

Now, with the lever latched in this position, we will assume that the pull of the valve gear tends to drag the crosshead forward. A slight forward movement causes the valve to be still further displaced toward the left, and opens first the admission port to the front end of the cylinder, increasing the pressure on this side of the piston. If the difference of pressure is still insufficient to hold the valve gear, a further movement exhausts all air from the back side of the piston. A movement of 3/16 of an inch from true central position of the crosshead will thus instantly bring to bear the full force of the reverse cylinder which, with 120 pounds reservoir pressure, equals 10,200 pounds —amply sufficient to hold the largest engine with dry valves.

On account of the exhaust lap being double the outside lap, ordinary stresses are taken care of without releasing air from the cylinder. This has led to an erroneous conception of the cushioning type of gear. It is believed by many, through careless explanations sometimes published, that a reverse gear resists the movement of the valve gear by reason of a high balanced pressure on both sides of the piston. Nothing could be further from the fact. The pressure can balance only when there is absolutely no strain on the reverse gear. It will be readily understood that even though the pressure on each side of the piston be infinity, the gear would exert absolutely no holding power. Suppose five men are trying to push an automobile forward, and the same number exerting the same force to push it in the opposite direction. It is easily seen that then another man could start it in either direction he chose, just as readily as though no other force was being exerted upon the machine.

The office of the floating lever is to effect an automatic variation of pressures on opposite sides of the piston sufficient to exactly balance the forces acting to displace the piston.

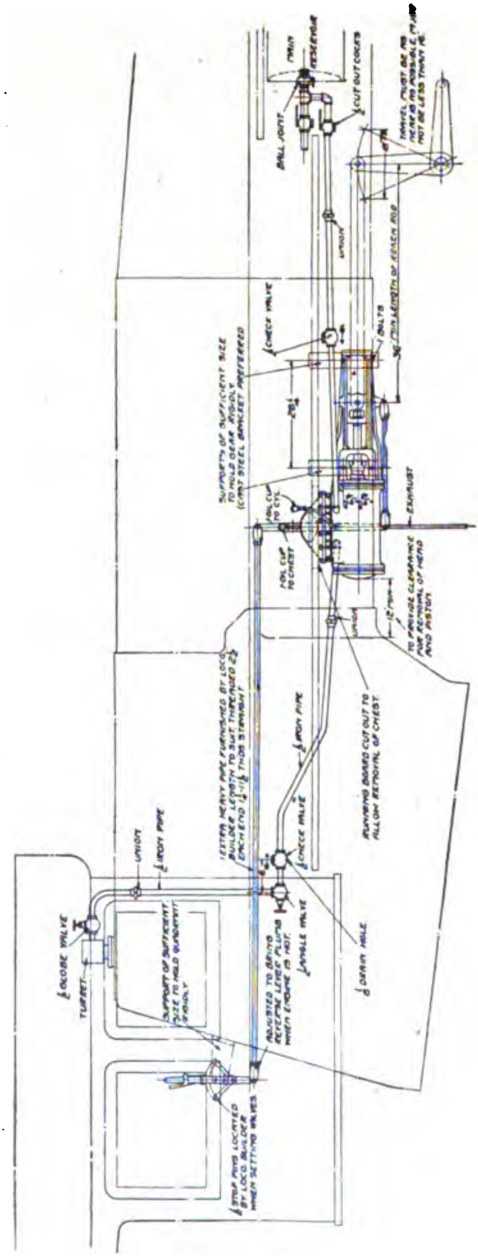

MAIN
RESERVOIR
BALL JOINT
CUT OUT COCKS
UNION
CHECK VALVE
BOLTS
EXHAUST
UNION
IRON PIPE
CHECK VALVE
ANGLE VALVE
DRAIN HOLE
GLOBE VALVE
TURRET
UNION
IRON PIPE

FIG. 165.

The Ragonnet gear is, under normal conditions, operated by air, and is applied under the running board, as shown in Fig. 165. But, in an emergency, the gear may be operated by steam, as is also clearly illustrated.

As may be seen, the cab lever is located conveniently for the engineer, and, as far as possible, the lever arm is in line with the combination lever on the cylinder, thus avoiding the necessity of off-set in the reach rod.

This gear is manufactured by the Franklin Railway Supply Co. of New York City.

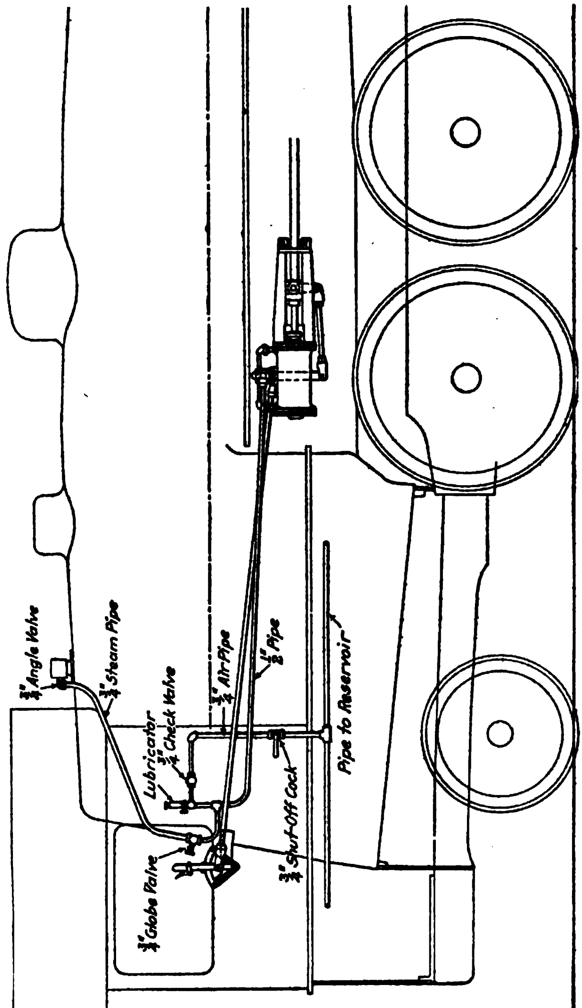
The Alco Gear.

The Alco gear, developed by the American Locomotive Co., for use with either steam or air pressure, is shown, as applied to the locomotive, in Fig. 166 (Type E).

In this gear, as in the Ragonnet, no usual features are presented in the cylinder, piston, piston rod, crosshead, and crosshead guide, which closely follow ordinary locomotive practice.

Referring to Fig. 167, showing the type E gear, it may be seen that the control valve is of the flat rotating type, bearing on a corresponding flat face, which obviates the necessity of a stuffing box for the valve stem, and makes it uniformly easy to operate at all times. The valve and its seat have ports arranged so that the cylinder is filled with air or steam, as the case may be, on both sides of the piston, in any stationary position of the latter, causing a quick response to any movement of the hand lever or crosshead.

Motion is given to the operating valve by the control lever in the cab, through a system of linkage anchored at one end to the main crosshead, and at the other end to the operating lever.



3/4" Angle Valve
3/4" Steam Pipe
Lubricator
3/4" Check Valve
3/4" Air Pipe
1/2" Pipe
Pipe to Reservoir
3/4" Shut-Off Cock
3/4" Globe Valve

FIG. 166.

In operation of the gear, movement of the control lever shifts the valve from its central position, which position is taken by the valve when at rest. With valve in its central position, admission is line and line, and exhaust ⅛-inch lap. Admission being line and line permits keeping the pressure built up on both sides of the piston. A ⅛-inch movement of the valve opens exhaust from one side of the piston, causing the piston and crosshead to immediately move out or in, as the case may be. This move-

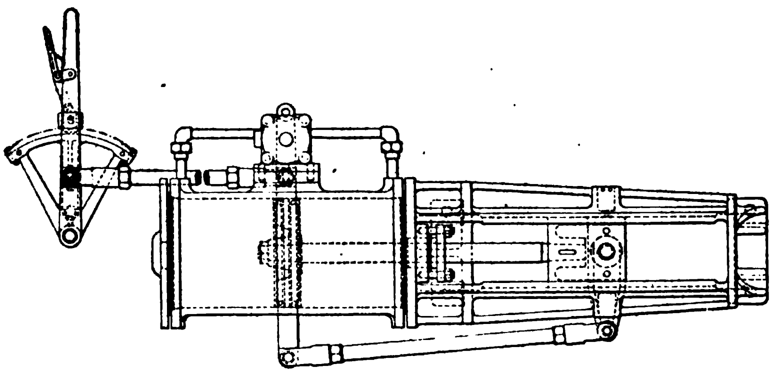


FIG. 167.

ment of the crosshead returns the valve, by means of the linkage, to its central position, thereby closing the exhaust from the cylinder and causing the piston to come to rest at the desired position corresponding to the position of the control lever on its quadrant.

Type D, shown in Fig. 168, is another Alco reverse gear, employing a device for locking the gear. The locking device consists of a locking cylinder which is located above the operating cylinder, and arranged with its piston under a constant spring load, which tends at all times to compress the main crosshead between the upper and lower guides. The lower guide is hinged at one end of the operating cylinder and connected at the other end to the locking cylinder through a bell crank attached to the upper guide. On raising the operating lever latch, pressure is admitted to the locking cylinder, and the spring load on the crosshead is released, thus permitting the crosshead to move. On releasing the operating lever latch, the process is reversed and the crosshead becomes locked in the desired position.

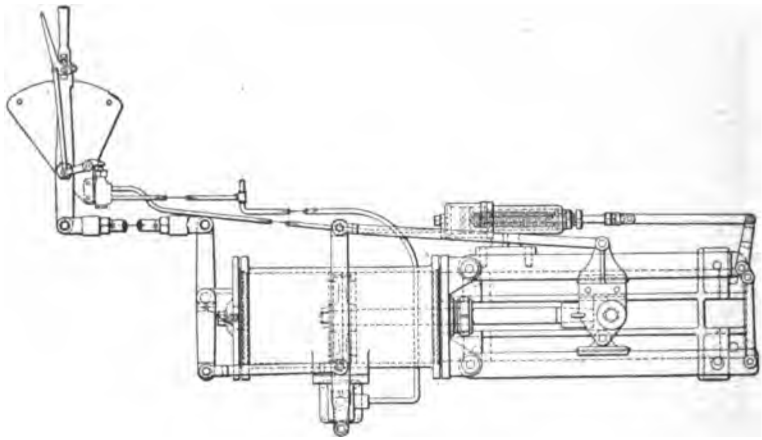


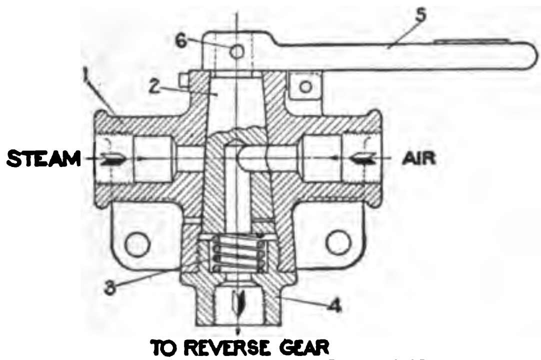
FIG. 168.

Movement of the rotary valve is controlled by stops on the valve body. When the operating lever in the cab is moved sufficiently to bring the valve arm up against a stop, any further movement of the valve and operating lever is delayed until movement of the piston starts the valve toward its central position. These stops control the movement of the operating lever when the gear is without pressure, and any effort toward further movement of the control lever will result in straining the parts, and should be guarded against.

Reverse Gear Air and Steam Throttle Valve.

This appliance, Fig. 169, has been adopted by the United States Railroad Administration for all standardized Government engines for the purpose of supplying the operating air pressure for the power reverse gear.

Under normal conditions air is admitted to the reverse gear, but should anything happen which would interfere with the supply of air pressure, the air connection may be cut off and



No. of Part.	Name of Part.
1	Body.
2	Key.
3	Spring.
4	Cap.
5	Handle.
6	Pin.

FIG. 169.

steam supplied to the reverse gear by throwing the lever No. 5 over to the steam position. This is accomplished by means of a three-way cock operated by handle No. 5, which cock, in the position as illustrated, admits air but shuts off the steam and in the opposite position is reversed and the air is cut off and steam admitted.

This device was developed by the Nathan Manufacturing Co., of New York City.

BOILERS.

The locomotive boiler, including its firebox, consists of such apparatus as to convert water into steam, for the purpose of utilizing the steam's energy in driving the locomotive.

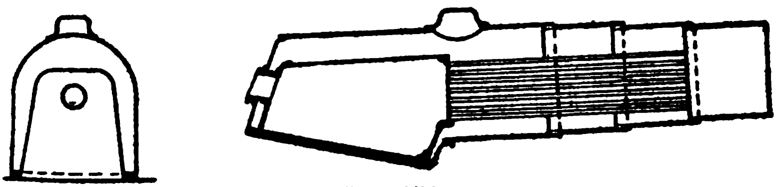


FIG. 170.

Since the era of the early locomotives, which, of course, all employed boilers of some fashion, many improvements in design, construction, and operation have been made, so that now, with good care, the locomotive boiler will give years of service without developing defects of a serious nature.

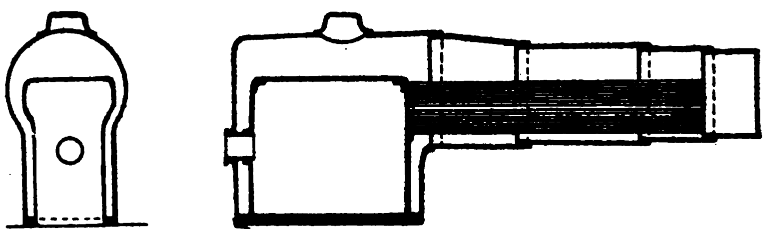


FIG. 171.

On modern power, the designs of boilers used may be classed as:

The Straight Top boiler, shown in Fig. 170.

The Wagon Top boiler, as illustrated in Fig. 171, and

The Extended Wagon Top, which, in Fig. 172 is employed with a round top firebox.

Fig. 173 shows a Wagon Top boiler, similar to that illustrated in Fig. 172, except that it contains what is known as the Belpaire firebox.

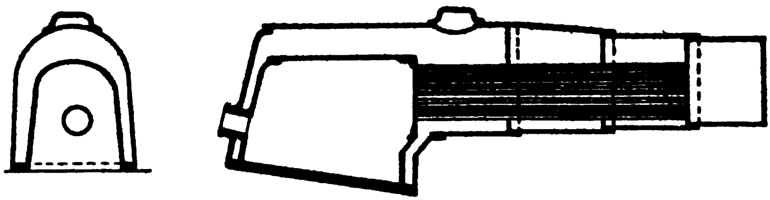


FIG. 172.

The combustion chamber firebox, as illustrated in connection with the Extended Wagon Top boiler, in Fig. 174, is being used considerably on locomotives of large proportions and high tractive force. This type aids in protecting the flues from leakage, and also tends to improve combustion.

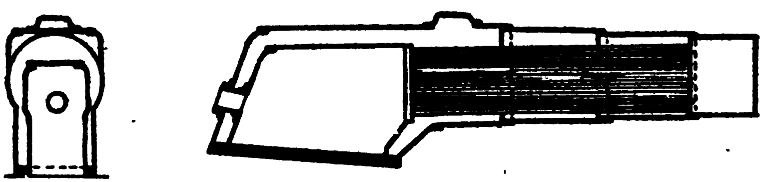


FIG. 173.

These various designs of boilers all consist of a shell, a steam dome, front tube sheet, smoke box and outside firebox, joined with a foundation ring, varying in dimensions according to the water space required, small and large flues (the large ones being used chiefly for carrying superheater units), and staybolts and crown braces.

Shell. The cylindrical shell is constructed in sections, as illustrated, and is enlarged in diameter at the rear to take the firebox. Leading out from the front extension, or smoke box, is the stack. It will be observed that the wagon top boilers have a sloping course, or section of plates, next to the firebox, which is conical, and tapers down in front to the diameter of the main shell in front.

Dome. Behind this, and on top of the boiler shell, is the dome, made of steel and riveted onto the boiler. This dome is used to obtain dry steam for the cylinders. The dry steam in the boiler will naturally rise to the highest point, which is the dome, made of steel, and riveted onto the boiler. This dome is and cylinders. The throttle valve is located in the dome, and,

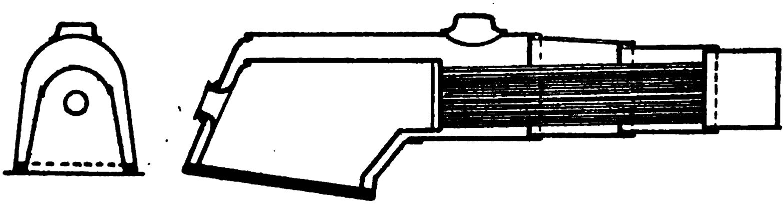


FIG. 174.

also, the safety valve, and other attachments, are usually placed on the dome of the boiler.

Tubes. The tube sheets are the circular sheets which are fastened vertically inside the shell, and into which the flues are made fast by expansion, or by beading. While the front flue sheet is comparatively cool, the rear flue sheet is subject, almost directly, to the most intense heat of the firebox, and is very apt to develop leaks between it and the ends of the flues, due to the distortion of the sheet when subject to the high temperatures. They, therefore, should be subject to frequent inspection, and should be repaired promptly when necessary.

On the smoke box is located the stack, and such apparatus as is employed to create the draft sufficient to give complete combustion in the firebox, are also placed in the smoke box.

This subject is treated at length under the heading "The Smoke Box," page 434.

Firebox. The bottom of the firebox is secured by a heavy foundation ring, commonly known as the mud ring (because much of the impurities of the water are deposited on it), which is shaped to conform to the outlines of the firebox. Both inner and outer sheets are riveted to this ring, leaving a space between

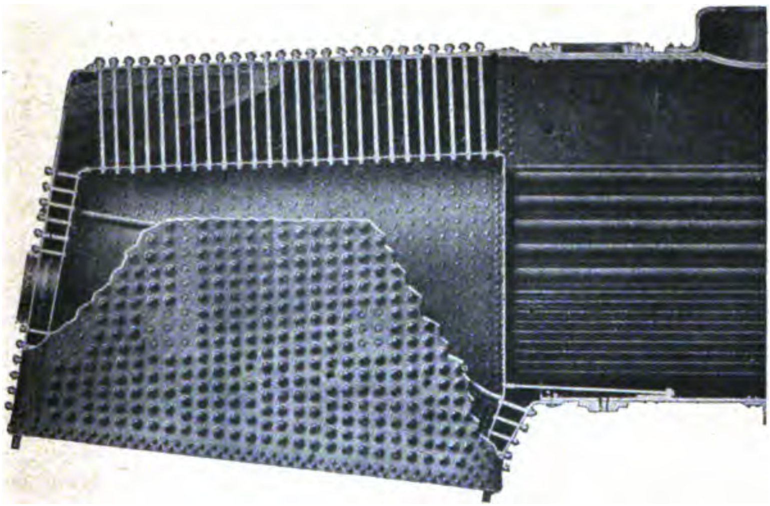


FIG. 175.

them, called the leg of the boiler. This enables the firebox to be entirely surrounded by water, while, below this mud ring are attached the grates, and beneath these the ash pan, of the firebox.

The firebox is also held in place inside the main boiler by means of braces and bolts, called stay-bolts, which are designed for the purpose of allowing some play between the inner and outer walls, to eliminate the distorting effects of unequal expansion. For the inner, or firebox, wall will expand to a greater extent than the boiler shell surrounding it, because it is subject to much higher temperature, and some means must be employed to give a semi-rigid construction in this respect. A very good form of bracing the firebox within the boiler is illustrated in Figs. 175 and 176. In these illustrations, a modern arrangement of flexible bracing by means of Tate flexible stays, is made clear. Under our chapter on Staybolts, page 430, it will be seen that this is accomplished on modern locomotives by the use of a type known as "flexible" staybolts, as shown applied in Figs. 175 and 176.

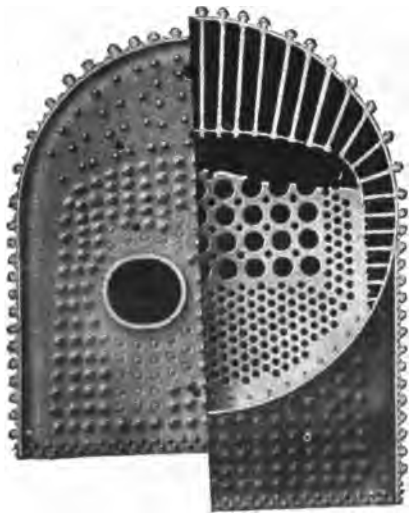


FIG. 176.

In operation, the heated gases of combustion pass out of the firebox, are carried forward through the flues, and, after passing through the smoke box, are ejected at the stack. In this manner, the water in the boiler, brought into contact with the heated outer surfaces of the flues, is raised in temperature until it is converted to steam. The steam thus formed will rise to the top of the boiler, and the cooler water will tend to move downward, thus causing circulation within the boiler.

Incrustation of Boilers.

The principal source of trouble to be coped with in the operation of the locomotive boiler is incrustation. This consists of solid matter—the impurities in the water—being precipitated by the evaporation of the water. Gradually these deposits accumulate, until, after a time, an almost solid scale surrounds both stay-bolts and flues, and the points where the staybolts tap into the inner plates of the firebox. This incrustation of scale prevents water from reaching the area it covers, and, consequently is responsible for overheating of the affected parts, for this scale is composed principally of such carbonates and sulphates as are poor conductors of heat.

The prime requisite of prevention of incrustation is good feed water. In such sections as pure (or nearly pure) water is available, and is used, incrustation is comparatively slow. But, where it is necessary to use water containing considerable quantity of impure ingredients, this formation of scale accumulates very rapidly, and must necessarily be removed frequently, or it will cause undue overheating of the parts, as well as corrosion.

Sometimes, even the space between the flues is calked with scaly matter, shutting off from this space the water, for which the space was primarily intended. Thus, not only is the steaming ability affected, but there arises great danger of collapsing of the flues.

It therefore becomes necessary that boilers should be thoroughly cleaned out, either chemically or mechanically, at such periods as conditions warrant.

STAYBOLTS.

When boiler pressures were increased to the figures of present day practice, it was soon realized that a staybolt that would permit a certain amount of expansion and flexibility was necessary, and the flexible staybolt, embodying all the essential features of the bolt now in common use, was introduced. Difficulties experienced in manufacturing in duplicate quantities, and the general reluctance toward complication, delayed the general use and the further development of the flexible staybolt at that time.

Later, when the so-called "wide firebox" type of locomotive boiler was generally adopted, staybolt maintenance became such a very important item that attention was again given to the flexible staybolt.

The recognized necessity for this device, and the improvements in manufacturing facilities, soon brought out several types which are now in general use.

Advantages.

In general, the advantages of the flexible staybolt are as follows:

1. It permits, with practically no bending stress in the staybolt itself, a slight lateral relative movement of the two stayed surfaces.

2. It permits a longitudinal movement of the staybolt, allowing the head to be lifted from its seat, thus providing for excessive local expansion such as occurs over tube sheets, etc.

Types.

The above mentioned advantages have brought out two general types:

1. One arranged with detachable caps on the outside of the boiler shell and
2. One without caps but arranged with internal flexibility and screwed directly into the boiler shell.

FIG. 177.

The type in most general use permits the required lateral movement by means of a special seat under the staybolt head, and vertical motion is provided for in the clearance between the top of the staybolt head and the under side of the sleeve cap. Inspection is made by the removal of the sleeve cap only.

The Alco flexible staybolt, shown in Fig. 177, is of this type.

It is the practice of the American Locomotive Company to apply flexible staybolts as "expansion stays" over the back tube sheet whenever the size of the boiler is sufficient to warrant their use, the number of bolts depending on the size of the boiler. They also apply them on boilers having internal combustion chambers, arranging for about six horizontal rows on each side on about the center line of the boiler. This application is shown in Fig. 178.

FIG. 178.

This type of flexible staybolt, employing the sleeve, is also handled by the Flannery Bolt Company, of Pittsburgh, Pa., know as the Tate staybolt, and is illustrated in Fig. 179.

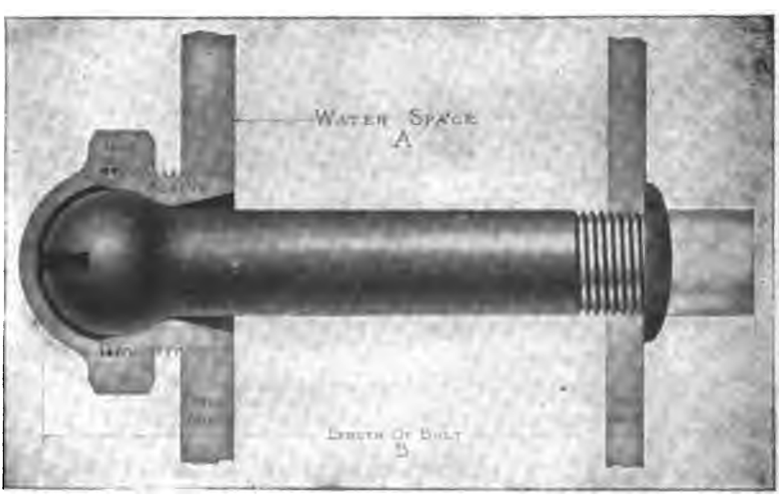


FIG. 179.

The latest form of staybolt connection, as manufactured by the Flannery Bolt Company, is the F. B. C. assemblage, shown in Fig. 180.

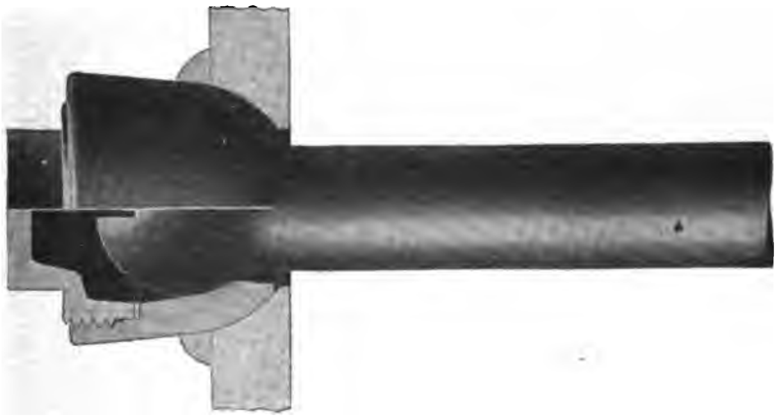


FIG. 180.

This assemblage is unusual in that the outer sheet connection is made secure by an electric arc weld, which, it is claimed, adds much to the strength of the connection, as well as elimin-

FIG. 181.

ating all leakage due to improper thread fits, and permitting the use of smaller holes in the shell plates.

A type of flexible staybolt, which does not employ the sleeve, is manufactured by the United States Chain & Forging Co., of

Pittsburgh, Pa., and is shown in Fig. 181. This bolt is of the flush head type, presenting an unobstructed surface. The method of securing flexibility of the bolt may be clearly understood by reference to the illustration.

The staybolt manufactured by the American Flexible Bolt Co., of New York, obtains its flexibility from the fact that it is, as the illustration shows, Fig. 182, of the "twisted" type. That

FIG. 182.

is, the two halves of the body section, being twisted one-third of a turn, give flexibility in every direction, while the solid ends provide the same holding power as a solid bolt. This bolt is manufactured by first slotting the body, rounding the sharp edge of the slot, and twisting the entire body of the bolt.

The Smoke Box.

That portion of the boiler ahead of the front flue sheet, and into which the flues do not project, is known as the smoke box of the locomotive. See Fig. 183.

The smoke box contains the stack 1 in Fig. 183, for the purpose of carrying to the atmosphere the gases of combustion, and also contains the various appliances used to create a proper and sufficient draft on the fire.

The method of employing the exhaust steam from the locomotive cylinders to act upon the fire, indirectly, and to give it the necessary draft, is, in general, a universal one. This exhaust steam is carried by a pipe from each cylinder to a contrivance called the exhaust stand, indicated by 2 in Fig. 183.

This stand is surmounted by the exhaust nozzle, 3, a passageway of reduced diameter, through which the steam is caused to pass, before being exhausted from the stack. As the steam is released from the exhaust nozzle, it has a very high velocity, due to the pressure built up behind it by the exhaust stroke of

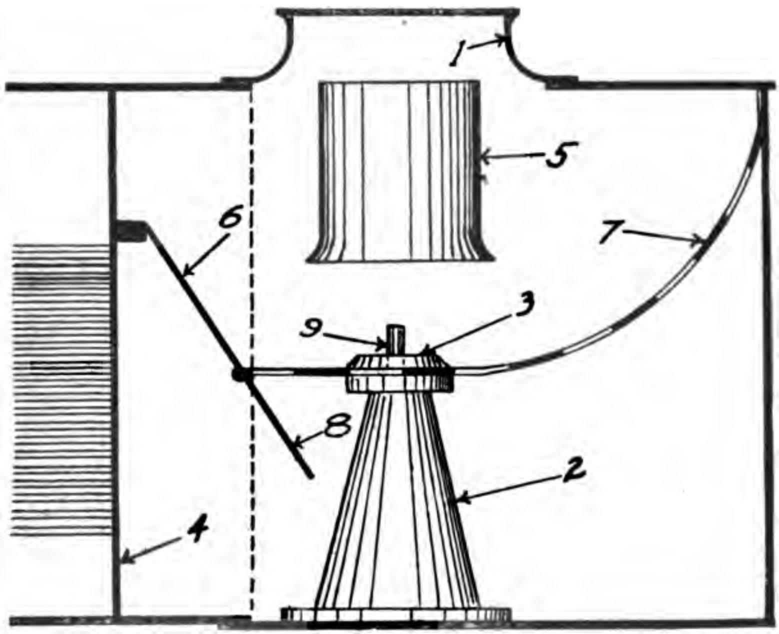


FIG. 183.

the piston, and due also to the fact that its volume is decreased (it is choked) in passing through the nozzle. This steam must now traverse an open space in the smoke box, directly ahead of the ends of the flues in the flue sheet, 4, before entering the stack extension, or petticoat pipe, as the case may be. It might be said here that there is a growing tendency toward the use of the inside stack extension (5 in Fig. 183), thus doing away with the petticoat pipe so widely used some years ago.

This rapid exhaust of steam through the stack, and its extension, by way of the exhaust nozzle, causes a tendency to create an almost continuous vacuum to act upon the fire, through the flues, sucking air up through the fire grates, into the flues, and, consequently out the stack.

In this way the gases of combustion are made to heat the water surrounding the flues, before being exhausted. Of course, these gases carry with them sparks, and cinders, which are caused to be deposited in the smoke box, as we shall explain, rather than passing out of the stack, and, perhaps, causing damage by fire along the right of way.

From the top of the front flue sheet, above the flues, a deflector plate, or diaphragm, 6, in Fig. 183, extends diagonally downward toward the exhaust nozzle. The lower end of the deflector attaches to a horizontal netting of steel, 7, which extends just below the exhaust nozzle, and then upward diagonally to the upper front end of the smoke box, and prevents the larger sparks from passing out of the stack. The purpose of the diaphragm is to deflect the gases downward, and to deposit the sparks and cinders on the floor of the smoke box. The diaphragm has attached to it, just below its connection with the netting, a movable section, 8, which may be raised or lowered, to equalize the draft on the fire.

For the purpose of supplying sufficient draft to the fire when the throttle is closed, and there is no exhaust of steam through the nozzle, a blower pipe is used. This pipe, 9, is set in the smoke box, above the netting, and at the side of the exhaust nozzle. By means of a valve in the cab, live steam from the boiler may be blown through this pipe, and out the stack, and has the same effect as the steam exhausted through the nozzle—it causes a draft through the flues, and on the fire. Intelligent use of the blower will eliminate much black smoke, especially when standing still, as in terminals, where this smoke would be very undesirable. This smoke elimination is due to the fact that the fire requires a draft to complete combustion, otherwise the unburned combustible products would pass out of the stack in the form of black smoke.

Alignment of the Exhaust Nozzle and Stack.

Under this heading, it might be well to first state that the shape and size of the exhaust nozzle opening are very important.

The most common forms of these openings are the round and the square hole. The latter, in the adjustable type, is usually made so that, when the size of opening is increased, the hole is elongated into a rectangular shape. A common form of adjustable nozzle is known as the "waffle iron" type.

Then, too, there is the "dumb bell" opening, the shape of which corresponds to the outline of a longitudinal section through the center of the ordinary dumb bell.

The size of the opening, however, is most important, as a very slight change in nozzle area has a pronounced effect on the draft, and, therefore, on the steaming ability of the locomotive. Much has been said on this subject, regarding the proper size of opening, and conditions vary so that it is inadvisable to endeavor to state just what opening will give the best results. A too large opening will not produce sufficient draft, and too small an opening will pull holes in the fire due to too strong a draft, and will also cause back pressure in the cylinders. In this very important connection, it should be remembered that there is only one correct size of opening under given conditions (unless the opening is variable) and it should be endeavored to have an opening of this size, and no other.

Remember, also, that in decreasing the otherwise correct area of nozzle to eliminate the bad effects of leaks in the smoke box, which tend to destroy the vacuum and reduce the draft, you are merely correcting one error by introducing another. This evil should be corrected within itself—the leaks should be repaired. They may be readily located, when the engine is standing, by holding a lighted torch to all seams and joints in the front end, at the same time using the blower to create a draft. Of course, if there be a leak, the flame from the torch will be drawn toward the leaking seam, or joint.

Regarding the adjustment of the draft appliance, the exhaust pipe should throw the steam exactly to the center of the stack; that is, pipe and stack should be absolutely in line, and central with reference to each other. Also, the exhaust pipe must be perfectly plumb when the engine is level. For, if the jet of steam from the nozzle does not strike on all sides of the stack, but leaves a portion of the stack untouched, it has a very bad effect on the fire. This error is readily apparent, for the condensation of steam in passing upward from the nozzle will show on one side of the stack, or its inside extension; or, sometimes, even the outside of the stack on the side toward which the exhaust steam is thrown, will be bulged.

The nozzle and the stack, as well as the petticoat pipe, or inside stack extension, should be so adjusted with reference to each other that the steam will not only pass out centrally, but so that the jet of exhaust steam will completely fill the stack—otherwise a perfect vacuum can not exist, and the draft will be affected.

COMBUSTION (FIRING).

While it is not essential, it is certainly a great advantage for the engineman to have a working knowledge, at least, of the general principles of combustion; especially as regards locomotive firing.

Coal is a substance made up of carbon, hydrogen, nitrogen, oxygen, sulphur and ash; about 75 to 80 per cent carbon, 5 per cent of hydrogen, and the remaining 15 to 20 per cent of the other ingredients, which are waste, being non-combustible, or non-burning. The amount of carbon per cent, which is the chief element in all fuel, varies with the quality of coal—the greater the carbon percentage, the better the grade of coal.

The carbon in the coal is made to unite with oxygen taken from the atmosphere, causing combustion, or burning. This is nothing more or less than the rapid chemical combination of the carbon in the coal and the oxygen in the air, producing heat.

As the atmosphere is not composed entirely of oxygen (in fact, oxygen constitutes but one-fifth of the atmosphere, by volume, and little more by weight), each pound of oxygen requires over four pounds (4.32) of air; or, in four pounds of air is contained one pound of oxygen. Consider, now, that each pound of coal burned requires from 1.6 to 3 pounds of oxygen. This would mean, therefore, from seven to thirteen pounds of air; but, as there is some oxygen contained in the coal, a little less is needed from the atmosphere. The amount of air required, therefore, to burn one pound of coal, varies from 5 to 12 pounds, depending upon the grade of coal used. These figures are merely approximations.

Now, in firing, it is natural that, to supply sufficient oxygen, air in excess of the amount actually required will be supplied, as it is not to be assumed that just enough air can be supplied to furnish the exact amount of oxygen necessary. But, with

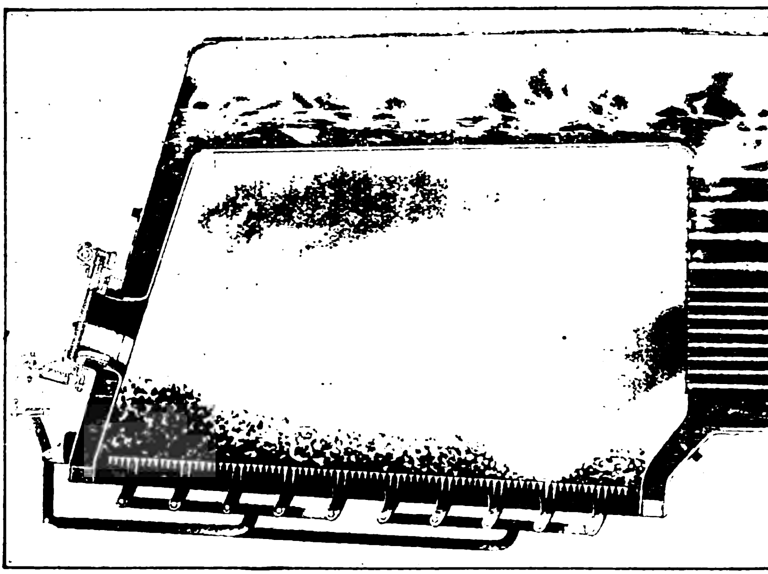


FIG. 184.

intelligent firing, this excess of air need not be great, and should not, as it only cools the fire, and acts to lower the efficiency of the locomotive furnace.

A large excess of air supply is due, usually, to an open fire-box door, or to a large hole in the fire. A light, even fire makes it impossible to get too much air through the grates, but holes in the fire, or thick banks of green coal, as shown in Fig. 184, which represents a typical condition of the fire, causes uneven distribution of air, with the result that complete combustion is not possible. The effect of an open firebox door is clearly understood. A shaft of cold air rushes through the flames and gases, and enters the flues. There is but little mixing of this air with the gases, as in the case of holes in the fire, and the air enters the flues at a comparatively low temperature, cooling them, and causing them to contract. This contraction sets up unequal strains in the flue sheet, and eventually causes flue leaks, or failures.

The fireman should endeavor to carry, therefore, a light, level fire on the grate, a little heavier at the sides, perhaps, so that the air can not come in quite as fast near the sheets as in the center of the firebox. He should fire as lightly and consistently as possible, rather than less frequently and in greater quantity, so as to avoid banks of green coal, with the evil results explained. For, when a large bank of coal is placed on the fire, it reduces the temperature of the fire before it has had time to complete combustion of itself, and excludes air, where it is most needed, from entering the firebox through the grates, while a steady, light firing brings about a constant temperature and complete combustion.

Remember, too, that too little air causes incomplete combustion, due to a lack of oxygen, and that too much air causes a cooling of the fire. Both cause fuel waste. It might be said that, while about ten pounds of air are required to burn a pound of coal, a fair average of the amount of air supplied, in locomotive practice, is approximately fifteen pounds. And air that has passed through the grates to the firebox is the preferred air supply, as it has become heated, and is more ready to unite with the coal gases, and take its part in combustion, than is air admitted above the fire. The result of this latter source of air supply, through the firebox door, has been explained.

It is also good practice to use coal which has been crushed, or broken, into lumps of three or four inches in diameter, as this exposes more fuel surface, thereby aiding combustion, and allays, to an appreciable extent, the formation of clinkers.

Coal is often wet down for the purpose of cleanliness—to keep coal dust from flying—and to prevent fine coal from being drawn through the tubes, but, in this connection, remember that it takes heat to evaporate the moisture. Aside from this fact, the steam generated thereby occupies about 1,800 times the volume of water at atmospheric pressure. At a reduced pressure, and high temperature, conditions existing in the firebox, the volume would be correspondingly larger. Therefore, water introduced into the firebox through any medium, upon becoming steam (and if not dissociated) occupies space that would otherwise be occupied by combustible gases. That is, the efficiency of the draft is impaired. This same effect manifests itself in firing on damp or rainy days.

By combining a knowledge of the above facts with a little sound judgment, the fireman should be enabled to fire his engine in an efficient, satisfactory manner, and reduce greatly the fuel losses often prevalent in this connection.

The Security Arch.

The firebox arch, as manufactured by the American Arch Company, of New York City, has been designed to do away with a great many of the fuel losses due to improper firing, and to improve combustion, even when the firing is accomplished as perfectly as is possible.

By comparing the illustration herewith presented, Fig. 185, with that in the foregoing pages, Fig. 184, the results obtained by the use of this brick arch are readily seen, and require no explanation. The mixing obtained by passing the gases through the restricted opening above the arch promotes combustion, heats the air up to the temperature of the gases and flame, and

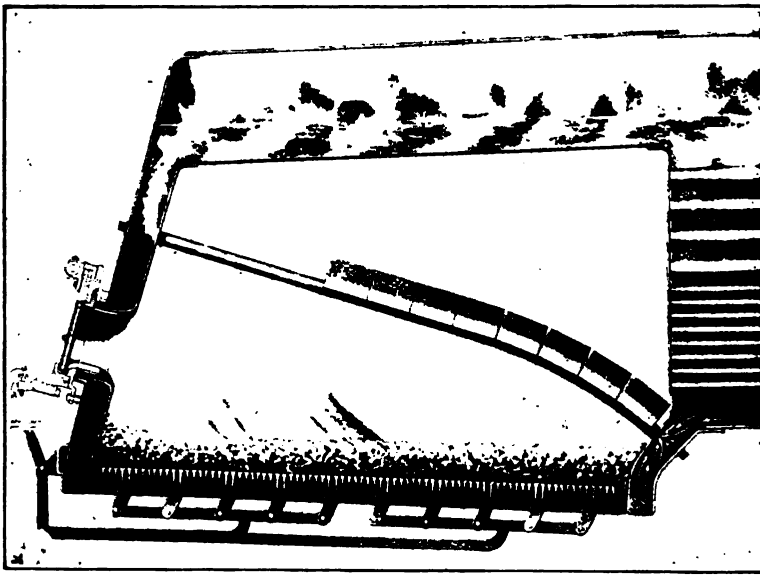


FIG. 185.

causes a more uniform temperature at the flue sheet, reducing liability to flue leakage.

Also, in using the arch, the hydrocarbons arising from banks of the fire are diffused with the air available above the fuel bed, reducing the formation of soot and smoke, and their consequent waste of fuel.

The firebox arch minimizes the efficiency loss due to a poorly distributed fuel bed, as existing in Fig. 184. The complete-ness of the resulting combustion depends upon the amount of oxygen present and the length of the combustion space. Long combustion chambers give long flame ways, provide more time for combustion, and, in a measure, compensate for a small excess of oxygen above the fuel bed, but the volatile combustion arising from the fuel bed of the bituminous coal cannot be burned completely in a combustion chamber firebox of the greatest practical length, unless the gases are mixed by some mechanical means. The arch illustrated is one means of bringing about this condition, and shows the beneficial results obtained from using the arch, and from firing lightly and evenly. It has been demonstrated that in actual practice the arch reduces the emission of black smoke, and its consequent losses, by from 30 to 50 per cent.

Fig. 187 will give the reader a representation of the construction, location and operation of the complete locomotive furnace—firebox grates, ash pan, arch, flues and mechanical stoker.

THE LOCOMOTIVE MECHANICAL STOKER.

On many of the more modern, and larger, locomotives, the limit of capacity for work of the engine is not the ability of the water in the boiler to absorb heat, but the ability of the firebox to supply all the heat the boiler might use in heating the water. And, in a great many cases, the fire in the box is limited in heat giving capacity, not by the air which may be admitted through the grates to aid combustion, but actually by the ability of the fireman to supply fuel in sufficient quantity to carry a fire of maximum efficiency. It is in this regard that the automatic stoker shows its extreme usefulness. The term automatic, however, is perhaps misleading, for it must be remembered that the mechanical stoker does not dispense with the fireman. He must attend to the firing of the stoker, and must occasionally make use of the shovel, and the rake as well.

In the past decade, mechanical firing of locomotives has become a general practice on American railroads and is already gaining general acceptance throughout the railroad world. Approximately 5,000 engines have been equipped with stokers to date, and the impetus of the influences contributing to these applications is so great that stokers have been made standard equipment for certain classes of engines on the majority of the larger American lines.

Originally brought forward simply as a labor-saving device, the mechanical stoker for locomotives has become an aid to greater railway revenues because of the various economies arising from the use of power units larger than can be fired by hand.

The increased hauling capacity of large locomotive units is the greatest reason for their adoption, but the limit of human ability to supply the coal necessary for maximum boiler pressure over any extended period prohibited their use until successful devices for supplying the required coal mechanically had been developed. Thus, mechanical stokers have not only increased the hauling capacity of many previously hand-fired engines, but a great and growing number of locomotives would never have been built had it not been possible to fire them successfully by mechanical means.

Mechanical locomotive stokers of two general types have been applied from time to time for other than experimental purposes,—namely, over-feed and under-feed, depending upon the method of coal introduction.

Of the earlier examples the Crawford stoker was the foremost under-feed type—over 400 Crawford stokers being in service today,—and the most prominent of the earlier over-feed stokers was the Street, first marketed in 1909,—over 1,500 of which type are now in use.

Lately, the over-feed type, exemplified by the Duplex, Standard and Hanna stokers, is in greatest use. Primarily, the over-feed stoker consists of a screw conveyor for carrying coal from the tender, an elevating system for bringing it to a point above the grate surface, a distributing system for spreading it over the grate surface, and a regulating system to regulate firing rate and distribution. A small steam engine located on the engine frame drives the mechanism, and on the three types mentioned there is a crushing system to enable the stoker to fire run-of-mine coal. Successful stoker operation with any type machine demands the following qualifications:

1. Ability to successfully fire the engine to capacity.

2. Durability of the apparatus to secure positive operation.

3. Ability to use any coal suitable for locomotive fuel.

4. Flexibility sufficient to enable the fireman to place fuel in any part of firebox as required.

5. Ability to do at least 90% of all manual labor in taking coal from the tender and distributing it over the grate.

6. Construction so that fireman can at any time inspect or rake the fire, or do shovel-firing if necessary while the stoker is in operation.

7. Definite, marked rates of feed, and capacity in excess of maximum requirements, together with ability to feed coal at any set rate, maintaining that rate, regardless of steam pressure, or grade and condition of coal fed.

8. Accessibility of apparatus so that foreign substances in the coal, such as bolts, spikes or rocks can be removed in a minimum time to prevent clogs or breakage.

9. Low maintenance cost (not over 50 cents per 100 locomotive miles) and,

10. Simplicity such that the average fireman can understand and successfully operate the mechanism after one or two instruction trips.

No locomotive stoker is automatic. The fireman with any successful stoker must regulate its firing and have absolute control of distribution to the firebox in order to take care of the fire's varying needs. The constant, uniform delivery of coal to the firebox, as performed by any successful stoker, is proved the best method of coal distribution,—as specified by the United States Bureau of Mines. It keeps the coal supply uniform with the air supply, allowing thus the complete combustion of all volatile matter.

Any locomotive which cannot be fired to maximum capacity by hand over an extended period of time is a proper engine for stoker application. In temperate climates where grades are not excessive this includes all engines of 50,000 pounds tractive effort and over, both freight and passenger. In regions subject to excessive heat and extremely heavy service, stokers have been found necessary on engines of 35,000 pounds tractive effort.

Mechanical stokers are in successful service on Mallet, Santa Fe, Mikado, Decapod, Consolidation, Pacific and Mountain type locomotives at the present time.

The Duplex Stoker.

The Type "D" Duplex Stoker, illustrated in Fig. 186, has proved competent to fire to capacity the largest locomotives no matter what locomotive coal is used, from screenings to run-of-mine.

From study of Fig. 186, operation of this stoker is easily understood. The coal from the tender is carried forward by the helicoid conveyor screw, 2, to the crushing zone, 4, where it is broken to a size suitable for firing. Thence it passes to the transfer hopper where the dividing rib, 18, apportions it to the elevator casings, 19 and 10, in quantities suited to the timely requirements of the fire. The elevator screws, 11, carry the fuel upward to the distributing tubes, 15. These tubes extend through holes in the backhead on either side of the fire-door, and on a level with it. Constant steam jets 16 and 17 blow the coal through the tubes 15, fitted with distributing corrugations which deflect and spread the coal over the entire surface of the fire in two overlapping zones, so that the greatest amount of coal falls along the center line of the fire-box where the rate of combustion is greatest. The angle at which the coal is directed to the fire-box area is such that a high percentage of the coal is burned in suspension. The height of

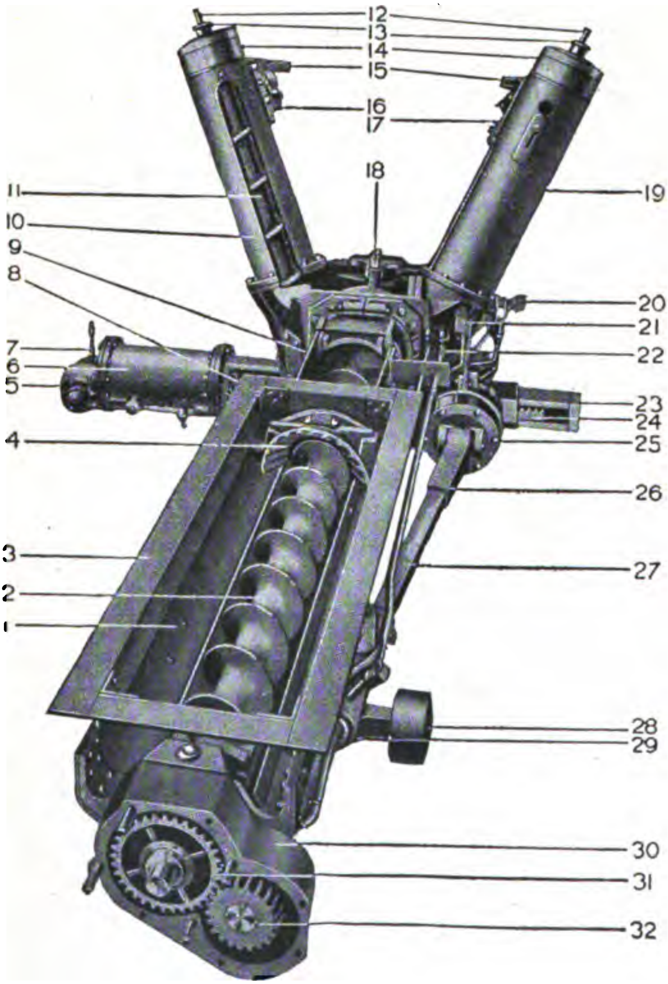


Fig. 186.

the distributing points together with the air cooling prevents any part of the distributing mechanism from being burned out

FIG. 187.

by the intense heat of the fire before a reasonable period of use has transpired.

The conduction of the coal to, and distribution of it from points practically outside the fire-box renders this stoker exceptionally accessible in every way, so that it is always possible to make repairs or adjustments without dropping the fire or cutting the engine from service. These features are more discernible in Fig. 187, showing a cross-section of a locomotive fire-box equipped with the Duplex stoker.

Likewise, by reason of its dual, or Duplex, system of coal introduction, the stoker has a maximum of flexibility, so that either or both of the elevating screws may be run forward, stopped or reversed independently, or either screw may be run in forward speed while the other is stopped or reversed.

The tank conveyor screw can also be run forward, backward, or kept stationary, independently of the elevator screws. This renders simple the concentration of distribution to any spot or spots in the fire-box, in conformance with the fire's varying needs.

The basis of stoker success is a slow engine of high-power and fool-proof operation. For this reason the Duplex engine is the reverse-head of a Westinghouse air-pump whose speed is only 15 R. P. M. and which requires an average of 18 pounds of steam for successful operation.

This engine, 5, in Fig. 186, reciprocates a rack which meshes with gears to drive the elevator screws and a shaft 26, geared at 32 to drive the conveyor screw.

Fig. 188 shows an interior cab view of a U. S. Railway Administration Mikado Type Locomotive equipped with the Duplex stoker and portraying clearly the non-interference of this stoker either with the fire-door, the conventional arrangement of cab

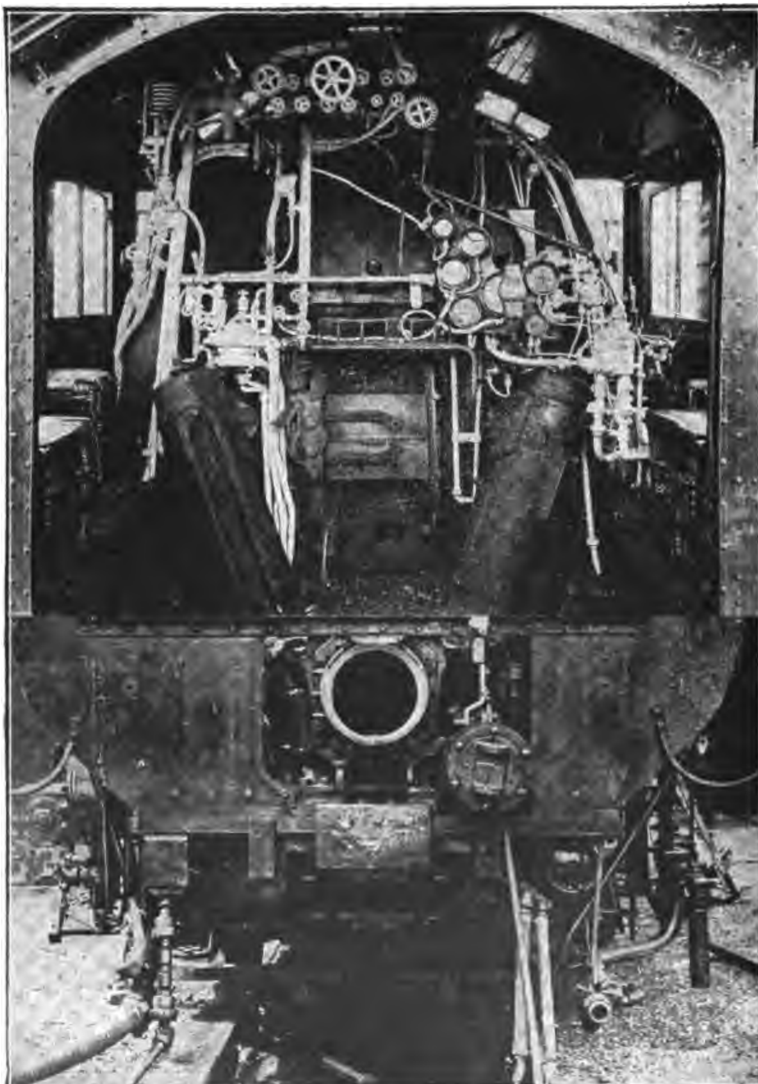


FIG. 188.

fittings or the grates of the fire-box. That the Duplex stoker has passed the experimental stage is evidenced by the fact that on U. S. R. A. locomotives alone, it is doing daily service on Mallet, Santa Fe, Mikado, Mountain, Pacific and Consolidation types.

The Duplex stoker is manufactured by the Locomotive Stoker Company of Pittsburgh, Pa.

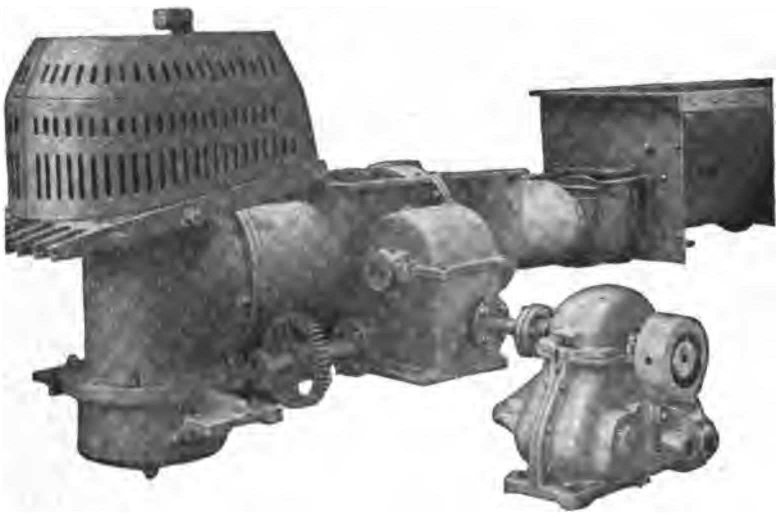


FIG. 189.

The Standard Stoker.

The complete assemblage of the Standard apparatus is shown in Fig. 189; on the right the housing for the helicoid screw, which is placed below the deck of the tank, or tender. This screw takes coal of any size, including run-of-mine, forward from the tender, as long as there is coal in the tender, and without necessity of outside agitation, and delivers it to the vertical conveyor, shown on the left, after it has passed through a crusher in the forward part of the helicoid screw housing, and an intermediate conveyor. This vertical conveyor lifts the fuel up to a point at the lower surface of the fire-door, where, by means of jet valves, indicated by the arrow in Fig. 190, it is fed to the fuel bed on the scatter principle, keeping a thin, level fire. The fuel is delivered so close to the fuel bed that combustion is as complete as possible.

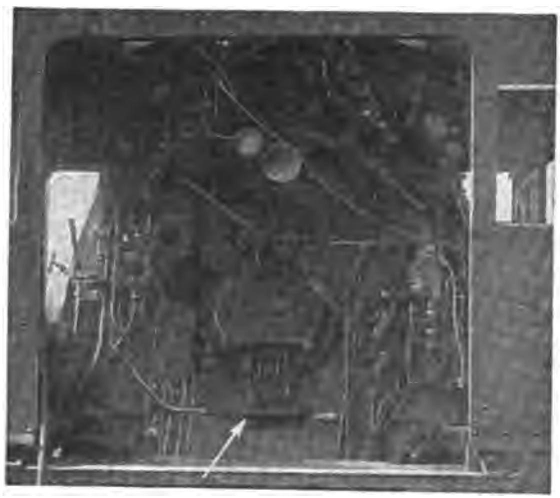


FIG. 190.

The vertical conveyor enters the fire-box from within, and eliminates all cab obstruction. The jet valve, indicated in Fig. 190, is the only visible part of the apparatus in the cab, as every part is built under the deck of the cab and the tender. This, as well as an absence of noise of operation, makes for a silent appliance.

The grate area is not lessened by the fact that the vertical conveyor enters the fire-box from within, for, as shown in Fig. 189, this conveyor is provided with a protecting grate area, which really increases the effective grate area of the fire-box by over two square feet.

Fig. 189 also shows the operating engine, a two-cylinder steam engine of high efficiency and ample surplus power, as required by an engine subject to this service, which is operated by steam generated in the locomotive boiler.

This type of stoker is manufactured by the Standard Stoker Company, of New York City.

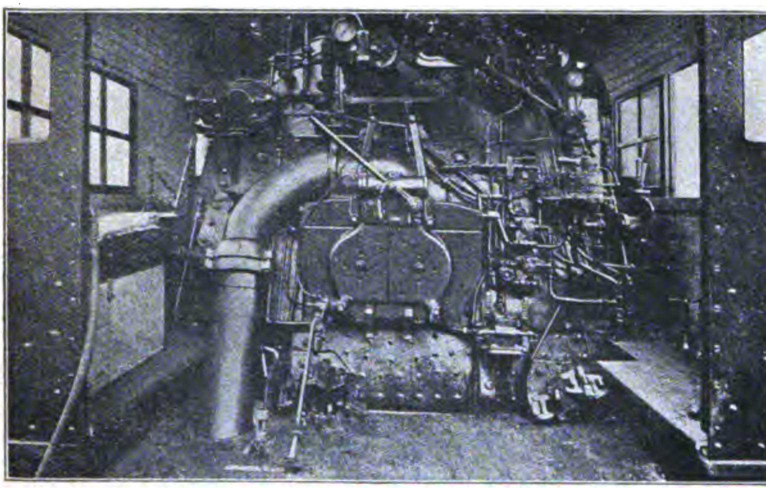


FIG. 191.

The Hanna Stoker.

The Hanna stoker, also, is located under the decks, but is readily accessible for thorough inspection from the outer sides of the locomotive framing, without removing the deck.

Type S-1 is shown in Fig. 191, applied to a locomotive, with a vertical conveyor delivering the fuel to the top of a door cabinet, this door cabinet taking the place of the usual door frame. The usual swinging or pneumatic door can be applied.

The complete type S-1 is shown in Fig. 192, with the exception of the operating engine, which is hidden from view in the illustration, and the names of parts which are given on page 457.

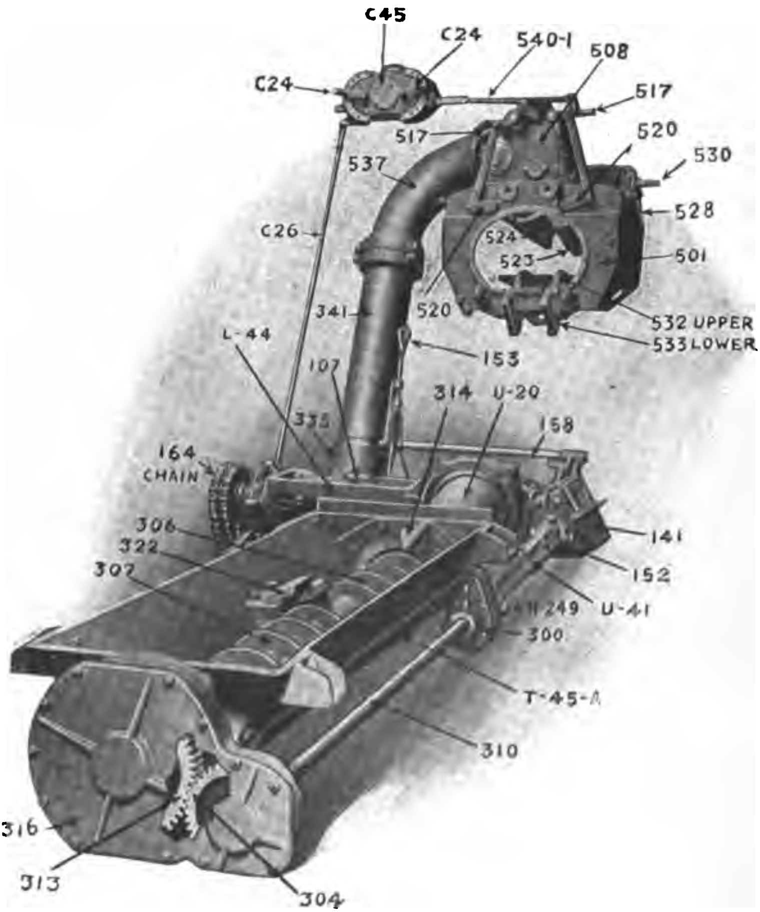


FIG. 192.

In operation, the coal falls by gravity into the combined crusher and conveyor hopper, located in the bottom of the tank bunker. From this point it is pushed forward through a flexi-

C24	Adjusting Arm Lever	L44	Locomotive Hopper
532	Blast Chamber (Upper)	528	Ridge Plate Shaft Handle
533	Blast Chamber (Lower)	316	Rear Housing
341	Casing (Vertical Worm)	307	Rear Worm
322	Center Bearing	158	Shifter Shaft
164	Chain (Hopper Drive)	153	Shifter Plunger
C45	Control Case	U20	Spout (between Engine and Tender)
540-1	Control Case Reach Rod	508	Spout (Cabinet Delivery)
C26	Control Connecting Rod	537	Stoker Neck
306	Crusher Worm	T45-A	Tender Hopper
523	Distributing Wing (Right Hand)	U41	Universal Drive Shaft
524	Distributing Wing (Left Hand)	335	Vertical Worm Base
501	Door Cabinet	517	Wing Handle
152	Gear Box	526	Wing Throw-out Lever
141	Gear Box Cover	300	Front Bearing (Jack Shaft)
107	Hopper Knife (Locomotive)	313	Gear (Crusher)
314	Hopper Knife (Tender)	304	Pinion (Jack Shaft)
310	Jack Shaft	U249	Grease Cap

ble pipe to the feed hopper located on the locomotive. It is next delivered by a worm to the base of the vertical conveyor, by which it is carried upwards to a point of delivery to the fire bed. From here it drops downwards to pivoted distributing wings, or vanes.

These wings move alternately, continually changing the point of delivery on the distributing plate for final distribution to the grate. This is accomplished by means of continuous steam blasts which are driven out flat over the distributing plate from a series of nozzles, radially directed, and operated in connection with a fan-like sheet blast emanating from a lower chamber. These two blasts in connection with back corner channels cut in the face of the distributing plate, cover the entire grate surface with an even layer of coal.

The pair of wings mentioned may be operated so as to deliver coal at either side, or in the center of the fire. They may also be operated manually to fire the locomotive in any manner desired.

This stoker is made by the Hanna Locomotive Stoker Co., of Cincinnati, Ohio.

Each of the above stokers incorporates a number of peculiarities in detail of construction and operation, which it is unnecessary to mention here, but all are examples of modern, simplified practice, and are, in general, of the same type.

SUPERHEAT.

The practice of superheating the steam used in operating the locomotive, that is, adding heat to the steam after it is formed in the boiler, and before it enters the cylinders of the engine to perform its work, has of comparatively recent years come into general use on railroads in this country and abroad. Considerable experimental work had been done, however, before 1900, to the extent that Dr. Wilhelm Schmidt had produced a boiler and motor, in 1894, which used superheated steam of relatively low pressure and of a temperature of about 700° Fahrenheit. So successful were these experiments that within five years after the first superheater of this type was applied to the locomotive, the system was in quite general use.

In the attempts to use superheated steam in locomotive practice dry steam or moderate degrees of superheat were frequently tried at the start, but without the marked economical results that are obtained by the use of a high degree of superheat. The first forms of superheater experimented with were of the smoke box type, but they were open to the serious objection of obstructing the front end. In the form where the steam passed through the tubes of the superheater there was a high cost of maintenance, due to the corrosion of the tubes, the cutting action of the cinders, and the difficulty in keeping the superheater clean so that a uniform degree of superheat could be maintained.

Following these were other types of smoke box superheaters, so arranged that the steam passed around the tubes while the gases passed through them. All forms of smoke box superheaters seriously obstructed the front end, and, at the same time, made very difficult and expensive the construction, installation, and maintenance of the steam pipe and other connections in the smoke box. However, irrespective of the mechanical defects of these types of superheaters, they have been practically abandoned because of the small economies obtainable from the use of steam with the low degrees of superheat they are capable of producing.

The fire tube superheater, capable of superheating steam continuously to a temperature of 600° Fahrenheit, or over, is now generally adopted by railroads.

The principal advantages derived from the use of superheated steam results from the increased volume of steam delivered per unit of power evaporated, and from the prevention of cylinder condensation. To bring this out clearly a brief consideration of the principal differences in the characteristics of saturated and superheated steam will be of interest.

Saturated steam has the same temperature and pressure as the water from which it is evaporated, and with which it is in contact in the locomotive boiler. For each pressure the steam has a definite constant temperature. At 170 pounds boiler pressure, for example, the steam always has a temperature of 375° Fahrenheit and a volume of 2.47 cubic feet per pound. If more heat is added to the boiler, it is transmitted to and used in evaporating more water, but does not increase the steam temperature as long as the pressure remains the same. If heat is taken away from saturated steam in doing work or by cooling, as in the cylinders and in the steam passages to them, part of the steam condenses. The amount of the steam condensed is almost proportional to the heat abstracted, and this condensed steam, or water, is inert so far as capacity for further work is concerned. When the steam, however, has left the boiler and passed into the superheater, it is separated from the water. If heat is now added, its temperature and volume are increased, although its pressure remains the same, and it becomes superheated steam.

Superheated steam, which partakes more of the nature of a perfect gas than does saturated steam, is a poor conductor of heat and has a larger volume than has an equal weight of saturated steam. For example, steam at 170 pounds pressure, and at 200° Fahrenheit superheat, has a volume of 3.27 cubic feet per pound against a volume of 2.47 cubic feet per pound for saturated steam at the same pressure. When superheated steam is cooled off in the cylinders, it loses part of its superheat, but it remains steam—that is, *it does not condense*—until all of the superheat has been absorbed. Tests show that in Mallet compound engines without superheaters the condensation in the receiver pipe is sometimes more than 20%, while in simple saturated engines using short cut-off, cylinder condensation increases to over 35% of the weight of steam admitted to the cylinders. This means that for every 100 pounds of steam delivered to the cylinder only 65 pounds are available to perform work. The condensation loss, as stated above, can be overcome by the use of highly superheated steam, which means an average reduction of 35% in the amount of water, and of 25% in the amount of coal, used per ton mile.

In using superheated steam the engineman must be particularly careful to see that the valves and cylinders are well lubricated. The lubricator steam pipe connection should be at least as large as recommended by the makers, and the steam should be taken from a point which will insure full boiler pressure at the lubricator under all conditions. If the superheater be properly applied, and the engine given ordinary care, however, no trouble should arise through difficulty in properly lubricating the valves and cylinders of the locomotive.

It is a very common tendency to carry water at too high a level, when running a superheater locomotive, chiefly, perhaps, because its presence can not be detected in the cylinder. It should be remembered that in doing this, one is merely using the superheater as an auxiliary boiler, to evaporate a quantity of water, rather than making it *super*heat the steam already formed. The locomotive should always be operated in such a manner that only dry steam from the boiler will pass into the superheater through the throttle and drypipe. The lowest water level consistent with safety is the practice which should be followed.

It might be well to add that, because the superheat saves water, it is natural that the water is not used up so rapidly, which also results in a tendency to carry the water at too high a level.

The proper operation of the damper, too, is necessary. The damper controls the draft, and, therefore, the flow of gases through the large flues. It is placed just below the bottom row of large flues, usually on the same level with the table plate, and is operated by a small cylinder bolted onto the smoke box. This cylinder is connected either to the steam pipe or the blower, the latter being the case in switching locomotives, and works automatically upon the opening of the throttle or the blower valve. When in closed position, on road locomotives, the damper is held by a counterweight. The damper, as thus operated, protects the superheater units from overheating, when there is no steam passing through them. Failure of the damper to operate properly, materially reduces the steaming capacity of the boiler and, as a consequence, reduces the degree of superheat. For example, if the damper fails to open, it will obstruct the passage of gases through the tubes and flues above it, considerably reducing the boiler evaporation and preventing the effective superheating of the steam passing through the units.

If these things are guarded against, however, the superheater is the most economical improvement in connection with locomotive practice yet made, and its usage will result in very satisfactory service.

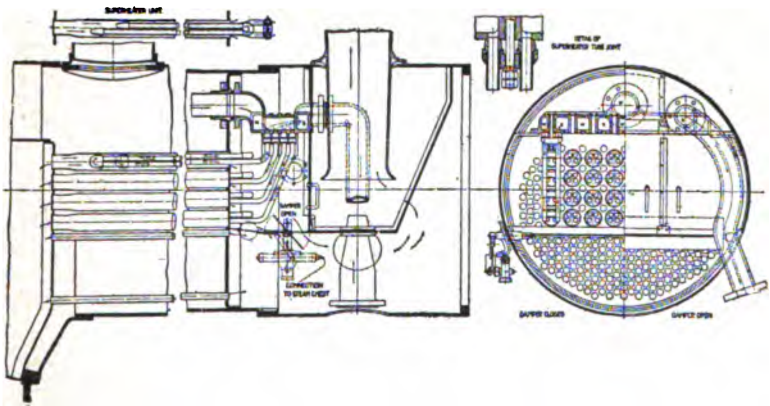


FIG. 193.

Because it is by far the most extensively used superheater in this country, we shall present the Schmidt system of superheating, which is manufactured by the Locomotive Superheater Company, of New York, so as to enable the reader to become familiar with the construction and operation of the modern superheater system.

The Schmidt Superheater.

This superheater is of the "fire tube" type, of which there are several forms, differing principally in the arrangement and location of the header castings. We shall describe the best known

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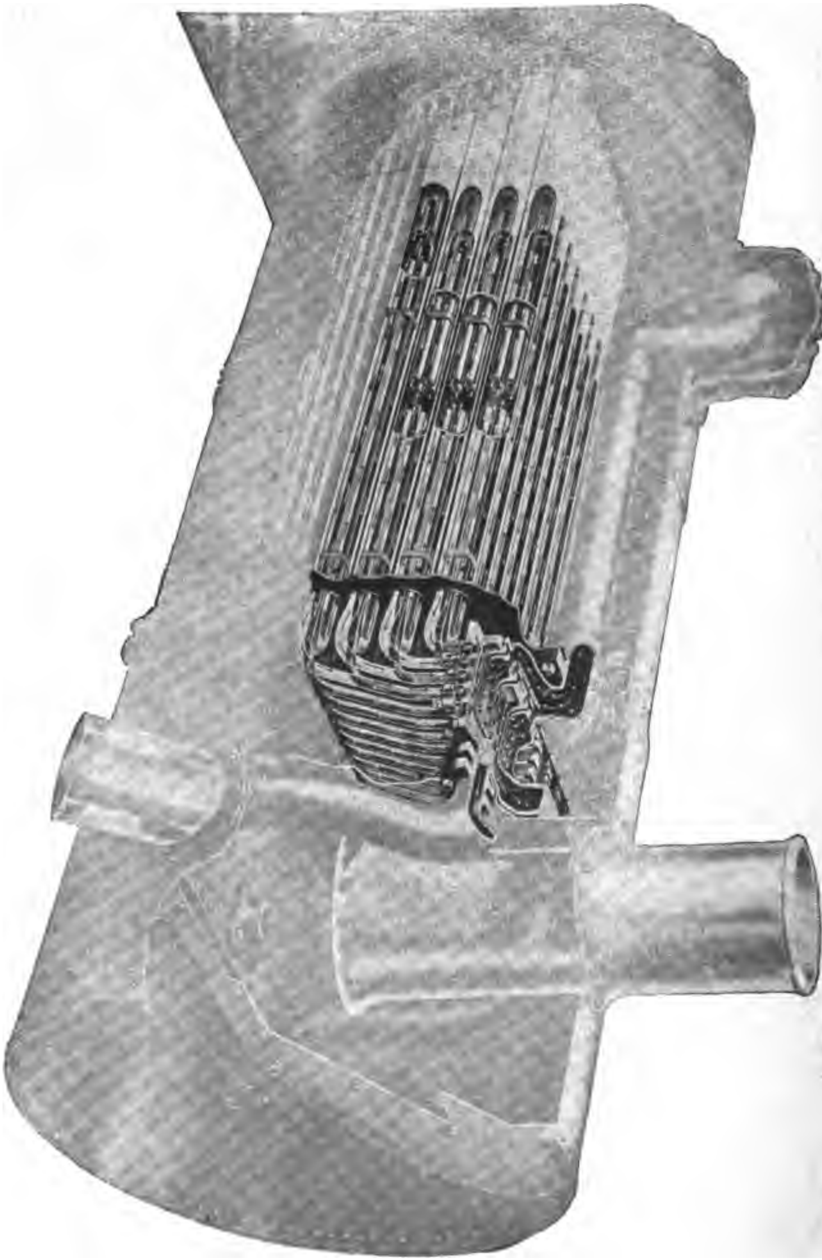


Fig. 194.

of them, type "A," shown in Fig. 194. On referring to the illustration, it will be seen that the superheater proper consists of groups of four pipes of about 1½" outside diameter located inside of flues of about 5½" outside diameter, placed in the upper portion of the boiler, having their back ends swaged down to 4½" outside diameter, in order to secure better circulation of water next to the firebox tube sheet.

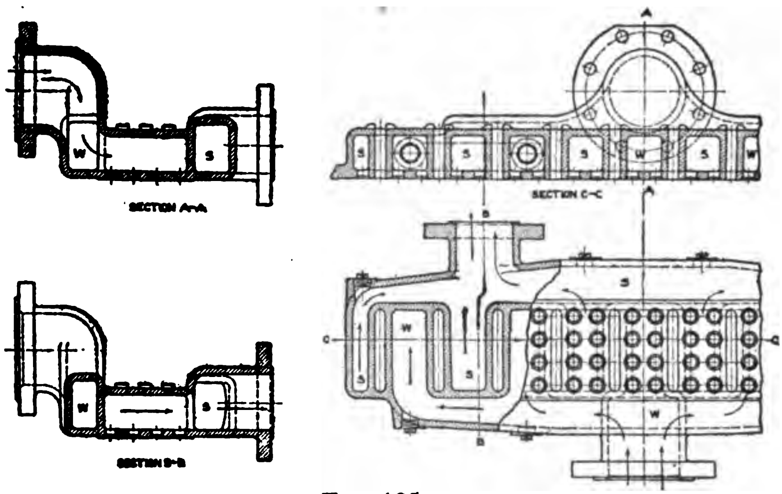


FIG. 195.

Each group of pipes, called a superheater unit, is formed of four seamless steel tubes, connected at the ends by a forging process which forms return bends from the material of the pipes. There is thus formed a continuous pipe, as shown in Fig. 193. To insure the proper flow of steam through these units, the header, shown in Fig. 195, is provided, which takes the place of the ordinary "Tee" or "Nigger" head. This header is so designed with internal walls that the steam entering it from the drypipe must pass through the passages marked "W" to the pipes of the superheater units, where it is superheated by the heat of the firebox gases and made to pass through the flues, and around the tubes. These pipes extend back to within a short distance of the firebox, thereby exposing the steam to the direct effect of the high temperature of the fire. On leaving the units, the superheated steam is returned to the header on the opposite side of the partition walls, to the passages marked "S," connecting with the steam pipes and steam chests. The direction of the steam flow is indicated by the arrows in the foregoing figures, which serve to make more clear the operation of the system.

In connection with this description, the operation of the damper, shown in Fig. 194, is necessary.

In the general operation of the type A, as well as other types of superheaters, the passage of steam is simple and unobstructed, to insure freedom of flow and a minimum tendency toward wire drawing. When the throttle valve is opened, saturated steam passes through the dry pipe and into that portion of the superheater header designed to receive the saturated steam. From this section of the header, which is in communication with one end of the superheater units, the steam passes downward through one pipe of the unit, backward toward the firebox tube sheet, forward and backward again, then forward and upward into another compartment of the header designed to receive the superheated steam. From this compartment in the header it then passes into the steam pipes and the cylinder, where the effective work is done.

As the steam passes through the steam pipes to the steam chest, a small part of it is led by means of a small copper pipe, to the damper cylinder, which operates the damper controlling the flow of gases through the large flues. In road locomotives the position of this damper is *closed* when the engine is ***not working*** steam. Upon opening the engine throttle steam is admitted to the damper cylinder "E," Fig. 194, through pipe "E-1," which is in direct communication with the steam pipe or steam chest and the damper cylinder. This automatically opens the damper by means of a crank and link connection between the damper piston and the damper shaft. Simultaneously with the operation of the damper the counterweight "E-2" is raised to the topmost position, from which position it will fall when the steam is cut off, and the damper will close. Thus the damper permits the passage of hot gases through the flues when the engine is working steam, and prevents these gases passing through the large flues and coming in contact with the superheated units when there is no steam in them to absorb the heat.

Under certain conditions, such as are found in switching or other similar classes of service, a damper cylinder designed to operate under different conditions than the one described above, is used. This design permits the damper to remain open at all times except when the blower throttle is opened, when it is automatically closed.

This system of superheating may be applied to all types of existing locomotives. If slide valves are used on the locomotive, the change to piston valves is advisable. Over 35,000 locomotives are now equipped in North America.

Finally, to obtain the best results from any superheater, observe the following precautions:

1. Keep the tubes and flues clean.

2. Keep the water in the boiler at such a level that it will not be carried over into the superheater. The superheater is built to superheat steam, not to evaporate water.

3. Keep the damper, damper cylinder and rigging in good operative condition. Don't operate a locomotive with a damper tied open, nor permit one to be operated in this condition.

4. Make sure that a good joint is obtained between the ball ends of the units and their seats in the header when repairs are being made, and keep the unit clamp bolts tight to prevent steam leaks.

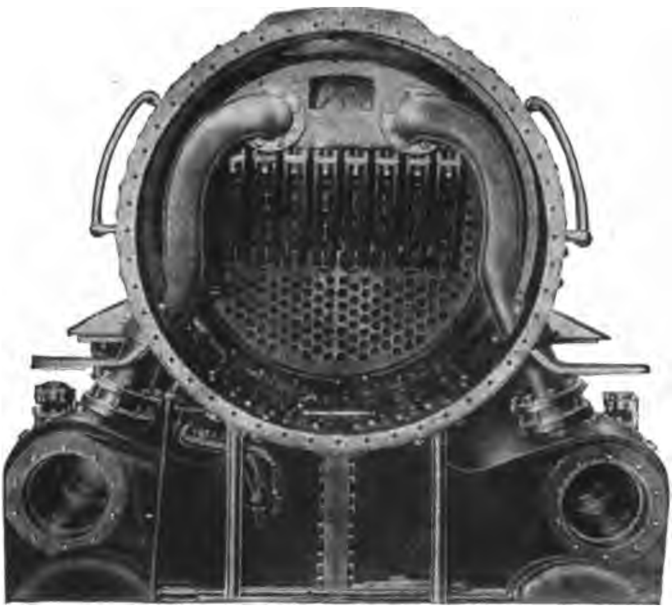


FIG. 196.

5. Apply the correct number of bands and supports and see that they are maintained.

THE PYROMETER.

With the use of superheated steam, it is considered an advantage, in view of the most efficient and economical operation of the locomotive, to know for a certainty the temperature of the steam. If the fireman is not firing correctly, if the engineman is carrying water too high, if the flues are filled up, or if the locomotive in almost any way falls off in performance, the temperature of the steam will drop. But, if the superheater is functioning properly, and conditions are correct, normal steam temperature will prevail. By the use of the pyrometer, which is nothing more or less than an instrument employed to measure the temperature of steam, the engineer may know, at all times, the exact temperature of the superheated steam in the steam chest.

When a superheater locomotive is standing or drifting with the throttle closed, there is, of course, no superheat being obtained, and the indicating hand of the pyrometer instrument in the cab is located at the left-hand side of the dial, reading between 350 and 390 degrees, assuming that the boiler pressure carried is 200 pounds or less.

As the throttle is opened and the engine starts to work, steam from the boiler passes through the superheater pipes and the superheating process begins. As the engine starts, the pointer will move from left to right on the scale, showing an increased temperature in the steam chest, and, as the engine is worked harder, the superheat added to the steam increases until, under average conditions, the indicator registers between 600 and 650 degrees.

When the pyrometer operates in this manner, it is an indication to the engineer that the locomotive is being handled so that the maximum saving that the superheater makes available is being obtained. If it fails to operate in this manner it shows him that either the locomotive is not being operated to produce the best results or that it has not received the proper attention in the roundhouse.

The instrument marketed by the Locomotive Superheater Co., shown in Fig. 197, is of the electrical type, making use of the thermo-couple principal, by means of which an electric current proportionate to the difference in temperature between the hot and cold junctures of the couple is generated. The indicator itself contains a millivolt-meter actuated by the current generated at the thermo-couple and moves a pointer on the dial indicating the actual temperature of the steam. The pyrometer equipment for each locomotive consists of three parts; a steam fixture (containing the hot junction of the thermo-couple), which screws into the steam pipe or steam chest of the engine; an indicator, which is placed in the cab, and a cable connecting the fixture to the

FIG. 197.

indicator. The cold junction of the thermo-couple is at atmospheric temperature and variations in the atmospheric temperature are automatically compensated for. The hot junction of the thermo-couple is directly exposed to the flow of superheated steam and the slightest variation in the steam temperature immediately affects the current generated, showing this change instantaneously upon the dial in the cab.

The Brown pyrometer, as shown in Fig. 198, is also of the thermo-couple type. The complete Brown instrument, consisting of three units, is clearly shown in the illustration. The thermo-couples are generally installed in the steam pipes or in the steam chests. Compensating leads carry the cold end of the couple back to the instrument located in the cab, where the temperature is practically constant. The instrument dial is open-faced, grad-

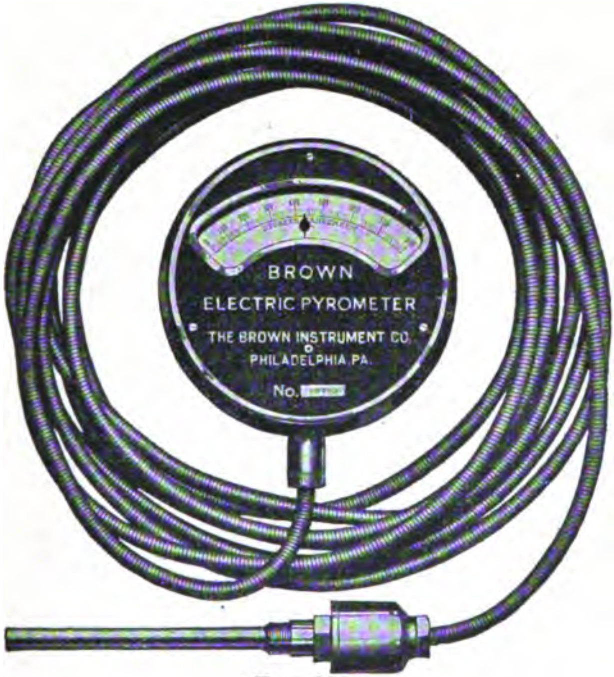


FIG. 198.

uated from 0 to 800 degrees Fahrenheit, is easily read, and is water tight, and unaffected by the action of gases.

FEED WATER HEATING.

Feed water heating, as applied to locomotives, is not a new, or an untried, experiment. In fact, the application of certain devices used to pre-heat the water, before allowing it to pass to the boiler, dates back as far as 1802, in England, and, in the United States, to 1835. The structural and operative principles of these early types were basically the same, in that they used a tubular heater, to which the feed water was supplied by a pump, as in our modern heaters, of which we shall describe and illustrate one American and one English system, as examples of modern feed water heating.

The Locomotive Feed Water Heater.

The object of feed water heating is the reclamation of heat, the source of supply being the heat which would otherwise go out the stack as exhaust steam, and the medium for getting this heat back to the boiler being the feed water. That is, a part of the hot exhaust steam is made to transfer its heat to the cold feed water, before the latter enters the boiler.

Description of Feed Water Heating Equipment.

Feed water heating equipment on the locomotive consists of a steam-driven boiler feed pump drawing water from the tender and forcing it through the feed water heater and then into the boiler. The pump is usually mounted on the left side of the boiler alongside the air pump. The heater is usually placed

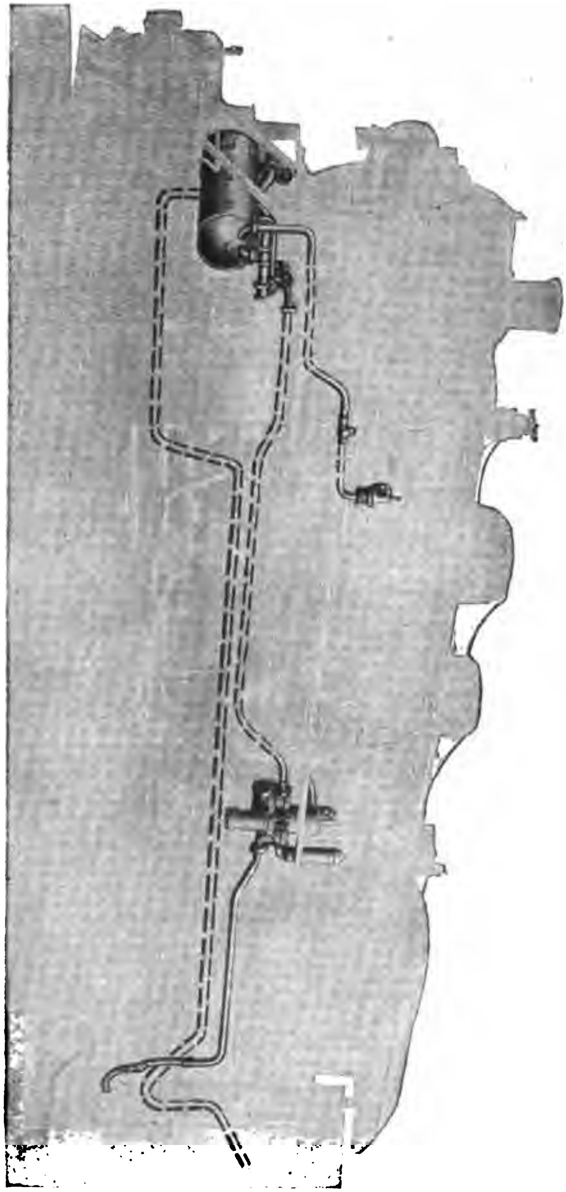


Fig. 199.

across the frames just ahead of the cylinders. In the case of Mallet type locomotives, it is fastened to the boiler shell. Reference to the illustration (Fig. 200) will show the names and the arrangement of the various parts of the equipment. It should be noted especially that the pump handles only cold water from the tender. This feature is very important for reliable service of the pump.

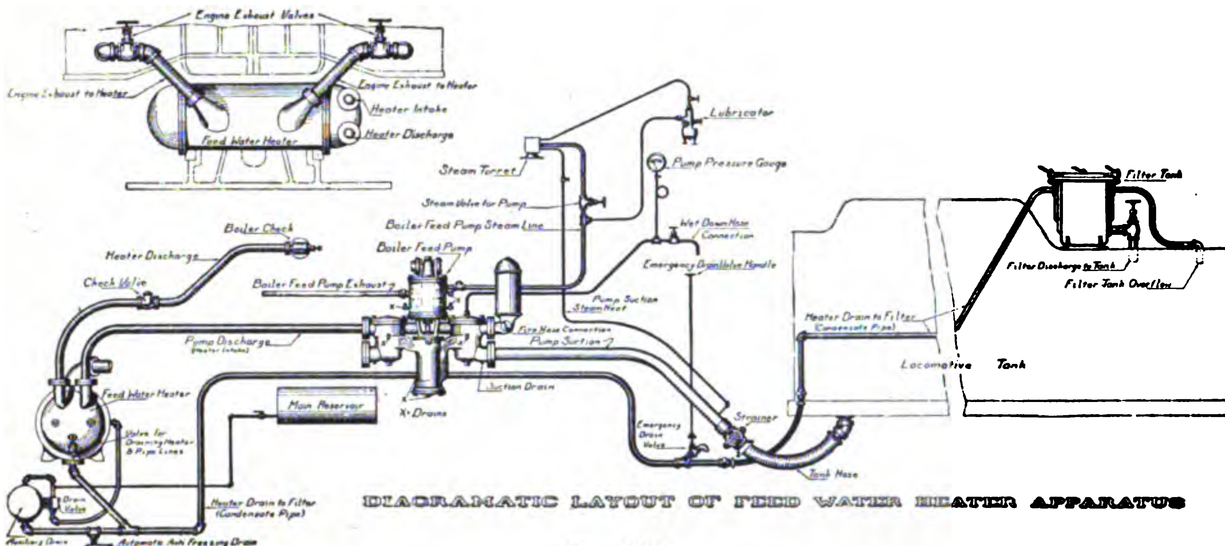
Pipes carrying the exhaust steam from the valve chambers to the heater should be as short as possible. The heater is therefore placed in a well-protected and otherwise vacant location just ahead of the cylinders, where it is easily accessible for inspection. See Fig. 199.

A throttle valve controlling the speed of the pump is conveniently located in the cab. Lubrication of the pump is provided from the main engine lubricator.

Exhaust steam condensed in the heater passes through a drain pipe in the bottom of the heater and is carried to a filter, where the oil is removed and the pure water returned to the tender tank or pump suction to be used again. An auxiliary drain tank is provided, arranged as shown in the illustration, to elevate the condensate to the filter when the back pressure of the locomotive is not high enough to do it.

Arrangement of Feed Water Heating Equipment on a Locomotive.

This illustration shows the arrangement of the various pipe connections, and gives the names of the different pipes. The sizes shown are typical, but may not be exact for any particular locomotive. The same is true of the location of the pump, which can be applied to the right side of the boiler if desired, or can be located in a number of places other than shown here if necessary.



DIAGRAMATIC LAYOUT OF FEED WATER HEATER APPARATUS

FIG. 200.

The heater should be close to the cylinders, but can be back of them instead of in front, if there seem to be advantages in that location, or if it can be located on the running board, or on top of the boiler if desired.

Where the heater is to be applied to an existing locomotive it is recommended that it be located in front of the cylinders and under the overhang of the boiler. This is generally an unoccupied position which is ideal for the heater. The pump can be located on either side of the boiler but it is generally applied on the left side. It occupies practically the same space as a cross compound air pump but it is supported by a standard 9½-inch air pump bracket. Generally the only change required for the application of the pump is the moving of one of the air pumps, or a change in position of one of the air drums.

All piping is located where it can be properly and securely held, and if once properly installed it gives no trouble from leaking or freezing. Generally, in the application to older locomotives, the connections for the exhaust steam leading to the heater are made directly through the front head of the valve chamber or through an opening cut into the exhaust passage at the top of the valve chamber. A valve is provided in each exhaust steam pipe to permit them to be easily closed when testing superheat units.

The Pump.

A boiler feed pump has been developed which, it is claimed, has been operated under the most difficult conditions with a slip of not over 4 per cent. and a steam consumption of not over 70 pounds per water horse-power at any point of the range of capacity. It will deliver well over 50 pounds of water against a pressure of 250 pounds, with one pound of steam at a pressure of about 135 pounds. This means that the pump will consume less than 2 per cent. of the capacity of the boiler and that the amount to be deducted from the theoretical saving of the feed water heater is less than 2 per cent.

Between 65,000 and 70,000 pounds, or from 7,800 to 8,400 gallons of water an hour can be handled by this pump at its maximum rate. A normal continued rate of 50,000 pounds or 6,000 gallons an hour is maintained with quiet action.

Reference to Fig. 201 will show the general features of construction of the pump. The steam end is essentially the standard 9½-inch locomotive air compressor of the Westinghouse Air Brake Company. All standard 9½-inch air pump parts for the steam end are used in the water pump.

On the water end the effort was made to develop a rugged construction with every part easily accessible for inspection and of the simplest and strongest design.

Valve chambers containing the suction and discharge valves are located on either side of the water cylinder in order to get ample volume for the free passage of the water and at the same time keep the pump within clearance limits for attachment to the side of the locomotive. The valves in the back valve chamber are those controlling the water going to the cylinder underneath the water piston, while those in the forward chamber control the water in the upper part of the water cylinder. These valves are located in groups of five, each side containing two sets, one for suction valves and one for discharge valves. Each set of five valves is arranged in a separate removable deck which is held in place by a central bolt. The stops for the suction valves are part of the upper or discharge deck and the stops for the discharge valves are attached to the valve chamber cover.

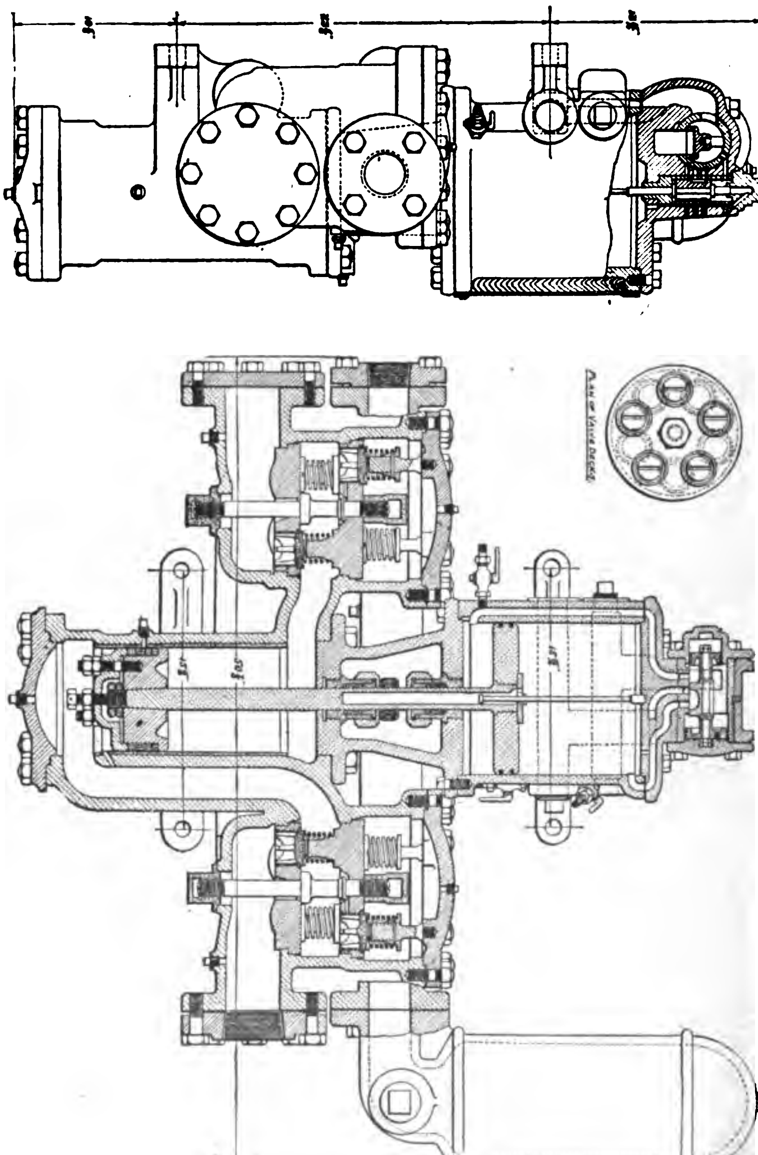


FIG. 261.

By using this large number of valves in each deck it is possible to get the desired area of opening with a very small lift of the valve. It is this feature that accounts for the high efficiency of the pump and also results in greatly reduced maintenance cost. With the low lift the valves do not pound themselves and seats, and start leaking, nearly as soon as would valves of larger size.

Bronze valves and valve seats are used, and over each is a light bronze spring insuring quick and positive action to each valve.

An accepted standard water packing is used on the water piston and the construction at this point is extremely simple. It incorporates a follower plate which carries an adjusting screw in the center for following down the packing, if necessary. Experience has shown that very little attention is required at this point. The packing should need renewing only at the shopping of the engine.

This pump has a fire hose connection at the bottom of the air chamber. The opening at this point is closed with standard pipe plug inside of the bushing. This can be removed with a monkey wrench and a standard fire hose connected.

Suitable provision is made for a careful draining of all parts of the pump, in case the locomotive is to be laid up dead in cold weather.

The Feed Water Heater.

A prime requirement of a feed water heater on a locomotive is that it shall give the greatest possible rate of heat transfer between the exhaust steam and feed water per square foot of heating surface, accompanied by a reasonably low resistance to the passage of the water through the heater. This is necessary in order to get the minimum weight and size of the heater, and careful tests have demonstrated that the corrugated agitator type of heater here illustrated, and used in this system of feed water heating, answers these requirements.

Reference to Fig. 202 will show that the constructional features of the heater consist of a number of brass tubes each 5/8 inch outside diameter which are expanded into thick sheets at either end. These tubes are 4 feet or more in length between tube sheets and are arranged in groups so that the water makes the passage of four different lengths of tubes in going through the heater, giving it a travel of over 16 feet in the tubes.

Within each tube is a spirally corrugated agitator made of thin brass stripping. These agitators have stops at each end which overlap and interlock and prevent the agitator from turning in the tube. The water in passing through the tube and around the agitator in constantly impinging against the inner surface of the tube. This results not only in the high transfer of heat for a given area, but also scours the inside of the tube and keeps it clean.

Heavy cast iron headers are bolted to the tube plates and are arranged with partitions which compel the water to pass through the four passages in succession. These headers are easily removable for inspection, if desired. One of the tube plates is larger than the other and is arranged to be secured at the end of the cast iron body which encloses the whole nest of tubes. The smaller tube sheet is allowed to move freely in a longitudinal direction within the body, and thus provides for the difference between the expansion of the brass tubes and the cast iron shell. The end of the shell is enclosed by a casing to prevent the escape of the exhaust steam around the floating header.

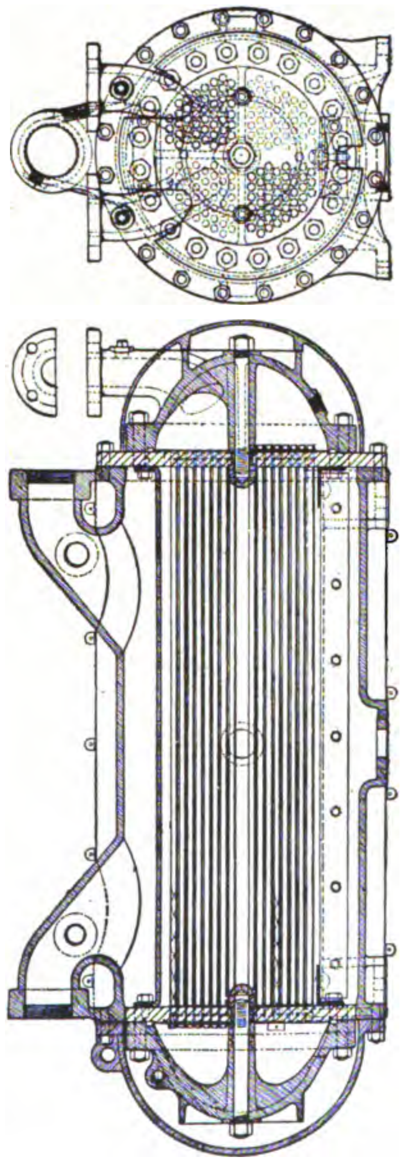


Fig. 202.

Feet for securing the heater to the locomotive frames are formed on the body and elbows are provided for connection with the pipes leading from the exhaust passages in the cylinder. A drain is provided at the bottom of the shell for clearing away the condensed steam. It will be noticed that there is a baffle plate at the top of the nest of tubes to prevent the impinging of the steam directly on the tubes and a nest guard is provided at the bottom to prevent damage to the tubes when the nest may be removed for any purpose.

Inlet and outlet water connections are provided at one end of the heater, the arrangement being for a flanged pipe connection to allow the removal of the header without disturbing the pipe lines.

A thick coating of asbestos lagging is provided around the body of the heater and this is covered with a sheet steel jacket.

A point of special interest in this heater is the form of joint used for the connection of the tubes and the tube sheets. This joint includes a groove in the tube sheet about midway between the two faces, and as the brass tube is rolled the metal is cold-flowed into this groove. Experience has shown that by employing this construction a leaky tube at its connection with the tube sheet is practically unknown.

A drain is provided at the inlet end of the heater which, when opened, will remove the water, not only from the heater but also from the pipe leading to the pump and the connection as far back as the check valve at the boiler. Thus by opening the drain on the end of the heater all of the line between the pump and the boiler check valve is easily drained.

Connections are provided to receive the exhaust steam pipe from the boiler feed pump and also from the air pumps at the back of the elbows on the top of the heater body.

This design of heater has been developed after three years' practical experience in locomotive service and is the result of a most careful study of roundhouse and shop conditions as well as regular road service.

Instructions for Operation.

Instructions required by an engine crew for operating a feed water heating apparatus are surprisingly simple.

Before leaving the terminal the engineer should see that the valves in the exhaust steam line leading from the cylinders to the heater are wide open. He should see that the boiler check valve is open, and that the tank valve is wide open and also that the drain at the heater and those on the pump are closed. After coupling to the train the lubricator should be open to feed about two drops a minute and the steam valve leading to the pump should be slightly cracked in order to clear the pump of water and insure that it is in operation. Thereafter the only attention required is simply to open and close the valve located on the steam line leading to the pump and by its adjustment determine the amount of water that is being fed into the boiler. The gauge in front of the engineer will show that the pump is feeding water and the pulsation of the hand gives a clear indication of the speed at which the pump is running.

On leaving the locomotive it is recommended that the emergency drain at the lowest point of the line between the heater and the filter to be opened. The steam valves to the pump and the lubricator are closed tight. Outside of this, nothing is required except in the case of cold weather, when the drain on the left end of the heater is opened to drain the pipe line, and the steam heater valve turned on to prevent the hose and suction pipe from freezing.

It has been found from long experience that a very slight amount of attention is required to the apparatus at the roundhouse. It is advisable that the strainers in the suction line be cleaned at frequent intervals, and occasionally the packing on the water end of the piston rod of the pump needs tightening down to prevent leakage.

It has been found that the air chambers on the pump will occasionally fill with water, due to the absorption of the air in the chamber, and in this case the procedure is to open up the strainer cap on the suction line with the tank valve closed, and then by opening the squirt hose line and running the pump slowly, all the water in the system will be discharged and the air chambers will be cleared and ready for service again.

Experience has demonstrated that there is absolutely nothing to do to the heater in the roundhouse under ordinary conditions, and the pump will need but very minor attention at any period between shoppings of the locomotive.

This system is the product of The Locomotive Feed Water Heater Company, of New York City.

The Weir Feed Water Heater.

This feed water heating arrangement is, in basic principles of construction and operation, similar to the appliance previously described, and differs only in detail. The arrangement of the various units of the system is shown in Fig. 203. It may be seen from the illustration that the spray valve delivers the feed water to the boiler at its highest point, in the dome. By this means the feed water reaches the water surface of the boiler at practically the steam temperature, and secures increased efficiency and decreased upkeep of the boiler, due to uniformity of temperature maintained.

Without this valve, shown in Fig. 204, the heater installation would be handicapped by the fact that the water fed to the boiler when the engine is standing would be comparatively cold. When chalky waters are prevalent, the heating of the feed water to the steam temperature has also a pronounced effect on the character of the scale, which is deposited as a fine sludge, and may be readily washed out..

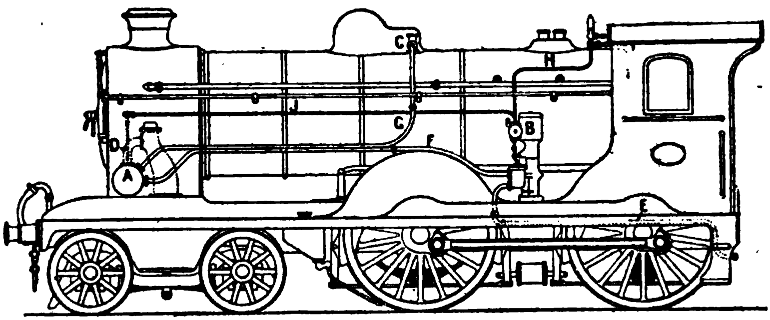


FIG. 203.

Names of Parts as Applied.

A Heater
B Pump
C Spray Valve
D Exhaust Steam to Heater
E Pump Suction
F Pump Discharge
G Discharge from Heater
H Pump Steam
J Pump Exhaust

The body of the valve is of gunmetal, the spindle of steel, and the valve and seat of monel metal. The valve is so constructed as to be easily accessible for examination at any time.

The Pump.

The pump employed is shown in detail in Fig. 205, and the names of the parts are given. It is of the single cylinder type, double acting and vertical (horizontal type are sometimes employed), and is fitted with a special lubricator grease cup, and thermometer, and the length of the stroke of the pump is adjustable.

Feed Heater.

The heater is of the multi-flow surface heater type, with rolled brass tube plates, and solid-drawn copper tubes. This heater, shown in Fig. 203, is known as an "Impulsive" heater, and heats the water by means of steam taken from the exhaust,

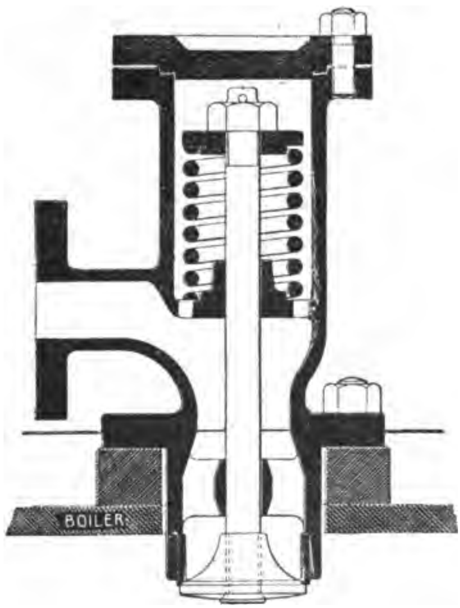
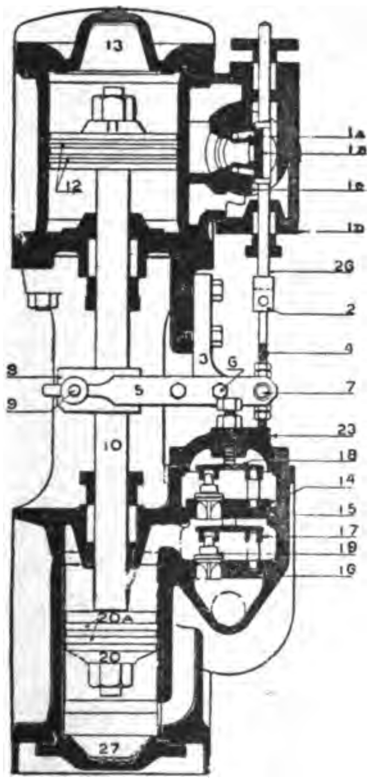


FIG. 204.

which, passing through a branch from the blast pipe, impinges on the outside of the heater tubes, through which the feed water is passed. The orifice in the blast pipe is not altered in fitting the heater connection, so that the effective exhaust area is increased, with a consequent reduction in back pressure.

The manufacturers, G. & J. Weir, Ltd., of Cathcart, Glasgow, claim a fuel economy of not less than 10% with the use of this heater, when properly installed, as well as a decided increase in locomotive capacity, especially in heavy service. The London, Brighton & South Coast Railway, of England, makes extensive use of this device, as do many other roads on the continent and abroad, with a resultant economy of power operation.



No. of Part.	*Name of Part.*
1A.	Main slide valve.
1B.	Auxiliary slide valve.
1C.	Main slide valve Bells.
1D.	Bottom cover of piston valve chest.
1E.	Liner for piston valve chest.
2.	Double joint.
3.	Front stay.
4.	Bottom spindle.
5.	Valve gear levers.
6.	Front stay bush.
7.	Ball crosshead.
8.	Main crosshead.
9.	Crosshead pin.
10.	Piston rod.
11.	Piston body.
12.	Piston rings.
13.	Cylinder cover.
14.	Discharge valve seat.
15.	Discharge valve seat ring.
16.	Suction valve seat.
17.	Suction valve guard.
18.	Discharge valve guard.
19.	Water valves.
20.	Bucket.
20A.	Bucket rings.
23.	Valve chest cover.
26.	Auxiliary valve spindle.
27.	Pump bottom cover.

FIG. 205.

In conclusion, it might be said that the latent heat from *one* pound of steam will heat *ten* pounds of water by nearly 100 degrees. So all the water the boiler uses can be heated as hot as this exhaust steam will heat it with but a small part of the exhaust (it runs from 12% to 16%, depending on how cold the tank water may be). The remainder is amply sufficient to produce the required draft without any changes in the nozzle.

The water can be heated in this way to 220 degrees if the back pressure is 5 pounds, and then enters the boiler at that temperature instead of 60 degrees. This, of course, simply means that the heat that would otherwise be supplied by the coal to heat it from 60 to 220 degrees is saved; this is about 160 British Thermal Units per pound of water, or well over 10% of the heat from the coal.

BOILER FITTINGS.

THROTTLE VALVES.

The Alco Throttle.

The present design of the Alco throttle was first brought out by the American Locomotive Company to facilitate entrance into locomotive boilers through the main dome, thus eliminating the separate inspection manhole. A separate inspection manhole

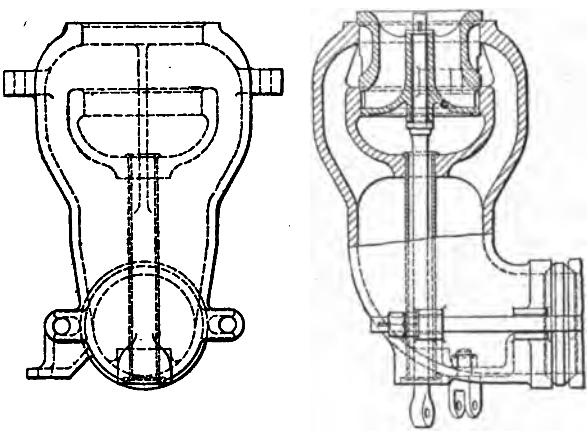


FIG. 206.

adds another large hole on the top center line of the boiler, and added weight and complication of the boiler design became necessary in order to maintain the proper factor of safety.

In operation and maintenance this throttle does not differ from the usual double seated valve.

However, by operating the throttle valve bell crank by a shaft extending through the side of the dome, the operating lever is removed from the backhead to the outside of the firebox, permitting location of the lever in the most convenient place for the engineer, and giving additional room on the boiler backhead for fittings. This outside type of valve is shown in detail in Fig. 206, while the ordinary type for inside connection is illustrated in Fig. 207.

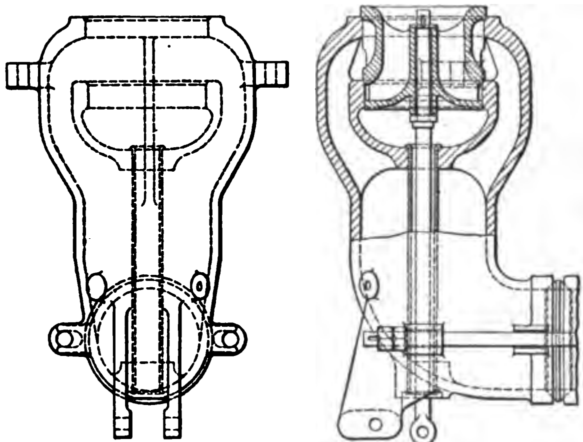


Fig. 207.

This arrangement also allows the use of the Alco operating lever, which gives a maximum slow opening force on initial movement, which changes on further opening to a rapidly increasing throttle lift for any given lever movement.

Description.

Although the following description applies to the Alco throttle and operating lever combined, it should be remembered that the Alco throttle itself may be operated by any suitable operating mechanism.

The essential features are as follows:

1. The usual double seated throttle valve.
2. A vertical stand pipe.

3. A horizontal operating shaft extending through the side of dome.

4. A specially designed operating lever and quadrant.

The throttle valve itself does not differ from the usual double seated type in common use. However, a modification has been made from the usual "L" head type stand pipe to a construction having the throttle valve on the same center line as the stand

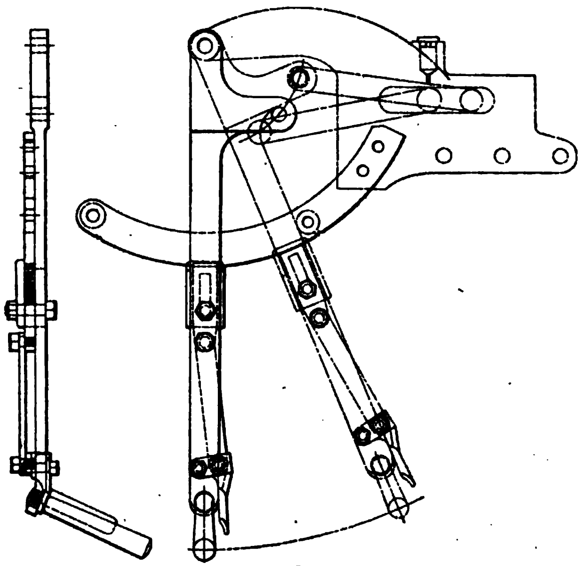


FIG. 208.

pipe. This arrangement, by reducing the space taken up by the stand pipe, permits easy entrance to the boiler through the main dome. In order to further simplify and compact the general design, the throttle valve is arranged to lift from the bottom.

An arrangement of links and bell crank operates the throttle valve from a horizontal shaft which extends through the side of the dome. This shaft is supported by ample bearings and is arranged with a suitable stuffing box. Motion in this stuffing box being rotary, instead of sliding, greatly reduces the difficulties experienced with the usual throttle rod stuffing box.

The specially designed operating lever is arranged with an offset connection through links to a sliding block in such a manner that a maximum opening leverage is obtained with a slow initial throttle lift, as is clearly shown in Fig. 208. As the movement of the lever continues, the leverage decreases and the rate of throttle lift rapidly increases, since after opening, the throttle valve becomes practically balanced and little power is required to continue it's movement.

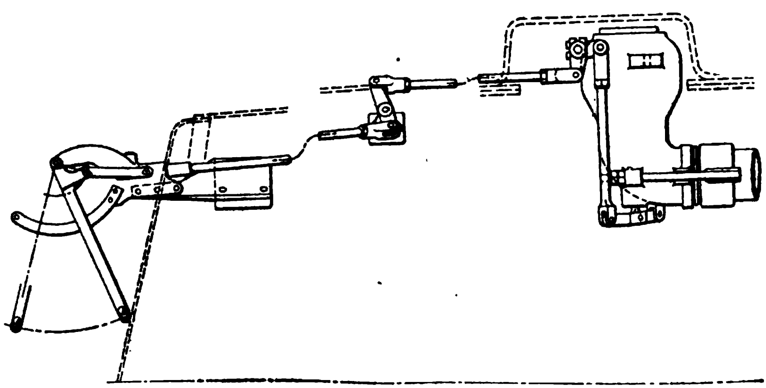


FIG. 209.

The throttle lever bracket, sliding block guide, and quadrant support are integrally combined in a form best suited to the engine design. Very often the lever is placed in a vertical position very convenient to the engineman and also giving maximum clearance in the cab. This entire bracket is fastened to the outside of the firebox and gives a very rigid and secure support to the entire mechanism.

In order to offset the trouble experienced with long throttle rods due to boiler expansion, a rocker is introduced at approximately the center of the throttle rod. We have illustrated this method of application in Fig. 209. This construction equalizes the effect of expansion and prevents any tendency to unseat the throttle valve.

Application.

The application of the Alco throttle is not confined to any particular size or type of engine. However, in combination with the Alco throttle lever, it is particularly adapted to large engines with limited cab room, especially if the engine is equipped with an automatic stoker located on the backhead.

Chambers Throttle Valve.

The Chambers Throttle Valve is designed to meet the requirements of throttle control in large modern power, especially on superheated steam locomotives. It is designed to overcome the difficulties encountered in the increasing unbalanced areas of double-seated throttle valves, and to provide a fully steam-balanced, single-seated valve, which practically eliminates regrinding, thereby insuring a tight throttle; a valve which will take all the steam at a high point in the dome and in such a manner that it will not syphon water into the dry pipe and superheater units; a valve which need not be removed for internal boiler inspection and having all parts standardized and interchangeable, and readily accessible without removal from the boilers; a valve which has a positive and flexible drifting feature.

The Chambers Throttle Valve, in its present form, Fig. 210, has been adopted as standard on all classes of locomotives authorized by the United States Government.

Description.

It is composed of five principal parts, which are Throttle Box, Removable Ring Seat, Main Valve, Balancing Drifting-Valve, and Balancing Piston.

In the Throttle Box there is a chamber which, for convenience, is called the Balancing Chamber, in which the balancing piston operates. The Ring Seat, when removed, allows the balancing piston, which has the same area as the Main Valve, to be inserted in the Balancing Chamber. The Ring Seat is then in-

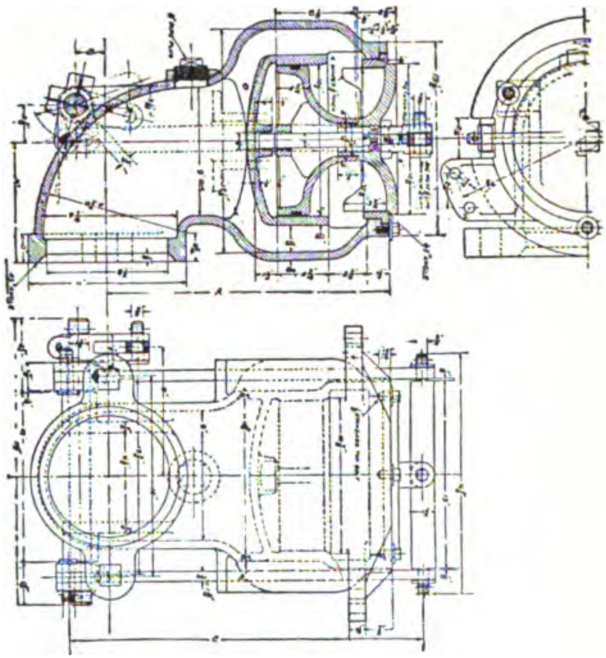


FIG. 210.

serted, making a joint on the shoulder, near the top of the box. The Main Valve is then inserted, seating on the ring. The Balancing Drifting Valve is then secured to the valve stem in such a manner that it is steam tight and is seated upon the Main Valve.

Operation.

The Throttle Valve is operated by means of a rod connected to a crank arm located inside of the boiler. As the Throttle

Lever is moved from closed position, the crank arm moves backward, and, in so doing, actuates the inside lifting arms, with a corresponding movement upward. This, in turn, raises the side bars, which connect with the top cross bar, and to which is attached the Balancing-Drifting Valve. This valve is first raised 1/8", before the Main Valve begins to open, and by means of the valve stem, which raises simultaneously, the upper face of the balancing piston, is brought in contact with the provided shoulder on the Main Valve. The Balancing-Drifting Valve having an unbalanced area of only 2¼ square inches, a very slight pull on the throttle lever is required to open it. As this action takes place, steam is admitted under the Balancing Piston and fully balances the main valve.

Continuing the movement, the Balancing Piston, which has already moved up to a contact, engages the Main Valve, which in turn is lifted from its seat on the balancing ring.

In closing, the Main Valve seats before the Balancing-Drifting Valve. The steam used in drifting is that which leaves the Balancing Chamber through a small hole in the wall of the balancing piston, thence to the superheater and cylinders.

This device is manufactured by the Chambers Valve Co., of New York City.

LOCOMOTIVE INJECTORS.

Introduction.

An injector is a device through which the energy of a jet of steam is imparted to a more slowly moving body of water, forcing the latter into the boiler, against the weight of the water and the pressure of steam contained in the boiler.

This ingenious device is the invention of an eminent French engineer, Henri Jacques Giffard, and was first patented by him on May 8, 1859. The knowledge of the capacity of a moving jet of steam to produce a vacuum in properly formed ducts, in order to raise water and convey it from one place to another, may be traced back long before that date, so that, strictly speaking, he was not the original inventor of steam jet instruments in general. What he really invented, or discovered, was the detail of the outlet orifice, or overflow space, which, for boiler feeding purposes, is invaluable, and which is usually located between the discharge end of what is termed the condensing nozzle and the receiving end of what is known as the delivery nozzle.

In starting an injector, more water, as a rule, enters the instrument under certain conditions than the injector is capable of delivering against back pressure of the steam in the boiler, and if there were no overflow, or communication to the atmosphere, which permits the surplus water to escape, the accumulation of water in the nozzles would disrupt the continuity of the jet and prevent the prompt starting of the injector. This disruption of the jet, popularly termed the "breaking," or refusal of the injector to work, would result in the steam being blown back into the tank from which the water is taken, because the steam would naturally follow the line of least resistance. By providing for an overflow space between the condensing and delivery nozzles, the surplus water is afforded an opportunity to escape, until the jet of combined steam and water has attained a sufficient velocity and over-pressure to open the boiler check, and deliver the jet into the boiler. After the apparatus has been started, and is in operation, the waste of water may be avoided by cutting down the supply, until no more water is seen at the overflow, or the overflow is stopped automatically, according to circumstances and the type of injector used.

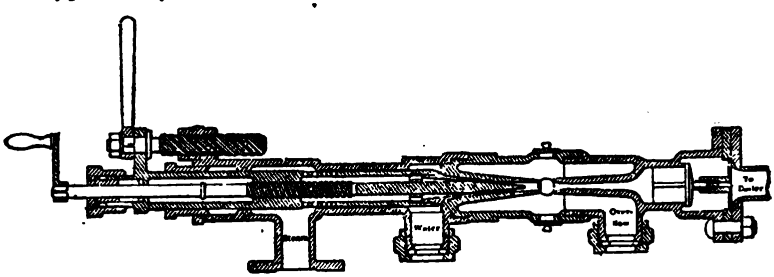


Fig. 211.

The underlying principles concerning the operation of the instrument were so well understood by Giffard, that his original model, as illustrated in Fig. 211, was remarkably similar, even in such detail as the nozzle proportions, to the various forms in use today. Giffard was, therefore, deserving of the grand mechanical prize for 1859, awarded to him by the Academy of Science, of Paris, France.

During the same year (1859) the injector was introduced into England by Sharp, Stewart & Co., while the American patents were given to William Sellers & Co., of Philadelphia, Pa., who commenced the manufacture of the injector in 1860.

Improvements in the general design and in the constructive details of the nozzles and operating mechanism rapidly developed, so that today the manufacture of injectors forms a conspicuous branch of industrial activity.

Theory of the Injector's Action.

There is a difference of opinion regarding the theory of the injector's working, one theory being that the high velocity of the steam, as it issues from the boiler, strikes the column of water, and, mingling with it, carries the water along with it into the boiler. This mixture forces its way into the body of water in the boiler, which, though under the same pressure as the steam which operates the injector, is a passive body and cannot resist the velocity of the inflowing water.

This is, in brief, the "velocity" theory, which is the one generally accepted by engineers and professors, and which can be enlarged upon as much as desired by the use of figures dealing with kinetic energy, etc., if desired.

There are a few, however, and among them the makers of well-known injectors, who do not agree with this theory, and they appear to have good grounds for their opposition. They argue that if the velocity theory were correct, injectors could be made with the steam and discharge tube of the same diameter (or with the steam tube possibly a trifle larger, to allow for friction of steam before reaching the water), as the velocity would be there just the same, to force the water into the boiler. In reality, an injector will not work under these conditions, but the area of the steam tube must be considerably in excess of the area of discharge tube, and this difference in area will be found to vary as the difference between the steam pressure and the pressure the injector forces against increases.

The experience of those who oppose the generally accepted theory is, that the injector's action may be likened to that of a steam pump, the forcing of water being dependent upon the excess of total pressure in the steam tube over that opposing the flow of water from the discharge tube.

These theories have long been the subject of debate, but we do not believe that an extended discussion of the subject in this work would be appropriate, or beneficial.

Definitions and Classifications.

The injector is an apparatus for supplying water to steam boilers and it performs its function usually by means of three nozzles, which are: (1) a steam nozzle, through which the steam from the boiler enters the injector; (2) a combining, or condensing nozzle, in which the steam and water meet, the steam condensing and transmitting its energy to the stream of water; (3) a delivery nozzle, in which the combined mixture of water and steam attains its maximum velocity, which is subsequently reduced by increasing cross sections, to the velocity and pressure in the boiler feed pipe.

Generally, injectors are divided into two classes: "Non-lifting" and "lifting."

Non-lifting injectors cannot raise the water, which must therefore flow from the supply tank to the injector by gravity.

Lifting injectors both raise the water and force it into the boiler. They should be located just above the highest level of the water in the supply tank.

Injectors may further be sub-divided into following classes:

1. Single jet injectors containing a single set of nozzles.
    a. With central movable lifting nozzle attached to the operating lever.

b. With an independent lifting arrangement.

2. Restarting injectors.

3. Double jet injectors containing two distinct and separate sets of nozzles, one set for lifting the water and the other for forcing it into the boiler.

The single jet injector with central lifting jet has the advantage of simplicity of construction and operation.

The independent lifting arrangement involves a few additional parts and separate operation of the lifting valve, but these seeming disadvantages are more than offset by the almost absolute reliability of starting or priming under the most adverse conditions. This is due to the fact that this separate lifting arrangement is entirely independent of and therefore uninfluenced by any other part of the injector.

The restarting feature becomes extremely desirable when a temporary interruption of the water supply occurs. This may take place when the water rolls considerably, with low water in the tank, under which conditions the restarting injector picks up the water automatically as soon as the supply is restored, without the necessity of first shutting off and then starting again by repeated manipulation.

The self-adjusting feature is of undoubted advantage and utility, making hand adjustment of the steam and water unnecessary.

Supplementary inlet valves which admit a supply of cold water from the suction pipe directly into the overflow chamber, cooling the nozzles and thereby tending to prevent formation of scale, are very desirable, but should be combined with an emergency cut-off so as to make it possible to counteract any unexpected defect in the operation of these valves.

Injectors may further be classified as open overflow injectors which have one or more ports in the combining nozzle, opening into the overflow chamber, these being closed against the admission of air by the use of check valves, usually called "heater cock checks," and into closed overflow injectors which can be started only by means of an opening placed beyond the delivery nozzle, which opening must be closed forcibly before the jet can be diverted into the boiler.

The capacity of an injector is measured in gallons of water per hour that it can deliver to the boiler at a certain pressure and under certain conditions of operation, the maximum capacity being the greatest volume or weight of water which can be discharged through the delivery nozzle at any given steam pressure and condition of feed, and the minimum capacity, the least volume or weight of water that can be continually delivered without waste through the overflow. This is often expressed in a percentage of the maximum capacity.

The numbers which may be seen on most injectors indicate the exact diameter of the smallest orifice of the delivery tube, expressed in millimeters, which is equal to .03937 inches.

Action of the Injector.

The action of the injector is as follows: A jet of steam is admitted through the steam nozzle to the combining and condensing nozzles. The water that has been brought to the injector by priming condenses this jet. The steam issues from the steam nozzle at a high velocity. When the water condenses it, its velocity is imparted to the water, and the water and condensed steam pass from the combining and condensing nozzles to the delivery nozzle.

The impact or blow of the steam exerts against the underside of the boiler check a pressure sufficient to overcome the boiler pressure acting against the upper side, the result being that the water lifts the check valve and enters the boiler.

In short, the action of the injector is the resultant of the high velocity with which the jet of steam strikes the water entering the combining tube, transferring to it its momentum and forming with it a continuous jet having sufficient velocity to overcome the pressure of the boiler.

In order to give the reader a more thorough understanding of the injector, special reference will now be made to the various well-known types which have met with such general approval, and the development of which has advanced simultaneously with that of general motive power.

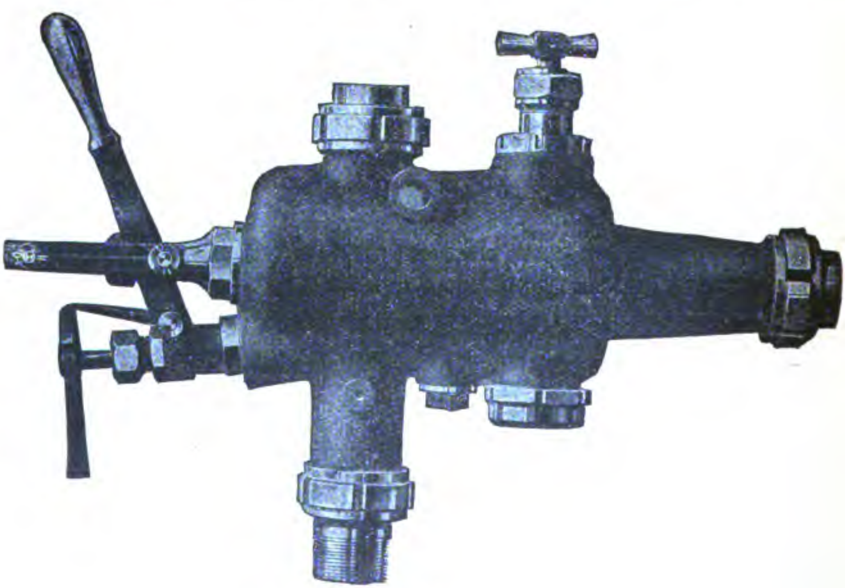


FIG. 212.

The Sellers Injector.

We herewith present two standard types of the Sellers improved self-acting injectors, which possess all the features of this

FIG. 213.

firm's well-known injector of 1887, with the added advantage of largely increased capacity at 200 and 250 pounds steam pressure, which are applicable to all kinds of steam boilers. These injectors should be set horizontally, and may be placed on a locomotive wherever it is most convenient to the engineer. The class M type is shown in Fig. 212, and the class N by Fig. 213.

Both forms are of the restarting, self-adjusting type—that is, if the water supply be temporarily interrupted the injector will start automatically when the supply is resumed, and no regulation of the water supply is required to prevent overflow above 40 pounds.

It is claimed that they will lift promptly even when the suction pipe is hot, and at 10 pounds steam they can lift the water supply two feet; at 30 pounds, five feet; and at high pressures, twelve to twenty feet.

They are interchangeable with older types, and the tubes and other parts can be easily removed for cleaning or repairs.

Method of Operating.

To Start—Pull out the lever slowly.

To Stop—Push in the lever slowly.

Regulate for quantity with the water valve. To use as a heater, close waste valve and draw starting lever. *In starting on high lifts and in lifting hot water,* pull out the lever slowly.

These injectors are made by William Sellers & Co., of Philadelphia, Pa.

Monitor Injector, Type "XX."

The general characteristics of this type of instrument are the same as those of the Monitor Injector Type "R." In the type "XX," however, removable seats are provided for the steam and lifting valves, making the body practically indestructible. The water valve in this type of instrument is an ordinary screw valve of simple and positive construction.

Operation.

Referring to Fig. 214, the action is as follows: Valve 113 is opened first, which admits steam into the lifting tube 118-A and the lifting steam nozzle 118-B. The steam passing through nozzle 149 creates a vacuum in the injector body and suction pipe, which draws water from the tank into the body. This water passes through the nozzle, heater cock check 134, and finally through the overflow 149. When this takes place valve

108 is opened, admitting steam into the forcing steam nozzle 125. The steam mixing with the water, forces the same through the nozzles, 126, 127 and 128, and past the line check valve 131 into the boiler. The valve 113 is then closed, and the quantity of water required is regulated by means of water valve 119.

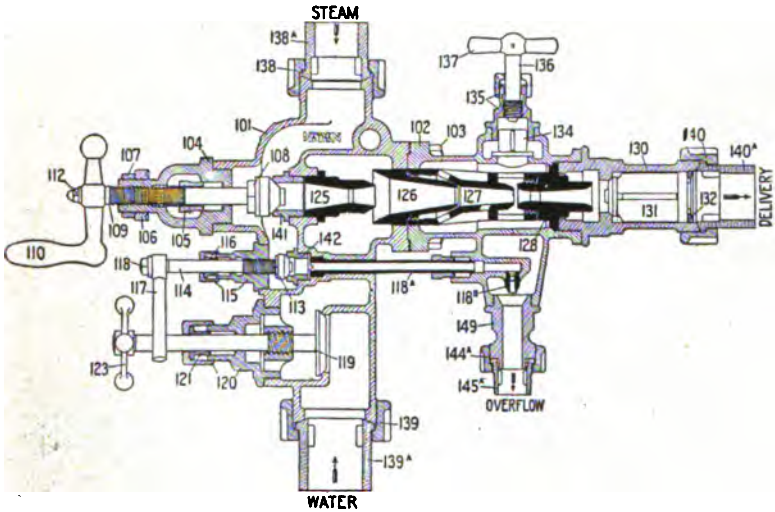


FIG. 214.

The heater cock check 134 should be open except when it is desired to warm the water in the tank. In that case, spindle 136 is screwed down on top of this check to keep it to its seat, and the steam valve 108 is slightly opened.

This injector is also provided, as are all Nathan types of lifting injectors, with an oiler plug at the suction end for the reception of an oiler to lubricate the nozzles in bad water districts.

This injector is a product of the Nathan Manufacturing Co., of New York, N. Y.

Simplex Injector, Type "R."

This type of injector meets the most severe requirements of modern locomotive practice. It is simply constructed and contains only a few operating parts, as may be seen from the illustration, Fig. 215. It is self-regulating; that is, after being started at the highest operating pressure, the latter may drop down to

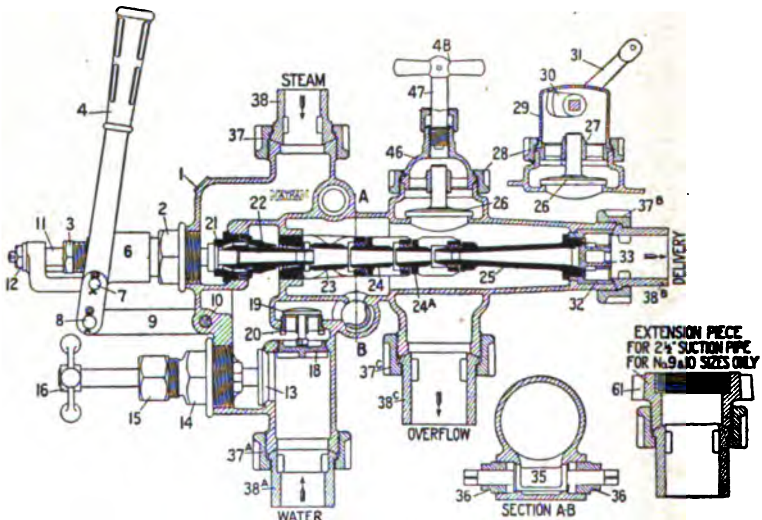


FIG. 215.

about forty pounds before there is any waste at the overflow. It is also restarting, that is, if from any cause the supply of water should be temporarily interrupted, the injector restarts automatically as soon as the water supply is restored. The reducing capacity is 50% of the maximum capacity under ordinary variations of lift and feed water temperatures.

Operation.

The action is as follows: Steam from the boiler is admitted to the lifting nozzle 22 by drawing out the starting lever 4 slightly, and without withdrawing the plug on the end of the steam spindle 11 from the steam nozzle 21. Steam then passes through the small openings around the steam nozzle, and discharges into the overflow chamber, lifts the heater cock check 26 and issues from the overflow to which the overflow pipe is attached.

When water appears at the overflow, the lever 4 is drawn back as far as it will go, which opens the steam nozzle 21 and allows the full supply of steam to enter the intermediate nozzle, forcing the water through the delivery nozzle 25 into the boiler.

At high steam pressure a vacuum is produced in the overflow chamber which draws an additional supply of water into the nozzles through the inlet valve 19, and through the supply openings between the nozzles, which additional water is forced into the boiler, thereby increasing the capacity of the injector under ordinary conditions of operation.

In other injectors provided with the inlet valve, the injector does not prime properly, or not at all, if for some reason this valve leaks, but in all the Simplex injectors the cut-out or emergency valve 35 is provided, which in such cases enables the inlet valve to be cut out and the injector to be operated until there is an opportunity to grind in or otherwise repair the defective valve.

The quantity of water needed is regulated by means of the water valve 13.

The heater cock check 26 is closed down only when it is desired to warm the water in the tank, in which case it is accomplished by the screw handle 47. At all other times the heater cock check 26 must be allowed to open to its full extent.

This injector is also manufactured by the Nathan Mfg. Co.

Nathan's 1918 Special Non-Lifting Injector.

In its general characteristics, method of operation and results, this type of injector is the same as the type "H-W" non-lifting Simplex injector.

However, in the 1918 body, Figs. 216 and 217, the flanges are screwed on to the body, in place of being cast on. Also, the

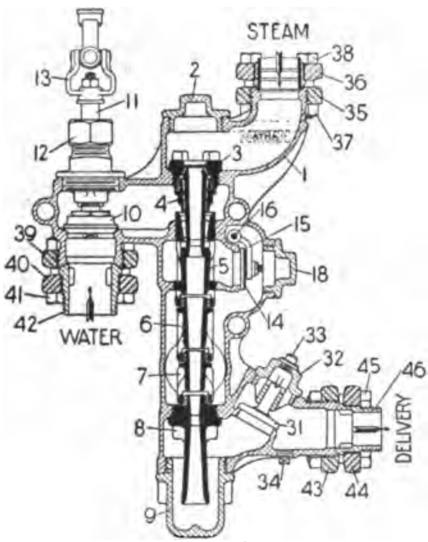


FIG. 216.

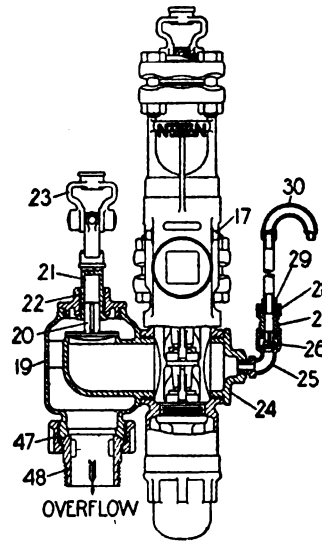


FIG. 217.

steam admission branch is offset on one side of the body and the top of the body is covered by the cap, 2, so that the steam nozzle, 3, may be taken out without disturbing any pipe connections.

This type is also provided with a tell-tale arrangement consisting of a check valve, 26, placed in an elbow, 25, and attached to the side cap, 24, on the body, on the opposite side from the overflow chamber, 19. From this tell-tale a small pipe is run into the cab, terminating in an elbow nozzle in front of the operator. The purpose of this arrangement is to indicate to the operator when the injector flies off for some reason that this occurrence took place and that the injector is not operating properly. When the injector flies off, part of the steam escaping through the overflow will escape through the tell-tale check, 26, and a small stream will be blown through the tell-tale nozzle, 30, indicating the flying off of the injector. Thus warned, the operator will shut off the injector if the same does not start again automatically, provided that the flying off was caused by a temporary interruption of the water supply.

The action of the injector is readily seen by studying the illustrations. Types A and B are similar, except that type B is of larger size and greater capacity than is type A.

This model is another manufactured by the Nathan Mfg. Co.

The Hancock Inspirator.

The Hancock Inspirator is a double tube apparatus, working with a positively closed final overflow valve, and was first manufactured in Boston, Mass., in 1874. It consists of one apparatus for lifting or priming and another for forcing, each having its own steam nozzle and combining tube, so arranged that the lifter delivers the water to the forcer under pressure. This feature may be clearly seen by an inspection of Fig. 218.

The fundamtental advantage of this construction, obtained with no other design, exists in the automatic regulation or governing of the water supply. Both steam nozzles receive approximately the same steam pressure when the inspirator is working, the lifter tubes raising the water and delivering it under pressure into the intermediate chamber. The forcer receives this water and delivers it into the boiler. If the steam pressure rises, the lifter tubes will lift and deliver more water to the forcer, but inasmuch as there is a correspondingly greater amount of steam passing through the forcer nozzle, proper care is taken of this increased supply of water.

As a result, the Hancock Inspirator has an extraordinary range of steam pressure, working equally well at 35 or 350 pounds pressure without any adjustment whatever.

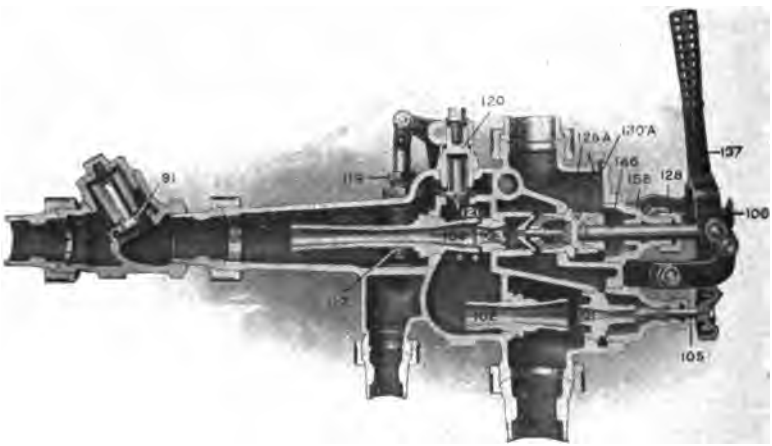


FIG. 218.

In addition to this advantage, the receiving of water by the forcer side under pressure makes the inspirator much less susceptible to "breaking" under severe conditions, such as high suction temperatures or excessive back pressures.

The positively closed final overflow, of course, makes it impossible for any water to "spill" or be lost while in operation, thus insuring the delivery of all the water into the boiler. As the overflow nozzle is coupled to the overflow pipe, escapement of steam in the cab is avoided.

Operation.

To start the inspirator, pull back the lever No. 137 into the first or lifting position. This will draw the lifter valve No. 130 from its seat and admit steam through the forcer steam valve No. 126 to the lifter nozzle No. 101. The discharge of steam from this nozzle into the lifter combining tube No. 102 entrains the air in the suction pipe, finally producing sufficient vacuum to lift the water. The flow of water passes through both the intermediate overflow No. 121 and the forcer combining tube No. 104 and out of the final overflow No. 117. A further movement of the lever opens the forcer steam valve No. 126 and admits steam to the forcer steam nozzle No. 103, while at the same time the final overflow valve is approaching its seat, producing a consequent increase of pressure in the delivery chamber. This pressure closes the intermediate overflow valve and when the final overflow valve No. 117 is brought to its seat the inspirator will be in full operation.

The intermediate overflow valve operates automatically and its function is to afford relief to the lifting portion of the apparatus when priming.

The variation in the capacity of the inspirator is accomplished by means of the regulating valve No. 105. This valve reduces or increases the amount of steam admitted to the lifting nozzle of the apparatus.

To use as a heater, disengage the rod No. 106 from the stud in the lever No. 137 and pull it back until the final overflow valve is closed. The operating lever No. 137 is then pulled to the priming position, the amount of steam necessary for a heater being governed by the regulating valve No. 105. If desired, the operating lever No. 137 may be pulled back to its limit of travel, thus closing the final overflow valve; and the amount of steam necessary for a heater be adjusted at the throttle on the fountain.

This apparatus is made by the Hancock Inspirator Co., and is licensed for use by Manning, Maxwell & Moore, of New York City.

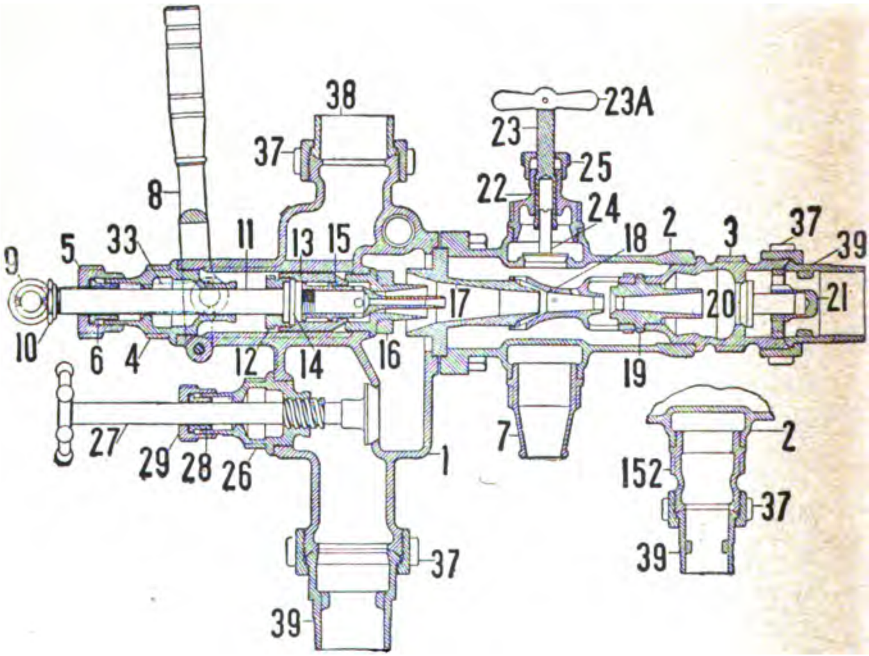


FIG. 219.

The Ohio Injector.

We present a sectional diagram of the Ohio Standard injector, type A, Fig. 219, which will serve to make clear to the reader its details of construction and operation. It is of the lifting type, and is supplied with the necessary details which make it usable as a heater. This is accomplished by closing the overflow valve

24, by means of its handle 23A, and then pulling back the lever 8 until the resistance of the primer 15 is felt, and let stand at this

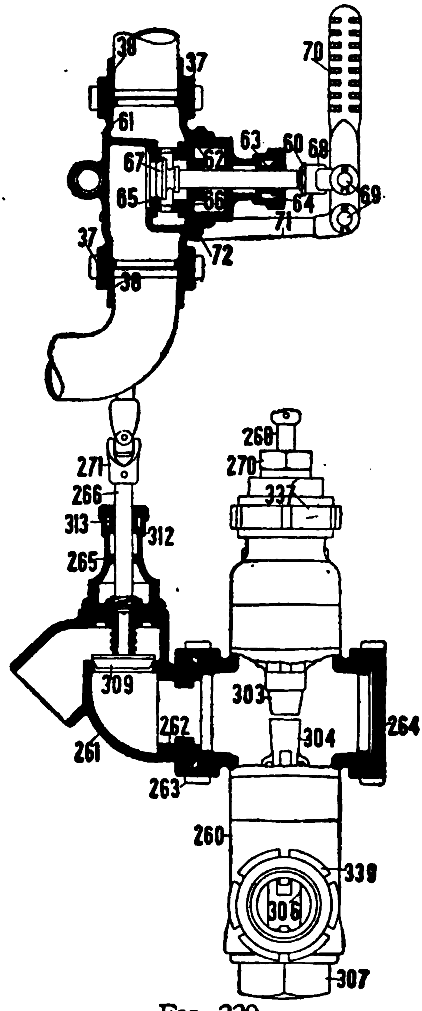


Fig. 220.

point. However, the overflow valve must not be closed *except* when the injector is to be used as a heater.

The Chicago Injector.

This injector is of the non-lifting type, as shown by Fig. 220, with lever steam throttle valve, and it is of importance that the injector should always be placed below the bottom of the water tank, so that the water can flow to the injector, and a strainer with an area of not less than twice that of the water pipe should be used in connection with it. If the injector is used with a siphon in the tank rather than a tank valve, all joints must be made perfectly tight, in order to get the desired result.

Both the Ohio and Chicago injectors are made by the Ohio Injector Co., of Chicago, Ill.

The Edna Injector.

The Edna injector, type L, as shown in detail in Figs. 221 and 222, is very extensively used on locomotives. It is of the lifting type, and this, together with the nozzle tubes' special construction, insuring long service, makes it practical for service conditions on the locomotive.

It is made in all usual sizes, and, if required, is furnished with universal joints and extension rods for mounting outside the cab.

Operation.

To start the injector in operation, pull lever 4, Fig. 222, out slowly, and to stop, push the lever in until it stops. If it is wished to use the instrument as a heater, close the heater cock by means of the handle 48, or, if using cam attachment, move the cam lever, 31, to its closed position. This is the position

FIG. 221.

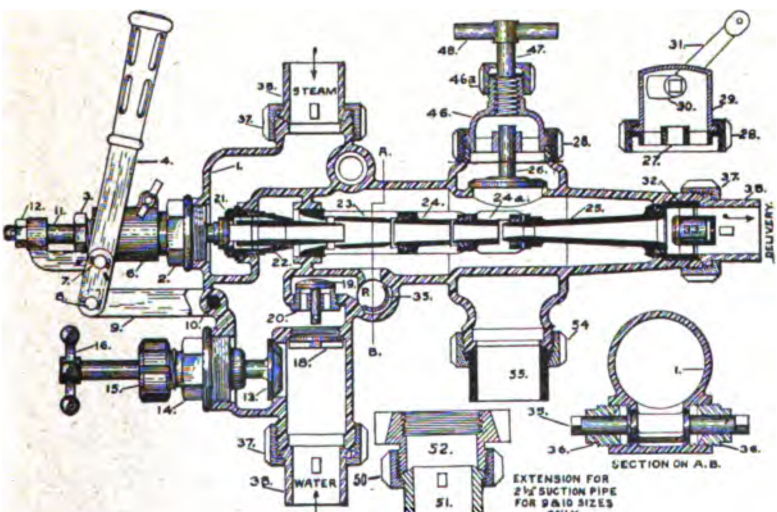


FIG. 222.

when the handle is pointing straight back. Now pull out handle 4 until the desired amount of steam is admitted to the feed water, or suction pipe.

Another type of injector made by the Edna Brass Mfg. Co., is that illustrated herewith, Figs. 223 and 224, known as type M. It is also of the lifting type, adaptable to locomotive service.

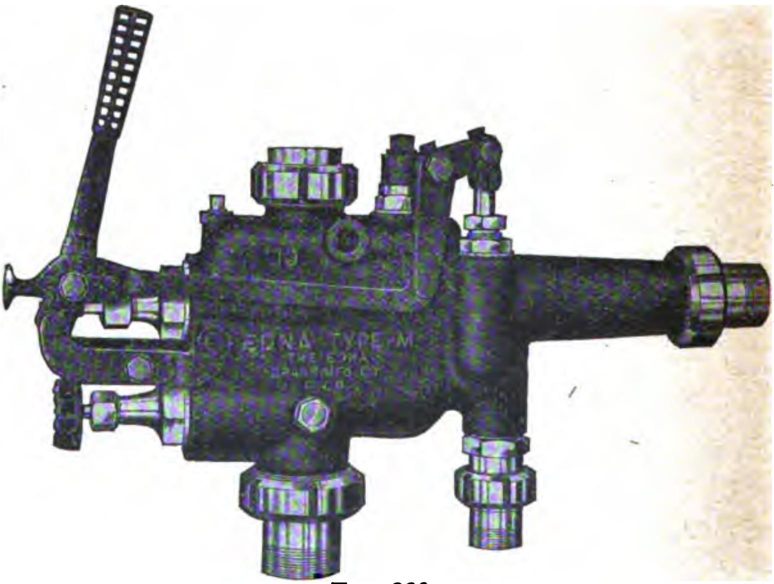


FIG. 223.

This instrument interchanges with the Hancock Inspirator, type A, and all its parts, also, are interchangeable.

Extensions for mounting outside the cab may also be obtained for this type.

The compound forcer steam valve, 126 A, is an improvement, designed to interchange with the old style valve 126.

An unusually wide range of steam pressure may be used with this particular instrument, as it operates equally well at 35 or 350 pounds, without adjustment.

To start the injector, draw back lever 137 to lifting position, until water appears at the overflow. Then draw the lever back until it stops. To stop, move the lever forward until it stops. The type M may be used as a heater by lifting the connecting rod 106, and drawing it back until the overflow valve, 117, is closed. Then draw back lever 137 to lifting position. This, in most cases, will give all the steam necessary for the heater.

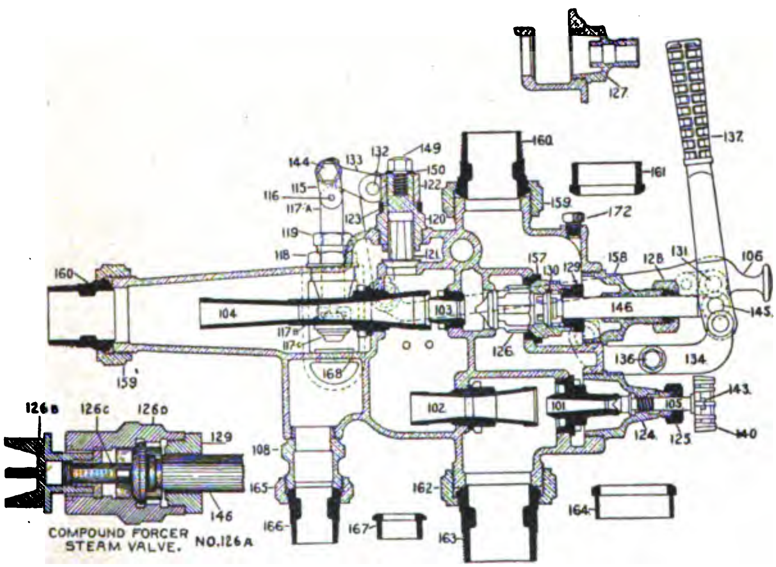


FIG. 224.

This injector, as well as all the other types, should be placed at a point above the highest water level. Steam should be taken from the highest point of the boiler, as dry steam gives the most satisfactory results.

The Edna Alarm Valve.

The Edna Alarm Valve, Fig. 225, is a "safety first" valve, being provided with a closed drain valve which automatically closes when for any reason the injector should break or not work. This prevents any escape of steam at the alarm valve.

This alarm valve can be used on any non-lifting injector by putting suitable pipe connections in the branch pipe. The alarm valve body, 2, can be made to suit any connection.

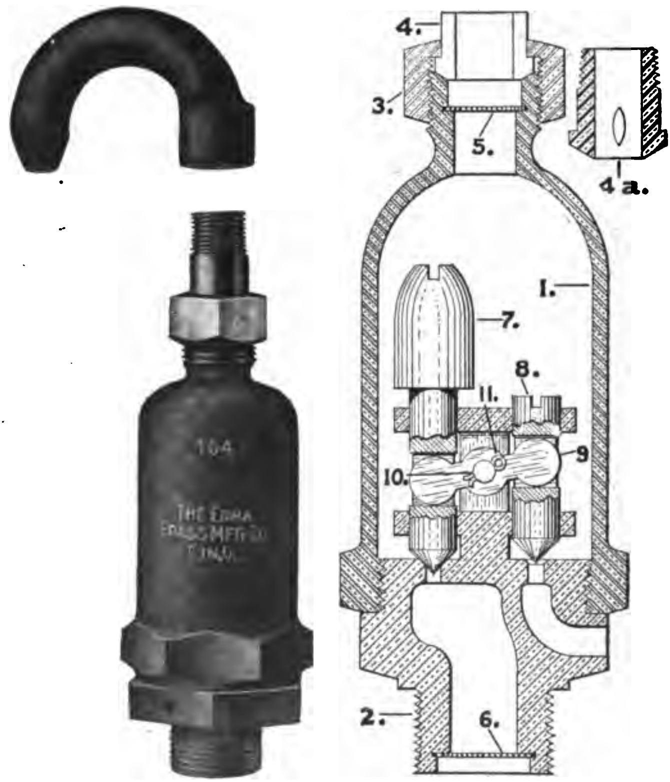


FIG. 225.

This valve will instantly show, by a small jet of steam blowing in the cab, that the non-lifting injector has failed to work, thus giving the engine crew instant warning that the injector is not working, thereby preventing any chances of injury to the boiler, or delay to the train.

Direction for Application.

Place the alarm valve on the injector, when provided for, if not provided for on the injector body, place it next to the injector, and back of the check valve, if used.

Pipe the valve from the top to a point convenient in the cab where it can be seen by both enginemen. Secure the pipe by suitable clamping. Attach the nozzle, and turn it so that it can be readily seen, and will not blow where it would be objectionable.

The alarm valve is automatic, and will not freeze, being self-draining.

To Repair.

Remove coupling nut 3; also remove alarm valve and hold piece 2 lightly in vice. Remove alarm valve body top. Examine alarm drain valve 8 and grind if necessary. This can be done by lifting drain valve from its place and holding it by hand in opposite direction from its natural position. Grind seat on top of alarm valve body bottom part 2 by dropping alarm valve disc valve in place. Very little grinding is necessary. Care must be exercised not to change proportion of disc valve 4, as it regulates opening of drain valve 8.

Injector Failures.

Injectors are, of course, subject to failures. Difficulties are sometimes experienced in the working of the same, and in discussing the reasons for such failures together with their remedies, it will be assumed that the injector itself is in fairly good condition. It will also be assumed that the suction pipe, hose, strainer, tank valve and boiler check valve are accessories of the injector, as a disorder in any one of these affects the injector's working.

The Injector Does Not Lift the Water.

This may be caused by any of the following defects.

1. *Suction pipe stopped up.*

This may be due to a clogged strainer, a collapsed hose lining, or the pipe itself being clogged at some point. This naturally prevents the water from flowing readily through the pipe and is a frequent cause of the injector not priming properly.

In case the suction pipe itself is clogged, the remedy is to close the overflow valve and open the steam valve just sufficiently to blow steam back through the pipe and force out the obstruction.

In case the trouble is caused by a collapsed hose lining, the same remedy may be applied, and if it is not followed by good results the trouble may be overcome by turning the hose from end to end. The hose should have an easy curve from the tank connection to the suction pipe and should have no kinks in it, as these reduce the area of the opening and obstruct the supply of water.

Any obstruction in the strainer may be removed by taking out the strainer waste plug and allowing some water to pass through, which will wash out the obstruction.

If the water pipe is provided with a strainer in the pipe itself and not accessible from the outside, the hose must be taken down and the strainer removed and cleaned. It is best, however, not to use any of these inaccessible strainers, but to have one either in the tank or between the hose and suction pipe which is accessible and can be cleaned without the breaking of joints.

2. *Suction pipe or its joints leaking.*

This allows air to enter into the pipe and prevents the formation of the necessary vacuum to lift the water. The suction pipe may be tested for leaks by closing the overflow valve and the tank valve, and then opening the steam valve just sufficiently to produce a slight pressure in the suction pipe and hose. The leaks will allow the steam to escape and indicate their location.

3. *Water in the supply pipe or tank too hot.*

This may be caused by a leaky steam valve in the injector, or by a leaky boiler check or line check valve. The obvious remedy for this is to grind in the valves, and re-seat them in addition, if necessary

Injectors with hot water in the suction pipe may be started by closing down the overflow valve just long enough to blow steam back into the suction pipe and force the hot water back into the tank, then quickly open the overflow valve. Cool water then enters the suction pipe, replacing that which is hot.

In case the water in the tank has been heated to such a high temperature by blowing steam back into the tank to prevent the safety valve from blowing, or to warm the water in extremely cold weather, so that the injector does not prime, then the tank cover must be opened and, if possible, cold water added to cool off the hot water in the tank. It may be stated here that the injector handles warmer water better at low pressure than at high pressure, so that should the water be too hot for the standard boiler pressure, the steam may be throttled down until the injector just barely starts with dry overflow, and so supplies water to the boiler at a reduced capacity until fresh water can be put into the tank and the temperature reduced.

4. *Obstructed nozzles.*

The openings between the condensing and delivery nozzles, and the spill openings in the condensing nozzle, may be obstructed by incrustation, preventing the ready exhaust of the air from the interior of the injector in starting. The remedy for this is to remove the nozzles and clean them. If they are badly incrusted they should be placed in a bath consisting of one part of muriatic acid to ten parts of water. Care should be taken that the nozzles are removed from the bath as soon as gas bubbles cease to be given off. As long as there is alkaline matter to combine with the acid, there is no danger of the metal being attacked, but after this is consumed, the acid attacks the nozzles, and the interior surfaces will be pitted and roughened, which affects the working of the injector.

5. Referring particularly to the Monitor type "R" and the Monitor type "XX" injectors, the jet valve tube 18a or 118a may be obstructed. If the obstruction is slight, blowing of steam through the tube by opening valve 13 or 113 may blow it through the lifting nozzle 18a or 118b. If, however, the obstruction is larger than will pass through the opening of the lifting nozzle, it may be necessary to remove the jet valve bonnet 15 or 115 and push a piece of iron or steel into the tube 18a or 118a to break up the obstruction.

It is quite obvious that a lifting steam nozzle which is badly worn or otherwise defective, prevents the proper lifting, and the remedy for this is the removal of the nozzle.

The Injector Lifts the Water, but Does Not Force It Into the Boiler.

It may happen that the injector lifts the water, but does not force it into the boiler when the steam valve of the injector is opened, or it may force part of the water into the boiler and part of it out at the overflow. This may be due to any one of the following causes:

1. *The suction pipe or strainer partly obstructed, hose lining collapsed, or a kink in the hose.*

Under these conditions the injector does not lift sufficient water to condense the steam passing through the steam nozzle. The remedy for these conditions has already been pointed out.

2. *The Tank Valve is partly closed.*

This may be due to oversight or its handle may be so placed by mistake that the tank valve is closed when it is supposed to be open. The obvious remedy is to open the tank valve or put on the handle properly.

3. *Too much or too litte water being supplied to the injector for the prevailing pressure.*

In case of too much water, the same issues from the overflow only slightly warm, and the remedy is to reduce the supply by means of the water valve until the injector works properly and with dry overflow. In case of too little water, the same issues from the overflow at a high temperature mixed with vapor. The remedy is to increase the supply of water or cut down the supply or steam until the injector works properly.

4. *The delivery nozzle of the injector is cut, worn or obstructed.*

In event of the first two defects, replacement is the only remedy. The obstruction in this or in any other of the nozzles may be removed by taking out the steam valve, then inserting a piece of iron through the nozzles to break up the obstruction. If this cannot be accomplished, removal of the nozzles is necessary. Quite frequently it is found that a piece of coal or waste from the tank, or scale from the pipes, is carried into the nozzles, obstructing the passages just sufficiently to prevent the water from readily passing through. For this reason it is necessary that pipes, particularly steam pipes, be thoroughly blown out before they are connected up, and that the tank be kept clean.

5. *Leaks in the Suction Pipe.*

Leaks in the suction pipe may not be sufficient to prevent the injector from lifting the water, but the quantity passing into the injector may not be sufficient to condense all the steam, which has the effect of reducing or disturbing the vacuum in the combining tube.

Another effect of such leaks may be that the water in the combining tube not passing through in a solid stream, is not properly acted on by the steam and the jet does not properly form. If the leaks are only slight, they simply have the effect of cutting down the capacity of the injector, and in such cases it works with a rumbling sound. How the suction pipe may be tested has already been pointed out.

6. *Boiler check valve sticks.*

If the boiler check is stuck entirely closed, the injector raises the water, but throws it out through the overflow and eventually breaks.

If the boiler check is stuck partly closed, then the injector puts some water into the boiler and throws some through the overflow.

In case the boiler check cannot be opened wide, the wasting of water may be stopped or reduced by cutting down the water supply to the injector and working it at a reduced capacity. The reduced supply may then pass through the partially opened boiler check without wasting at the overflow.

When the boiler check is stuck closed it may sometimes be opened by lightly tapping on the cap or on the casing of the boiler check, but this must be done carefully so that the casing will not be injured. Another method of opening the check which may succeed, is to start the injector and gradually close down the water valve to diminish the supply of water and thereby increase the pressure which the steam exerts against the check.

If the boiler check sticks open and the line check valve does not prevent the water from flowing back from the boiler when the injector is shut off, it can sometimes be reseated by opening the frost cock in the delivery pipe or partially disconnecting the union of the delivery pipe and tapping lightly on the cap, or the casing of the valve.

If the above defect cannot be remedied, then the overflow and the water valves of the injector should be shut to prevent the water flowing back from the boiler through the injector. The second injector must then be relied upon for feeding the boiler.

To find out whether a leak through the injector comes from a leaky steam valve or a leaky boiler check, the main steam valve of the injector, located on the boiler, should be closed. If it is the steam valve of the injector that is leaking, this stops the flow of steam at the overflow, but if the leak comes from the boiler check, steam and hot water continue to pass out at the overflow.

If the boiler check is provided with a stop valve, the communication between the boiler and the check valve may be closed, the cap over the check valve taken off, the check valve itself taken out and the obstruction which caused the sticking removed.

7. *Delivery pipe frost cock opening too large.*

In this event the delivery pressure may be so reduced that it is not sufficient to open the boiler check, when waste at the overflow occurs.

8. *Overflow valve leaking.*

A leaking overflow draws air into the injector. If the leak is great enough, it diminishes the capacity of the injector and also disturbs the jet.

The remedy is the removal of the overflow cap and the grinding in of the check.

9. *Tank cover too tight.*

It is necessary that air freely enter the tank. If this is not the case, water from the tank is withdrawn by the injector faster than the air enters. The air in the tank expands as the water is withdrawn, and the pressure exerted by it diminishes. This may take effect to such a degree that the injector does not receive the necessary supply of water, thus reducing its capacity or breaking the injector. It is necessary, therefore, to see that the tank cover is not air tight.

BOILER CHECK VALVES.

It is necessary that, when the injector is not in use, some means be provided to prevent a return flow of water from the boiler. The check valve, or boiler check, is employed to prevent the return flow, for, when the injector is shut off, the boiler pressure acting upon the check valve automatically closes it. The boiler check should fit perfectly tight upon its seat, and should be given the required opening; otherwise the injector is apt to fail, or to give unsatisfactory results.

Some of the check valves in more general usage are described and illustrated herewith, so as to give the reader an understanding of the principles of construction and operation of the modern locomotive boiler check.

The Nathan Combination Boiler Check and Stop Valve.

This type of boiler check has been adopted by the United States Railroad Administration as standard for all of the standardized government engines.

It consists of a stop valve attached to the boiler by means of a large and substantial steel flange No. 19 and ball joint No. 20, arranged as in Fig. 226.

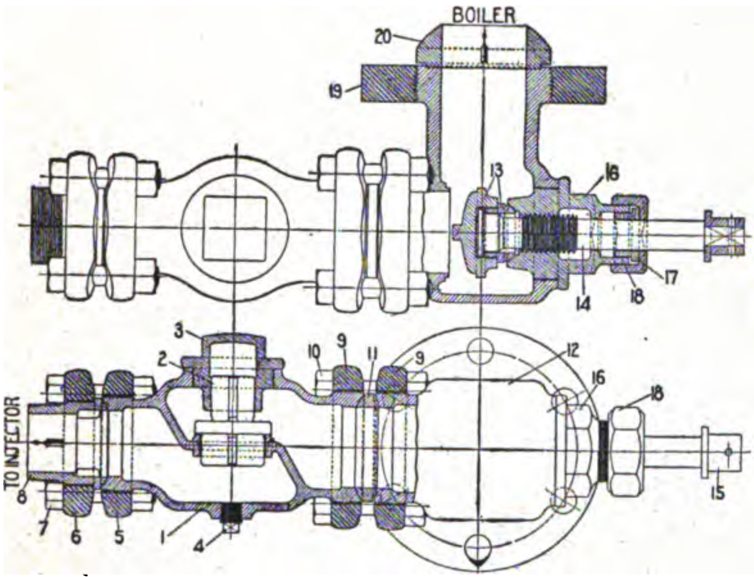


FIG. 226.

No. of Part.	Name of Part.
1	Check Valve Body.
2	Check Valve.
3	Check Valve Cap.
4	Check Valve Drain Plug.
5	Check Valve Body Flange, Inlet end.
6	Check Valve Companion Flange, Inlet end.
7	Check Valve Bolt and Nut, Inlet end.
8	Check Valve Tailpiece.
9	Check Valve Body Flange, Delivery end.
10	Check Valve Bolt and Nut, Delivery end.
11	Check Valve Ball Joint, Delivery end.
12	Stop Valve Body.
13	Stop Valve Disc and Nut.
14	Stop Valve Spindle.
15	Stop Valve Knob and Pin.
16	Stop Valve Bonnet.
17	Stop Valve Gland.
18	Stop Valve Packing Nut.
19	Stop Valve Boiler Flange.
20	Stop Valve Boiler Ball Joint.

The check valve proper is attached to the stop valve by means of flanges No. 9 and joint ring No. 11.

The stop valve is of 2½" size in all cases, whereas the check valve itself is of either 2" or 2½" size, according to whether the check valve is used with lesser or greater capacity injectors, which were previously described. The construction of this boiler check makes it possible to remove the check valve for purposes of inspection, repair or renewal without disconnecting the stop valve part from the boiler which makes the arrangement very convenient for quick adjustment if anything should be the matter with the check valve.

The pipe connection to the injector at the inlet end of the check valve part is also of the flanged type, as are all the pipe connections for the feed water arrangement for government engines.

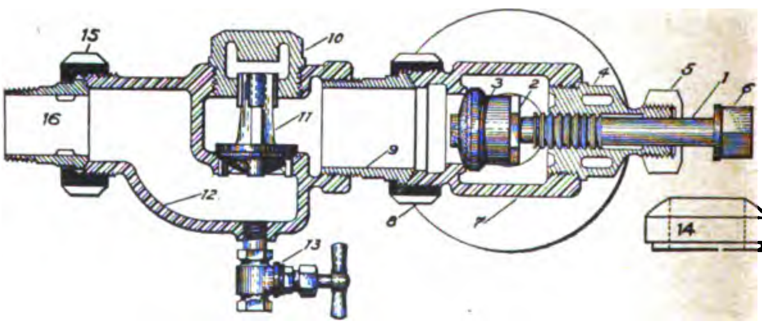


FIG. 227.

The Edna Combined Stop and Check Valve.

The Edna combined stop and check valve is designed to be applied to right or left side of the boiler, for the union between the stop valve and check permits of this application. The stop valve is so constructed that, when it is open, the disc (3) seats on the stop valve bonnet (4) (Fig. 227), allowing the stem (1) of the stop valve to be readily packed with steam in the boiler. When the stop valve is closed, the check valve (11) may be removed completely for repair or regrinding.

The Edna Double Boiler Check Valve.

This valve, Fig. 228, is designed for use when both injectors are placed on the right side of the locomotive, as both feed pipe connections are at right angles with the boiler. It is provided with shut-off valves, so that the check valves and seats may be removed, or ground, while steam is in the boiler. The valve is

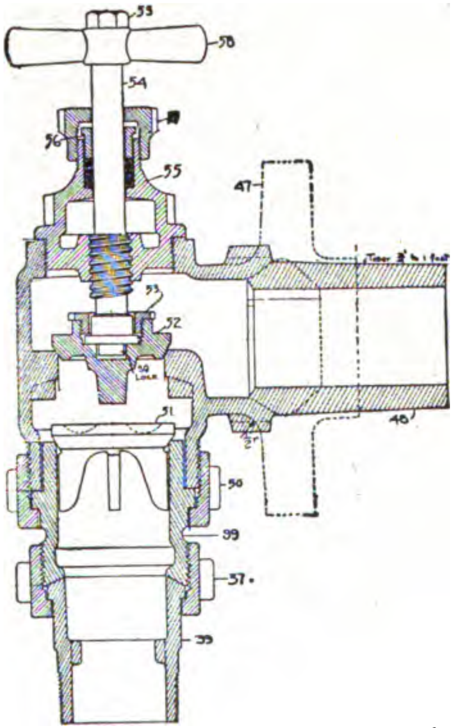
FIG. 228.

also provided with relief plugs, to release pressure which may be trapped between the shut-off valves and checks.

These two check valves are made by the Edna Brass Mfg. Co., of Cincinnati, Ohio.

The Ohio Combination Boiler Check and Stop Valve.

This instrument, made by The Ohio Injector Co., of Chicago, Ill., is furnished with either the straight or the angle delivery casing, and is also equipped, as desired, with a flanged body, or with a body with screw shank. Both arrangements are shown in the illustration, Fig. 229.



List of Parts.

Number
47 Body with Flange
49 Body with Screw Shank
49 Delivery Casing—ANGLE
99 Delivery Casing—STRAIGHT
50 Delivery Casing Union Nut
51 Check Valve
52 Positive Valve
53 Swivel Nut
54 Spindle
55 Hub
56 Packing Gland
57 Packing Nut
58 Handle
59 Handle Nut
37 Coupling Nut
39 Coupling

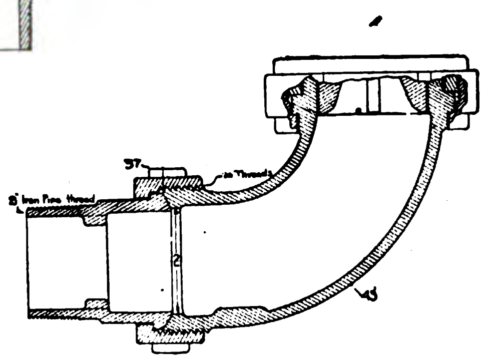


FIG. 229.

The Sellers Combined Check and Stop Valve.

The construction and operation of this device is clearly shown in the illustration Fig. 230. The instrument is furnished with

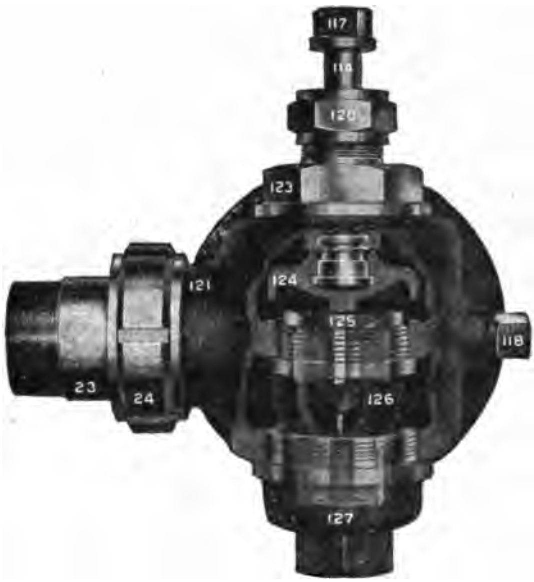


FIG. 230.

either flanged connection or screw shank. The main check valve and its seat can be removed from the body while the boiler is under pressure.

This check and stop valve is the product of Wm. Sellers & Co., of Philadelphia, Pa.

LOCOMOTIVE LUBRICATORS.

In the early days of the locomotive, the cylinders were lubricated by oil plugs, the use of which necessitated the shutting off of steam pressure before the cylinders could be lubricated. It was for this reason that the lubricator, which feeds oil to the cylinders under steam pressure, continuously, came into such general use that, at the present time, all locomotives are equipped with them.

The first attempts were made, between 1875 and 1885, to apply the hydrostatic displacement principle for oiling valves and cylinders of the locomotive. This principle utilizes the pressure due to a head of water, together with the difference in the specific gravities of oil and water, to force the oil from the reservoir to the pipes.

Considerations of convenience and accessibility made it desirable to take the steam supply from the boiler and connect the oil pipes to the top of the steam chest. This mode of connecting up the lubricator resulted in full boiler pressure prevailing at all times at the inlet end of the lubricator, while at the outlet end, the pressure was variable according to whether the engine was running under steam or with throttle closed, as when drifting on a down grade or running into a station.

This condition necessitated the use, and resulted in the invention of what is termed the "Equalizing Feature." This consists in a steam chamber located near the delivery end of the lubricator, which chamber is at all times subject to direct boiler pressure, and from which the steam and oil are permitted to escape through a passage of such small size that the pressure at the outlet is maintained at practically that of the boiler. By this means the pressure at the outlet and inlet of the lubricator are balanced, or equalized, irrespective of the pressure in the steam chest and in the oil pipes.

When the boiler pressure of locomotive engines began to be materially increased, it was discovered that under certain conditions the lubricator did not deliver the oil into the steam chest or cylinder, owing to the fact that the back pressure from the steam chest at the delivery end of the oil pipes was greater than the pressure exerted by the small volume of steam passing through the choke plugs at the receiving end.

Numerous steam chest attachments were introduced to assist the lubricator in delivering oil under adverse circumstances, consisting of automatic valves, which under variations of pressure in the steam chest, vary the opening proportionately, increasing it for the delivery of steam and oil when the engine is steaming and choking it when steam is shut off the chest.

Changes in the conditions of locomotive practice, particularly in continually increasing boiler pressure, necessitated changes in the construction of lubricators to make their operation reliable and safe for enginemen. Glasses particularly received attention, as the tubular form became more and more unreliable and dangerous. A constructive change of vital importance consisted, therefore, in replacing the tubular sightfeed and gage glasses by a new form of glass popularly known as "Bull's Eye."

These glasses have the form of a thick, round disc. They do not break, and even if they should crack, they do not fly around, but remain embedded in their casings, so that the enginemen are absolutely safeguarded against injuries which otherwise might result. The fact that these glasses are safe under high pressure permits the elimination of parts otherwise necessary, reducing the number of parts and, therefore, the cost of maintenance.

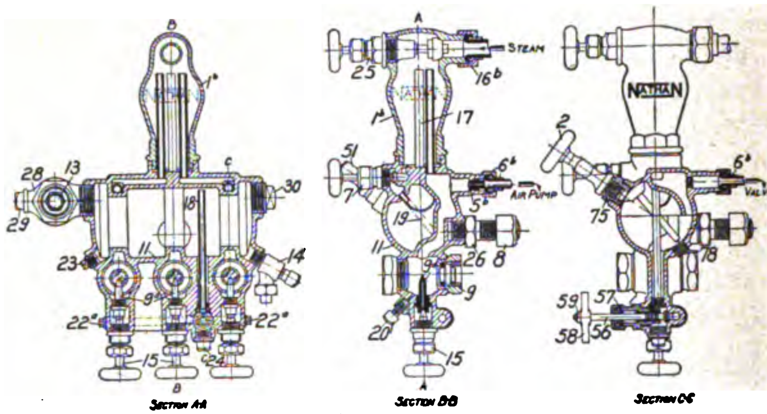


FIG. 231.

The Nathan Bull's Eye Lubricator.

Type "166 F-3."

This lubricator, Fig. 231, contains three feeds, two of which lead to the two steam chests and one to the steam cylinder or the air pump. It is made in four sizes, Nos. 8, 9, 10 and 11, of 2, 3, 4 and 5 pints capacity respectively.

Specific Characteristics.

The lubricator is provided with two filling plugs, one near each end of the lubricator, so that either one may be used as is found most convenient. The filling plugs do not seat on the body, but on removable bushings. In case the seats on the bushings wear out, only the bushings need replacement. These bushings are provided with left-hand threads in the body.

General Description.

The body of the lubricator 11 is made of one single cylindrical casting, with the sight feeds 9 and the regulating valves 15 located at the bottom of the lubricator, all being in one casting. An oil pipe 18 extends from the oil channel, connecting with the regulating valves nearly to the top of the reservoir, and supplies the oil to each of the regulating valves as long as there is any oil in the reservoir. The water pipe 19 extends from the water valve 7 to the bottom of the oil chamber, so that when the water valve is open, water from the condenser passes freely into the oil chamber and transmits to the oil the pressure due to the head of water in the condenser.

The condenser 1 is kept filled with water up to the top of the equalizing pipes 17, which are contained in the condenser, by the condensation of steam from the boiler. Any excess of water passes down these tubes with live steam. These equalizing pipes are screwed into the passages connecting with the outlets from the lubricator and supply live steam to these passages, which keeps them full of condensed water up to the level of the reducing or choke plugs 5, from which point the excess of water, oil and steam leaves the lubricator to pass through the oil pipes to the steam chest and cylinders. This supply of steam from the equalizing pipes also balances the steam pressure on the water in the condenser and on the water in the sightfeed and outlet passages, and the duty of the choke plugs is to restrict the flow of steam from these passages, so that the steam pressure back of the choke plugs is equalized with that in the condenser, whether engine is working steam at full boiler pressure or whether steam is shut off. Three openings are provided to clear out the oil passages to the regulating valves, which openings are covered by the plugs 22A at the ends of the oil channel, and plug 24 underneath the oil pipe 18.

Hand oilers 3 are provided, one for each oil pipe leading to the cylinder or steam chest, which hand oilers are provided with spring covers to keep out foreign matter.

The steam chest oil plug is not furnished with the lubricator, for any standard plug of this character is adaptable. It must be provided at its lower end with a bore of not less than 3/32 inch, or is more than 1/8 inch in diameter. This is absolutely necessary for the proper functioning of the lubricator.

Operation.

The lubricator is first filled with clean strained oil through the filling plug 2, and when the oil chamber is full, the plug is replaced and the water valve 7 opened immediately, irrespective of whether or not the lubricator feeds are started. This opening of the water valve immediately after the cup is filled is very important in order to prevent the bursting of the oil chamber through the force of expansion of the oil as it becomes heated. The steam valve of the lubricator is then opened, which operation fills the sight-feed chambers with water. The lubricator may then be started feeding by opening the regulating valves 15 more or less, according to the feed desired.

To renew the supply of oil after it has been fed from the lubricator, first the regulating valves, then the water valve is closed. If the water valve is tight, then it is not necessary to close the steam valve at the boiler. The drain cock 14 is then opened and the water removed from the reservoir, after which the filling plug is opened. When the water is entirely out of the cup. the drain cock is closed and the reservoir filled with oil, after which the filling plug is replaced and the water valve opened immediately.

To use the hand oilers, which should be done only when the engine is running on a down grade with the throttle closed, steam is shut off from the lubricator by closing the steam valve at the boiler, then the valve of the hand oiler is opened, the cover turned to one side and the oil poured in. When the oil has run out of the hand oiler, the valve of the same is closed and steam turned on from the boiler to the lubricator. This at once carries the oil to the steam chest.

The steam valve at the boiler should always be opened before the engine begins to work, whether the feeds are started or not, and should be kept open as long as the engine is doing service of any kind, whether steaming or drifting, unless using the hand oilers as described before.

The water valve should be open at all times except when filling the cup as described.

In making the steam connection from the boiler to the lubricator, the valve should connect at a point where dry steam can be obtained, since when the water gets from the boiler into the lubricator it interferes with its proper performance, and muddy water soon cuts the valves and their seats, causing leaks. This type of steam valve as represented in the illustration, Fig. 231, is provided with a pipe ring 32, to which the dry pipe may be attached, leading from the point on the boiler where the valve is attached to a point inside of the boiler above the highest water level.

The lubricator shown in Fig. 232 is also a Nathan product, made by the Nathan Manufacturing Co., of New York. It has six feeds, instead of three, as in the previous type, and, as the illustrations make clear, is of slightly different construction.

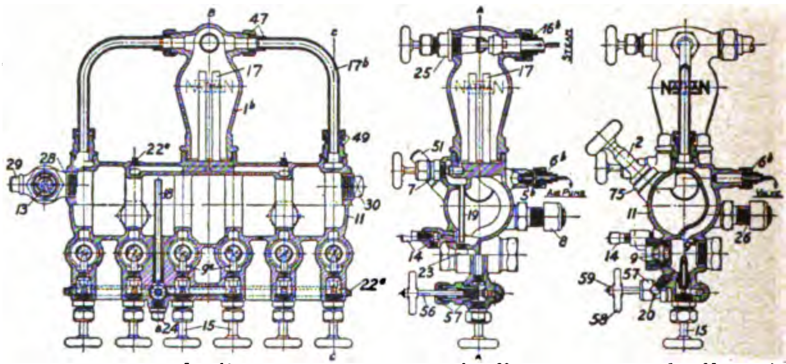


FIG. 232.

No. of Part.	Name of Part.
1b.	Condenser.
2.	Filling Plug.
5b.	Reducing Plug.
6b.	Delivery Nut and Tail-piece.
7.	Water Valve.
8.	Stud Nut.
9.	Sight Feed Glass and Casing.
9a.	Feed Nozzle.
11.	Body.
13.	Gauge Glass and Casing.
14.	Waste Cock.
15.	Regulating Valve.
16b.	Steam Connection.
17.	Equalizing Pipe.
17b.	Outside Equalizing Pipe.
18.	Oil Pipe.
19.	Water Pipe.
20.	Sight Feed Drain Valve.
22a.	Body Plug (small).
23.	Body Plug (large).
24.	Oil Pipe Plug.
25.	Steam Valve.
26.	Support Stud.
28.	Gauge Glass Bracket.
29.	Cleaning Plug (Gauge Glass).
30.	Gauge Glass Cap.
47.	Coupling Nut and Tail-piece.
49.	Bonnet and Packing Nut.
51.	Plug for Filling Hole.
56.	Oil Cut-Out Valve Spindle.
57.	Oil Cut-Out Bonnet and Nut.
58.	Oil Cut-Out Valve Handle.
59.	Oil Cut-Out Valve Handle Nut.
75.	Removable Seat for Filling Plug.

Chicago Locomotive Lubricator.

The accompanying illustration, Fig. 233, shows a five feed type Chicago lubricator, which is made by the Ohio Injector Co., of Chicago, Ill.

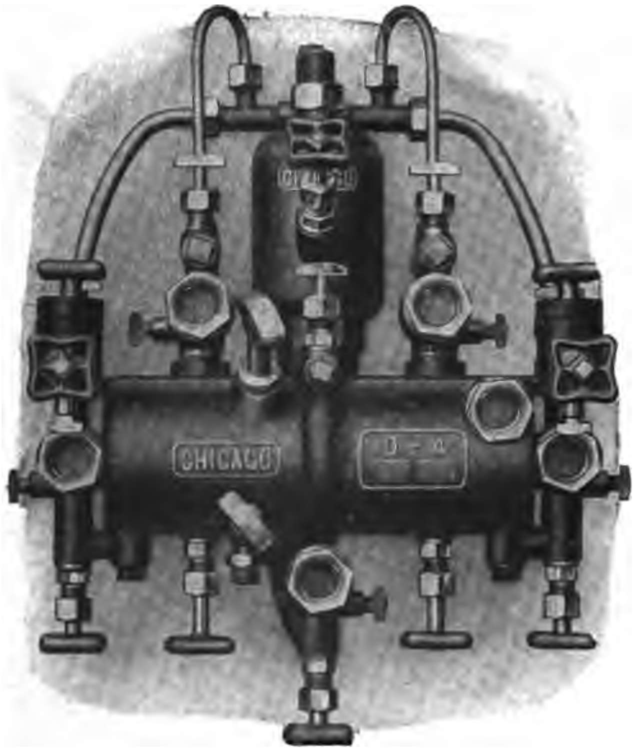


FIG. 233.

This lubricator is equipped with solid sight feed glasses, and is of five pint capacity. The details of construction of the lubricator are made clear by reference to Fig. 234.

To operate:

1st. Open steam valve at boiler. Open valves 62 one turn. Open water valve 73 three turns. Note the feed glasses to see if filled with condensation. After the glasses are filled regulate feeds with valves 31, 175 and 176.

2nd. To blow out glasses: Close feed valve 31 or 175 or 176 and pressure valve 62; open valve 33 to exhaust pressure, after which regulate flow of steam through glass with valve 62.

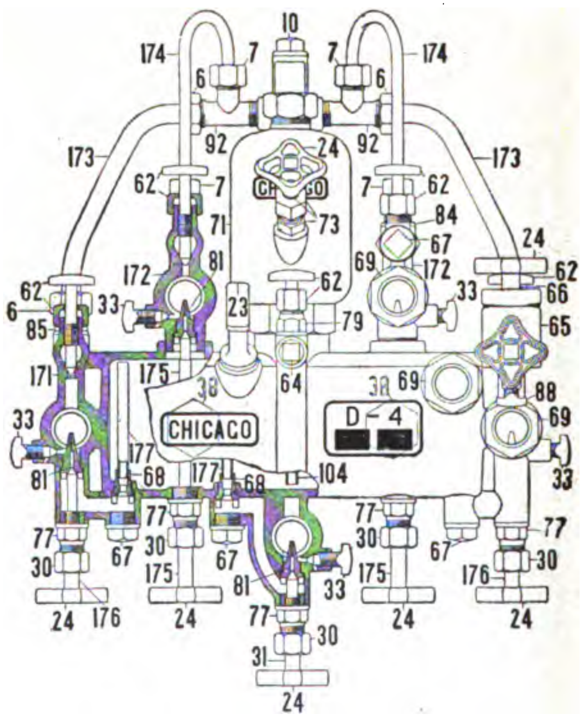


FIG. 234.

3rd. To fill glasses with water: Close valve 33 and open valve 62 one turn.

4th. To operate auxiliary oil cup 65, close pressure valve 62. See that valve 78 is closed. Open auxiliary drain valve 88 to free cup of water. Open auxiliary filler plug 66 and fill. After cup is filled close cup tight and open feed valve 78 wide.

One filling of auxiliary oil cup will feed one hour. This auxiliary cup can be operated with steam on lubricator and engine throttle open.

5th. To remove gaskets or glasses with steam pressure on lubricator: Close valve 62; open valve 33 to drain, then remove packing nut 69, follower washer 70, and gasket 89 with small packing hook. After gasket is removed glass will come out.

6th. To renew glasses: Put washer 70 in first to make a seat between metal and glass; then place gasket on glass, insert large end of glass first; put another follower washer 70 in place on top of gasket and screw packing nut until the necessary resistance is felt.

7th. To set lubricator feeds: By counting in the way here given no mistake can be made. (o-drop) 1, and 2, and 3, and 4, and 5, and 6, and (o-drop). This count will give you ten drop feed per minute. Count in the same way up to twelve, and this gives you five drop feed per minute. Always use an "and" between every count up to twenty. Regulate feeds to suit engine requirements.

The Edna Bull's Eye Lubricator.

This lubricator is of five pint capacity and is equipped with a special type filling plug, Fig. 235, common to all lubricators made by the Edna Brass Mfg. Co., which is very readily replaced, obviating the necessity of cutting new threads, inserting new bushings, or scraping the body of the lubricator on account of worn-out plug threads. A new feature of this lubricator is the straight cored oil passage, which is entirely surrounded by live steam, and hot oil, maintaining an even temperature therein, regardless of exterior temperature. Edna lubricators are made in sizes 1 pint to 6 pints, and with one to six feeds.

Directions for Application.

Secure the lubricator to boiler head or on top of the boiler in most convenient place. Connect at a point on boiler giving dry steam. The steam pipe should have not less than ¾-inch

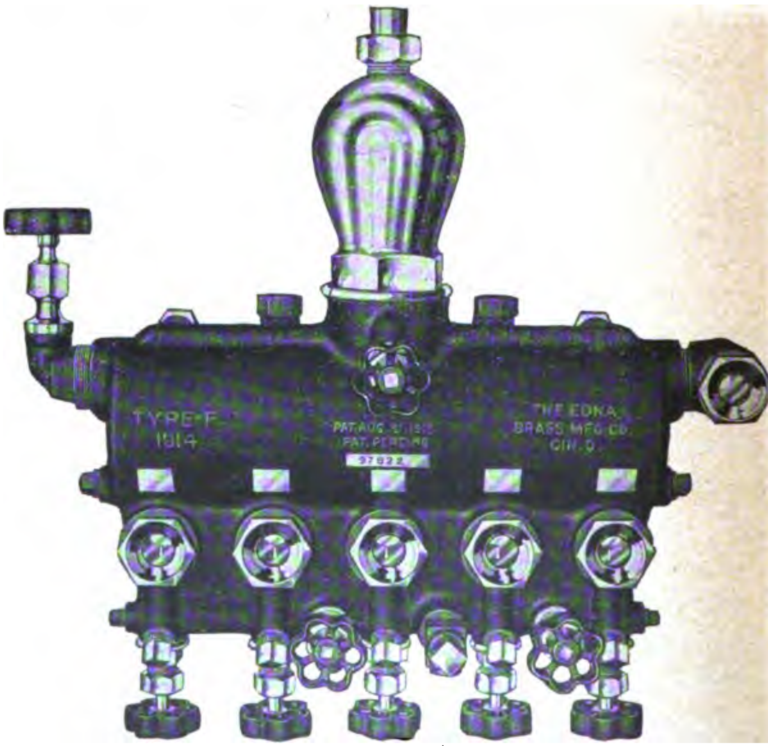


FIG. 235.

area if of iron pipe, and not less than ⅝-inch area if of copper pipe, and the steam valve should have corresponding areas.[1]

The oil pipes must have a continuous fall from lubricator to same, carefully avoiding all bends that will form traps.

Directions for Use.

When the lubricator is set up, all valves should be opened and steam turned on to blow-out.

Then close steam, and all other, valves and fill through filling plug No. 2, Fig. 236; when full screw filling plug to seat and open steam valve No. 25, and water valve No. 7, and wait until sight feed chambers are full of water, then open oil cut-off valve No. 7-B, and regulate the feeds with regulating valve No. 15, to number of drops per minute required.

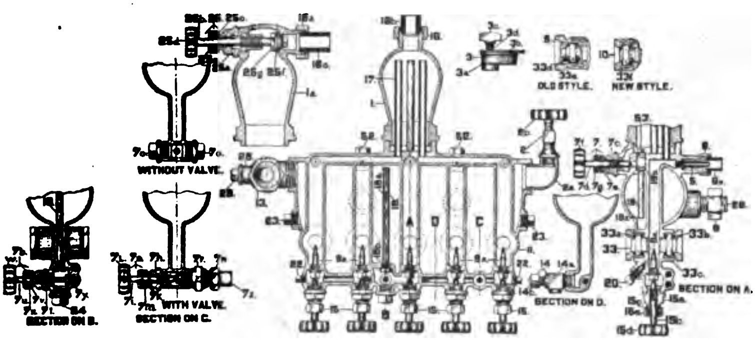


FIG. 236.

To stop the feeds to steam cylinders close oil cut-off valve No. 7-B. Do not close the regulating valves No. 115 to steam cylinder for this purpose. To stop feed to air pump, close regulating valve No. 15.

To Renew Supply of Oil.

Close air pump regulating valve No. 15, oil cut-off valve No. 7-B and water valve No. 7, and draw off water through waste cock No. 14, then fill as before, after which open water valve No. 7 at once, and then regulate the air pump feed only, and open the oil cut-off valve No. 7-B for the cylinder feeds.

Detroit Bull's Eye Lubricator.

The first sight feed hydrostatic locomotive lubricator placed on the market was manufactured and sold by the Detroit Lubricator Company, of Detroit, Mich., in 1877.

DESIGN.

The Oil Control Valve.

The introduction of an oil control valve in the oil passage between the reservoir and the sight feed regulating valves in the Detroit bull's eye lubricator places in the hands of the operator a means of instantly starting, stopping or throttling the rate of feed, does away with the necessity of shutting off the feed regulating valves at a terminal or in refilling on the road and, consequently, the necessity of opening and readjusting these valves after refilling, or at the commencement of a service movement. Under the old method of closing, opening and readjusting the regulating valves, the frequency with which this occurred not only shortened the life of the device, but the length of time consumed in this operation made it impractical to require the operator to shut off his cylinder feeds during temporary stops, with the result that much oil was wasted and the oil mileage decreased.

This oil control valve has a lever handle and index plate, and is so designed that from the "Closed" position a half turn to the "All Open" position will open all feeds, or a quarter turn the feed to air pump only, and vice versa. By moving the handle away from the "All Open" or the "Pump" position, the rate of feed to the different points can be throttled.

The duty of the modern air pump is so severe that it requires almost constant lubrication from the time it leaves a terminal to its return to the roundhouse. It is, therefore, most important that the air pump feed be left working while the locomotive is temporarily at rest at a station or on a siding. A quarter turn of the oil control valve handle provides for this.

Once the feed regulating valves have been adjusted, the use of the oil control valve makes the starting of the lubricator an instantaneous operation and insures the correct amount of lubrication. The operator knows that as soon as the lubricator has reached its proper temperature the oil will feed at the rate required.

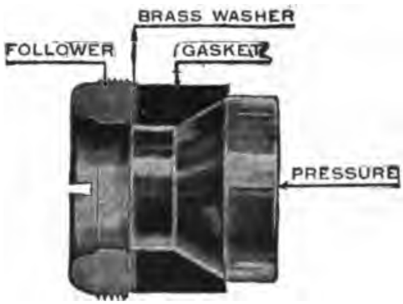


FIG. 237.

The adjustment of the lubricator feeds in night service or where the device is inconveniently located is always a matter of more or less difficulty. As the oil valve does away with the opening and closing of the different feed valves when starting and stopping the device, it becomes a great convenience to the operator, as under all conditions, night or day, he can by the sense of touch alone place the lever handle of this valve in any one of its three positions.

Sight Feed Glasses.

In the bull's eye type of lubricator the shape and dimensions of the glass are of necessity vital factors. This glass, as illustrated in Fig. 237, has been designed to overcome the possibility of breakage and consequent danger to the enginemen, with a shape permitting it to be packed successfully and at the same time refract the maximum amount of light into the sight feed chamber. It is manufactured from a special mixture, has a slightly amber color, and is not affected in any way by the action of steam and oil. It is more than one inch thick and will not break under any conditions of service.

FIG. 238.

Three Feed Standard.

This type of lubricator is used for lubricating the steam chest valves and cylinders, and one air pump, on simple locomotives in average service. It is of three pint capacity, and the operation of the oil control valve may be easily seen in the illustration, Fig. 238. Feed No. 2 lubricates the pump, and is choked at the lubricator, while feeds Nos. 1 and 3 lubricate the steam chest, and are choked at the steam chest.

Eight Feed Standard.

The lubricator here illustrated, Fig. 239, is the largest standard type manufactured by the Detroit Lubricator Company. It has eight feeds, and is of eight pint capacity. It is used for lubricating the high pressure steam chest valves and cylinders, the low pressure cylinders and the air pumps on compound locomotives.

FIG. 239.

The two middle feeds may also be used, one for lubricating the air pumps and the other for delivering oil into the low pressure receiver pipe, thus providing lubrication for the low pressure valves.

To Fill or Refill.—Move the oil control C to "Closed" position, close water valve D and steam valve B, Fig. 240. Open drain valve G and fill with clean strained valve oil. If lubricator is under pressure, proceed as before, but remove filler plug slowly to allow pressure above the oil to escape and the air to enter. Fill the reservoir full. If there is not sufficient oil for this purpose, use water to make up the required quantity. This method will expel the air and enable the feeds to start without exhausting the water from the condenser or materially lowering its level.

Steam Valve.—The regular boiler valve must be left wide open at all times, and the steam valve B at top of condenser must be left wide open while the locomotive is in service.

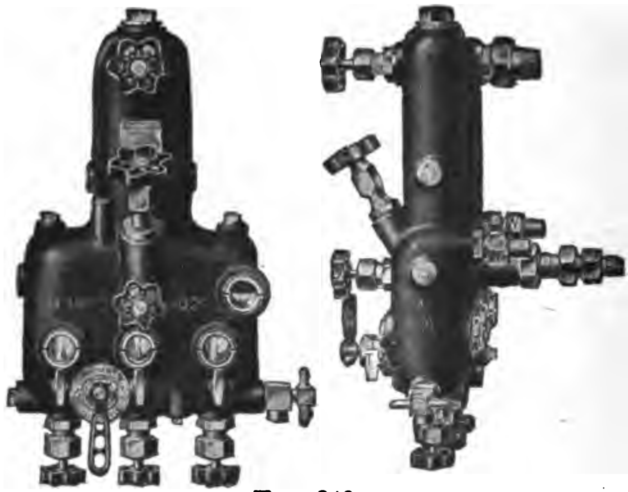


FIG. 240.

To Start Lubricator.—Always start the lubricator about fifteen minutes before leaving the terminal. Be sure that the regular boiler valve is open, then open wide the steam valve B at the top of the condenser and keep it wide open while the lubricator is in operation. Allow sufficient time for the condenser and eight feed glasses to fill with water. Open the water valve D. Three turns will give full port opening. Open the oil control valve C.

In adjusting the lubricator feeds for a class of service after the oil valve C is opened, regulate the cylinder and pump feeds by means of valves E, E and L. After these valves have once been adjusted, do not use them in the ordinary operation of the lubricator. The oil control valve is employed to throttle the feeds and to shut off part or all of them.

Setting Feed Valves.—Adjust the feed valves to the maximum number of drops required for the hardest level track service in the class. Then the number of drops can be decreased for the lighter service by throttling with the oil control valve.

To Shut Down the Lubricator.—For short stops, close oil control valve C only. For terminal stops, close oil control valve C first, then water valve D, and last the steam valve B.

Steam Chest Plugs.—Steam chest plugs are used in connection with the Detroit locomotive lubricators, and whether they be straight, angle or T-angle type, it is equally important to see that their chokes are in place and not worn beyond the limit specified. As these chokes retain the balance of the lubricator, frequent attention should be given them. The steel chokes in the steam chest plugs are reversible and should be replaced by new ones when the holes in both ends are enlarged to an area of 3/32 inch.

A 5/16-inch brass ball is employed in choke at lubricator to balance the feeds to the air pump, intercepting valve and mechanical stoker.

Schlack's Lubricator.

The Schlacks lubricator, formerly known as the McCord, is the product of the Locomotive Lubricator Co., of Chicago, Ill., This lubricator, as may be seen from the illustration, Fig. 242, is built upon an entirely different principle than those previously described.

The lubricator has a lever, shown very clearly in Fig. 242, which is operated by a link connected to an extension on the combination lever of the locomotive valve gear. The arrangement is very clearly shown in the illustration, Fig. 241, of the lubricator as applied to the locomotive. This lever operates a ratchet that turns a cam shaft in a reservoir, which holds approximately eight pints of valve oil.

The stroke shaft, actuated by a cam, carries two pumps, one each for right and left steam pipe, so arranged that they deliver

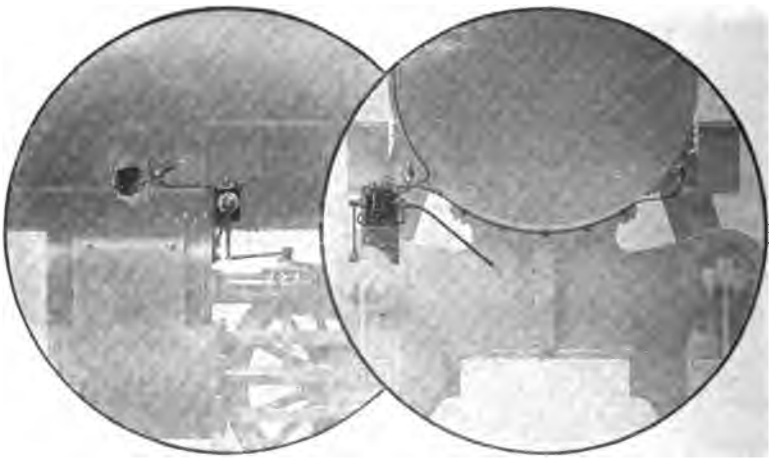


FIG. 241.

oil from the reservoir into the steam at every revolution of the drivers. The reservoir is attached to the back head of the left valve chamber. The length of the lubricator lever, that controls the amount of oil delivered, is determined by a formula considering the valve travel, at average running cut-off, and the cylinder dimensions. A diaphragm check valve, Fig. 243, attached to the delivery end of each oil pipe, and screwed into each steam pipe of the locomotive, maintains a constant oil pipe pressure in excess of the maximum working steam pressure. This insures positive feed.

The oil in the reservoir is kept at a practically constant temperature by a steam supply through a small pipe against the boiler under the lagging from the fountain in the cab, thus ensuring an oil delivery consistent with the pump movements.

FIG. 242.

FIG. 243.

This lubricator does not have to be started before the engine starts, but delivers oil the instant the locomotive moves. There is no valve, nothing to turn on or off before starting or after stopping. The lubricator pumps will deliver against five thousand pounds pressure per square inch, which also insures positive lubrication.

It is absolutely automatic, in that it requires no attention from the engine men.

SAFETY POP VALVES.

The function of the pop safety valve, which has been in use on locomotives for many years, is to prevent the boiler pressure, or pressure of steam in the boiler, from rising above a predetermined, definite point, and to do this automatically, and reliably, under all conditions.

Description and illustration of the details of construction and operation of a few makes of these valves will suffice, for all are, in general, greatly similar, although, of course, each maker advances certain claims of superiority peculiar to his own product.

The pop valve is made in two general types, the open and the muffled. Both are similar in operation, and the muffled type is used merely to eliminate to a great extent the undesirable noise accompanying the action of the open type in relieving the boiler pressure.

The prevalent idea that a safety valve is working correctly when repeatedly opening and closing, without reduction in pressure of two or three pounds, is erroneous. This practice places the valve on a balance, producing rapid wear, which soon results in a humming or singing valve. Destruction of the valve can be prevented by pop regulators, if adjustment is made at once. If allowed to run on a balance, the wings will wear rapidly, seats and lip likewise, and the pop regulators will then have no influence.

It is desirable, too, in the valve, to construct it so that the steam blowing through the valve will be ejected as nearly vertically as possible, rather than to have it spread on all sides, thus obstructing the view ahead from the cab window.

Crosby Locomotive Pop Safety Valves.

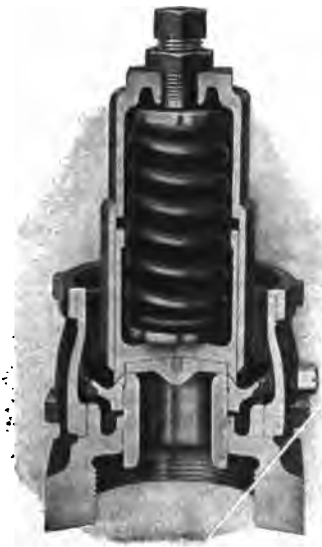


FIG. 244.

FIG. 245.

The Crosby type L, muffled safety valve, shown in Fig. 245, is the product of the Crosby Steam Gage & Valve Co., of Boston, Mass.

It is claimed for this valve that its discharge capacities are very great, whereas the blow down is very low in comparison, and is very easily adjusted.

CROSBY "HIGH EFFICIENCY" (TYPE L) POP VALVE.

The construction of this valve, in detail, may be seen from the interior view of the type L, open valve, as illustrated, Fig. 244. This valve is very simple, and is easily understood and repaired.

The disc of the valve is guided from above in a cylindrical chamber, avoiding all guides in the inlet space in the throat of the valve. The flow of steam is unimpeded and by reason of the careful design of the curve of the steam passage through the valve seat, uniform pressure of the steam is maintained at all points upon the valve disc, and upon the regulating ring, so that the greatest possible lifting effect, to give large steam discharge opening, is obtained from the available steam pressure.

To regulate the blow-down: Remove the large hex-head brass screw plug in the side of the valve casing, and with a steel wire or screwdriver, turn the ring one or two notches, to the right to *increase* blow-down, to the left to *decrease* blow-down; then replace the screw plug to lock the ring in set position.

There have also been provided removable bushings, to form the disc and seat in this valve, so that these parts can be renewed with little trouble and cost.

Ashton Pop Valve.

Here are illustrated two types, Figs. 246 and 247, of the Ashton Increased Lift Valves, designed to meet the requirements of the modern locomotive of increased size and boiler capacity. In operation, when the predetermined pressure in the boiler is reached, the steam raises the valve against the tension of the spring, as shown, and the steam is allowed to escape to the atmosphere. The greater the pressure of steam, the higher the valve will lift off its seat, and when the steam pressure again drops to the maximum allowed, the valve will reseat, due to the spring tension.

To change the "pop," slack the check-nut on either one, or both, of the top regulators (2 and 3), and screw down these regulators to increase the pop; to decrease it, screw the regulators upward; then tighten the check-nuts when the "pop" is set as desired.

To change the set pressure, unbolt and remove the top cap, thus exposing the pressure screw, which can be seen in the illustration. Slack the check-nut, and turn the pressure down for increased, or upward for less, pressure, then set up the check-nuts. It is not desirable to change the set pressure more than fifteen pounds—new springs should be applied if this is necessary.

FIG. 246.

FIG. 247.

This type of valve is manufactured by The Ashton Valve Co., of Boston, Mass.

Consolidated Pop Valves.

Figs. 248 and 249 show sectional views of the Consolidated open and muffled pop valves; these valves are similar with the exception that one is of the open, and the other of the muffled, type.

In operation, it may be seen that these valves are similar greatly to the general design of pop valve. However, the blow-back adjustment is obtained by means of a special adjusting ring and pin. The method of adjustment is self-apparent, if the illustration be studied. The details of pressure adjustment, too, are clearly shown.

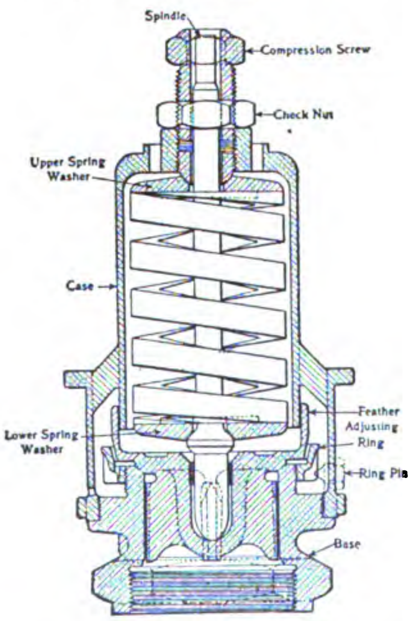


FIG. 248.

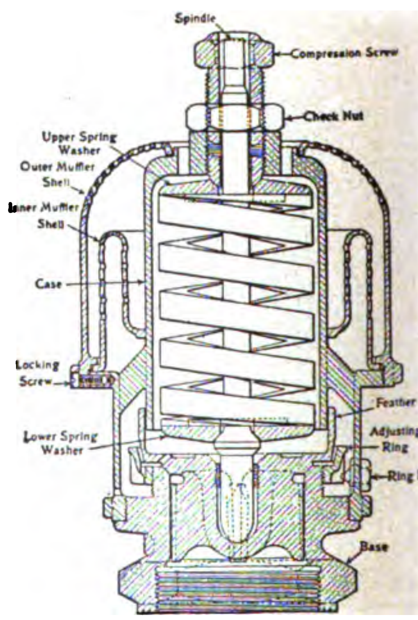


FIG. 249.

The muffler shell may be removed by removing the locking screw in the lower rim of the outer shell, and then unscrewing the shell from the valve case. Thus, when the muffler is removed for cleaning, the muffled type valve may be used on a few trips, while the cleaning is being done, just as an open type, and will operate efficiently in this manner.

This valve is made by the Consolidated Safety Valve Co., of New York City.

American Pop Valves.

This valve is of the muffled type, manufactured by the American Steam Gauge & Valve Mfg. Co., of Boston, Mass., and is notable for its very few parts, the muffled type having but ten

FIG. 250.

FIG. 251.

parts. The open type is similar, in fact, the parts are interchangeable. As all of the adjustments are at the top of the valve, and are easily accessible, an explanation is unnecessary—they are clearly shown in the illustrations, Figs. 250 and 251.

These are not all of the safety valves for locomotives on the market, but an understanding of the essential features of the valves illustrated and described herein will suffice to give the reader a general idea of modern safety valve practice.

Two or more valves are ordinarily used on the boiler of a locomotive, for the sake of preventing boiler explosion in case one of them fails. They need not, of course, be of the same make. When two are used, it is customary to set one to relieve the boiler at a slightly less pressure than the other, so that, unless the pressure is excessive, but one valve will be in operation, while, if the pressure approaches the danger point, the boiler is relieved by the combined efforts of both safety valves.

STEAM GAGES.

Every locomotive must be equipped with at least one steam gage, to accurately indicate the working pressure of the steam in the boiler. The gage should be located so that it will be kept comparatively cool, and can be easily and conveniently read by the enginemen.

The modern pressure gage is a development from the invention of Eugene Bourdon, who was granted a patent in France, in 1852. The essential element is a tube of resilient material, such as steel or bronze, oval or flattened in cross section, and curved, by rolling or bending, into the form of a great arc of a circle. If such a tube be closed at one end, and air, steam or liquid admitted into its interior, the tube will tend to straighten under the effect of the internal pressure, moving the free end of the tube which carries with it connecting linkage adapted to revolve the pointer or hand upon the face of the gage dial. When the interior pressure is reduced or released, the tubular spring takes again its original position and the hand is carried back to indicate absence of pressure at the zero position on the dial.

In 1873, George H. Crosby made the further improvement of combining the action of two such tubes so that the vertical as well as the horizontal movement of the free ends of the tubes could be utilized in joint effect upon the gage hand. By this means gage tube springs could be used that were stouter and stiffer than would otherwise be possible to employ. This was important in gages to be used on locomotives, where the vibration transmitted to the gage hand would prevent accurate reading of the boiler pressure. The steadiness and accuracy of modern locomotive gages is largely due to this invention. The

FIG. 252.

FIG. 253.

use of such tube springs avoids the danger of freezing and bursting, as they will drain because the tubes do not have long curved ends in which condensed water is trapped; and the stiffer and heavier material is not subject to overstrain, or setting, under ordinary steam pressures. Such gages are accurate and sensitive, but free from vibrations of the pointer.

Every gage should be protected by a siphon or trap, filling the tube springs with water. A steam gage should never be screwed to the boiler connection until proper precautions have been taken to avoid admitting steam into the gage tubes.

It is desirable to see that a proper air space is provided between the back of the gage and its support, so that excessive heat will not affect the gage readings.

The Crosby Steam Gage.

The Crosby locomotive steam gage is shown in Figs. 252 and 253. In this gage the index mechanism and the linkage, transmitting the movement of the tube-spring to the hand, are

FIG. 254.

FIG. 255.

mounted upon an extension of the socket. Thus the operating parts of the gage are independent on the case and free from any effect of strain or distortion caused by twisting or turning the case in connecting the gage, as sometimes occurs. Small feet or lugs are cast on the back of the case to insure free circulation of air behind the gage, so that the effect of heat from the boiler is not directly transmitted to the tube springs and linkage to cause errors in the pressure indication on the dial. There is a packed joint in the edge of the gage case to exclude dust and moisture.

Ashton Steam Gage.

Figs. 254 and 255 show the Ashton gage, No. 52, of the Lane type, employing double springs of solid drawn seamless tubing. The movement is non-corrosive, and the gage may be had in any size from 4½" to 8½" dial, but in locomotive practice, 6" or 6½" is standard. This gage is made by The Ashton Valve Co., of Boston, Mass.

FIG. 256.

FIG. 257.

American Steam Gage.

Figs. 256 and 257 show the standard type of locomotive gage made by the American Steam Gauge & Valve Mfg. Co., of Boston, Mass. It is furnished in standard 6 or 6½ inch dial, or, if desired, in a 5 inch dial.

This gage is of the double spring Bourdon type, and is designed to meet the requirements of service where vibration is severe, as in the locomotive. The spring is of the solid drawn bronze tube construction.

There are numerous other makes of steam gages on the market, but all follow the same general principles of construction and operation, so that a description of each would be superfluous.

It might be added, in conclusion, that it is required, with any make or type of locomotive steam gage, to employ a siphon, of ample capacity to prevent steam entering the gage. The pipe connection should enter the boiler direct, and must be maintained steam tight between the boiler and the gage.

Crosby Pressure Gage Tester.

Accuracy is the essential feature in all pressure gages, and it is important that gages should be tested and calibrated at frequent intervals. The only satisfactory way in which a pressure gage may be tested and adjusted is by means of a dead weight gage tester, in which the pressure of known standard weights is transmitted directly to the interior of the gage tube springs. The body of the gage tester holds a considerable volume of oil and the pressure of the weights is transmitted to this by means of a piston 1/5 sq. in. in area, accurately fitted to slide vertically in a carefully finished cylinder. This piston, with its tray of weights, permits no leakage of oil, and yet will revolve and move freely without appreciable friction. The gage is connected to the oil chamber by a short, hollow arm and pressure may be brought upon the oil, to support the load of weights, by means of a screw plunger acting in the chamber at the side of the body of the instrument.

Directions for Using.

To use the instrument, proceed as follows: Set the handle of the three-way cock horizontal, with the word "open" upward. See that the screw plunger is in as far as it will go, then remove the cap from the top of the cylinder and pour oil from the can into the cylinder until it is nearly full and the oil shows at the top of the arm, then gradually withdraw the screw plunger by turning the hand wheel, and continue pouring in oil until the plunger is out nearly to its limit and the bore of the cylinder

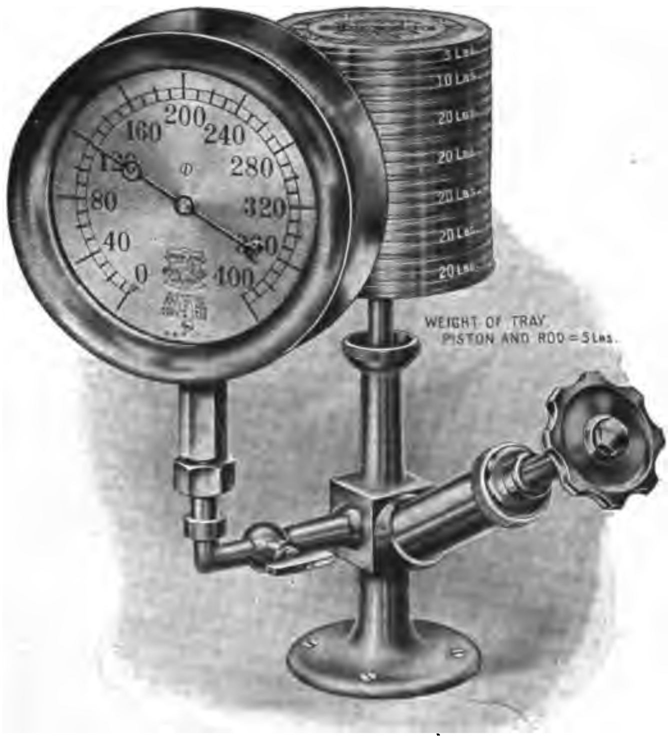


FIG. 258.

is nearly full; then replace the can on the base, with its open nozzle under the cock.

Then couple the gage to the arm, as shown in Fig. 258, using one of the connections furnished. Two wrenches are furnished, which may be used at the same time, one to hold the bushing and the other to turn the union nut. Or the gage itself may be turned, taking care not to strain the socket by too much pressure upon the gage case. If the threads on the gage socket are worn, leakage may be prevented by winding in a few threads of hemp fiber.

Now insert the piston, which, with its tray, will show on the gage dial a pressure of five pounds. Place the weights—one at a time—on the tray, which should be gently rotated to insure perfect freedom of movement to the piston. Each weight added will exert a further pressure on the gage equal to the number of pounds marked on it.

Note by pencil marks on the dial the position of the gage hand or pointer at each pressure, thus indicating any variation from the dial markings. Tapping the gage case lightly with the finger or pencil will insure true readings. If the gage requires correction, remove the pointer by means of a hand-puller or small jack (not using for this a screw-driver or similar tool that might bend the pinion staff on which the hand is fixed), and reset the gage hand to the true position, at some fair working pressure, or readjust the linkage of the gage mechanism, if this be necessary, to secure uniform calibration at all pressures.

Before proceeding to test the gage by the weights, it is well to force the piston down in its cylinder, by pressing with the palm of one hand upon the tray, exerting thus sufficient pressure upon the oil to cause the gage hand to move around the dial almost to its maximum graduation, several times. This expels the air from the gage connections and at the same time insures that the working parts of the gage move freely. Remember that the piston and its tray are equivalent to five pounds pressure on the gage.

If, in testing a large gage, the piston descends nearly to its full length, screw in the plunger by means of the hand wheel and the piston will be forced upward by the oil and more weights can then be added, as may be required by the limit of pressure marked on the gage dial.

When the test is completed, remove the weights—one at a time—and as the piston rises, turn the hand wheel back to withdraw the plunger to make room for the returning oil.

When all the weights have been removed, lift out the piston and tray, and turn the cock handle to a vertical position, with the end marked *X* upward, which will allow the oil in the gage (but not that which is in the cylinder and pump) to drain into the can. Then set the handle of the cock horizontal again, with the word "open" upward, before attaching another gage.

When all the testing has been completed, the piston should be carefully wiped and replaced in the box. The oil may be left in the machine, but it should be cleaned out thoroughly every few weeks in order to prevent the oil from becoming gummy or depositing a sediment. Use nothing but clear, light mineral oil, such as a good machine oil, entirely free from corrosive ingredients. It should be filtered to keep it free from grit. Dirt is often drawn out of the gage tubes into the oil during the testing, and therefore care must be exercised. Keep the screw cap on the top of the cylinder when not in use, to exclude dust.

When it is desired to drain the whole machine of oil, set the handle of the cock with the "drain" side upward and it will all run out. Then turn the cock with the "open" side upward and so leave it. When setting the cock for another test, see that the open nozzle of the can is under it to catch the drip.

WATER GLASS GAGES AND GAGE COCKS.

The water glass provides a visual determination, available at a glance, of the water level in the boiler. While the gage cock, the use of which is also required on the locomotive, offers a sure means of determining the water level, the water glass gage is used universally, due to its convenience. The instrument, indicating, as it does, the exact boiler water level, is very important, especially when used in connection with the superheater locomotive, as the information it gives will prevent waste of fuel, loss of power, damage to the locomotive, and, often, serious accident.

The Edna Water Glass.

The Edna type M reflex water gage, complete with gage cocks and drain, is illustrated in Figs. 259 and 260. It is made in nine sizes, with observation glasses from 3 1/16 to 12 1/2 inches in length, and is guaranteed to stand a pressure of 300 pounds.

As numbered in Fig. 261, the names of various parts are: 1, Body, Back Part; 2, Body, Front Part; 3, Glass; 4, Gauge Stem (½", ⅝" and ¾"); 5, Copper Washer; 7, Cap Screws; 8, Fibre Gasket; 9, Asbestos Gasket; 10, Top Gauge Cock Body; 10A, Bottom Gauge Cock Body; 11, Packing Nut; 12, Gland; 13, Gauge Stem Packing Nut; 14, Gauge Stem Gland; 15, Gauge Cock Stem; 16, Gauge Cock Handle; 17, Stop Ring; 18, Top Gauge Cock Plug; 19, Drain Cock Nipple; 20, Drain Cock; 21, Yoke. The yoke, 21, may be moved to the top or bottom of the gage by loosening the set-screw, 7, and the front may then be removed to apply new glass, without removing the gage from the engine.

This device is the product of the Edna Brass Mfg. Co., of Cincinnati, Ohio.

FIG. 259.

FIG. 260.

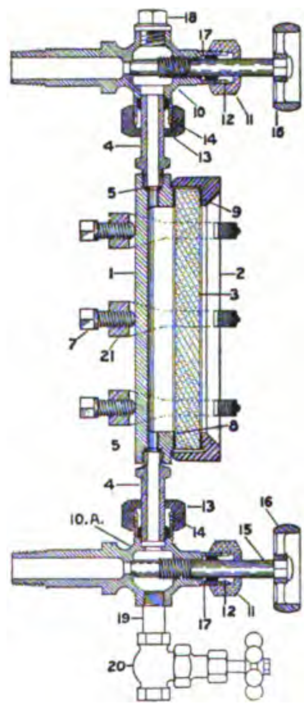


FIG. 261.

The Sargent Safety Water Gauge.

In the Sargent safety water gage is a water level indicator employing the usual form of tubular water glass, but in which the protective features are embodied as a unit structure with the gage itself. It consists of a strong encasing element having three heavy sight panels, arranged in triangular form close in about the tube and held fixedly in place with uniform bearing on all sides through the use of cement especially selected for

FIG. 262.

the purpose. Fig. 262 illustrates these points clearly. Into the ends of the body structure are threaded the gage glass stuffing boxes, making a tight assembly that is at once rigid, compact, and altogether secure against any possibility of inflicting injury through the flying of broken glass or of hot water and steam.

The strength of the gage is greater from the further fact that the space wherein the tube is contained is reduced to the point where practically nothing in the way of expansion of steam takes place when a tube gives way. Inability on the part of the steam to expand, results in lessened velocity of impact against the sight panels of the gage, and a correspondingly lessened disruptive force to be resisted by the protective element.

This added strength is utilized to resist the full pressure of the boiler, except for such relief as is afforded by a single small vent hole at the lower end of the casing, this vent being provided purely in the nature of a warning that the tubular glass is broken and is in need of replacement. The broken glass remains in the casing until removed, there being no possibility of its escape, nor of any sufficient escape of water and steam to prove dangerous, and hence no need of a vent to discharge it through the floor of the cab, as when the gages or gage protectors of the separable part construction are used.

Sargent Renu Gage Cock.

The cost of blowing down a locomotive boiler becomes the greater as the size of the locomotive increases. When, as is often the case, necessity for blowing down hinges on relatively so minor a matter as a leaking gage cock, the trouble, expense and delay involved are all out of proportion to the extent of the defect to be remedied.

The Renu cock is designed to overcome these difficulties. This is done by providing in addition to the main closure, a second seat against which the pressure of the boiler may be held while the main seat is undergoing repair. In this gage cock the blow of steam or water when the cock is open is through a nipple threaded into the side of the bonnet. This latter is held to the body by means of a coupling nut which serves, at the same time, to hold in place the renewable composition disc against which fits a shoulder on the stem of the cock in order to effect its main closure. The stem screws into the body by means of left-hand threads so that by turning the stem to the right in the usual manner, the shoulder is brought into contact with the disc, shutting off the flow. Temporary closure is effected by turning the stem to the left, thus bringing the rounded end of the stem tight into the coned seat in the body of the cock. See Fig. 263.

FIG. 263.

The lead threads on the stem engage with the body rather than with the bonnet, as in ordinary cocks. This leaves the bonnet detachable for the renewal of the main seat, while at the same time the pressure of the boiler is held with as much security to the occupants of the cab as is afforded by the ordinary gage cock under its most favorable conditions.

The flow of water or steam when the cock is open is through slots milled across the threads on the stem, through the renewable seat disc to the chamber in the bonnet into which is tapped the nipple for flow. One and one-quarter turns only are required to completely close the cock against either the permanent or the temporary seat.

This device is made by the Sargent Company, of Chicago, Ill.

Sargent E. S. E. Reflex Gage.

A very good application of a water glass gage and gage cocks is shown in Fig. 264, which illustrates the Sargent Reflex gage of the Klinger, or Reflex type, adapted to locomotive service.

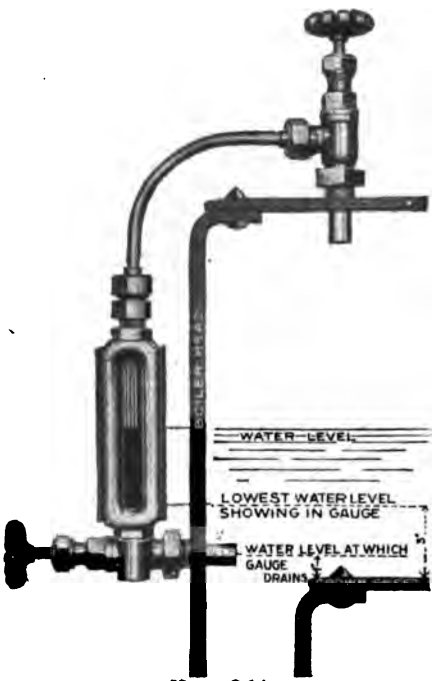


FIG. 264.

While this gage embodies some unusual features of construction, it is in general of the ordinary Reflex type, but it is very rigid of construction, eliminating the possibility of leaks and blows. It is finished with or without stems, but, by attaching the gage directly to the bottom gage cock, as illustrated, it is possible to locate the cock at least one inch above the top of the crown sheet, and still be able to show the minimum reading of three inches above that level. This is of advantage in that, were the top cock to become stopped up, the water could not fall below the level of the crown sheet without draining the cock, and thereby indicating the danger.

This appliance is manufactured by the Sargent Company, of Chicago, Ill.

Blow-Off Valves.

It is essential that the water from the bottom of the boiler should be blown off as often as necessary to remove the suspended matter that would otherwise deposit in a hard scale or cause foaming or priming. The blow-off valve must give a full, free opening for quick discharge so as not to impede the rapid flow of water, and must be designed so that it cannot become stuck from incrustation of mud or lime around the disc or stem.

The Johnstone Blow-off valve, well known on most railroads of the country, has been in use for over 25 years. It is made by the Crosby Steam Gage & Valve Co., of Boston, Mass., and was invented by a railway superintendent of motive power. In this valve there is no wedge or gate, but a simple flat disc is loosely encircled by a stirrup or ring, by which it is raised or lowered. Wings or ears on the stirrup serve as guides in the vertical slots in the valve body and prevent the valve, in closing, from passing beyond the seat. The disc does not wedge into place or jam, but is held to its seat by the pressure of the water alone and therefore is free at all times. The straight lever lifts the disc from the seat into the upper part of the body, leaving a full straightway opening as shown in Figs. 265 and 266. The sliding of the valve maintains close contact between the disc and seat faces which do not become scarred or worn by foreign matter lodging on them, as might otherwise happen.

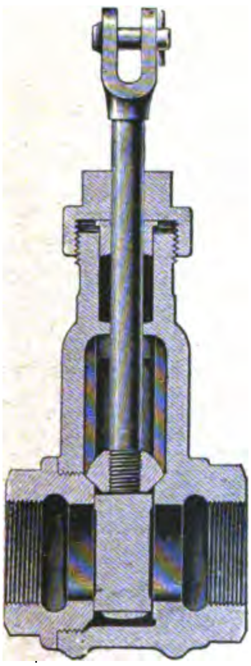


FIG. 265.

FIG. 266.

The disc is loose and freely adjusts itself upon the seat. It is a simple, durable construction, well proven by long service. The removal of the outlet bushing gives access to the interior of the valve; the body is in one piece. The disc is reversible, giving a new seating surface if one side becomes worn. If the surfaces of the seat or disc become scarred, they can be easily refaced by rubbing upon a face-plate.

The improved Johnstone valve, shown in Figs. 267 and 268, made by the Crosby Steam Gage & Valve Co., carries the same idea into a sliding blow-off valve with a quick-opening quarter-

FIG. 267.

turn lever for opening and closing. This valve is made of two halves, bolted together, and therefore can be easily taken apart

FIG. 268.

for cleaning or reseating. The spindle joint is tight and has unusually long bearings. The disc is solid and cannot become wedged nor affected by bad water.

Sentinel Low Water Alarm.

The purpose of this device is to insure against operation of the locomotive with low water, and its attendant casualties. The operative principles of the alarm are based on the expansive properties of metal under increases of temperature. A body of water, when drawn from a steam boiler, and isolated so as to prevent circulation, will quickly chill to a temperature considerably below that of the steam within the boiler, and will also cause a drop in the temperature of its container.

If, now, the chilled water be replaced with live steam, the temperature of its container will also increase, with a corresponding definite expansion.

The essential parts of the mechanism are, as illustrated in Fig. 269: A crosstee, A, which is attached to the shell of the boiler, preferably just over the high point of the crown sheet. An internal drop pipe, B, communicating with the crosstee, and extending to any predetermined water level (preferably 1" on passenger engines and 1½" on freight engines, above the highest point of crown sheet). An expansion element, or tube, C; the tube head, D; the alarm valve, E; the lever, F; and the whistle. G. These parts are also shown clearly in Fig. 270.

In operation, the water level of the boiler is such as to seal the open, lower end of the drop pipe B, so that any steam which may have been imprisoned in the drop pipe, crosstee, expansion element, and tube head, will have condensed, and these parts are consequently filled with water which has been drawn up from the boiler. As long as the open end of the pipe B is sealed (that is, so long as the level of the boiler water is above the predetermined point), the water in these parts is isolated, and will remain at a temperature below that of the steam and water within the boiler.

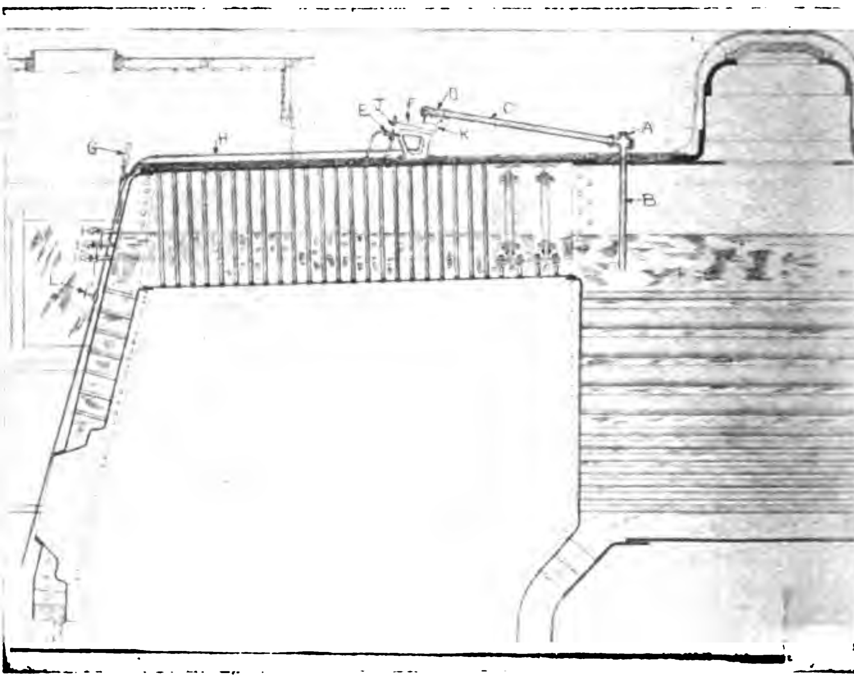


FIG. 269.

But, if the water level in the boiler falls below the end of the pipe, leaving it open, all of the water held in the various parts of the device will flow out, due to its weight, and these parts will instantly fill with the higher temperature steam. The expansion element will consequently respond to the change in temperature, and the contact stud, in the outer end of the level F shown at J

FIG. 270.

will bear down the stem of the alarm valve, and admit boiler steam to the whistle, this lever pivoting at the point K.

When the normal level of water in the boiler is restored, the pipe B will again be sealed, and steam will be imprisoned in the device, to be condensed, drawing up water which will chill the expansion element C, causing it to contract; then after this the contact stud J will be withdrawn from the valve, shutting off steam from the whistle.

As a device of this nature should be frequently tested, to be reliable, this requirement is met by the test pipe H. By opening the valve L (located on the back head of the boiler), the chilled water in the expansion element is blown out, and replaced with hot water from within the boiler. Thus, within a few seconds, and with the blowing off of less than a gallon of water, the operative condition of the entire mechanism may be ascertained. This test should be made by the enginemen upon their arrival at the locomotive, and before entering terminals.

This device is handled by The Pilliod Company, of New York City.

Whistles.

It is the aim of the manufacturers of locomotive whistles to design them in such a manner that they will give a clear, penetrating, and yet an agreeable sound.

It has been the practice to locate the whistle on top of the boiler, just behind the dome, but it has often occurred to the writer that, because this location of the whistle is very injurious to the ears of the enginemen, it would be advantageous to place the whistle in front of the steam dome on the boiler.

The illustration, Fig. 271, shows the Crosby single-bell chime whistle, equipped with renewable seats, and adpted in particular to locomotive service.

The peculiar merit of this whistle consists in its producing three distinct tones, pitched to the first, third and fifth of the common musical scale, which harmonize and give an agreeable musical chord.

Fig. 271.

Fig. 272.

It is more penetrating than the common whistle and can be heard at a greater distance. It obviates the harsh, disagreeable noise or shriek of common whistles, and thus overcomes one of the great annoyances of railway travel, and also serves to distinguish passenger from freight trains.

This method of casting three chambers of different length in a single bell was originally patented and introduced by the Crosby Steam Gage & Valve Co. of Boston, Mass., by whom the whistles, Fig. 271 and 272, are manufactured.

In any whistle, when the valve is badly worn, the cup is of little value, but, in this valve, the seats are the same as those employed in the Crosby Spring-seat globe valve, the disc seat A, and the valve seat B, and are renewable at small expense, thus lengthening the life of the whistle as a whole.

To overcome the difficulty found in operating whistles of large size on locomotives using high steam pressures, such whistles are

FIG. 273.

FIG. 274.

sometimes made with the Crosby compound valve, in which the only valve which must be operated by hand is a small one, held by a spiral spring pressed to the foot of the lever.

A slight pull on the lever pushes inward the small valve and permits steam to flow into a chamber behind a large piston, which automatically forces the large valve from its seat and sounds the whistle. By this device the largest whistles under the highest steam pressures may be operated with ease and rapidity.

Fig. 273 shows the Ashton Bell Chime whistle, with side valve. This valve is operated by a plunger, which, in turn, is acted upon by the lever shown, to sound the whistle. It is made for any size steam pipe, and with a bell of any diameter up to 12 inches. This whistle is manufactured by The Ashton Valve Co., of Boston, Mass.

The American Steam Gauge & Valve Mfg. Co., of Boston, Mass., are the makers of the whistle shown in Fig. 274, which is constructed with particular adaptability for locomotive service. This whistle is operated, in general, just as the usual chime whistle, and has, incorporated in the fulcrum, an adjustable nut, used to govern the opening of the valve.

Bell Ringers.

Air and Steam Operated.

Automatic bell ringers have come into such general use, and are required by law in so many states, that it has been deemed advisable to describe and illustrate a few of the better known types, using either air or steam as the operative force.

THE GOLLMAR BELL RINGER.

This bell ringer, which, in a slightly different form, has been in use for over twenty years, and is illustrated in Fig. 275, is operated by either steam or air.

Construction and Operation.

Its construction and operation are as follows. There are two openings for pipes; the upper one is the inlet, the lower is the exhaust. Pressure is admitted through upper opening, opposite an annular groove is valve 18, through which four holes are drilled, admitting the pressure under the single-acting piston 10; this causes piston 10 to rise, forcing the bell to swing. Piston 10 has a stroke of 1¼ inches

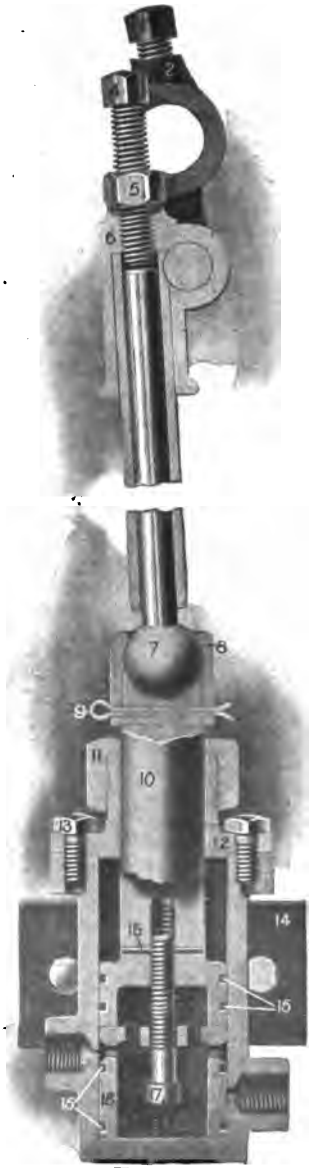


FIG. 275.

when at its extreme travel; crank 2 has a stroke of 4 inches. The connecting rod is in two sections, 6 and 7, which allows the crank 2 to make a complete revolution without causing piston 10 to move. When the ringer is started to work the piston 10 will be driven upward, causing the bell to swing, and valve stem 17 will raise valve 18, closing inlet port and using pressure expansively by traveling the length of the lap before it will open the exhaust port. The bell, having received an impulse, will continue its motion after the piston 10 has reached the upper end of its stroke, the crank box 6 sliding on rod 7. The impetus which the bell receives being expended, it will fall; the governor bolt 4 will strike the end of rod 7, the piston 10 will be forced downward, coming in direct contact with valve 18, closing exhaust port and opening inlet port after cushioning on the pressure remaining under piston 10 after exhaust is closed. It will be seen that valve 18 is only operated at the terminations of the piston 10 stroke. Throw of the bell should be regulated by means of governor bolt 4 and jamb nut 5. If bell swings too high or turns over, back out the governor bolt 4. If bell does not swing high enough, screw in governor bolt 4.

This bell ringer is marketed by The United States Metallic Packing Company, of Philadelphia, Pa.

THE SECURITY BELL RINGER.

This bell ringer, air operated, is handled by D. R. Niederlander, of St. Louis, Mo. It consists, as shown in the illustrations, Figs. 276 and 277, of a rigid piston and a movable cylinder, the top of which is crown shaped for contact with a roller, which operates the bell. There are no toggle joints, or adjusting rods, so that when the bell cord is used, the ringer is not brought into action. The working parts are entirely protected from cinders, and from the consequences of snow or water freezing, as in open top bell ringers.

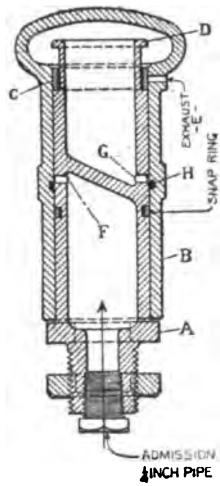


FIG. 276.

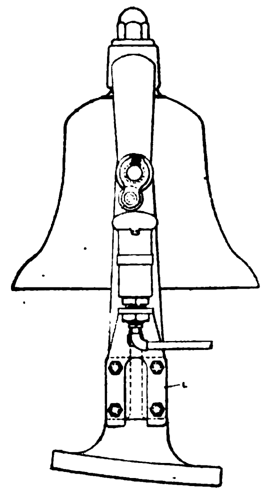


FIG. 277.

Application and Operation.

Adjustment of piston "A" is provided for by a threaded bracket and large lock nut. This bracket can be bent to suit bell frame or turned downward and attached to base bracket. Air is admitted through standard pipe connections at bottom of piston, and finds its way through passage "H," causing cylinder to rise automatically, thereby shutting off the intake on its upward movement, while exhaust ring "C" is carried up by the cylinder, until interrupted by retaining ring "D," when exhaust port "E" is opened, immediately permitting cylinder "B" to return of its own weight until exhaust ring "C" again rests on shoulder of piston "A," when a slight pressure of the roller completes the downward operation of cylinder "B," thereby closing exhaust port "E" and opening communication with intake "F."

In this bell ringer is a specially ground bottom snap ring; and a wide top exhaust ring "C" for controlling exhaust. These two snap rings, together with the liberal area of both piston and cylinder, constitute the working parts, thus insuring maximum service, there being practically nothing to get out of order.

For heavyweight bell the clearance under roller should be one-sixteenth of an inch, and for light bell one-eighth inch space should be allowed. Crank arm should be set at an angle of about five degrees ahead or in back of center line. to avoid dead center when bell is at rest.

Track Sanders.

Some means by which sand may be delivered to the rail at its point of contact with the driving wheel is necessary to the satisfactory operation of the locomotive. This stream of sand deposited in front of the drivers prevents their slipping, or sliding, and in this manner aids in getting a train under way, and in bringing it to a quick stop. This is especially true when the rails are wet.

The ordinary sand lever arrangement is unsatisfactory, as it is difficult to operate, and it is also very wasteful of sand; often depositing it so unevenly as to cause added resistance to trains, rather than aiding in their starting.

The pneumatic sander, however, eliminates these objectionable features, and, by its ability to deliver sand in any quantity desired by the engineer, is not only more economical of sand, but is much more efficient.

There are several of this type of sander on the market, but the author believes that a description of the one most widely used will, with the aid of illustrations, make clear the principles of construction and operation of pneumatic sanders in general.

It might be added that it is advisable to thoroughly screen the sand before placing it in the sand box, so as to remove all foreign matter.

THE LEACH SANDER.

Type A, shown in Fig. 278, is the most widely used of the several pneumatic track sanders on the market. This extensive use

FIG. 278.

is due to its adaptability to service conditions, and because of its simplicity of construction and small number of connections. A comparatively small amount of piping, too, is necessary.

The accessibility of the sander for inspection and repair is due to the fact that it is located wholly outside the sand box. This eliminates the necessity of removing all the sand from the box to make repairs to the sander.

The blast caps are removable, so as to be readily replaced when worn by the action of the sand blast. The air nozzles, controlling the blast, must be kept in good order to prevent the blast from becoming deflected from the blast cap.

The sand traps are attached to the box in a convenient manner, as shown in Fig. 278, and the sand is supplied thereto through independent outlets from the box, and is discharged from the traps into and through the usual hand lever controlled

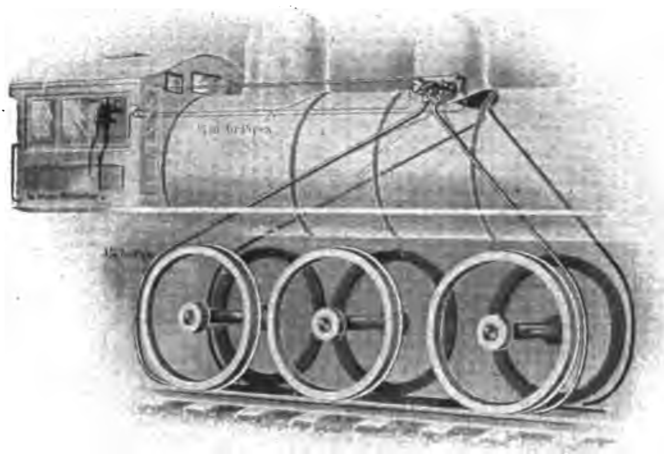


FIG. 279.

pipes to the rail, the hand lever attachments being available for use if desired. The sand pipes should be clamped securely at the bottom, and so bent as to deliver the sand directly to the point of contact between driving wheel and rail. Fig. 279, showing the application of the complete device, illustrates this point.

The discharge pipes, usually 1¼ inch, must be fitted at such a pitch that sand will flow through them by gravity when the lever is used. The amount of sand discharged is controlled by the adjustment of the engineer's valve. This valve is provided with a warning arrangement, which prevents leaving the valve open unintentionally, and, consequently, wasting sand.

The details of the sand trap are clearly shown in Fig. 280, which illustrates the type A double sander.

This track sander is made by The United States Metallic Packing Co., of Philadelphia, Pa.

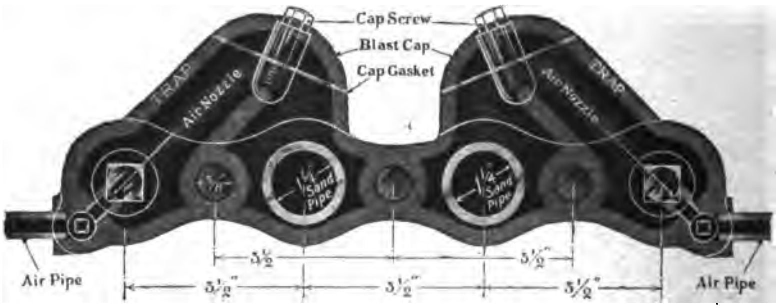


FIG. 280.

THE STEAM ENGINE INDICATOR.

The steam engine indicator, invented about 1785, by James Watt, and long kept a secret, was for many years after its secret became known, strangely neglected by most makers and users of steam engines.

The earlier forms of the instrument, which preceded that invented by Richards, were so imperfect and so ill-adapted to engine running at other than very low speeds, that their indications were often misleading, more often unintelligible, and seldom of much value beyond revealing the point of stroke at which the valves opened and closed: a most valuable service, alone worth the cost of an indicator, but only a small part of the service to be obtained from a really good instrument.

The general principles on which the best types of the steam engine indicator is designed, may be briefly stated as follows:

A piston of carefully determined area is nicely fitted into a cylinder so that it will move up and down without sensible friction. The cylinder is open at the bottom, and fitted so that it may be attached to the cylinder of a steam engine, and have free communication with its interior, by which arrangement the under side of the piston is subjected to the varying pressure of the steam acting therein. The upward movement of the piston—due to the pressure of the steam—is resisted by a spiral spring of known resilience. A piston rod projects upward through the cylinder cap and moves a lever having at its free end a pencil point, whose vertical movement bears a constant ratio to that of the piston. A drum of cylindrical form and covered with paper is mounted with the cylinder in such a manner that the pencil point may be brought in contact with its surface, and thus record any movement of either paper or pencil. It might be said here, that while graphite was formerly used as a marking point, and ordinary paper as drum paper, modern practice recommends metallic paper (a paper coated with salts of lead), and, as a marking point, a piece of soft brass. This makes it possible to obtain very fine lines, and little pressure on the marking point is required, thereby reducing friction between pencil and paper, with its resultant error.

The drum is given a horizontal motion coincident with and bearing a constant ratio to the movement of the piston of the engine. It is moved in one direction by means of a cord attached to the crosshead and in the opposite direction by a spring within itself. Special drums have been made which permit taking a continuous series of diagrams to show varying conditions in the operation of gas and oil engines.

How and Where to Attach the Indicator.

The indicator should be attached close to the cylinder whenever practicable, especially on high speed engines. If pipes must be used they should not be smaller than half an inch in diameter and as short as possible; if long pipes are needed they should be slightly larger than half an inch and covered with a nonconducting material.

Diagrams should be taken from both ends of the cylinder of an engine. If the diagram from one end is satisfactory it is not safe to assume that one taken at the other end will be equally so; it is often otherwise, owing to the varying conditions usually

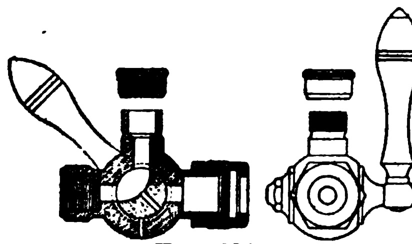


Fig. 281.

found. The lengths of thoroughfares, the points of valve opening and closure, and the lead, are variable and should be carefully adjusted to secure the best results, and this can be done through the instrumentality of an indicator.

When one indicator is employed, it is generally attached to a three-way cock (Fig. 281), which is located midway in the line of pipe connecting the holes at either end of the cylinder; by this arrangement diagrams can be taken from either end simply by turning the handle of the three-way cock. In such a case the second diagram should be taken as quickly as possible after the first, so as to be under like conditions of speed, pressure and load.

The indicator *can* be used in a horizontal position, but it is more convenient to take diagrams when it is in a vertical position, and this can generally be obtained, when attaching to a vertical engine, by using a short pipe with a quarter upward bend. No putty or red lead should be used in making any joints, as particles of it may be carried by the steam into the indicator, and great harm result therefrom; if a screw thread fits loosely, wind into the threads a little cotton waste, which will make a steam-tight joint. The indicator should never be set so as to communicate with thoroughfares where a current of steam will flow past the orifice leading to the indicator, as the diagram taken under such conditions would be of no practical value.

The cylinders of most modern steam engines are drilled and tapped for the indicator and have plugs screwed into the holes, which can readily be removed and the proper indicator connections inserted. But when this is not the case, the engineer should be competent to do it under the directions here given.

When drilling holes in the cylinder the heads should be removed if convenient, so that one may know the exact position of the piston, the size of the ports and passages, and be able to remove every chip or particle of grit which might otherwise do harm in the cylinder or be carried into the indicator and injure it. When the heads cannot be taken off, it can be arranged so that a little steam may be let into the cylinder, when the drill has nearly penetrated its shell, so that the chips may be blown outward—care being taken not to scald the operator.

Each end of the cylinder should be drilled and tapped for ½-inch pipe thread. The holes must always be drilled into the clearance space, at points beyond the range of the piston when at the end of the stroke, so as not to be obstructed by it, and away from steam passages, to avoid strong currents of steam. By placing the engine on a dead center, it is easy to tell how much clearance there is, and the hole should be drilled into the middle of this space; the same process should be repeated at the other end of the cylinder.

Before attaching the indicator to an engine, be sure to blow steam freely through the pipes and cock to remove any particles of dust and grit that may have lodged in them.

The most common practice of application is to drill and tap holes in the *side* of the cylinder at each end and insert short ½-inch pipes with quarter upward bends, into which the indicator cocks may be screwed; on some horizontal engines it may be more convenient to drill and tap into the top of the cylinder at each end and screw the cocks directly into the holes.

When this mechanism is properly adjusted and free communication is opened with the cylinder of a steam engine in motion, it is evident that the pencil will be moved vertically by the varying pressure of steam under the piston, and as the drum is rotated by the reciprocating motion of the engine, if the pencil is held in contact with the moving paper during one revolution of the engine, a figure or diagram will be traced representing the pressure of steam in the cylinder; the upper line showing the pressure urging the engine forward, and the lower line the pressure retarding its movement on the return stroke.

To enable the engineer to more correctly interpret the nature of the pressures, the line showing the atmospheric pressure is also drawn in its relative position, which indicates whether the pressure at any part is greater or less than that of the atmosphere.

From such an indicator diagram may be deduced many particulars which are of supreme importance to engine builders, engineers, and the owners of steam plants.

Indicator Diagrams.

An indicator diagram, or indicator card, as it is commonly referred to, as produced by the action of the indicator applied to the engine cylinder, is the result of two movements, namely: a horizontal movement of the paper in exact correspondence with the movement of the piston, and a vertical movement of the pencil in exact ratio to the pressure exerted in the cylinder of the engine; consequently it represents by its length the stroke of the engine on a reduced scale, and by its height at any point, the pressure on the piston at the corresponding point in the stroke. The shape of the diagram depends altogether upon the manner in which the steam is admitted to and released from the cylinder of the engine; the variety of shapes given from different engines, and by the same engine under different circumstances, is almost endless, and it is in the intelligent and careful measurement of these that the true value of the indicator is found, and no one at the present day can claim to be a competent engineer who has not become familiar with the use of the indicator, and skillful in turning to practical advantage the varied information which it furnishes.

A diagram shows the pressure acting on one side of the piston only, during both the forward and return strokes, whereon all the changes of pressure may be properly located, studied, and measured. To show the corresponding pressures on the other side of the piston, another diagram must be taken from the other end of the cylinder. When the three-way cock is used, the diagrams from both ends are usually taken on the same paper.

Analysis of the Diagram.

The names by which the various points and lines of an indicator diagram are known and designed, are given herewith, and may be readily understood by reference to Fig. 282.

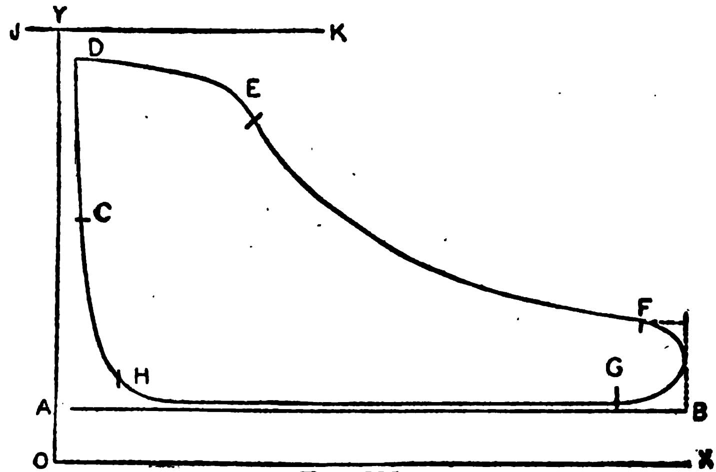


Fig. 282.

The closed figure or diagram, C D E F G H is drawn by the indicator, and is the result of one indication from one side of the piston of an engine. The straight line A B is also drawn by the indicator, but at a time when steam connection with the engine is closed, and both sides of the indicator piston are subjected to atmospheric pressure only.

The straight lines O X, O Y, and J K, when required, are drawn by hand as explained below, and may be called reference lines.

Diagram Lines Explained.

The admission line C D shows the rise in pressure due to the admission of steam to the cylinder by the opening of the steam valve. If the steam is admitted quickly when the engine is about on the dead-center this line will be nearly vertical.

The steam line D E is drawn when the steam valve is open and steam is being admitted to the cylinder.

The point of cut-off E is the point where the admission of steam is stopped by the closing of the valve. It is sometimes difficult to determine the exact point at which the cut-off takes place. It is usually located where the outline of the diagram changes its curvature from convex to concave.

The expansion curve E F shows the fall in pressure as the steam in the cylinder expands behind the moving piston of the engine.

The point of release F shows when the exhaust valve opens.

The exhaust line F G represents the loss in pressure which takes place when the exhaust valve opens at or near the end of the stroke.

The back pressure line G H shows the pressure against which the piston acts during its return stroke. On diagrams taken from non-condensing engines it is either coincident with or above the atmospheric line, as in Fig. 282. On cards taken from a condensing engine, however, it is found below the atmospheric line, and at a distance greater or less, according to the vacuum obtained in the cylinder.

The point of exhaust closure H is the point where the exhaust valve closes. It cannot be located very definitely, as the change in pressure is at first due to the gradual closing of the valve.

The compression curve H C shows the rise in pressure due to the compression of the steam remaining in the cylinder after the exhaust valve has closed.

The atmospheric line A B is a line drawn by the pencil of the indicator when its connections with the engine are closed and both sides of the indicator piston are open to the atmosphere. This line represents on the diagram the pressure of the atmosphere, or zero of the steam gage.

Reference Lines Explained.

The zero line of pressure, or line of absolute vacuum, O X, is a reference line, and is drawn by hand, 14.7 pounds by the scale, below and parallel with the atmospheric line. It represents a perfect vacuum, or absence of all pressure.

The line of boiler pressure J K is drawn by hand parallel to the atmospheric line and at a distance from it by the scale equal to the boiler pressure shown by the steam gage. The difference in pounds between it and the line of the diagram D E shows the pressure which is lost after the steam has flowed through the contracted passages of the steam pipes and the ports of the engine.

The clearance line O Y is another reference line drawn at right angles to the atmospheric line and at a distance from the end of the diagram equal to the same per cent. of its length as the clearance bears to the piston travel or displacement. The distance between the clearance line and the end of the diagram represents the *volume* of the clearance and waste room of the ports and passages at that end of the cylinder.

If the lines of the diagram are broken or irregular, it is probably due to dirt or grit in the instrument which should be immediately removed. The piston and cylinder should be kept wiped clean and frequently oiled. Never use any polishing material on the cylinder or on the piston of the instrument. Leakage of steam past the piston, unless it is sufficient to add to the atmospheric pressure above the piston, will not affect the accuracy of the diagrams. The large relief holes in some types of indicators will prevent any such accumulation of pressure above the piston. The one thing essential is absolute freedom from friction in the movement of the piston and pencil linkage without any lost motion or looseness in the joints. The taper pins which form the bearings of the Crosby pencil linkage afford means for preventing the lost motion.

Crosby Indicator.

By closely studying the following explanation, as well as by referring to Figs. 283 and 284, the construction and operation of the Crosby steam engine indicator are easily understood. The parts numbered in the following text refer to Fig. 283.

The cylinder, 4, in which the piston moves, is made of a special alloy, exactly suited to the varying temperatures to which it is subjected, and secures to the piston the same freedom of movement with high pressure steam as with low; and as its bottom end is free and out of contact with all other parts, its longitudinal expansion or contraction is unimpeded, and no distortion can possibly take place.

Between the cylinder, 4, and the casing, 5, is an annular chamber, which serves as a steam jacket; and being open at the bottom, can hold no water, but will always be filled with steam of nearly the same temperature as that in the cylinder.

The piston, 8, is formed from a solid piece of the finest tool steel. Its shell is made as thin as possible consistent with proper strength. It is hardened to prevent any reduction of its area by wearing, and then ground and lapped to fit (to the twenty-thousandth part of an inch) a cylindrical gage of standard size. Shallow channels in its outer surface provide a steam packing, and the moisture and oil which they retain act as lubricants, and prevent undue leakage by the piston. The transverse web near its center supports a central socket, which projects both upward and downward; the upper part is threaded inside to receive the lower end of the piston-rod. The upper edge of this socket is formed to fit nicely into a circular channel in the under side of the shoulder of the piston-rod, when they are properly con-

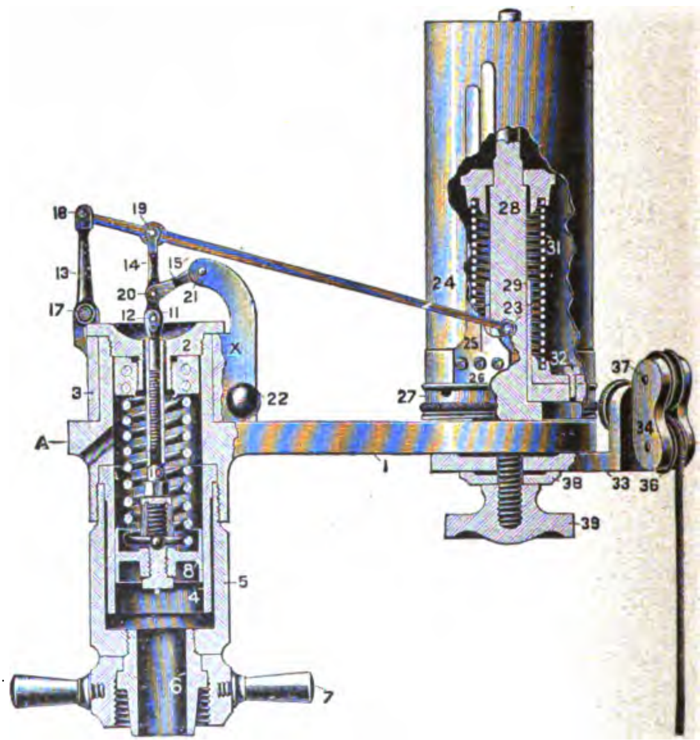


FIG. 283.

nected. It has a longitudinal slot, which permits the straight portion of wire at the bottom of the spring, with its bead, to drop to a concave bearing in the upper end of the piston-screw, 9, which is closely threaded into the lower part of the socket; the head of this screw is hexagonal, and may be turned with the hollow wrench which accompanies the indicator.

The piston-rod, 10, is of steel, and is made hollow for lightness. Its lower end is threaded to screw into the upper socket of the piston. Above the threaded portion is a shoulder having in its under side a circular channel formed to receive the upper edge of the socket, when these parts are connected together. When making this connection *be sure* that the piston-rod is screwed into the socket as far as it will go, that is, until the upper edge of the socket is brought firmly against the bottom of the channel in the piston-rod, before the piston-screw, 9, is tightened against the bead at the foot of the spring. This is very important, as it insures a correct alignment of the parts and free movement of the piston within the cylinder.

The swivel head, 11, is threaded on its lower half to screw into the piston-rod more or less, according to the required height of the atmospheric line on the diagram. Its head is pivoted to the piston-rod link of the pencil mechanism. This adjustment of the diagram upon the card is an advantage peculiar to the Crosby indicator.

The cap, 2, rests on top of the cylinder, and holds the sleeve and all connected parts in place. It has a central depression in its upper surface, also a central hole, furnished with a hardened steel bushing, which serves as a very durable and sure guide to the piston-rod. It projects downward into the cylinder in two steps, having different lengths and diameters; both these and the hole have a common center. The lower and smaller projection is screw-threaded to engage with the like threads in the head of the spring, and hold it firmly in place. The upper and larger projection is screw-threaded on its lower half to engage with the light threads inside the cylinder; the upper half of this larger projection—being the smooth, vertical portion—is accurately fitted into a corresponding recess in the top of the cylinder, and forms thereby a guide by which all the moving parts are adjusted and kept in correct alignment, which is very important, but is practically impossible to secure by use of screw threads alone.

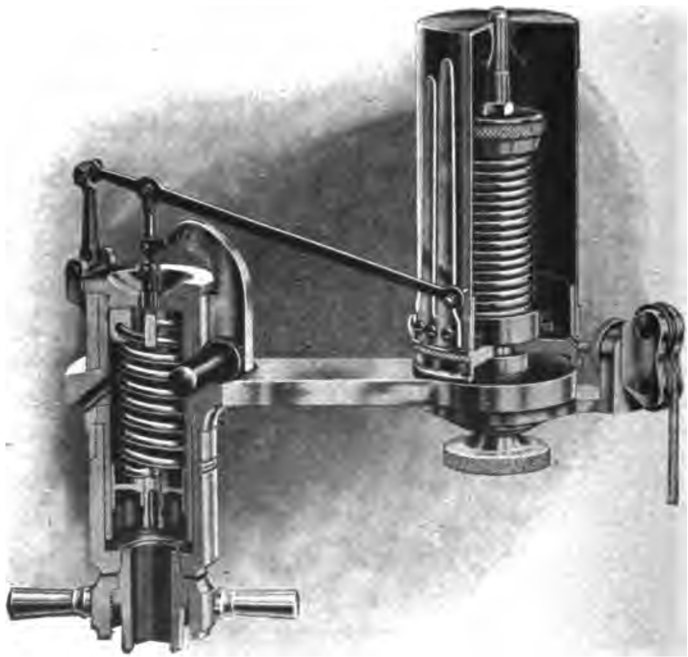


FIG. 284

The sleeve, 3, surrounds the upper part of the cylinder in a recess formed for that purpose, and support the pencil mechanism; the arm, X, is an integral part of it. It turns around freely, and is held in place by the cap. The handle for adjusting the pencil point is threaded through the arm, and being in contact with a stop-screw in the plate, 1, may be delicately adjusted to the surface of the paper on the drum. It is made of hard wood with a lock-nut to maintain the adjustment.

The pencil mechanism is designed to afford sufficient strength and steadiness of movement, with the utmost lightness; thereby eliminating as far as possible the effect of momentum, which is especially troublesome in high speed work. Its fundamental kinematic-principle is that of the pantograph. The fulcrum of the mechanism as a whole, the point attached to the piston-rod, and the pencil point are always in a straight line. This gives to the pencil point a movement exactly parallel with that of the piston. The mechanism is theoretically correct as well as mechanically accurate; the result is, therefore, mathematical precision in the pencil movement, not merely an approximation. The movement of the spring throughout its range bears a constant ratio to the force applied; and the amount of the movement of the piston is multiplied six times at the pencil point. The pencil lever, links, and pins are all made of hardened steel; the latter—slightly tapering—are ground and lapped to fit accurately, without perceptible friction or lost motion.

Springs. In order to obtain a correct diagram, the height of the pencil of the indicator must exactly represent in pounds per square inch the pressure on the piston of the steam engine at every point of the stroke; and the velocity of the surface of the drum must bear at every instant a constant ratio to the velocity of the piston. These two essential conditions have been attained at a great degree of exactness in the Crosby indicator, by a very ingenious construction and nice adaptation of both its piston and drum springs.

The piston spring is of unique design, being made of a single piece of the finest spring steel wire, wound from the middle into a double coil, the ends of which are screwed into a metal head having four radial wings drilled helically to receive and hold the spring securely in place, as shown in Fig. 285.

Adjustment is made by screwing the ends into the head more or less, until exactly the right strength of spring is obtained, when they are there firmly fixed. This method of adjusting and fastening removes all danger of loosening coils, and obviates all necessity for grinding the wires—a practice fatal to accuracy in indicator springs.

FIG. 285.

The foot of the spring—in which freedom and lightness are of great importance, it being the part subject to the greatest movement—is a small steel bead, firmly "staked" onto the wire, as shown in Fig. 286. This takes the place of the heavy brass foot used in some other indicators, and reduces the inertia and momentum at this point to a minimum, whereby a great improvement is effected. This bead has its bearing in the center of the piston, and in connection with the lower end of the piston-rod and the upper end of the piston-screw, 9 (both of which are concaved to fit), it forms a ball and socket joint which allows the spring to yield to pressure from any direction without causing the piston to bind in the cylinder, which occurs when the spring and piston are rigidly united. Designing the spring so that any lateral movement it may receive when being compressed shall not be communicated to the piston and cause errors in the diagram, is of extreme importance.

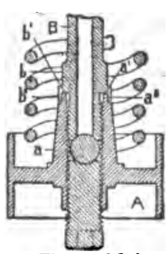


FIG. 286.

The drum spring, 31, in the Crosby indicator is in form a helix, while in other indicators it is a long volute. It is obvious from the large contact surfaces of a long volute spring that its friction would be greater than that of a short, open helical form of like power; and that in a spring of this kind, for a given amount of compression—as in the movement of an indicator drum—the recoil will be greater and exerted more quickly in the helical than in the volute form.

If the conditions under which the drum spring operates be considered, it will readily be seen that at the beginning of the stroke, when the cord has all the resistance of the drum and spring to overcome, the spring should offer less resistance than at any other time; and at the beginning of the stroke in the opposite direction, when the spring has to overcome the inertia and friction of the drum, its energy or recoil should be greatest.

These conditions are fully met in Crosby indicators, the drum spring being a helix having no friction, a quick recoil, and scientifically proportioned to the work it has to do. At the beginning of the forward stroke it offers to the cord only a very slight resistance, which gradually increases until at the end its maximum is reached. At the beginning of the stroke in the other direction, its recoil is greatest at the moment when it is most needed, and gradually decreases as the work it has to do decreases, until at the end of the stroke it is reduced to its minimum again. Thus, by a most ingenious balancing of opposing forces, the most nearly uniform stress on the cord is maintained throughout each revolution of the engine.

The drum, 24, and its appurtenances, except the drum spring, are similar in design and function to like parts of any indicator, and need not be particularly described. All the moving parts are designed to secure sufficient strength with the utmost lightness, by which the effect of inertia and momentum is reduced to the least possible amount. It is ordinarily 1½ inches in diameter, this being the correct size for high speed work, and answering equally well for low speeds. If, however, the indicator is to be used only for low speeds, and a long diagram is preferred, it can be furnished with a 2 inch drum.

All Crosby indicators (except some of the very old Standard Steam Engine indicators numbered below 3737) can be readily changed from right-hand to left-hand instruments as occasion may require.

Crosby New Type Indicator.

The indicator illustrated in Fig. 287 differs slightly from the ordinary instrument, in that it has an outside spring, set

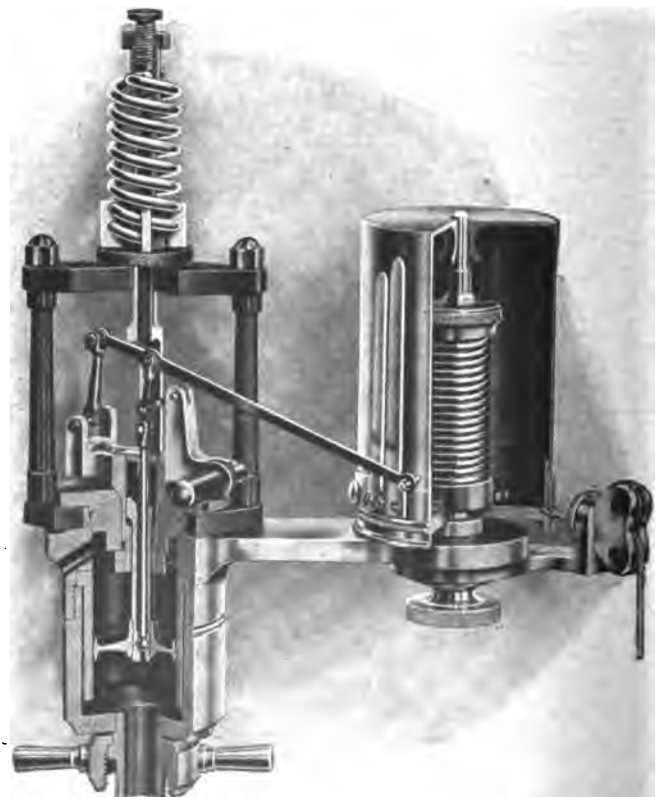


FIG. 287.

above the moving parts in the piston, where it will remain always cool, and will obviate errors due to heat affecting the spring. The piston, too, is larger than that ordinarily found in the indicator, and provides great active force with a very light pencil mechanism.

The piston is peculiar, for instead of being cylindrical, it is in form the central zone of a sphere. The spherical edge of the piston moves freely in the cylinder notwithstanding the fact that there may be eccentricity in the action of the spring. The piston is attached directly to the lower end of the piston rod which is connected to the upper end of the spring. The pencil mechanism is connected independently through the hollow piston rod by means of a rod having an adjustable ball and socket joint at the center of the piston, unaffected by any strains of the spring.

The Star Improved Indicator.

The Star Indicator, following the usual practice, consists of two principal units—the steam cylinder and the drum. The steam cylinder is in communication with the locomotive cylinder, so that its piston is subject to the same steam pressure as that within the engine cylinder at all times, and this pressure is recorded on a card which is wrapped around the outside of the drum. The drum, as is common to indicators, is driven from the engine crosshead by means of a cord, thereby receiving a reciprocating motion.

The sectional view of this indicator, as shown in Fig. 288, will make clear its construction and operation. As may be seen, it is of the exposed spring type, and is constructed with a very light piston assembly, making for accuracy. The Star Brass Manufacturing Co., of Boston, Mass., makers of the instrument, manufacture also an inside spring type indicator, which follows the usual accepted practice in construction.

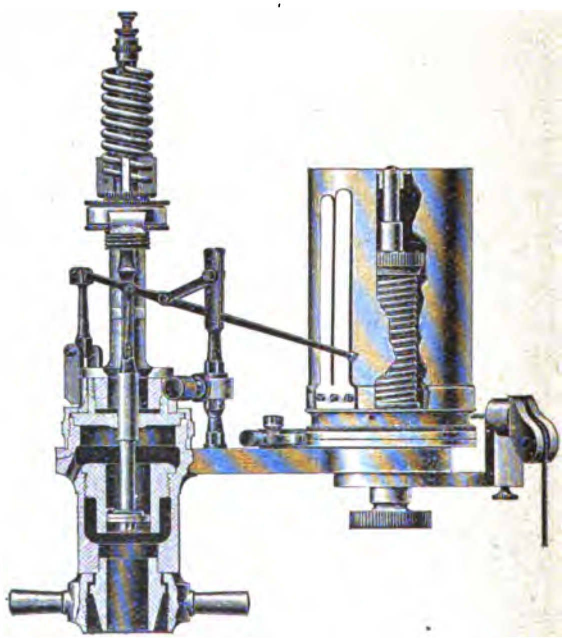


FIG. 288.

American Thompson Improved Indicator.

In Fig. 289, it will be noted, we present the American Thompson Exposed Spring Indicator, and, for the purpose of making clear its application, we have shown the Ideal reducing wheel in connection with it. The reducing wheel is fully described in the following pages.

The parallel motion feature of this indicator is said by the makers to be of extreme accuracy. The pencil's parallel movement is secured by a link attached to and governing the lever direct. The force required to guide the lever in its parallel movement is received on the pivots of the link alone, and the lever is guided by a short link, which is fastened to the stationary arm of the motion, and is connected to the piston rod by a yoke and screw.

The detent motion is a part of, and is contained within the drum and base of, this indicator. By its use several cards can be taken in the time ordinarily required for one, so that it shows great value on high speed work. Four cards in one minute have been taken in locomotive indicating, due to it.

FIG. 289.

This indicator is the product of the American Steam Gauge & Valve Mfg. Co., of Boston, Mass.

Tabor Indicator.

The Tabor steam engine indicator, type OP, is of the exposed spring type, and, as may be seen from the illustration, Fig. 290, does not greatly depart from ordinary practice, except in detail.

The cylinder of this instrument is made with inner and outer shell, making it possible to replace the inner, or wearing parts, without disturbing the outer shell.

The piston is one half square inch in area, the springs are of the Duplex type, and a stop device is provided, which prevents the pencil pressing too hard against the card.

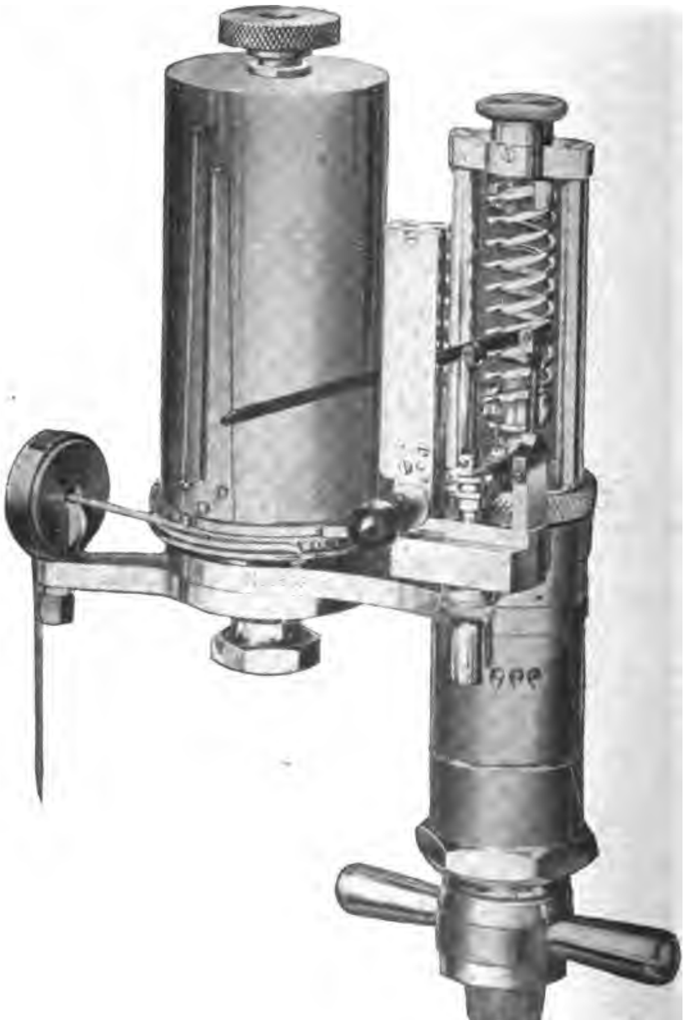


FIG. 290.

The parallel motion of the indicator is of peculiar construction. It consists of a vertical plate which is secured in an upright position, to the steam cylinder. In this plate is a curved slot in which works a roller revolving on a pin set in the pencil bar. The design of the slot and the location of the roller on the pencil bar exactly compensate for the pencil's tendency to move in an arc, and cause it to travel exactly parallel to the piston. This plate may be seen in the illustration, attached above the cylinder, and in front of the pencil bar.

This device is a product of the Ashcroft Manufacturing Co., of Bridgeport, Conn.

Drum Motion Devices.

The motion of the paper drum on the indicator may be derived from any part of the engine which has a movement coincident with that of the piston, but the crosshead is usually employed for this purpose.

The movement of the crosshead, actually, must be reduced, by appropriate mechanism, to the length of the diagram taken, and must be reduced in exact ratio to the piston motion, in regard to speed.

For this purpose are employed many devices, the best known being the pantograph, or lazy tongs, and the reducing wheel.

The pantograph, as illustrated in Fig. 291 gives theoretically perfect motion, but, owing to its many parts, is subject to erroneous results, as it may become shaky. However, it is widely used. The working end A receives motion directly from the crosshead, while the pivot B rests on a support. The cord is carried from the pin E directly to the indicator, and the motion of the drum is governed by adjustment of the pivots C and D which determine the position of the point E.

But the reducing wheel, as shown in Fig. 292, applied to the indicator, and detached in Fig. 293, has come into general use.

The illustration presented herewith shows the Crosby reducing wheel, which we shall describe as an example of modern reducing wheels.

It is attached directly to the cylinder cock of the steam engine, and has connected to it the indicator, forming a base, or support, for the latter.

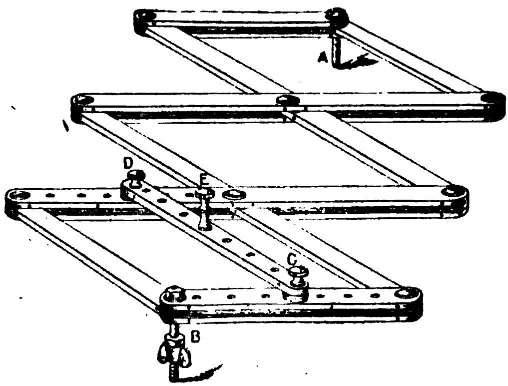


FIG. 291.

A cord, on a vertical pulley, transmits motion from the crosshead, through the reducing mechanism, to the drum of the indicator. Before attaching the cord to the crosshead of the engine, it should be drawn out to its full length to ascertain whether the cords on the indicator and reducing wheel have been properly adjusted. With the guide pulleys to the indicator drum directly over the stroke pulley of the reducing wheel, the indicator cord should be of the right length to prevent the paper drum from recoiling against its stop at the end of the stroke.

The reducing wheel is furnished with interchangeable pinion gears, and by this means the length of cord can be regulated within convenient limits for any length of engine stroke. The indicator diagram does not need to be of any definite length, but is usually taken between 2½ and 3½ inches long and about 1¼ or 1½ inches high.

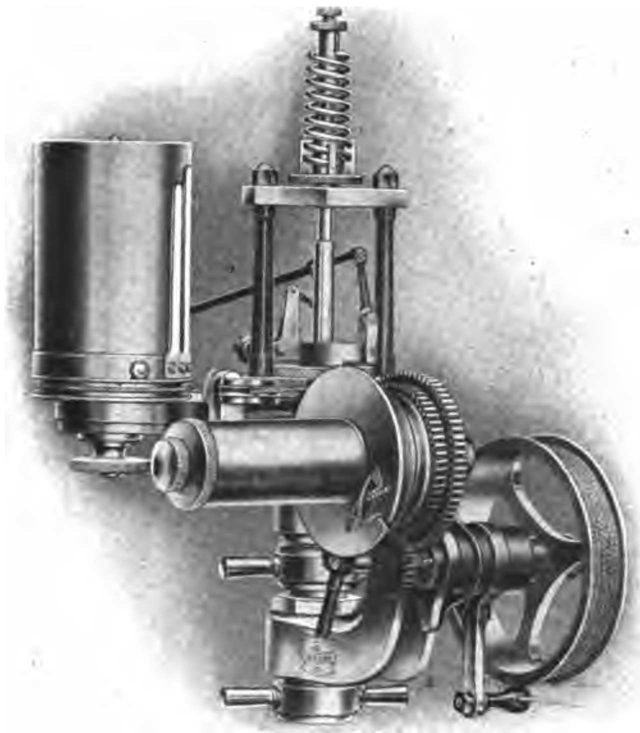


FIG. 292.

The long helical spring, inclosed in the tubular casing, affords the most efficient means for maintaining uniform tension upon the cord during the return stroke and is more active and more prompt than a flat volute spring could be in following the rapid change of speed of the crosshead, from maximum to zero, twice during each revolution of the driving wheel. The tension of the spring is easily adjusted, to meet various conditions of engine speed and length of cord, by means of the knurled head fitting upon a squared shaft.

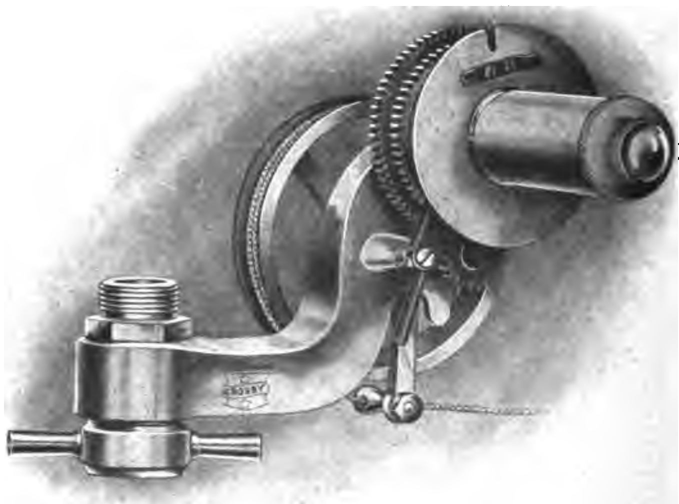


FIG. 293.

The cord supplied with the reducing wheel is specially made for the purpose and has the least possible amount of stretch. Ordinary cord may stretch three to ten times as much during each stroke and should not be used if accurate results are desired.

The detent device of the Crosby reducing wheel permits the taking of cards at convenient intervals, to show variable conditions in the operation of a steam engine to test the results of a change of load or of valve adjustment. By a movement of the small lever, the motion of the indicator drum may be instantly started or stopped without disconnecting the cord from the engine crosshead, and the Crosby detent mechanism allows the drum to continue to the end of its stroke, so that it may be started and stopped without shock to the instrument and without distorting or spoiling the diagram. The motion of the indicator drum accurately follows the travel of the crosshead at every point of the engine stroke and therefore this is the most convenient means for reducing the motion of high speed engines.

Planimeters.

In determining data from the indicator diagram, one of the principal objects is to ascertain the mean pressure on the piston of the engine through its entire stroke, in order to calculate the horse power developed. The pressure obtained in this connection is the "mean effective pressure" (M. E. P.); it is the difference between the average pressure on the forward stroke and that of the return stroke.

The M. E. P. is readily ascertained by means of the planimeter, shown in Fig. 294. This is an instrument used to measure the area of any plane surface, whatever be its outline, and a knowledge of mathematics is not essential to its satisfactory usage.

We have shown the planimeter in its simplest form, in which the point P is set at any convenient point near the surface to be measured, and acts as a pivot. The point F is made to carefully trace the outline of the enclosed surface, such as the indicator diagram. This will cause the small roller-wheel D to rotate, and the area traced by the point F will be recorded in square inches and decimals by the graduations on the roller-wheel D and the vernier E.

When the area of the diagram has been found, the M. E. P. is arrived at by calculations; the method being to divide the area by the length of the diagram, and multiply the quotient by the scale of the indicator spring used in obtaining the diagram.

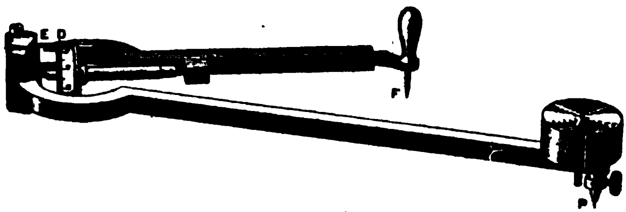


FIG. 294.

Suppose, for example, the area is found to be 6 sq. in., the length 4 in., and the scale of the indicator spring 50 lbs. per inch. Divide the area, 6, by the length, 4, and multiply the quotient, 1.5, by 50, which gives 75 lbs. per square inch, the M. E. P.

Besides the simple form of Crosby planimeter illustrated, there are other more complicated types, which will measure larger areas and give more accurate results, but they are alike in general construction and operation, differing only in detail.

The Locomotive Electric Headlight.

As a result of the universal operation of modern railway power at very high speeds, which necessitates, as a safety factor, sufficient illumination of the right of way, the electric headlight has come into very extensive use.

But to present a detailed explanation of the construction and operation of this installation would necessitate the use of many technical terms, confusing, rather than explanatory, to the ordinary person. For this reason we shall treat the subject broadly, sacrificing concise description to clearness and brevity.

The essential features of the system in general use are three. First we have the engine, known as a *turbine,* which furnishes the mechanical power to operate the *dynamo,* and is run by steam from the locomotive boiler.

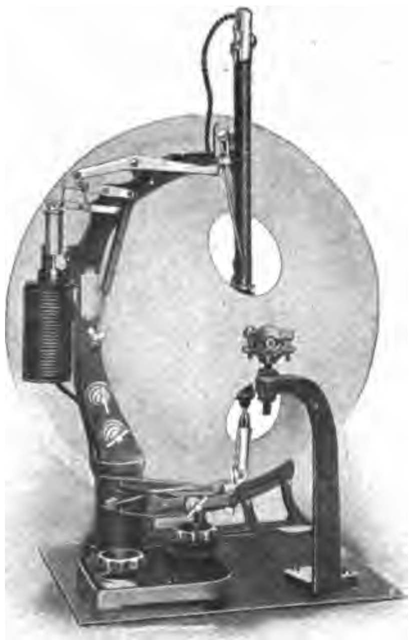


Fig. 295.

The *dynamo,* or generator, operated by the turbine engine mentioned, produces the light; that is, by means of its being operated by the turbine engine, the electric current is generated. Thus the name, *turbo-generator.*

The third basic, or essential feature, is the *lamp.* There are two kinds in use, the *incandescent,* such as the ordinary electric bulb found in the home lighting fixture, and the *arc lamp,* which consists of two vertical poles, usually one of carbon and one of copper, as shown in Fig. 295. These poles are set so that their points do not touch, and thus an arc is formed between them by the electric current passing through them, and this arc transforms the current into visible light.

FIG. 296.

This light is concentrated in the manner desired by a reflector, which is a part of the lamp. Both the reflector and the features of the arc lamp are shown in Fig. 296. The small incandescent bulb shown is merely for use in dimming the light, when desired.

This, in brief, is the complete assemblage. There are many points as to operation, lubrication, repair, and general care which vary with different types of lamp, so we shall not attempt to make them clear, as they might tend to confusion, but suffice it to say that the system described meets the most exacting requirements and is widely used.

As a point of interest, Fig. 297, is a reproduction of a photograph taken at night, without any lighting other than that of a Pyle-National electric headlight, properly focused. It will be observed that the right of way is very clearly illuminated, and that the beam of light is very penetrating, for the bridge shown in the illustration is a half mile distant from the locomotive.

The equipment herein described is the product of The Pyle-National Company, of Chicago, Illinois.

FIG. 297.

Automatic Cylinder Cock.

An engine standing in exposed places during freezing weather, soon fills the cylinder with condensed steam—especially if the throttle is leaking slightly—and, even though the regular cocks are opened when starting, the volume of water is too great to pass out during the piston stroke, and damage to the cylinder may result.

This is prevented by the use of the automatic cylinder cock, shown in Fig. 298. This cock is automatic in action, for, while the engine is running, the ball is lifted by steam pressure and held tightly against the seat, thus shutting

FIG. 298.

off any communication between the cylinder and the atmosphere. As soon as the throttle is closed, however, the pressure

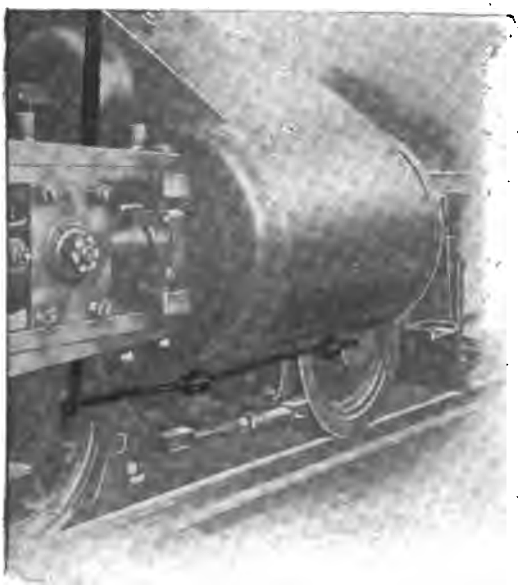


FIG. 299.

against the ball is released, so that it automatically falls away from its seat and lies at the bottom of the chamber, as shown, thus opening a clear passage between the inside of the cylinder and the atmosphere, so that any condensed water present is drained from the cylinder. Immediately upon starting, the ball is again seated by the inrushing steam, closing the cock, and preventing flying steam and oily water from escaping from the cylinder.

As Fig. 299 shows, the cock is provided with means of auxiliary operation from the cab. The plunger shown is bolted to a standard size slide rod, supported by ears cast on the body of the cock. This bar is then connected to the regular cylinder cock rigging, so that the engineer is enabled to operate the cocks by hand when the locomotive is running.

This device is handled by the Watertown Specialty Co., of Watertown, N. Y.

Extended Piston Rod Guide.

The purpose of this design, as manufactured by the American Locomotive Company, was to produce a self-centering guide for piston tail rods in locomotives which could be erected, removed and replaced without requiring lining, at the same time which would exactly coincide with the longitudinal axis of the cylinder. Another object in view is to provide a guide having such ample wearing surface that it would run for two or three years without requiring adjustments, lining or repairs of any kind. This is accomplished by making the guide so that it can be bored out and faced off at one setting on the machine. Its circular face registers with a corresponding face on the front cylinder head, and furthermore, the fact that the guide surface is struck from the center of the cylinder, any refinement in adjustment of the shoe on the front of the rod and crosshead at the back is unnecessary, because, while the main crosshead works in a flat guide, the tail rod shoe will swing around on the center of the cylinder so that it will always take a fair bearing without cramp-

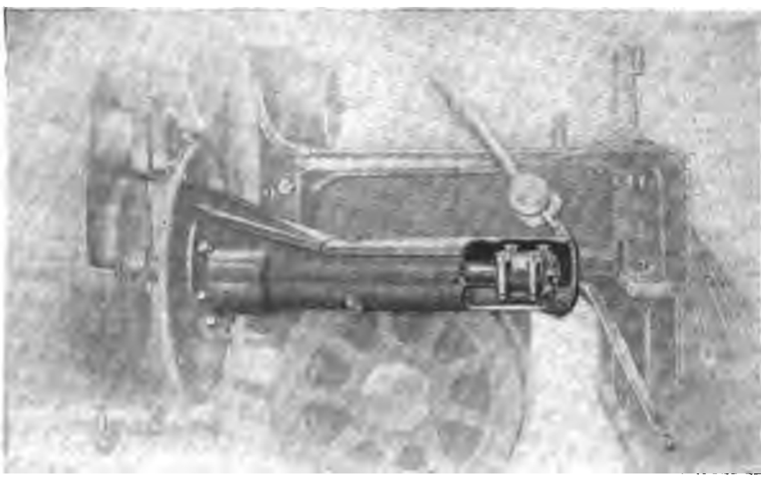


FIG. 300.

ing. The casing is made open at the top of horse shoe form with a corresponding opening in the flange, as shown in Fig. 300, so that it can be removed in case the engine stands on the forward center, without removing the pilot and the bumper. It is provided with a dust tight top and cover and with oil stops so that the shoe always runs in oil. Experience with these types on a large railroad of the northwest for nearly two years, indicates that the wear on the shoe and guides in two or three years will not exceed 1/16", so that relining will not be necessary between general shoppings. In cases where the wear has become pronounced so as to allow the piston head to trail or drag on the bottom of the cylinder, liners may be inserted under the shoes by slacking out four bolts, or where the shoe is made solid its surface can be retinned.

The purpose of the piston rod extension guide is to prevent excessive wear and friction on the bottom of the cylinder by keeping the surface of the piston head out of contact with the cylinder barrel. It is evident that if this can be done in a satisfactory manner without requiring constant watching or raising up of the bearing surface, that the life of the piston head and its rings will be prolonged. Furthermore the wear on the cylinder barrel will be substantially decreased. This is especially desirable where highly superheated steam is used. Another advantage is that cast steel piston heads without wearing surface of cast iron or bronze may be employed for this purpose, which makes a very satisfactory and light piston, provided the steel surface is not allowed to come in contact with the polished cylinder walls.

Valve Stem Guide.

This device, developed by the American Locomotive Co., consists of a guide made integral with the back head of a piston valve chamber, and is advantageous principally from the fact that it can be erected, taken down and replaced without lining up, thus insuring that the valve stem guide is absolutely in line with the piston valve chamber. It is self-supporting and self-centering, so that no bracing from the guides or from any other source than the cylinder is required, as the illustration, Fig. 301, makes clear. The combination lever is made straight without forks and is connected to the crosshead by a pin passing through

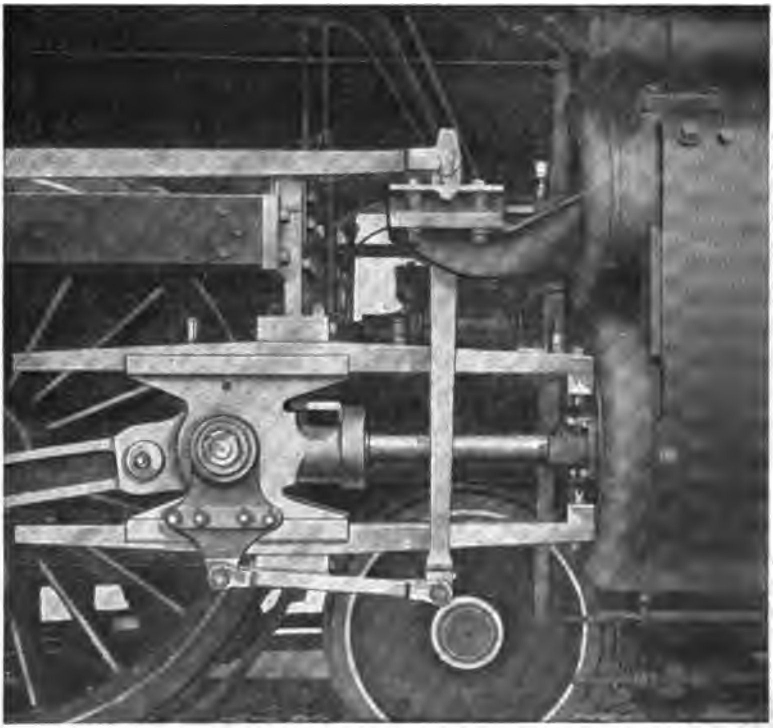


FIG. 301.

its wings, which affords considerable lateral stability usually lacking in other designs. In case of wear liners are provided on the top and bottom which can be removed or inserted, as may be required.

METALLIC PACKING.

The use of metallic packing for locomotive piston rods and valve stems has become practically universal. Especially is this type of packing necessary to provide against leakage on locomotives whose stuffing boxes are subject to the action of superheated steam.

Satisfactory results from any type of packing, more particularly when under the pressure of superheated steam, is dependent, to a great extent, upon thorough and proper lubrication, and it is very good practice to use a drifting valve, in connection with the throttle, to relieve the packing of pressure when drifting and to assist lubrication, thereby preventing the possibility of scoring the piston rod, with its resultant loss.

The King Tandem Type Piston Rod Packing.

This packing consists of two single King packing rings, 6, arranged in tandem on the rod, as illustrated in Fig. 302, and is very effective with superheat. The inner set is divided from the outer by the bushing 12. Each packing is entirely separate and independent, so that, should the inner set fail, the outer is in place on the rod, to take up the work. The inner packing ring is constructed of special (superheated) metal, and the outer of ordinary packing ring metal, such as is used in the single type packing assemblage, manufactured also by The United States Metallic Packing Company, of Philadelphia, Pa.

As numbered, the parts of this packing construction are: a swab holder, 1; swab holder discs, 2; gland, 3; sliding plate

FIG. 302.

ring, 4; sliding plate half pieces, 5; packing ring, 6; retainer half pieces, 7; retainer shell, 8; preventer, 9; spring, 10; swabbing, 11; and dividing bushing, 12.

King Locomotive Valve Stem Packing.

This valve stem packing is also a product of the United States Metallic Packing Co., containing only one ground joint, that between the sliding plate, 4, and the gland, 3, Fig. 303. The bevel of the packing ring, 6, is toward the steam chest, and bears against the retainer, 7. The support, 12, is not used in packing for engines having piston valves. The other parts of the assemblage are: 1, swab holder; 2, swab holder discs; 9, preventer; 10, spring; 11, braided cotton swab (small size); 13, oil cup.

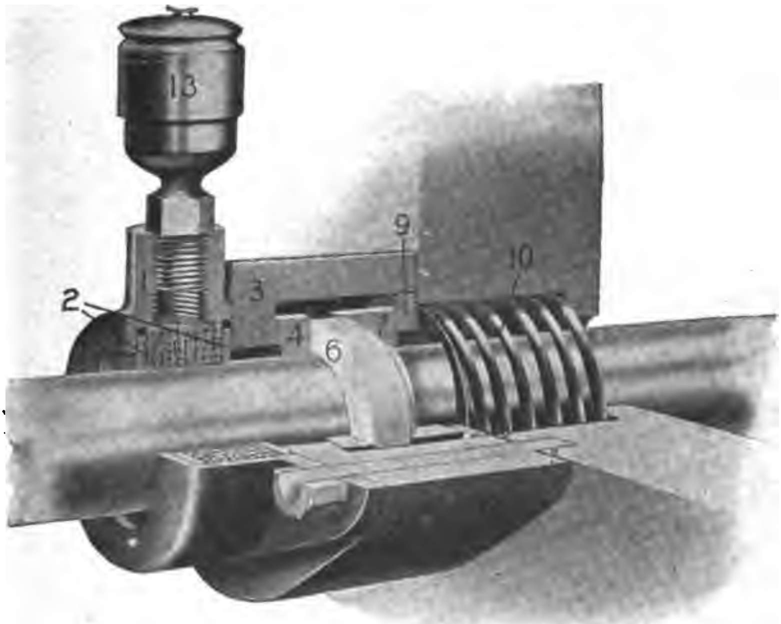


FIG. 303.

The Q and C Packing and Lubricator.

Another type of metallic packing, designed to give satisfaction with superheat, is the Q and C packing and lubricator assemblage, shown in Fig. 304. The lubricator is shown on the left, outside of the gland A.

The packing parts consist of a retaining sleeve, C; sleeve bushing, D; packing ring, E; packing ring shoe, E1; packing ring, F; spring stop, G; and compression spring, H.

When the locomotive is drifting the pressure on the packing is released, as there are no springs to hold it hard against the rod. The wear on the rod is reduced to a minimum by this feature, and scoring is prevented, even if the rod does not receive proper lubrication.

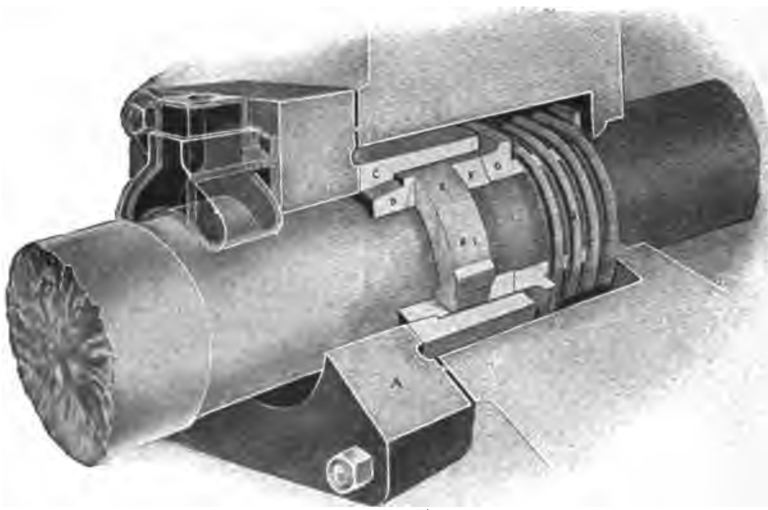


FIG. 304.

The Q and C Company, of New York City, are the makers of this packing.

The L. & K. Packing.

The illustration, Fig. 305, shows the L. & K. packing as applied to a piston rod. The parts are: A, back cylinder head; B, stuffing box bushing; C, L ring, forming grooves for packing; D, gland; F, packing segments; F1, feeding tongues on packing segments; G, split cast iron re-enforcing ring.

Parts B, C, and D are bored ½" larger diameter than the rod, which allows them to pass over any enlarged crosshead fit up to that size. The re-enforcing rings G have nothing to do with making a steam tight joint, but are to prevent steam pressure from forcing the soft packing rings through the openings between the piston rod and the bore parts C and D, and, therefore, the rings G are bored only 1/32 inch larger than the rod, to prevent scoring it.

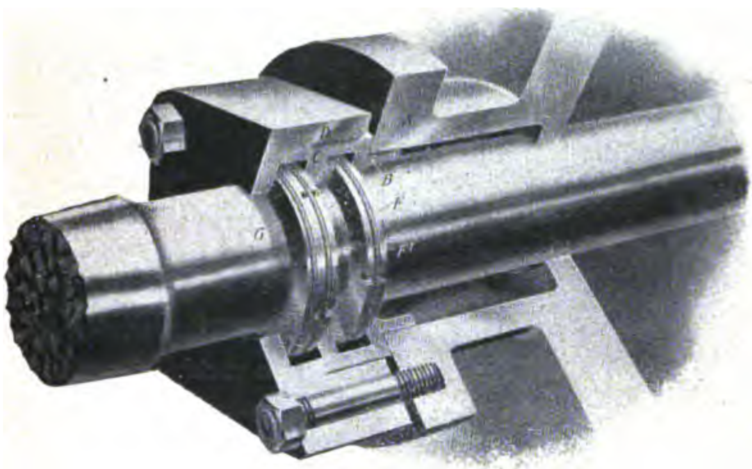


FIG. 305.

Fig. 306 shows the packing applied to the valve stem, and also illustrates that it is essentially the same as that applied to the piston rod.

For slide valves, the stuffing box bushing B^1 is made of bronze, and supports the weight of the stem and yoke, at the same time acting as a guide for the rod. However, for piston valves, the bushing B is used instead. It is made of cast iron. For valve stems, the re-enforcing rings, G, Fig. 305, are not necessary, as there is no enlargement of the rod, and, accordingly, the parts B, C, and D are bored only ⅛ inch larger than the valve stem.

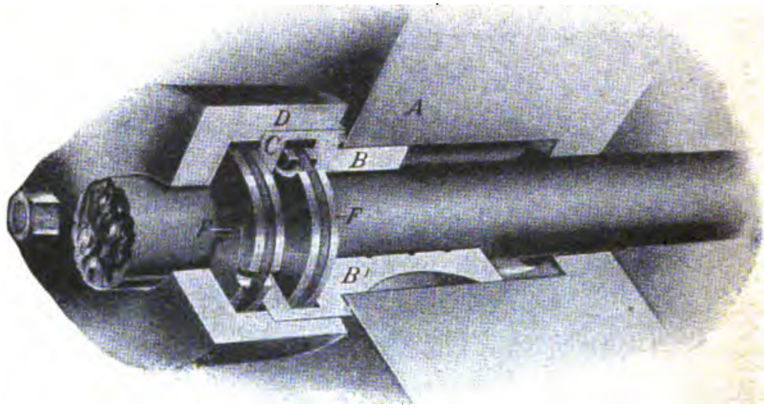


FIG. 306.

The Aurora Metal Company, of Aurora, Ill., manufacturers of the L. & K. packing, claim that, as the packing depends entirely upon the steam pressure, and not upon springs, to make tight joints, there is practically no friction on the rod when the engine is not working steam.

COMPOUND LOCOMOTIVES.

The compound locomotive is one having two or more cylinders so arranged that the exhaust from one is utilized in another, and made to perform additional work in moving the locomotive, before being exhausted, or discharged, to the atmosphere, through the exhaust nozzle and stack.

The era of compounding engines, as applicable to locomotives, is due to Mr. Anatole Mallet (pronounced Mallay), of France, who, in 1876, designed successful two-cylinder compounds for the Bayonne and Biarritz Railroad, of that country. Mallet was responsible for the earliest work of real practical value in compound locomotive designing.

Following Mallet, much was accomplished, regarding compounding, by von Borries in Germany, Wordsell and Webb in England, and Vauclain in America, as well as many others, all at about the year of 1889. Development continued until, after fifteen years, or in 1904, there were existing approximately three thousand compound locomotives.

As a better understanding of the various compounding systems was gained by experience, so that this type of locomotive was improved greatly both in construction and operation, its merits became more widely and thoroughly recognized, until, at the present time, it is an undisputed fact that the compound will do more work for a given amount of fuel consumed, than will the simple engine of like size and design. This, after all, is the object of compounding—increased economy—so it may well be said that the compound locomotive has proven successful.

There are several types of compound locomotives, which may be divided into two groups—two cylinder and four cylinder. In England, the Midland Railway built a few three cylinder types, in which one high pressure cylinder exhausted into two low pressure cylinders, but this method of compounding is very unusual. In the two cylinder type, called the cross compound, the high pressure cylinder, on one side, exhausts through a receiver in the smoke box, to the low pressure cylinder on the other side. A means of working the engine simple when desired, known as an intercepting valve, is provided.

Several different constructive arrangements are employed in the design of four cylinder compounds. They are described and illustrated in the following pages as the Vauclain, the Tandem, the Balanced, and the Articulated compounds.

The Articulated locomotive is one having two sets of cylinders driving independent sets of wheels. There are, then, two sets of frames, and these are joined by pivot joints, or hinges. The front end of the boiler is supported on a sliding bearing by the leading frame and wheels, which swivel about the pivot connection, and are in effect a truck. Thus the rigid wheelbase is materially reduced. In some designs, however, the forward part of the boiler is rigidly secured to the frames, and the swivel effect is secured by a flexible, or articulated, boiler.

With this type of locomotive, the Mallet Articulated, the term is applied without reference to the system of compounding; in fact, they are sometimes built with four simple cylinders, doing away entirely with compound features. However, the articulated compounds, as built by the American and the Baldwin locomotive companies, are described at length in the following pages.

The Mallet articulated compound is also built with three sets of cylinders and drivers (triplex), and the third group is placed under the tender. In this type, the two middle cylinders, driving the center group of drivers, are of the high pressure type. The right high pressure cylinder exhausts into a receiver which supplies the front pair of low pressure cylinders, while the exhaust steam from the left high pressure cylinder, through a receiver, operates the rear pair of low pressure cylinders.

The advantages of compounding a locomotive, as outlined above, and described in detail in the following pages, must be sufficient to warrant the added complications in design and maintenance—otherwise the compound would not be in such wide use. Its principal advantage is its ability to utilize a greater degree of expansion of the steam—to make use of the live steam through a much greater range of temperature before its final exhaust—and in this way to consume less steam for a given amount of work performed than is possible with a single expansion, or simple engine. In the two cylinder compound, the maximum of available power is limited by the diameter of the low pressure cylinder, which, in turn, is limited by the clearance of any particular road. The two cylinder type, however, possesses the disadvantage of poor balance, which is destructive to the roadbed. But the four cylinder type possesses even a better degree of balance than the ordinary simple locomotive, as the cranks are so set in reference to each other as to bring about a more uniform application of power to the drivers.

The Vauclain Four-Cylinder Compound.*

The original Vauclain four-cylinder compound was constructed in 1889. This type has nearly all the points of sim-

*Courtesy of the Baldwin Locomotive Co. of Philadelphia, Pa.

plicity shown by a single expansion locomotive. On each side a high and low pressure piston take hold of a single crosshead, and the steam distribution is controlled by a single balanced

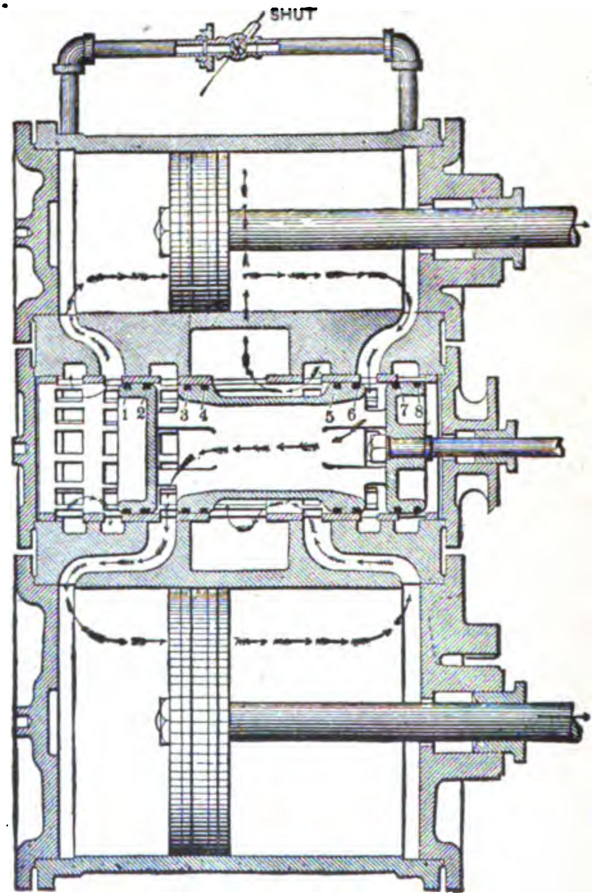


FIG. 307.

piston valve, so that the machinery is the same as in the single expansion, except that four cylinders and pistons are used instead of two.

The distribution of steam in the four-cylinder compound is shown by Fig. 307. The arrangement of the cylinders in relation to the valve is, for convenience of reference, somewhat distorted, the valve being shown in a vertical line between the two cylinders. The valve employed is of the piston type, working in a cylindrical chest located in the saddle casting, between the cylinders and the smokebox, and as close to the cylinders as convenience will permit. The steam chest is lined with a cast iron bushing, having port openings machined to exact dimensions, so that the admission of steam will be uniform under all conditions. By the use of this bushing, repairs can be made from time to time, and changes effected in the port openings, without the necessity of altering the cylinder castings. The bushing is forced into place by hydraulic pressure.

The function of the valve is to control the admission and exhaust of both cylinders. Live steam enters the chest at both ends of the valve, and is admitted to one end of the high-pressure cylinder. The exhaust from the high-pressure cylinder passes through the central hollow portion of the valve and supplies the low-pressure cylinder, while, at the same time, the steam in the opposite end of the low-pressure cylinder is allowed to escape under the valve to the final exhaust in the stack.

In order to obtain the maximum starting power in any compound locomotive, it is necessary to employ some means of admitting live steam to the low-pressure cylinder. The device for this purpose in the Vauclain four-cylinder type is a by-pass valve, which is opened to allow the steam to pass from one end of the high-pressure cylinder to the other, and from thence to the low-pressure cylinder.

The proper use of this valve is essential to the successful performance of the locomotive. The starting valve should never be open at speeds exceeding five miles per hour, unless the engine is drifting down grade with a closed throttle. Failure to comply with this rule will result in excessive repairs and high fuel consumption, and the locomotive will be "logy" in operation. The starting valve should always be closed before the reverse lever is hooked back.

Owing to the mild exhaust of a compound locomotive the fire is not torn when working the engine full stroke; hence in ascending grades, the reverse lever can be dropped forward, thus keeping up the speed without injury to the fire. If the engine is so heavily loaded that the speed drops to five miles per hour, and there is danger of stalling, the starting valve may be opened in order to keep the train moving; but this valve should be closed as soon as the difficulty is overcome.

If a locomotive of this type exhausts unevenly, or is lame, a careful inspection should be made of the starting valve levers and connections, as it is important that the two valves open and close simultaneously. The various parts of the valve gear should also be carefully examined, as bent eccentric rods or transmission rods, loose rocker boxes, etc., invariably cause trouble. If the valve gear and starting valve rigging are in good condition, the valve packing rings and piston packing should be examined for leaks or blows. The valve packing rings are numbered on the illustration. Admission and release of steam to the high-pressure cylinder is controlled by rings 1, 2, 7 and 8, and to the low-pressure cylinder by rings 3, 4, 5 and 6. These rings may be tested for blows as follows:

Rings 1, 2, 7 and 8. Place the valve in its middle position by means of the reverse lever, so that all the ports are covered. Open the throttle, and a leak past these rings will be shown by a steady escape of steam at the high-pressure cylinder cocks.

Rings 3, 4, 5 and 6. Place the reverse lever in full gear, with the starting valve open and driving brakes applied. Open the throttle, and a leak in the rings will be indicated by a steady blow through the exhaust nozzle.

To test the high-pressure piston rings, place the engine at about quarter stroke, admitting steam to the front end of the high-pressure cylinder. Keep the starting valve closed and the driving brakes applied. If steam leaks past the piston it will escape in a steady stream at the front cylinder cock.

To test the low-pressure piston rings, keep the engine and valve in the same position, and open the starting valve. A leak past the rings will be indicated by a steady blow at the back low-pressure cylinder cock.

The testing of the valves and pistons for leaks and blows should always be done when the cylinders are hot and well lubricated.

If, because of a breakdown, it becomes necessary to disconnect a locomotive of this type, the engine is handled exactly as a single expansion locomotive.

When the valve is placed in its central position, all the ports are covered, as in the case of an ordinary slide valve.

Two-Cylinder Compound.

The essential features of this design, brought out in 1898, are the intercepting and reducing mechanisms. These, when in normal position, permit the locomotive to operate by single expansion and so continue until changed to compound. The locomotive is therefore readily started in any position of the crank.

In Figs. 308 and 309, A is a double piston intercepting valve, located in the saddle casting of the high-pressure cylinder. In one direction the movement is controlled by a spiral spring, in the other by steam pressure. The function of the intercepting valve is to cause the exhaust steam from the high-pressure cylinder to be diverted, at the option of the engineer, either to the open air when working single expansion, or to the receiver when working compound. C is a reducing valve also placed in the saddle casting of the high-pressure cylinder, and like the intercepting valve, is moved in one direction by a spiral spring, and in

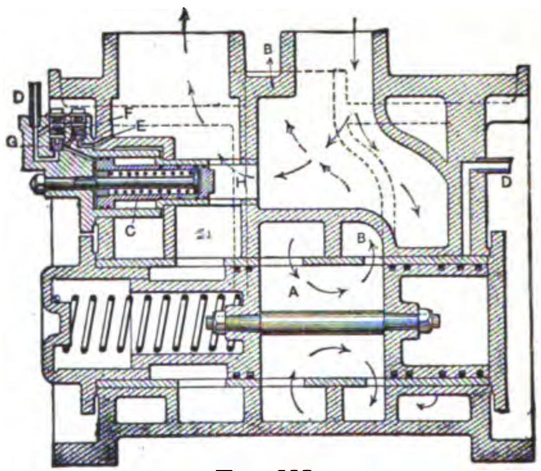


FIG. 308.

the opposite direction by steam pressure. The function of this valve is, in its normal position, to admit live steam into the receiver at reduced pressure while the locomotive is working single expansion. When the locomotive is working compound, this valve automatically closes, as it is evident that there is no further need of live steam in the receiver. A further function of the reducing valve is to regulate the pressure in the receiver so that the total pressure on the pistons of the high and low-pressure cylinders may be equalized. The steam for controlling the operation of both intercepting and reducing valves is supplied through the pipes D from the operating valve in the cab.

When not permanently closed by pressure in the pipes D the reducing valve C is operated automatically by the pressure in the receiver. To this end the port E is provided, communicating with the receiver and the space in front of the reducing valve; as the pressure rises the steam acts on the large end of the

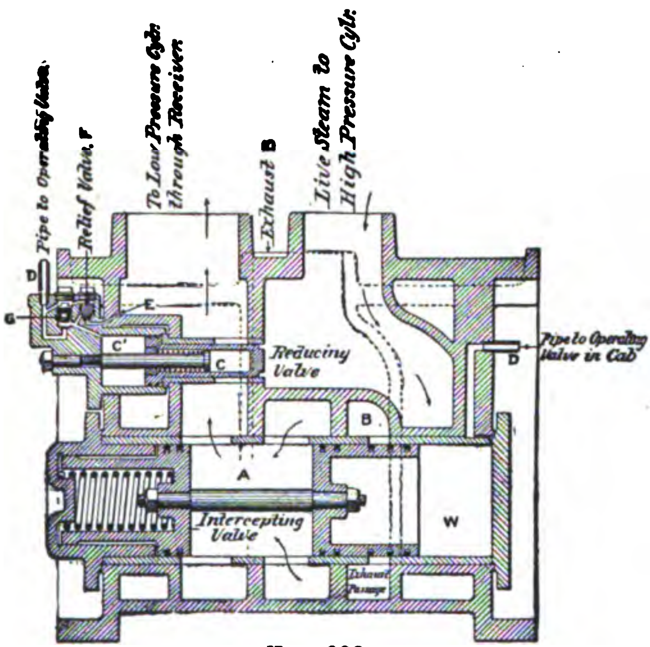


FIG. 309.

reducing valve, causing it to move backward and close the passage H through which steam enters the receiver, and thus prevent an excess pressure of steam in the low-pressure cylinder. Poppet valves F and G are placed in connection with the port E, one to prevent the escape of steam from the receiver to the pipe D when the locomotive is working single expansion, and the other to close the passage from pipe D to the receiver when working compound.

Normally the lever of the operating valve in the cab is in the position marked "simple." In this position no steam is allowed to enter the pipes D and no pressure will be exerted on the intercepting and reducing valves in opposition to the springs, and they will assume the positions shown in Fig. 308. The ports of the intercepting valve A stand open to receive the exhaust steam from the high-pressure cylinder and deliver it through the exhaust passage B to the atmosphere. The reducing valve is open, admitting live steam through passage H to the receiver and from thence to the low-pressure cylinder. The receiver pressure is governed by the automatic action of the reducing valve as previously explained. In this way the locomotive can be used single expansion in making up and starting trains for switching and slow running.

At the will of the engineer the operating valve in the cab is moved to the position marked "Compound." This admits steam to the pipes D and through them to the valve chambers W and C^1, changing the intercepting and reducing valves instantly and noiselessly to the positions shown in Fig. 309. The exhaust from the high-pressure cylinder is diverted to the receiver, the admission of live steam to the receiver is stopped by the closing of the passage H, and the locomotive is in position to work compound.

Both valves are of the piston type, with packing rings to prevent leakage. This insures an easy movement of the valves, and prevents the hammering action common to valves of the poppet type when automatically operated.

A locomotive of the two cylinder compound type is tested for leaks or blows in the same manner as a single expansion engine. The tests should be made when the engine is working single expansion at slow speed, with the cylinders warm and well lubricated. In case of a break-down, the engine can be disconnected as readily as a single expansion locomotive and in exactly the same manner; the main rod should be taken down, the crosshead blocked and the valve placed in its central position to cover all

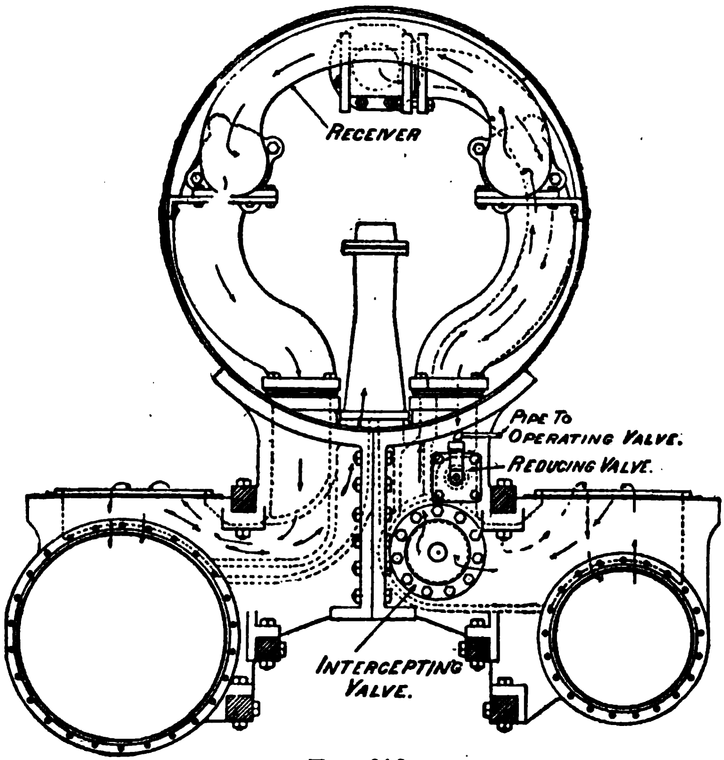


FIG. 310.

ports. In all cases, regardless of which side is disabled, the intercepting valve must be in position for working single expansion.

The Tandem Compound.

In this type of locomotive, designed in 1902, principally for heavy freight service, four cylinders are used, with a high and low-pressure cylinder and cylindrical valve chest on each side. The high-pressure cylinder is placed in front of the low-pressure, both having the same axis; that is, the center of the low-pressure cylinder extended becomes also the center of the high-pressure.

Each cylinder with its valve chest is cast separately and is separate from the saddle. The steam connections are made by a pipe from the saddle to the high-pressure valve chest, and the final exhaust takes place through an adjustable connection between the low-pressure cylinder and the saddle casting. The valve, which is double and hollow, admits steam to the high-pressure cylinder, and at the same time distributes the high-pressure exhaust from the front end of the high-pressure cylinder to the back end of low-pressure cylinder or *vice versa,* as the case may be, without the necessity of crossed ports. As shown in the accompanying illustration, Fig. 311, A is the high-pressure valve by which steam is conducted from the live steam openings through external cavities B and B to the high-pressure cylinder. The exhaust from the high-pressure cylinder passes through the opening C to the steam chest, which acts as a receiver; D is the low-pressure valve connected to the high-pressure valve by valve rod E. This valve in its operation is similar to the ordinary slide valve. The outside edges control the admission, and the exhaust takes place through the external cavity F. The starting valve connects the live steam ports of the high-pressure cylinder.

In Fig. 311, the valve packing rings are numbered 1 to 12. Rings 3, 4, 5 and 6 control the distribution of steam to the high-pressure cylinder, and rings 9, 10, 11 and 12 serve the same purpose for the low-pressure. Rings 1, 2, 7 and 8 prevent live steam from passing directly into the steam chest.

Rings 3 and 6 may be tested for leakage by placing the valve in its middle position and opening the throttle. If these rings leak, steam will enter the high-pressure cylinder and escape from the cylinder cocks.

To test rings 1, 2, 7 and 8 place the engine in full forward gear with the pin on the bottom quarter. Apply the driving

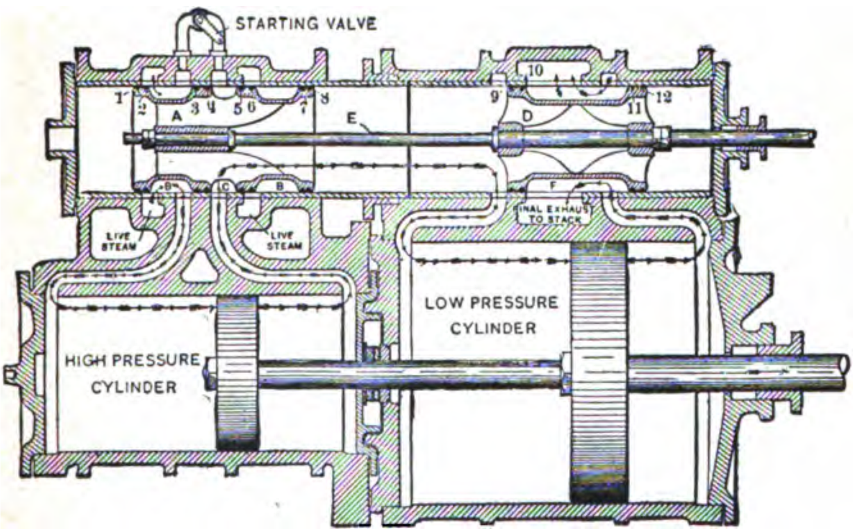


FIG. 311.

brakes and open the throttle. If these rings leak, steam will enter the steam chest and escape from the front low-pressure cylinder cock.

Rings 9, 10, 11 and 12 may be tested by placing the engine in full gear, with starting valve open and driving brakes applied. Open the throttle, and if these rings leak there will be a steady blow through the exhaust nozzle.

The high-pressure piston packing rings may be tested in the same manner as valve rings 1 and 2. Steam leaking past the piston will escape at the front low-pressure cylinder cock.

To test the low-pressure piston packing rings, keep the engine in the same position, and open the starting valve, which gives an increased pressure in the low-pressure cylinder. If the low-pressure piston rings leak, steam will escape at the back low-pressure cylinder cock.

As far as the use of the starting valve is concerned, an engine of this type should be handled like the Vauclain four cylinder type previously described. The effects of leaving the starting valve open are not as serious in the tandem type as in some others, but in order to realize the full economies due to compounding, this valve must be kept closed at speeds exceeding five miles per hour.

In the event of a break-down on the road, a locomotive of this type should be handled like a single expansion engine. It is impossible to use either cylinder independently of the other on the same side; therefore in the event of any damage the main rod should be removed, and the valve securely blocked in its middle position, thus covering all the ports.

The Balanced Compound.

In all two cylinder locomotives, whether single expansion or compound, and in four-cylinder types such as the tandem and the original Vauclain compound, the reciprocating parts are counterbalanced by rotating weights in the driving wheels.

This arrangement of balance becomes unsatisfactory, particularly for heavy locomotives, and when extremely high speeds are attained. By balancing the reciprocating parts against each other, the rotating balance in the wheels, used to complement these parts, can be eliminated, avoiding to a great extent the vertical shocks, and reducing the strain upon the track to that directly due to the weight of the locomotive. Consequently, with a self-balanced arrangement of reciprocating parts, the weight on the driving wheels may be increased without damaging the track, and higher speed is attainable without undue strain upon the working parts of the locomotive.

The balanced compound, as designed by S. M. Vauclain and first built by the Baldwin Locomotive Works in 1902, is intended to accomplish these results and simplify, as far as possible, the arrangement of the working parts.

The cylinders are a development of the original Vauclain four-cylinder compound type, with one piston valve common to each pair. Instead of being superimposed and located outside of the locomotive frames, the cylinders are placed horizontally in line with each other, the low-pressure outside and the high-pressure inside of the frames. Each valve is placed above and between the two cylinders which it is arranged to control. A separate set of guides and connections is required for each cylinder.

The two high-pressure cylinders being placed inside the frames, the pistons are necessarily coupled to a crank axle. The low-pressure pistons are coupled to crank pins on the outside of the driving wheels. The cranks on the axle are set at 90° with each other, and at 180° with the corresponding crank pins in the wheels. The pistons therefore travel in opposite directions, and the reciprocating parts act against and balance each other to the extent of their corresponding weights.

The distribution of steam is shown in Fig. 312. The live steam port in this design is centrally located between the induction ports of the high-pressure cylinder. Steam enters the high-pressure cylinder through the steam port and the central external cavity in the valve. The exhaust from the high-pressure cylinders takes place through the opposite steam port to the interior of the valve, which acts as a receiver. The outer edges of the valve control the admission of steam to the low-pressure cylinder. The steam passes from the front of the high-pressure cylinder through the valve to the front of the low-pressure cylinder, or from the back of the high-pressure to the back of the low-pressure cylinder. The exhaust from the low-pressure cylinder takes place through external cavities under the front and back portion of the valve, which communicate with the final exhaust port. The starting valve connects the two live steam ports of the high-pressure cylinder to allow the steam to pass over the piston.

In this type of locomotive the distribution of steam to the high-pressure cylinders is controlled by rings 5, 6, 7 and 8. To test these rings for leakage, place the valve in its middle position and open the throttle. Steam leaking past the rings will then escape from the high-pressure cylinder cocks.

If the rings controlling the distribution of steam to the low-pressure cylinder leak, there will be a steady blow from the exhaust when the engine is standing with starting valve and throttle open. This test should be made when the reverse lever is in full gear and the driving brakes are applied.

To test the high-pressure piston packing rings, place the engine with the outside main pin on the bottom quarter and the reverse lever in full forward gear. Open the throttle, keeping the driving brakes applied and starting valve closed. A leak past the high-pressure piston will result in a steady escape of steam at the front low-pressure cylinder cock.

To test the low-pressure piston packing, keep the engine in the same position and open the starting valve. If there is a leak in the low-pressure rings, steam will blow from the back low-pressure cylinder cock.

If the valve gear of a balanced compound locomotive becomes disabled on either side, the valve on that side should be blocked in its middle position, thus covering all the ports. If possible

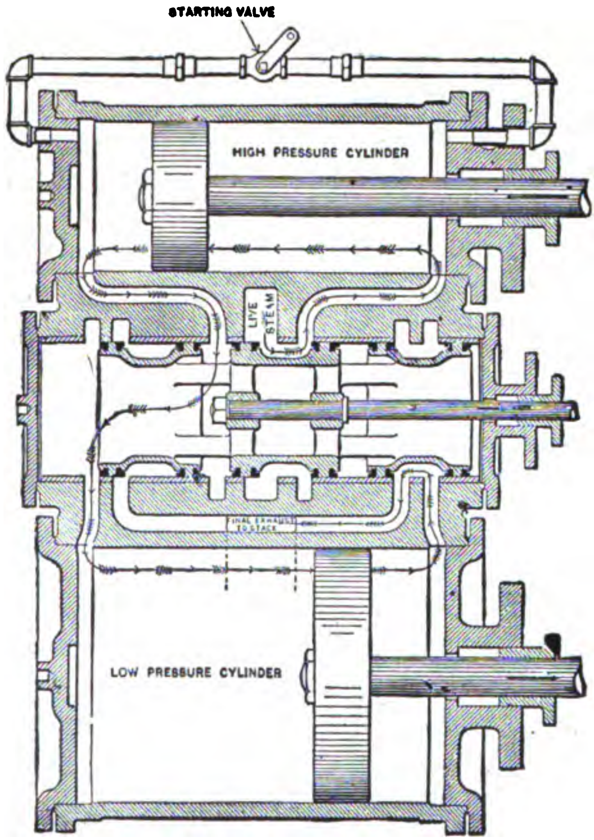


FIG. 312.

both main rods should be taken down, and the engine run with one side only.

It is possible, in an engine of this type, to run with either the high or low-pressure cylinder cut out on either side, provided the cylinder castings and heads, and the valve gear, remain intact. Thus if it is necessary, because of a bent piston rod, broken crosshead pin, or other cause, to cut out one of the high-pressure cylinders, the corresponding main rod should be removed and the crosshead securely blocked at about mid-stroke. By opening the starting valve, some power can then be developed on the damaged side. Similarly if the low-pressure main rod is removed, the low-pressure crosshead could be blocked at mid-stroke and the engine run with starting valve closed. In this case, especially when working full stroke, the back-pressure on the high-pressure piston would be excessive, and the blocking would have to be very secure to hold the low-pressure piston in place.

The Baldwin Articulated Compound.

This type of locomotive, as built by the Baldwin Locomotive Works, possesses fewer of what may be called compound features than any of the types previously described. This is because the Mallet type practically consists of two engines under one boiler, and each of these engines, as far as its steam distribution is concerned, operates like a single expansion locomotive. The rear group of wheels is operated by the high-pressure cylinders, and the forward group by the low-pressure cylinders. The front frames are pivoted to the rear frames, in order to make the wheel base sufficiently flexible. The receiver pipe, which conveys the high-pressure exhaust steam forward to the low-pressure cylinders, is provided with flexible joints so that it can accommodate itself to the position of the front frames when the engine is traversing curves.

The steam distribution in an engine of this type may be controlled by either slide or piston valves, and these are of the same construction as those applied to a single expansion locomotive. The valve motions for the high and low-pressure engines are controlled simultaneously, preferably by some form of power reverse mechanism.

In order to develop full starting power in a locomotive of the Mallet type, it is necessary to admit live steam to the low-pressure cylinders before the high-pressure exhaust has filled the receiver pipes. A starting valve is provided for this purpose; it is placed in the cab, and when opened, steam is admitted, through a small pipe, direct to the receiver pipe, and thence to the low-pressure cylinders. After the engine has made a few revolutions and the high-pressure exhaust is filling the receiver pipe, the starting valve should be closed. There is no advantage in keeping this valve open after the train is under way.

In the event of a breakdown, a locomotive of this type is handled in the same manner as a single expansion engine. Either one of the high or low-pressure cylinders may be cut out by taking down the corresponding main rod and blocking the valve in its middle position. The locomotive can then be run with the three remaining cylinders.

American Articulated Compound.*

An Articulated compound locomotive is one having two sets of cylinders which drive separate and independent groups of wheels and one of which ordinarily uses exhaust steam from the other. Both sets of cylinders are supplied with steam from a single boiler.

The rear group of wheels is carried in frames rigidly attached to the boiler in the usual manner; while the frames which carry the front group of wheels are not secured to the boiler; but support it by means of sliding bearings.

*Courtesy of American Locomotive Co. of New York.

There is a hinged connection between the frames of the front engine and those of the rear engine which permits the front group of wheels to swivel radially when the locomotive passes through a curve. The front group is thus, in effect, a truck.

Because of this feature from which its name "articulated" is derived, this type of locomotive may have twice as many driving wheels as a locomotive of rigid frame construction with no longer rigid wheel base.

Consequently, the articulated locomotive can be designed to pass through the same curves as any locomotive of the rigid frame type, and at the same time to provide twice the tractive power with no greater axle load or the same tractive power with one-half the axle load of the latter.

In addition to these advantages, due to its wheel arrangement, the articulated compound locomotive possesses all those derived from compounding the steam. Steam from the boiler is admitted to the first set of high-pressure cylinders which ordinarily drive the rear group of wheels; and, having done work in those cylinders, is then used over again in the second set or low-pressure cylinders, which are connected to the front group of wheels. From the low-pressure cylinders, the steam is exhausted to the atmosphere.

Connecting the high and low-pressure cylinders is a large pipe called the receiver, into which the steam from the high-pressure cylinders exhausts when the locomotive is working compound. From the receiver, the steam is admitted into the low-pressure cylinders by their valves in the usual manner.

The low-pressure cylinders have a larger piston area than the high-pressure cylinders, the relative size of the two being such that, at the ordinary working cut-off, the steam at the lower pressure per square inch, acting against the larger piston area, exerts the same force as the higher pressure steam acting on the smaller area. Consequently, the stroke of the high and low-pressure cylinders being the same, the two sets of cylinders ordinarily do practically equal amounts of work.

By using the steam successively in two cylinders in a compound locomotive, more of the steam is utilized in effective work than in a simple locomotive in which the steam is only used once; in other words, the same amount or volume of steam does more work. This results in a considerable saving of coal and water.

Recent exhaustive tests have shown that the use of the superheater in combination with compound cylinders gives increased economy in operation. Superheaters are consequently now being quite extensivey applied to articulated compound locomotives.

In every compound locomotive steam must be admitted direct from the boiler to the low-pressure cylinders in starting and until they are supplied with steam by the exhaust from the high-pressure cylinders. Provision is also usually made by which in cases of emergency, when additional hauling capacity is required, the locomotive may be changed from working compound into simple with an increase in power.

In the American articulated compound locomotive, a special mechanism called the intercepting valve performs these two duties. This valve is located between the receiver and the exhaust passages from the high-pressure cylinder. It is practically automatic in its operation and is described in the following pages.

Other locomotive builders use a by-pass arrangement for admitting live steam to the low-pressure cylinders in starting or working simple. By this arrangement communication is established between the two ends of the high-pressure cylinders by opening a valve operated from the cab. It lacks the advantage possessed by the intercepting valve of preventing increased back pressure on the high pressure pistons when the locomotive is working with live steam in both sets of cylinders.

In the American Locomotive Company's system of compounding, the intercepting valve is so designed that when the engine is working simple the exhaust from the high-pressure cylinder passes directly to the atmosphere and the valve cuts off communication between the receiver and the exhaust side of the high-pressure pistons. This relieves them of all back pressure except that of the steam exhausting to the atmosphere. In addition, the low-pressure pistons exert more power when working compound because the pressure of the live steam admitted to them, though reduced, is higher than the ordinary receiver pressure. This additional power added to that secured in the high-pressure cylinders from the reduction of the back pressure gives 20 per cent. total increase in power when working simple at slow speed.

Intercepting Valve.

Among the distinctive features of the American articulated compound locomotive, practically the only ones in which engineers are interested, as relating to the operation of the locomotive, are the intercepting valve, the power reversing gear and the by-pass valves.

The intercepting valve, the purpose of which has been already stated, is the same in principle as that used on the well-known two-cylinder cross-compound locomotive built by the American Locomotive Company, commonly known as the Richmond Compound. It differs from the latter only in certain modifications of the design, which the use of four cylinders, instead of two, necessitates.

Engineers who have operated the two-cylinder cross-compound of this build will be perfectly familiar with the construction and operation of the intercepting valve as applied to the American articulated compound locomotive. In any case, as the operations of this valve are all automatic, except that by which the locomotive is changed from compound into simple working, practically no special knowledge is required to handle the articulated compound.

Changing into simple working is effected by merely opening an operating valve in the cab, which controls the emergency exhaust valve. The emergency operating valve is an ordinary angle valve located in a small steam pipe running to the emergency exhaust valve. It is only used if the train is about to stall; that is, in an emergency; or in case of an accident in which one or more of the cylinders can be disconnected and the locomotive run in on the remaining cylinders.

The intercepting valve is located in the saddle of the left high-pressure cylinder, to the left of the vertical, and above the horizontal, center lines of the cylinders.

Its various parts are shown in Fig. 313. These parts assembled and in their relation to the steam passages in the cylinders, are shown in Fig. 314.

Parts 2, 3 and 5 of Fig. 313 constitute the intercepting valve proper.

Part 1 is the reducing valve, or sleeve, which fits on the stem of the intercepting valve along which it is free to slide longitudinally, as Fig. 314 shows.

The movements of all these parts are automatic.

Part 6 of Fig. 313, is the emergency or high-pressure exhaust valve, which, as previously stated, is the only part of the intercepting mechanism which is not entirely automatic in its operation; but is under the control of the engineer through an oper-

FIG. 313.

ating valve in the cab. It is located, as will be seen, in Fig. 314, at one of the outer ends of the intercepting valve chamber.

Fig. 314 is a reproduction of a working drawing of the intercepting valve assembled.

In this illustration, the valve is shown in two positions: the upper representing the position the parts automatically assume

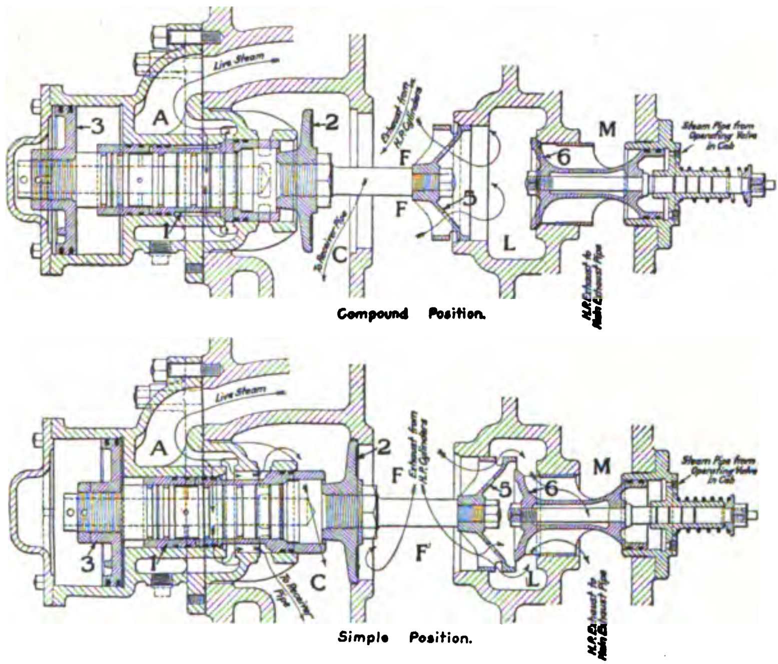


FIG. 314.

when the locomotive is working compound, and the lower their position when the locomotive is working simple.

The two positions may thus be studied in conjunction so as to most clearly understand the construction and operation of this system of compounding.

By referring to Fig. 314 the following is apparent:—Chamber "A" which surrounds the reducing valve (1) is in direct communication with the live steam passages of the high pressure cylinders.

Chamber "C" opens directly into the receiver pipe. Communication between chambers "A" and "C" is established through the reducing valve (1).

Chamber "F" connects directly with the exhaust passages of the high pressure cylinders.

Between chambers "F" and "C" is the intercepting valve (2). Chamber "F" is also connected with chamber "L" through the balancing piston (5) in which are a number of holes.

The emergency valve (6) establishes communication between chambers "L" and "M," the latter of which opens directly into the high pressure exhaust pipe, the small pipe which runs along the left side of the locomotive and connects to the main exhaust pipe.

Thus it will be seen that the reducing valve (1) controls the admission of the live steam from the boiler to the receiver pipe; the intercepting valve (2) opens or closes the receiver pipe to the exhaust from the high pressure cylinders; and the emergency valve (6) permits or prevents this exhaust from escaping through the main exhaust to the atmosphere.

When the locomotive is started in the ordinary way, the pressure in chamber "A" opens the reducing valve (1), and this movement in turn closes the intercepting valve (2).

Live steam is admitted to the receiver, and at the same time is prevented by the intercepting valve from backing up against the exhaust side of the high pressure pistons. These consequently start free from back pressure.

The reducing valve is so designed as to reduce the live steam entering the receiver to such a pressure that the low pressure cylinders will do the same amount of work as the high pressure cylinders.

Usually, this means a reduction to about 40 per cent. of boiler pressure. In the case of 200 pounds working pressure the live steam in the receiver would be reduced to 80 pounds.

If the pressure in the receiver rises above the amount to which the reducing valve (1) is designed to reduce it, the valve automatically closes and cuts off the admission of live steam to the receiver. It then remains closed until the movement of the low pressure pistons lowers the pressure in the receiver to the required amount when it again opens. The reducing valve can close without opening the intercepting valve (2), but these two valves cannot both be opened at the same time.

After one or two revolutions of the driving wheels, the steam exhausting from the high pressure cylinders into chamber "F" accumulates and its pressure rises sufficiently to open the intercepting valve (2) which in turn closes the reducing valve (1). The locomotive thus works compound; that is, the low pressure cylinders are supplied with steam by the exhaust from the high pressure cylinders. The upper view of Fig. 314 illustrates the intercepting mechanism in compound position.

The intercepting valve is usually so designed that when the pressure in chamber "F" has reached 30 per cent. of boiler pressure, it will open the valve against the steam in chamber "C" of 40 per cent. of boiler pressure, because the latter acts against a smaller area.

From the foregoing, it will be seen that, when operated in the usual way, the American articulated compound locomotive starts with live steam in all four cylinders and after a few revolutions of the driving wheels changes of itself into compound working.

If is is desired to prevent the locomotive from changing into compound after it is started, or to work it simple, and thus secure the maximum power of the locomotive at a critical point, it is only necessary for the engineer to open the emergency operating valve in the cab by turning it so that the handle points to the *rear*.

This opens the emergency valve (6), Fig. 314, and allows the exhaust from the high pressure cylinders to escape to the stack. The reducing valve (1) is immediately opened, there being no resisting pressure in chamber "F," and closes the intercepting valve (2). Live steam is admitted to the receiver which at the same time is cut off from the exhaust side of the high pressure pistons and increased back pressure thus prevented.

The balancing piston (5) is employed in order that chamber "F" may be exhausted instantaneously with the opening of the emergency valve; with the result that the intercepting valve is closed and the reducing valve opened before or at the same moment that the receiver is actually exhausted. This prevents any drop of pressure in the low pressure steam chest during the change from compound to simple.

The lower view of Fig. 314 shows the positions of the parts of the intercepting mechanism when the engine is working simple.

A brief summary of the conditions existing when the intercepting valve is in the simple position will explain the 20 per cent. increase in the normal maximum tractive power which, as already stated, is attained by working the locomotive simple. The high pressure pistons are relieved of the back pressure in the receiver amounting to about 30 per cent. of the boiler pressure which acts against them when the locomotive is working compound. On the other hand, the low pressure cylinders receive live steam of 40 per cent. of the boiler pressure instead of the exhaust steam from the high pressure cylinders at a pressure of only 30 per cent. of the boiler pressure, as ordinarily. The increase would be greater were it not for the wire-drawing of the steam through the restricted area of the ports of the reducing valve which are intentionally reduced for operation under this condition.

It is important to know and bear in mind that the reducing valve is so designed that at speeds of more than three or four miles an hour no increase in power is obtained by changing the locomotive into simple. This is done in order that the emergency feature may not be misused, with increased wear on the machinery and sacrifice of economy in fuel consumption.

The engineer must also remember that the locomotive having been changed into simple working by opening the emergency operating valve will continue to work simple until this valve is closed. The operating valve is closed when the handle points *forward.*

In changing from compound to simple when running, the sudden unbalancing of the intercepting valve tends to close it rapidly with the result that it would slam were not some provision made to prevent this. For this purpose, the piston (3) working in an air dash-pot at the outer end of the intercepting valve stem is employed.

Years of successful operation of this system of compounding, which has been previously applied to a large number of cross-compound locomotives, has proven that it is simple and reliable. Only a few simple rules are necessary for the guidance of the engineer in the proper care and operation of the locomotive.

Ordinarily, in starting, it is only necessary to open the throttle with the reverse lever in the position required for the weight of the train which is usually in the extreme notch, and with the cylinder cocks open.

If the train is about to stall, the locomotive should be changed into simple working by opening the emergency operating valve.

The intercepting valve should be given a liberal feed of oil for a minute before starting and occasionally during long runs when the throttle is not shut off for a considerable length of time. Outside of this, one drop of oil every 4 or 5 minutes is ordinarily ample when running.

This simple care will prevent any tendency for the parts of the intercepting mechanism to stick; but if neglected, the reducing valve might stick in a closed position which would prevent the admission of steam to the low pressure cylinders when the throttle is opened.

In the event that through neglect the reducing valve does stick, the difficulty can ordinarily be remedied by giving it a little extra feed of oil. If this fails, the cover of the dash-pot may be removed and the reducing valve moved in and out a few times with a bent piece of ¼-inch wire, after which it will probably clear itself when the throttle is opened.

By-Pass Valves.

These play an important part in the successful operation of the American articulated compound locomotive and although automatic in their operation, they should be understood by the engineer.

They are applied to the low-pressure cylinders, and establish communication between the two ends of the cylinder when the locomotive is running with the throttle closed. This permits free circulation of air from one end of the cylinder to the other; and thus prevents any injurious effects from alternating vacuum and compression, which would otherwise occur from the pumping action of the large pistons when the locomotive is drifting.

Fig. 315 illustrates the arrangement of the by-pass valves when assembled in their chamber, and their relation to the steam ports in the cylinders. Each cylinder is provided with a pair of these valves, which are located in chambers cast in the outside of the cylinders.

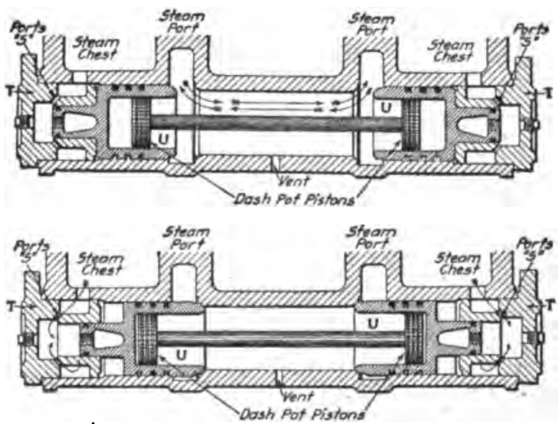


FIG. 315.

When the throttle is open, the pressure in the steam chest acting through the small ports "S" keeps the valves closed as shown in the lower view of the Figure 315. When the throttle is closed, they are automatically opened by the atmospheric pressure admitted through the air vent and connect the admission ports at either end of the cylinder as shown in the upper view.

With the by-pass valves, the locomotive drifts more freely when running with a long cut-off. It is strongly recommended, therefore, that in drifting the reverse lever be kept at ¾-stroke or more.

The by-pass valves require only ordinary attention, but as their duties are important they should not be entirely neglected. The principal care required is that they be kept clean.

When the locomotive is first put into service, the by-pass valves should be taken out and cleaned quite frequently, to keep them free from core sand which will undoubtedly work in.

The engineer can tell at once if the by-pass valves do not close when the throttle is open. This would not only cause a severe blow, but steam would also escape from the small pipe projecting from under the cylinder jacket midway between the ends of the cylinder. This pipe connects to the air vent in the center of the chamber containing the valves.

If the low-pressure engines thump and the locomotive does not drift freely, the trouble is probably due to the fact that the by-pass valves are stuck in the closed position by being gummed. In such event, they should be taken out and cleaned at the first opportunity.

If the locomotive is allowed to drift with the reverse lever hooked up, smoke-box gases may be sucked in by the cylinders and gum the by-pass valves. This possibility will be minimized if the reverse lever is kept in the position as recommended above, when drifting.

Periodical cleaning of these valves is recommended.

Vacuum and Relief Valves.

Vacuum valves are located in the high-pressure steam chests, or in some other convenient place which is in communication with the steam chests. The function of these valves is to admit free air into the steam chests when the locomotive is drifting, so as to avoid a vacuum and give a moderate flow of air through the cylinders.

The low-pressure cylinders are equipped with combined vacuum and relief valves, which, in addition to having functions similar to the vacuum valves of the high-pressure cylinders, also regulate the steam pressure in the low-pressure steam chests. These relief valves are set at 45 per cent. of the boiler pressure, and should be tested occasionally to see that they are properly set.

If they rise from their seats frequently when the locomotive is working compound, it may be due to a blow in either the valves or pistons of the high-pressure cylinders; as such a blow would increase the pressure in the receiver, causing the relief valve to open. In such a case, the high-pressure valves and pistons should be tested.

To test for blows, simply open the emergency operating valve in the cab; or, in other words, change the locomotive into simple working. Spot the locomotive and proceed the same as with a simple engine.

In Case of Accidents.

In case of any accident in which one or more of the cylinders may be disconnected and the locomotive run in with the remaining cylinders active, simply throw the emergency operating valve in the cab into the simple position and proceed as with a simple engine. Disconnect and block the disabled cylinder, or cylinders. This is the only rule to follow and the only one to be remembered, and covers all cases of accidents which do not entirely disable the locomotive.

Simplex Compound System.*

The Mallet system of compounding offers ready means of increasing temporarily the tractive power of the locomotive above

*Courtesy of the Franklin Railway Supply Co. of New York.

that of a simple locomotive. By this means, heavier trains may be started and handled over the ruling grade, than could be handled by a simple engine of the same weight and normal tractive effort.

This additional tractive power (which is only available for use at lower speeds) is accomplished by "simpling" the engine, that is, by exhausting the high-pressure cylinders to the atmosphere, which greatly affects the mean pressure in the cylinders through a reduction of back pressure, and by admitting live steam at reduced pressure directly to the low-pressure cylinders.

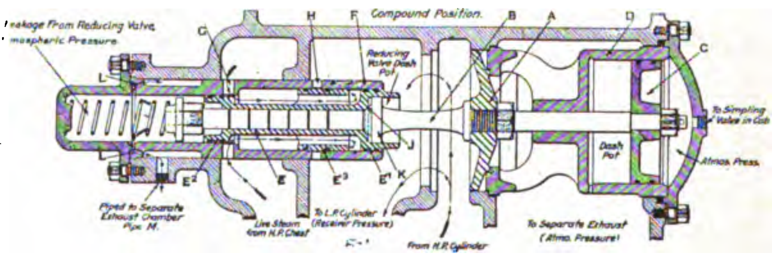


FIG. 316.

The result is a locomotive equivalent to two "simple" engines with abnormally large cylinders.

The Simplex system was designed for the purpose of overcoming certain mechanical defects in some of the older systems, as well as to reduce the complications found in some systems of compounding.

The Simplex intercepting valve is clearly shown in Figs. 316 and 317; Fig. 316 showing the valve in compound position and Fig. 317 showing it in simple position. The operation of the Simplex system is as follows: On the continuation of the stem, B, toward the right side of the engine, is a sliding reducing valve, E, which has a movement of one and a half inches on the stem, as well as a traverse of three inches with the main valve. This valve has three functions.

First: To admit steam at a reduced pressure to the receiver chamber when working simple.

Second: To force the main valve into compound position when pressure is released from the piston, C.

Third: To cut off the supply of live steam to the receiver when working compound.

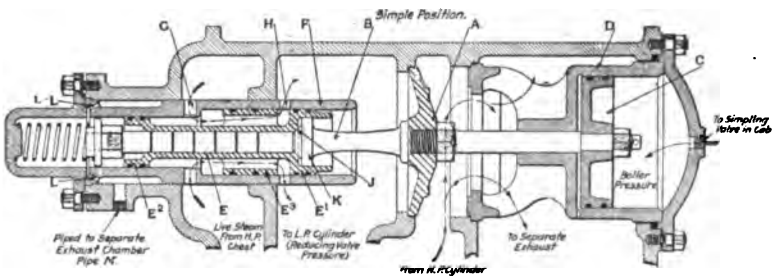


FIG. 317.

The reducing valve works in a bushing, F, having a double concentric bore. The ports, G, admit live steam between the end pistons of the reducing valve at all times when the throttle is open. The ports, H, admit steam at a reduced pressure to the receiver, when working simple.

The first function of the reducing valve is accomplished as follows: Fig. 316. Piston, E^1, being larger than piston, E^2, the live steam pressure admitted between these causes the valve to move in the direction of the large end, and to assume the position shown. As soon as sufficient pressure is built up in the receiver, this pressure reacts on the outer end of the piston, E^1, causing the valve to move in the opposite direction, cutting off the flow through the port, H. Thus, a constant pressure, depending on the relative diameter of the pistons, E^1 and E^2, is maintained in the receiver when working simple.

A second function of the reducing valve, that of moving the main valve into compound position, is effected through this same tendency of the reducing valve to move in the direction of its large end, acting on the ground joint, J, on the reducing valve

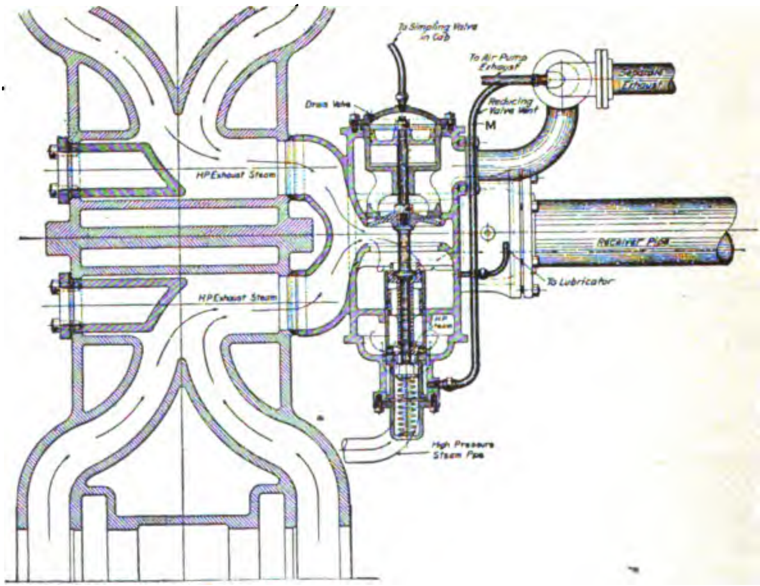


FIG. 318.

dash pot piston K. This movement is aided in its early stages by the reducing valve pressure in the receiver acting on the main valve.

A third duty of the reducing valve is performed by the intermediate piston, E^3, which closes the port, H, as soon as the stem, B, has moved sufficiently toward compound position, Fig. 316.

The small end of the reducing valve is vented to the atmosphere through the ports, L, and the pipe, M, Fig. 318, which communicate with the separate exhaust pipe. When working simple, should the high pressure engine slip, a considerable pressure is built up in the separate exhaust pipe. This, by reacting on the small end of the reducing valve, causes a high pressure to be temporarily admitted to the low pressure cylinders, thus maintaining the total draw bar pull as nearly constant as possible.

The Intercepting Valve may be applied to the high-pressure cylinders of a Mallet locomotive in any of three different positions, to suit the various types of high pressure cylinder saddle in common use. The preferred arrangement is to place the valve in a separate casting which is disposed transversely of the locomotive, in front of the cylinder saddle. This arrangement permits of the simplification of the high-pressure cylinders and saddles, as well as the placing of the valve in a position where it is more readily removable for cleaning and inspection. The Simplex locomotives built by the American Locomotive Company usually have the Intercepting valve located in the left-hand cylinder saddle, which is suitably cored for its reception. Locomotives built by the Baldwin Locomotive Works have the Intercepting Valve located in a separate cast steel cylinder saddle, which is in accordance with the usual Baldwin construction.

It will be seen that the Simplex system possesses flexibility of application to different designs of locomotives. In this design, all ground or packed joints holding live steam from leakage to the atmosphere have been eliminated.

OIL BURNING LOCOMOTIVES.*

Although petroleum was first discovered in commercial quantities in the United States in the year 1859, it is only within the past twenty years that it has been extensively adopted as a locomotive fuel. This is due to the fact that, previous to the development of the oil fields in Texas and California, the principal source of supply was Pennsylvania, where coal was found in abundance, and where there were no special ecnomic reasons for using oil fuel. In the southwest, however, coal is expensive and oil is now relatively cheap, and the railways have taken full advantage of the opportunity to reduce fuel bills, by changing their locomotives from coal to oil burners. An idea of the increasing extent to which oil is being used as fuel in railroad work may be obtained from the fact that in 1906, the consumption of fuel oil by locomotives amounted to 15,577,677 barrels, while in 1916 the annual consumption was estimated at 40,000,000 barrels.

In the year 1888, Dr. Charles B. Dudley presented to the Franklin Institute of Philadelphia, a comprehensive paper dealing with the subject of oil fuel for locomotives. Dr. Dudley founded his conclusions largely upon a series of experiments which had been conducted by the Pennsylvania Railroad Company. He determined that, based on the relative heat values of the fuels, one pound of oil was equivalent to one and three-quarters pounds of coal; while taking into account the various incidental

*Courtesy of the Baldwin Locomotive Works, Philadelphia, Pa.

economies due to the use of oil, one pound of the latter was practically equivalent to two pounds of coal. * * *

It is evident that the relative cost of coal and oil must vary greatly in different parts of the country. In any case, the costs to be considered are not those at the mine or well, but those at the point where the fuel is to be used, so that charges for handling, etc., are included. In some special cases, as in operating through long tunnels or forest regions, it is desirable, in order to eliminate smoke and sparks, to use oil-burning locomotives, even though the cost for fuel may be greater than when using coal.

In the paper previously referred to, eleven incidental advantages of oil over coal for locomotive work are given. These cover the ground very fully and are here repeated.

(1) Less waste of fuel. In a coal-burning locomotive unburned fuel escapes from the stack in the form of smoke, unburned gas and cinders, and it also falls through the grates. In a well-designed and properly handled oil-burning locomotive, however, there should be no loss due to these causes. (2) Economy in handling fuel. (3) Economy in handling ashes. (4) Diminished repairs to locomotives. (Recent experience hardly justifies this claim, as firebox repairs on oil-burning locomotives are greater than on coal burners.) (5) Economy in cleaning engines, due to the absence of cinders, ashes, etc. (6) Less waste of steam at safety valves, as the fire can be more easily controlled, to suit the demand for steam, than in a coal-burning locomotive. (7) Economy in cleaning ballast. This is particularly true of stone ballasted roads, where the ballast must be cleaned of cinders to prevent interference with the drainage. (8) Economy of space in carrying and stowing fuel, due to the fact that a pound of oil does not occupy as much space as a

pound of coal, while it is capable of generating more steam. (9) No fires from sparks. (10) Absence of smoke and cinders, a special advantage in passenger train service. (11) Possibility of utilizing more of the heat, since tubes are not choked up with cinders. On this account there is no reason why smaller tubes should not be used on oil burners than on coal burners, and more heating surface thus exposed. In practice, however, it is customary to use tubes of the same size for both coal and oil-burning locomotives, as the same engine can thus be easily equipped for burning either kind of fuel and the problem of repairs is simplified. * * *

A special design of boiler and firebox is not necessary to insure the successful use of oil fuel, and the required equipment is simple in construction and is easily installed. Such equipment consists, briefly, of an injector or atomizer through which the oil is fed into the furnace, a suitable fire-pan replacing the regular ash pan of a coal burner, and an arrangement of fire-brick for protecting the lower parts of the furnace sheets from the direct action of the flame. The supply of oil may be carried either on the tender or on the locomotive. The flow of oil is controlled by a plug cock in the feed pipe. A heater may be placed in this pipe as it is essential to have the oil warm enough to insure a steady flow to the burner. The general arrangement is shown in Fig. 319.

In order to secure complete combustion, and fill the firebox with flame, it is necessary to spray the oil into the furnace; and this is ordinarily accomplished by means of a steam jet. The burner used by the Baldwin Locomotive Works is rectangular in cross section, with two separated ports or chambers (one above the other) running its entire length. Oil is admitted into the upper port, and steam into the lower. A free outlet is allowed

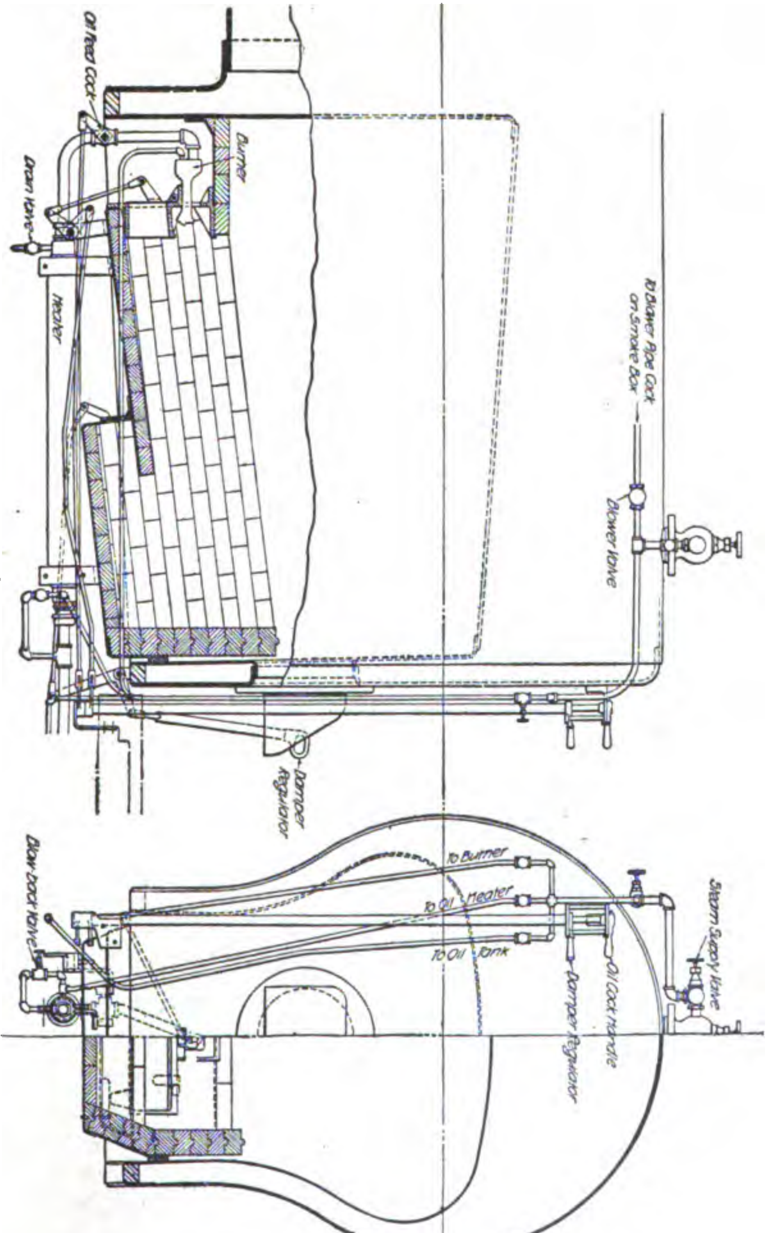

Oil Feed Cock
Burner
Drain Valve
Heater
To Blower Pipe Cock on Smoke Box
Blower Valve
Damper Regulator
Blow-back Valve
To Burner
To Oil Heater
To Oil Tank
Oil Cock Handle
Damper Regulator
Steam Supply Valve

FIG. 319.

for the oil at the nose of the burner; the steam outlet, however, is contracted at this point by an adjustable plate, thus giving a thin, wide aperture. This arrangement tends to wire-draw the steam and increase its velocity at the point of contact with the oil, giving a better atomizing. A permanent adjustment of the plate can be made for each burner, after the requirements of service are ascertained. The oil is carried into the firebox in the form of vapor, where, provided the air supply is properly ad-

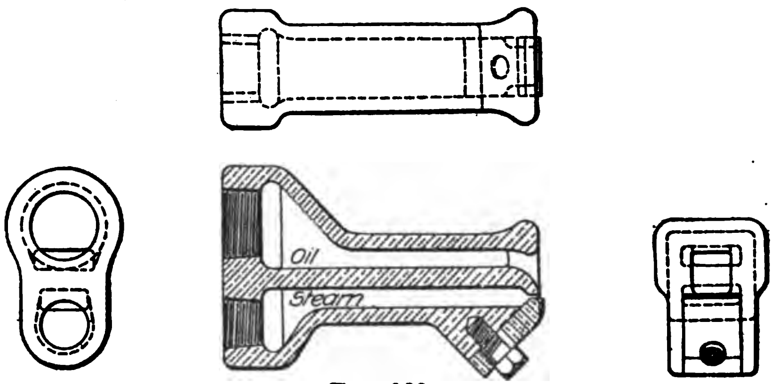


FIG. 320.

justed, it is completely consumed with the emission of little, if any, smoke.

The burner is made of brass, as an iron casting is found to be sufficiently porous to enable the steam to penetrate through to the oil passage, thus causing irregularities in the action of the flame. The adjustable plate is of copper, and it is held in place by a set-screw. The end of the burner is so shaped that the stream of oil is discharged in a downward direction. It therefore meets the steam jet before the velocity of the latter has been materially decreased. The arrangement of the device is shown in Fig. 320.

The fire-pan is supported in the bottom of the furnace, and is lined throughout with fire-brick. Where necessary, fire-clay and asbestos are used to insure an air-tight joint between the pan and the furnace walls. The front end of the pan proper is some distance back of the front water leg, and the intervening space is bridged over by a steel plate covered with fire-brick. The draft is controlled by means of a damper, which is placed in the front end of the fire-pan. The damper frame is of cast iron, and is substantially built; it has an opening in it near the top, through which the oil is injected. A second damper, admitting

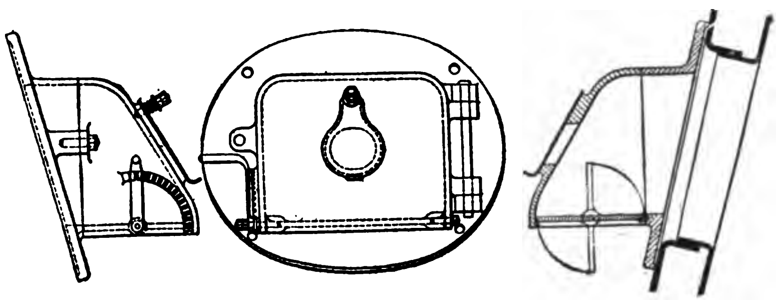


FIG. 321.

air in a horizontal direction, is placed in the pan at mid-length. Admission of air above the fire is effected through the fire-door, which is made in the form of a hood, as shown in Fig. 321. The bottom of the hood is mounted on trunnions and it can be rotated and held in any desired position by means of a handle which catches on a quadrant. With this arrangement the draft can be easily regulated, and there is little danger of the fire flashing back through the door; while the glare cannot blind the enginemen when running at night. The fire can be observed directly through a peep-hole, which is placed in the face of the door and is normally covered by means of a suitable lid. This peep-hole is also used when cleaning the tubes of soot. This is done by introducing sand into the furnace through a bent funnel, and the scouring action of the sand, when being drawn through the tubes, cuts out the soot, which is carried up the stack by the draft.

It was formerly the practice to place the burner in the rear end of the furnace and burn the oil under a brick arch. In service, however, when the engine was being heavily worked, the draft frequently lifted the flame over the arch, thus causing incomplete combustion and an excessive amount of smoke. The horizontal draft arrangement with burner placed in the front end of the furnace, as described above, has been found in practice to give very much better results.

Crude petroleum, such as is ordinarily used in locomotive work, is a rather thick, sluggish liquid, and it must be kept fairly warm to insure a steady flow to the burner. To this end, provision is made for turning live steam into the oil tank on the tender, and the oil is passed through a heater before it reaches the burner. This heater, as usually arranged by the Baldwin Locomotive Works, consists of a long steam jacketed pipe, through which the oil flows. The outside pipe is lagged to prevent radiation and the annular space between the pipes is kept filled with steam, drawn from the turret in the cab. Provision is made for cleaning out the oil pipe in the heater by blowing live steam through it. A drain is also provided for drawing off water from the jacket. The construction of the heater is shown in Fig. 322.

The oil feed cock is placed between the heater and the burner. This is a plug cock, as shown in Fig. 323, and the plug has an opening which is diamond in shape, thus giving a fine feed. The cock is controlled by a handle, which is placed in the cab within easy reach of the fireman. This handle works on a toothed sec-

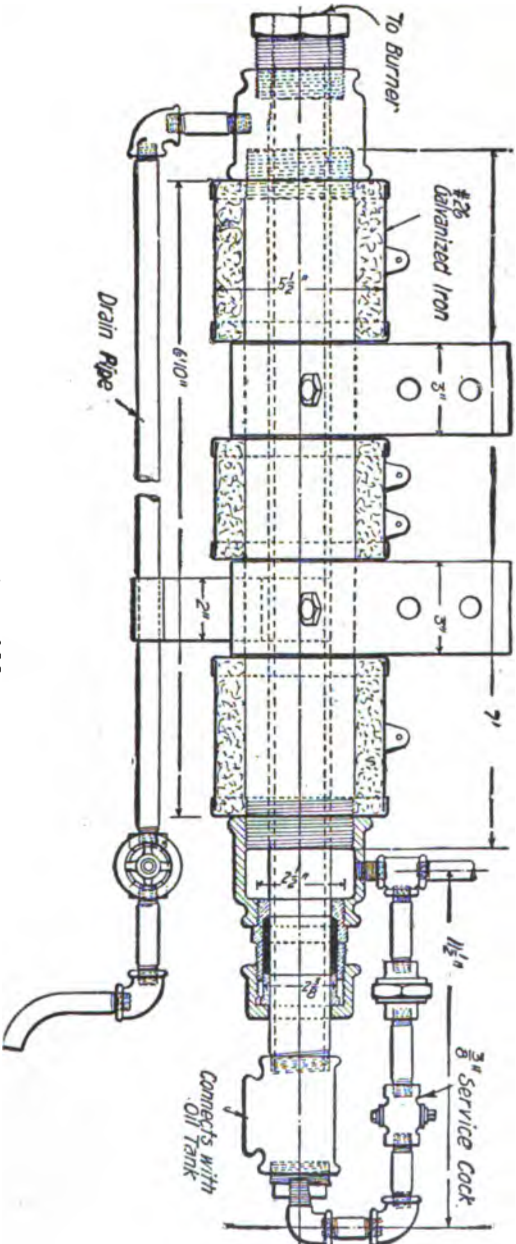

To Burner
#26 Galvanized Iron
Drain Pipe
6'10"
5½"
3"
2"
3"
7'
2½"
1½"
⅜" Service Cock
Connects with Oil Tank

FIG. 322.

tor so that a close adjustment of the oil feed is easily attained. The handles for operating the dampers are also placed in the cab, and are arranged, as shown in Fig. 319, so that the amount of air admitted to the fire can easily be varied to suit conditions. A plug cock is placed in the pipe line on the tender and is ar-

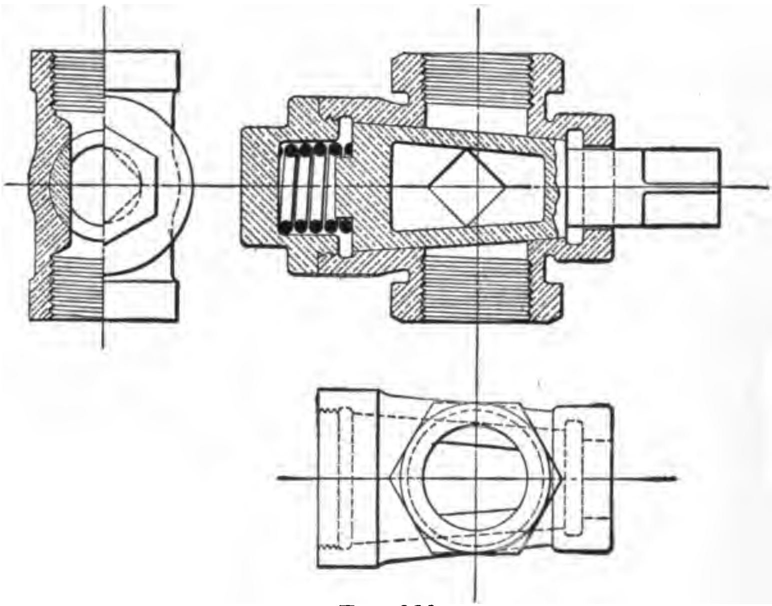


FIG. 323.

ranged to close automatically in case the engine and tender should become disconnected, thus avoiding waste of oil.

The front end arrangement of an oil-burning locomotive is simple in construction, as spark arresting devices are unnecessary, and no provision need be made for removing cinders from the smokebox. An adjustable petticoat pipe placed under the stack, is all that is required to secure proper regulation of the draft. The arrangement is shown in Fig. 324. A low nozzle is used, and the distance from the top of the petticoat pipe to the stack base should approximate four to eight inches, according to conditions. The drawing also shows the special cock in the blower-pipe, through which connection can be made to the steam pipe in the roundhouse when firing up.

The oil tank on the tender is arranged to fit into the fuel space, so that, by lifting the tank out, the tender can easily be used with

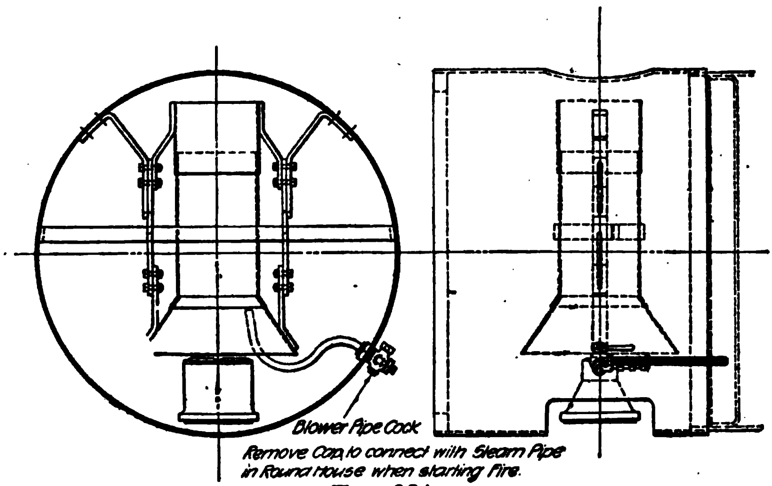


FIG. 324.

a coal-burning locomotive. The tank is provided with two tank valves, one for the oil connection leading to the burner, and the other to drain off water, which accumulates in consequence of using steam to heat the oil. The water settles in the bottom of the tank; hence the flange of the oil valve is extended upwards several inches, to prevent the water from flowing to the burner. Either a hose or flexible metallic connection is used between the locomotive and tender.

Fig. 325 shows the arrangement of the oil-burning equipment, as applied to a light-tank locomotive. The burner is placed immediately under the front end of the mud-ring, and draft is ad-

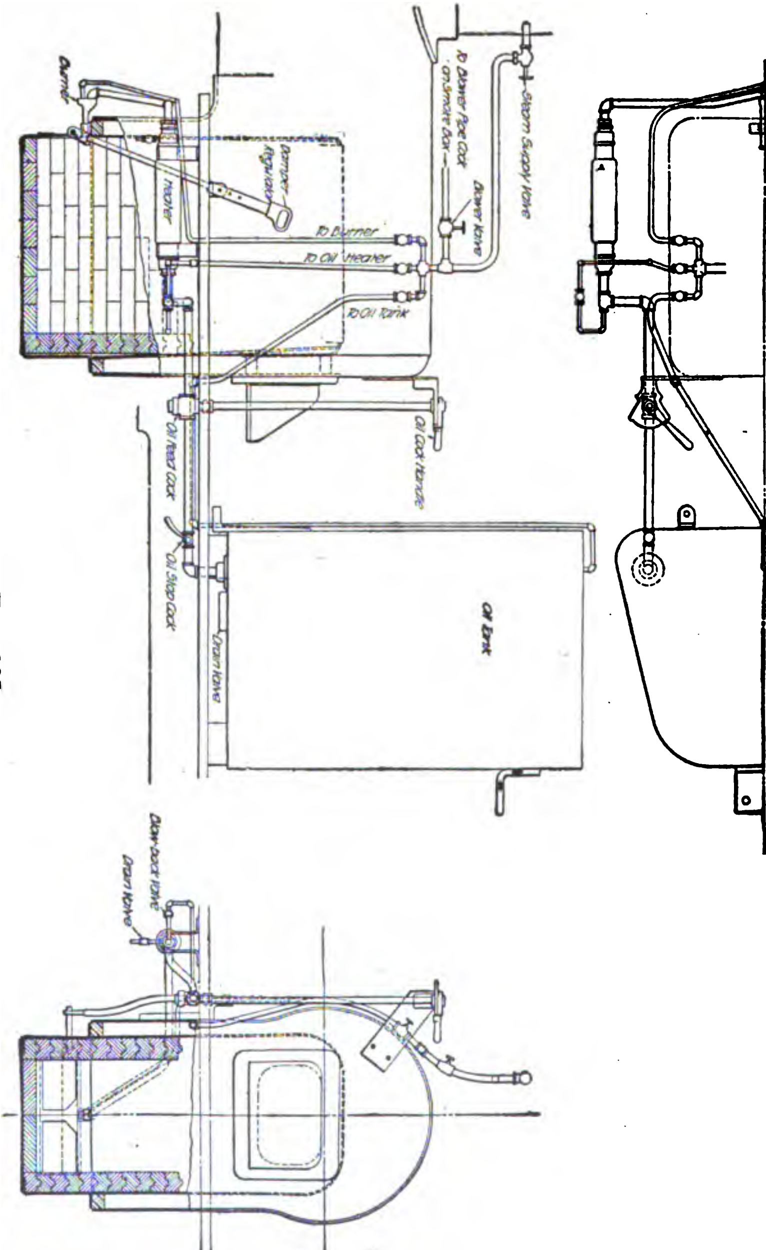

Burner
Damper Regulator
Heater
To Burner
To Oil Heater
To Oil Tank
Steam Supply Valve
Blower Valve
To Blower Pipe Cock on Smoke Box
Oil Cock Handle
Oil Feed Cock
Oil Stop Cock
Drain Valve
Oil Tank
Blow-back Valve
Drain Valve

FIG. 325.

mitted through a damper placed beneath it. The fire-pan is bricked at the bottom, sides and back. The drawing clearly shows the arrangement of the piping and controlling valves.

The two railways which have in service the greatest number of oil-burning locomotives in the United States are the Southern Pacific and the Atchison, Topeka and Santa Fe, and the arrangement of the apparatus, as used on these roads, differs somewhat from the Baldwin plan described above. On the Southern Pacific, the Von Boden-Ingalls burner, as represented in

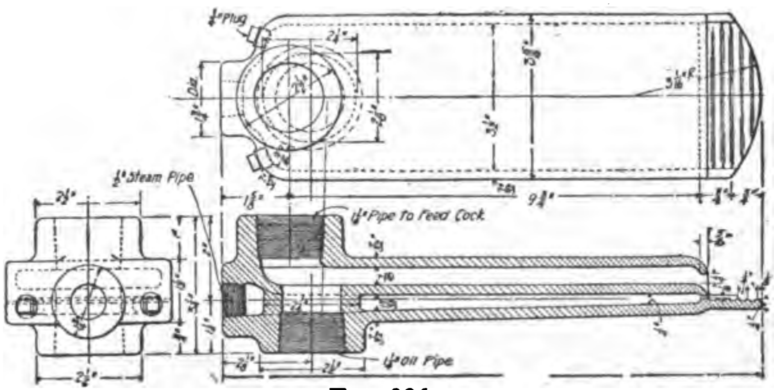


FIG. 326.

Fig. 326, is generally used. In principle, this burner is similar to the Baldwin burner. The adjustable plate, however, is omitted, and in front of the oil outlet is placed a corrugated lip, which retains any drippings from the burner, and is said to assist in atomizing the oil. The burner is placed in the front end of the fire-pan, as in the Baldwin system. Admission of air takes place through a number of horizontal tubes, placed under the burner; and these tubes can be covered by an external damper operated from the cab.

The Von Boden-Ingalls burner is so arranged that oil may be taken in either at the top or bottom of the oil chamber, as is the more convenient. The opening not in use is closed by a plug.

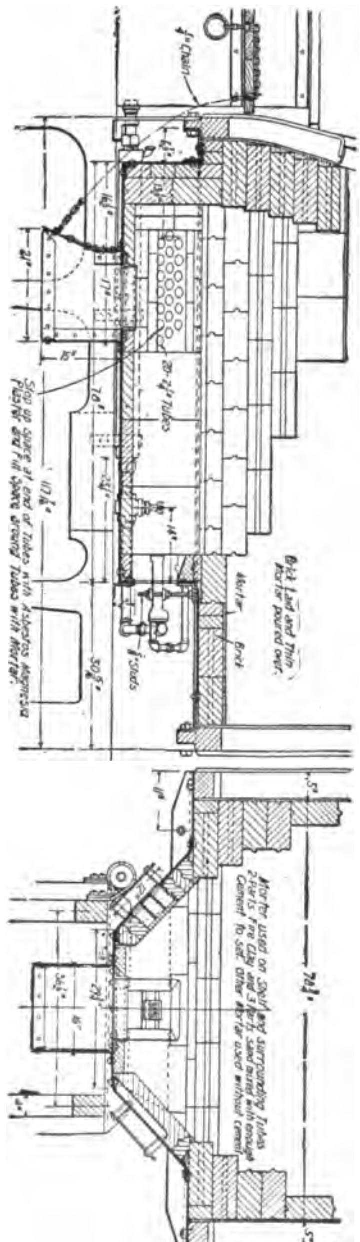
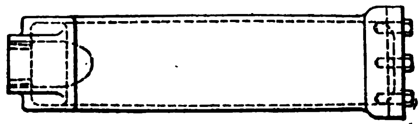

Brick Laid and Thin Mortar poured over.
Mortar
Brick
20-2¼" Tubes
Stop up space at end of tubes with Asbestos Magnesia Plaster and fill space around Tubes with Mortar.
Mortar used on Shelf and surrounding Tubes 2 Parts Fire Clay and 3 Parts Sand mixed with enough Cement to set. Other Mortar used without Cement

FIG. 327.

On the Atchison, Topeka and Santa Fe Railway the Booth burner is in general use. In this device, the steam aperture is made adjustable by means of a block which has in it a notch, the size of the desired orifice. This block stops off the steam passage and is secured to the burner by several tap bolts. The device is shown in Fig. 327.



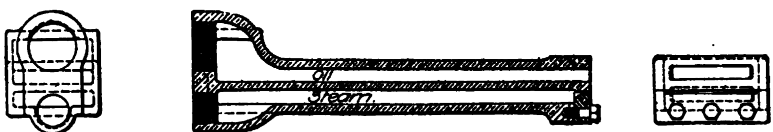


FIG. 328.

Fig. 328 shows the arrangement of the Booth burner as applied to a heavy Santa Fe type locomotive. The burner is placed in the front end of the fire-pan, and draft is admitted through a sheet-iron box, which is secured to the under side of the fire-pan and is provided with a damper. The position of the damper is regulated by a chain, which is run to a convenient point in the cab. This arrangement provides a vertical draft into the fire-pan and is successfully used; but in present practice the horizontal draft, previously described, is more generally employed.

The Handling of Oil-Burning Locomotives.

In order to realize the advantages and economies resulting from the use of fuel oil, it is essential that the apparatus be properly installed and maintained, and that the locomotive be handled intelligently. It is also important to remember that the fuel is highly inflammable, and that caution must be exercised when working about the engine and tender with lamps and torches. It is wise to steam out empty tanks which have contained oil, before entering them; as they are liable to contain gases which will not support human life.

The following extracts, dealing with the handling of oil-burning locomotives, are from a paper on Petroleum by Mr. Eugene McAuliffe, fuel agent of the Frisco Lines. This article applies more particularly to an arrangement in which the burner is placed at the rear of the furnace, but in general it is entirely applicable to the more recent practice.

The firing of an oil-burning locomotive differs very materially from the firing of a coal-burning engine, and more careful attention is necessary in oil burning than in coal burning to render the combustion economical. While the firing of an oil burner does not require great physical exertion, it absolutely demands that close attention be given it at all times in order to produce satisfactory results. To this end the fireman and the engineer must work together, and every time the engineer changes the throttle or reverse lever the fireman must change the fire. A fireman on an oil-burning locomotive must keep his eyes open, for he can make or waste more for the company than he could on a coal burner.

The nature of the case is such that no arbitrary instructions can be given as to how much steam should be used for the atomizer, how much the dampers should be opened, or the exact temperature to which the oil in the tender should be heated. These details must be left to the intelligence of the engine crew, and their experience should dictate that manipulation of these accessories is necessary under the various individual conditions.

In connection with oil burning, however, the following general rules are imperative, and must be observed in detail:

Filling Oil Tanks. Oil lamps or torches must not be brought within a distance of ten feet of any oil tank opening. A tender tank should not be supplied with a greater quantity of oil than will fill it to within two inches of the top, as the oil expands considerably when the heater is used, and may overflow. After the tank has been filled clamp the manhole cover down and keep it there until it is necessary to take another supply of oil. The safety oil valve on the tank should be kept in good order, and be operative at all times, and the valve connections should likewise be in place and operative.

Starting Fire. In starting a fire, the usual precautions should be taken to see that the boiler is properly filled with water. This should not be determined from the level of the water in the gauge glass alone, but the gauge cocks should be tested to see that water runs out. In firing up an engine in the roundhouse, steam connection is made to the three-way cock on the smoke arch, supplying both blower and atomizer at the same time. A locomotive already having twenty pounds or more steam pressure can operate its own blower and atomizer. Before starting the fire, care should first be taken to see that the bottom of the firebox in front of the burner is free from carbon, fire-brick or any other obstruction that would tend to interfere with the free passage of the oil from the burner to the front of the firebox. The front damper is next opened and the blower is put on strong enough to create the necessary draft. The atomizer valve is then opened long enough to blow out any water which may have been condensed in the steam pipe or burner, after which it is closed, and a bunch of old lighted waste is thrown in front of the burner. The atomizer is then opened sufficiently to carry oil to the burning waste, and the regulator is slowly opened until the oil is known to be ignited from the burning waste. This can be determined by observation through the firedoor, if open, or through the sand hole in the door, if the door is closed. After the oil is ignited the atomizer and oil valve are carefully regulated to be sure that all of the oil passing through the burner is being consumed.

Where steam is used from engine house steam pipe the fire should be watched until sufficient steam to operate the atomizer is generated in the boiler, when the roundhouse steam should be cut off. Care should be taken not to turn on too much oil, for an explosion might ensue, with possible danger of injury to the operator. Care should also be taken to see that the fire does not go out in firing up, for if it does and it is not noticed the oil will run down into the pit or elsewhere and may take fire or explode, and thus damage the engine or roundhouse and other property. Unless a reliable flow of oil is running to the burner, and the burner is in good shape, the throttle should not be opened to move the engine, or else the fire will probably be put out by the exhaust. When engines are standing under steam (on sidetracks, etc.), care should be taken to see that the fire does not go out while the oil valve is open, and thus allow the fuel oil to flow into the ash pan and onto the ground, as there is danger of the oil being ignited and setting fire to surrounding inflammable objects.

The fire should not be started in an engine until the oil in the tender is heated to a temperature that will insure a good flow to the burner. This can be done by means of the steam line in the roundhouse, if necessary, by connecting roundhouse steam pipe to blower pipe cock on the smoke arch and manipulating the proper valves to apply direct or indirect heat to the oil in the tank. In the event of the fire going or being put out, it should be relighted by using a piece of ignited waste. No attempt should be made to relight the fire from the heated bricks, and the practice of so doing should be prohibited, as it is almost sure to cause an explosion which will damage the brickwork and perhaps injure severely the person attempting it. In firing up where steam is not available, wood may be used until twenty pounds of steam is generated in the boiler. (Less than this pressure will not be sufficient to atomize the oil.) The wood must be placed in the firebox with great care, so as not to damage the brickwork, and in using wood for this purpose it should all be consumed before starting out. Further, the engine should be started carefully to prevent the wood sparks causing fires along the right-of-way or elsewhere. The condition of the fire should be observed by means of the sight hole in the furnace door. A fire having a bright clear color denotes proper combustion, while a fire burning with a dark, smoky flame indicates the reverse.

Use of Heater. When the tender tank is equipped with both direct and indirect heater, the direct heater should be used to such an extent as may be considered necessary, then closed, and the indirect heater should then be used to keep the oil warm. The direct heater must not be operated for a greater length of time than is absolutely necessary, as the amount of water condensed in the oil by its extravagant use is undesirable. The best results are obtained when the oil is heated to such a temperature that the hand can be held in contact with the outside of the tank without much discomfort; or, in other words, to about one hundred and twelve degrees. If too high a temperature is reached and maintained, some of the good qualities of the oil are lost by constant boiling, the burner does not work so well, and it is more difficult to regulate the flow to the burner. In case of failure of heater by hose bursting, the oil can be heated through the main oil hose by closing the firing valve and opening the blow back valve. After using the heater in this manner the blow-back valve must be tightly closed, otherwise the engine will not steam.

In getting an engine ready for service the hostler must in all cases see that the oil is properly heated, putting the heater in operation when necessary, and the fireman's first duty on taking charge of the engine should be to assure himself that the oil is heated sufficiently. He must see that the heater is properly adjusted and that the oil is in condition to flow freely to the burner. This is important in order to prevent delays. In using the heater care must be taken not to burst the hose by opening the valve to full flow.

Adjustment of Burner. The burner must be adjusted so that the blaze will not strike the top, bottom or side of the arch before striking the flash wall, as in either case it affects the engine's steaming and causes black smoke. In case the burner is cracked or the mouthpiece clogged up, a similar effect is produced, and it should be examined frequently for these defects. The engine should not be started until the fireman is at his post. It should be remembered that care of the firebox is as important as keeping up steam or making time, and large volumes of cold air should not be drawn through the flues at any time. The engine should be started carefully, so as not to slip the wheels, and the firing valve should be opened sufficiently to make sure that enough oil is being fed to produce a good fire, but not enough to cause a great volume of black smoke. The oil supply and steam for atomizing should be increased gradually as the speed is increased, and when the engine is hooked up the valves governing the burner should be regulated accordingly. Care and judgment must be exercised in the adjustment of the burner while drifting, to prevent as far as possible a waste of fuel; at the same time the fire must not be cut down so low as to allow the firebox to cool to such an extent as will result in leaky flues.

Black smoke. Black smoke should at all times be avoided. The production of a highly colored smoke is evidence of improper handling or of a defect in the brickwork or boiler. The soot formed by smoke is a nonconductor of heat and will make an oil-burning engine fail to steam quickly. The more black smoke that is made, the more it will be necessary to make. An accurate adjustment of oil and steam, together with the air supply, is necessary for thorough combustion, and especial care should be exercised to prevent black smoke when starting and stopping. At such times the fireman should work in absolute harmony with the engineer, and the starting of the fire must immediately precede the opening of the throttle, while the stopping of the fire should immediately follow the closing of the throttle, in order to prevent cold air being drawn through the flues in either case. The firing should not be forced, but the firebox temperature should be brought up as gradually as conditions will permit. Forced firing will overheat the plates, burn off rivet heads and cause leaks.

Handling Atomizers, Blowers and Dampers. The amount of steam necessary to atomize the oil depends upon conditions, and no definite amount can be determined. A slight change in the adjustment of the atomizer will often produce good results when an engine is not steaming well, and this must be left to the judgment of the person handling it. It is always necessary to use the blower (or roundhouse steam line) when firing up with oil. On the road the use of the blower should be avoided as much as possible, as it is detrimental to the flues and staybolts. Unnecessary use of blower is prohibited. Any draft through the firebox has a tendency to put the fire out; the stronger the draft the greater must the oil supply be. Otherwise there is danger of the fire being put out entirely before the throttle is closed. When the throttle is closed and the oil feed reduced, the atomizer should be cut down at once, so that it will just keep the oil from dropping onto the bottom of the pan. Dampers must be regulated to suit the conditions under which the engine is being worked. The admission of too much air is almost as bad as an insufficient supply. When the engine is standing or drifting the dampers must always be closed to prevent cold air being drawn in, causing leaks in the flues and staybolts.

Sanding Flues. Flues should always be cleaned out well after leaving terminals, or after an engine has been standing for some time. It is better to use the sand frequently and a small quantity at a time, than to use large amounts only a few times on a trip. Keep on sanding as long as black smoke follows the act of sanding, but do not continue unnecessarily. The sand funnel should be held in such a position that the sand will go over, instead of under, the arch and distribute the sand over all the flues. Some good things to observe in the management of oil-burning locomotives are: Use as little blower, as little sand and as little atomizer as possible.

When putting out the fire at terminals the main supply valve under the deck of the tank must be closed first, to allow the oil in the hose to be burned, and after all this oil has been consumed, the firing valve and dampers should be closed. The closing of the dampers is an important matter, to prevent the passage of cold air through the heated firebox and tubes after the fire has been extinguished.

Don't for Engineers and Firemen.

Don't think that the same adjustment of atomizers and dampers will apply to all engines. A slight change will often produce better results.

Don't think, because there is a cloud of smoke issuing from the stack, that you are firing in the proper manner. If an engine will not steam with a slight color at the stack, any further supply of oil is a detriment, as it clogs up the flues and causes the engine to steam worse.

Don't think that the front end arrangement composed of draft pipes, steam pipes, exhaust nozzles, etc., does not need the same attention that this arrangement on a coal burner needs. In the event of leaky steam pipes, draft pipes out of adjustment, or insufficient air supply, an undue amount of fuel, resulting in black smoke, will not assist in making steam.

Don't neglect the bottom of the ash pan and allow pieces of brick and carbon to obstruct the oil spray from the burner.

Don't put on the heater and forget all about it until the oil in the tank boils over.

Don't try to use all the sand in a few miles; the better practice is to use a small quantity at a time.

The fireman should not wait for the engineer to instruct him to shut off the oil supply. It is his duty to watch and to be governed by the engineer's movements of the reverse lever and throttle.

Don't use the blower indiscriminately; open it sufficiently to create the necessary draft. Too much opening is hard on the flues and sheets.

Don't move an engine when there is no fire in the box, if you wish to avoid leaky flues.

Don't neglect to drain the water out of the oil tank frequently.

Don't forget to examine the brickwork at the end of every trip.

Don't forget that the greatest possible care should be taken to maintain as even a temperature as possible in the firebox. It should not be either increased too rapidly by forcing the fire or reduced suddenly by permitting cold air to pass through the firebox and tubes. It is of the utmost importance that this even temperature be maintained to preserve the life of the firebox and flues, and to prevent engine failures by leakage of same.

Lastly, in view of the ease with which an extravagant waste of fuel can be effected in burning oil, it is especially urged that every effort be exerted to properly handle the engine and the burner and its accessories, in order to obtain an economical combustion, and guard against injury to boiler or firebox.

Where locomotives, because of inadequate terminal facilities or because of carelessness in ordering, are compelled to stand idle under steam, or when delayed on sidetracks en route, fuel oil will show an average saving of one-half the coal equivalent, as compared with that grade of fuel, while standing idle. This condition is due to the fact that while the oil jet can be instantaneously reduced or amplified, a heavy coal fire must be steadily maintained in order to insure prompt movement when called for.

The following additional instructions, covering the handling of oil-burning locomotives, should be given careful attention:

The cleaning pipe should be so adjusted that all the flues will be cleaned of soot. The bottom of the pipe should be on a level with the top of the exhaust, and the top of the pipe about four to eight inches from the top of the smoke-box.

The fire-pan joints should be kept tight, so that all air will enter the furnace through the dampers. The damper opening should be reduced when the fire is reduced.

The burner opening should be kept clean, and free from dirt and gummy accumulation, by blowing steam back through the burner before and after each day's work.

The blow-back valve should be kept tight, in order to prevent water from the heater from leaking into the oil supply.

In starting the fire, the steam valve should be opened first. There should always be a sufficient supply of steam to atomize all the oil that is fed to the burner.

The fire should not impinge upon the brickwork, as this will cause a deposit of carbon on the bricks. As a result, smoke will be formed, and pieces of burning carbon will be discharged out of the stack. The bricks should be kept clean and smooth at all times.

ELECTRIC LOCOMOTIVES.

Electric locomotives have, for many years, been employed in industrial haulage service, and also in light switching service, and have proven satisfactory for the work required.

Considerable attention is drawn, however, to the fact that the heavier types, weighing from twenty-five tons to over two hundred and fifty tons are, in many places, supplementing steam locomotives for regular passenger and light freight service. This is especially true within municipalities, and on road sections where tunneling is prevalent, and is due to the absence of noise and smoke, undesirable in such localities.

Even this practice of using electric locomotives for regular road service is not new, for, as early as 1895, the Baltimore & Ohio Railroad was supplied with a 96 ton General Electric Company locomotive, for use in the Baltimore tunnel. This locomotive had 62 inch drivers, and, with a 360 H. P. motor on each axle, developed a speed of nearly sixty miles per hour.

The accompanying illustration, Fig. 329, shows a modern type 100 ton electric locomotive, built by the General Electric Co., and used on the New York Central lines for operating in the Detroit River tunnel. This locomotive compares favorably with steam locomotives, as very few steam locomotives, other than the Mallets, have over 90 or 100 tons on the drivers.

However, even this is not the maximum of electric locomotive practice. In Fig. 330 is shown a 270 ton Baldwin-Westinghouse electric locomotive built by the Westinghouse Electric Mfg. Co.,

FIG. 329.

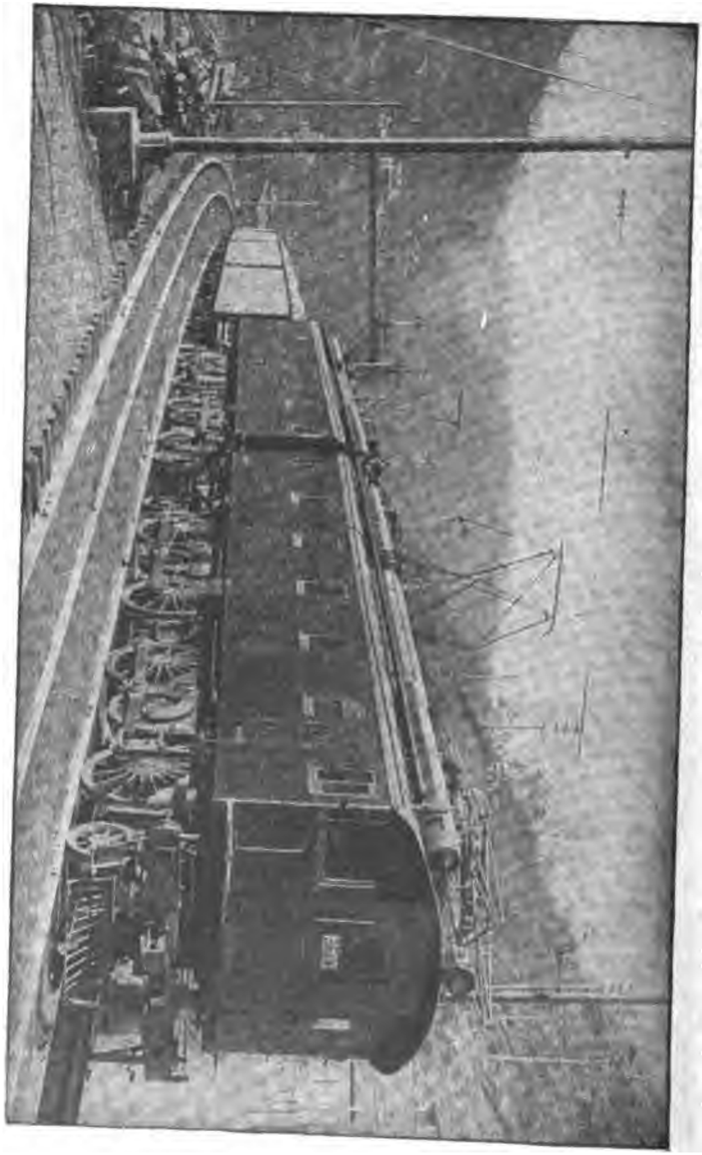


FIG. 330.

used by the Norfolk & Western in heavy freight traction. This locomotive is shown handling a 3,250 ton train, which task it accomplishes very well. The power developed by these electric locomotives is more remarkable than their speed, and the maintenance is very low. In fact, the maintenance of electric locomotives is about 55% of that of steam locomotives of equal weight, due principally to the fact that about one-half the cost of repairs of steam locomotives is centered in the boilers, while, of course, the electric locomotives eliminate this expense by reason of the fact that they employ no boilers.

Increased tractive effort is possible because of the fact that two or more locomotives may be coupled together, and operated by one crew. This flexibility of service is increased by the symmetrical arrangement of the wheels, which makes possible satisfactory operation in either direction. Again, these locomotives have the ability to maintain the rated speed and tonnage capacity irrespective of cold weather, which reduces the steaming capacity of steam locomotives.

Altogether, the electric locomotive possesses a number of very apparent advantages, which make it a serious contender of the steam locomotive, within certain limits of available source of power, and the expense of electrification of existing steam railway lines.

COMPRESSED AIR LOCOMOTIVES.

The first compressed air locomotive, which was of the single-expansion type, was constructed in 1873, but its range of use was extremely limited, as it could only travel a few hundred feet on one charge of air, with light loads; only ten or twelve of them were built during the following twenty years.

As a result of systematic development the details of construction were greatly improved from 1895 to 1908, and about three hundred of the single-expansion type were constructed during this period of thirteen years.

The introduction of the first two-stage compressed air locomotive, in 1908, attracted considerable attention at the time, and has since established a reputation for efficiency, durability and economy; as a consequence it is now being built in practical exclusion of the original single-expansion locomotive, so we shall disregard the latter in this chapter.

The Two-Stage Type.

This is a locomotive, such as shown in Figs. 331 and 333, in which compressed air is partially expanded in a high-pressure cylinder, until it becomes much colder than the surrounding atmosphere. This air is then passed through an interheater, in which it is heated to nearly atmospheric temperature by extended contact with the surrounding air. The expansion is then completed in a low-pressure cylinder.

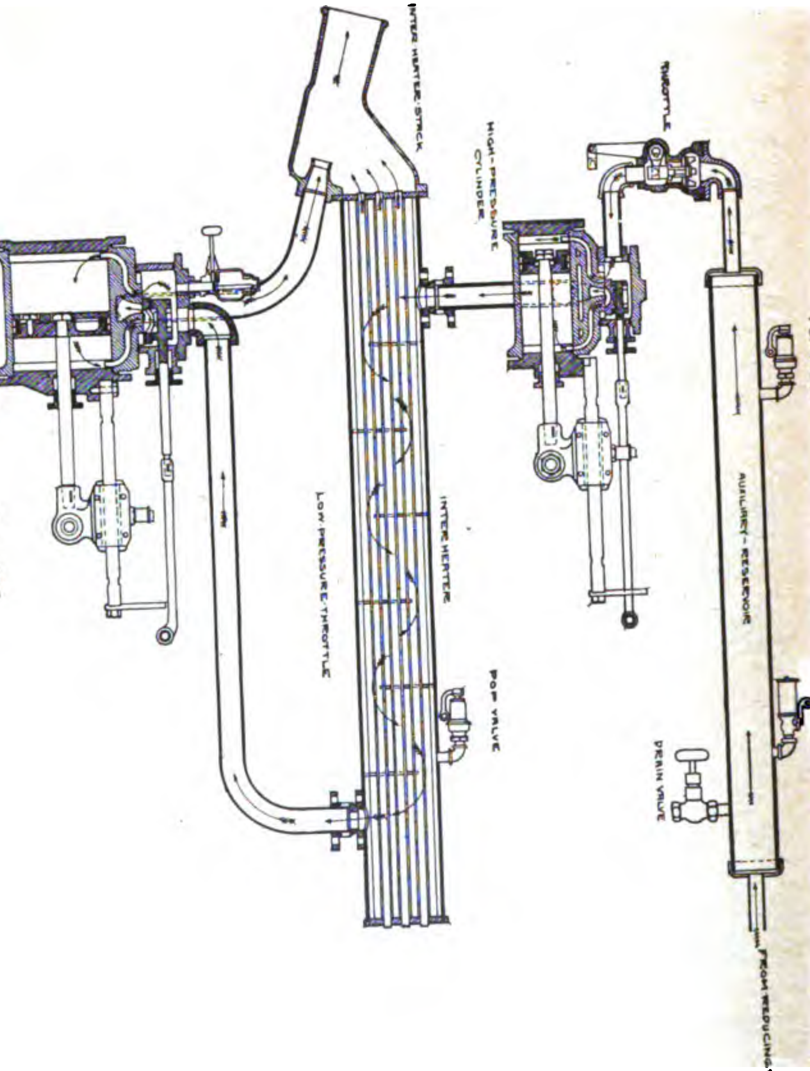
In a compressed air locomotive the temperature of the air entering the high-pressure cylinder is that of the atmosphere. It cools itself by doing work in the high-pressure cylinder. When it is thus cooled to a temperature much below that of the surrounding atmosphere, it can, in a properly designed interheater, absorb a great quantity of heat from the atmosphere, which is utilized to increase the quantity of work done in the low-pressure cylinder. The heat thus obtained, at no expense, not only increases the work done in the low-pressure cylinder, but makes the low-pressure work possible, because unless heat is obtained

FIG. 331.

from an outside source before the air enters the low-pressure cylinder, the temperature, due to further working of the air, would be so low as to interfere with lubrication and satisfactory operation.

The two-stage locomotive has therefore been designed to expand the air in two successive cylinders, with an atmospheric interheater located between them, through which the exhaust air from the high-pressure cylinder passes before it enters the low-pressure cylinder.

The construction of the interheater is clearly shown by Fig. 332. It consists of a cylindrical casing containing numerous aluminum or brass tubes of small diameter, around and between



THROTTLE
HIGH-PRESSURE CYLINDER
INTER HEATER STACK
POP VALVE
AUXILIARY-RESERVOIR
WHISTLE
DRAIN VALVE
FROM REDUCING VALVE
INTER HEATER
LOW-PRESSURE-THROTTLE
POP VALVE
LOW PRESSURE CYLINDER

FIG. 332.

which, guided by baffle plates, the air passes. The final exhaust air from the low-pressure cylinder is utilized in an ejector apparatus, similar to the smoke stack and exhaust nozzle of a steam locomotive, to draw a rapid current of the surrounding atmospheric air through the small tubes.

Thus the air leaving the high-pressure cylinder, cooled by the work which it has done to about 140 degrees Fahrenheit below that of the atmosphere, is brought into close contact with large quantities of air at atmospheric temperature for a sufficient length of time to raise the temperature of the partially expanded air from the high-pressure cylinder to within a few degrees of that of the atmosphere, before it is again utilized for doing additional work in the low-pressure cylinder.

A compressed air locomotive consists of: A main storage reservoir carrying a supply of compressed air at high pressure; a regulating valve adjusted to maintain any desired pressure in the small auxiliary reservoir, giving a uniform pressure for operating; and a throttle valve controlling the supply of air to the cylinders. The cylinders, slide valves, valve motion, connecting rods and driving wheels are similar to those of the steam locomotive, except for alterations in the piston packing rings, and the method of balancing the slide valves; while the links, frames and running gear are in all respects the same as for steam locomotives, except that in general all parts are somewhat heavier and the bearing surfaces somewhat more liberal.

The above outline applies to the mechanism only. There are radical differences in the temperature and condition of steam and compressed air in the cylinders, which call for different treatment.

Compressors supplying locomotives with air must deliver the air at a relatively high pressure—much higher than the pressure

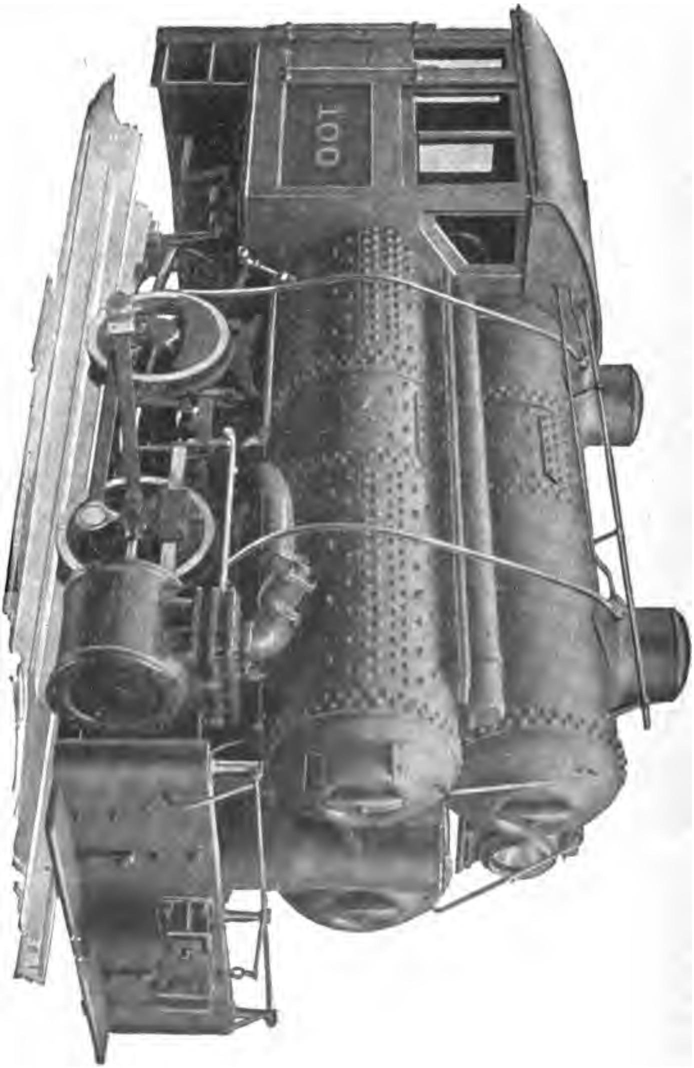


FIG. 333.

at which the air is admitted to the cylinders of the locomotive, in order that a sufficient quantity may be available to drive the locomotive a distance on one charge, which, when everything is considered, will give the best and most economical results.

Ordinarily a compressor capable of compressing air to a pressure of about 1,000 pounds per square inch is found most satisfactory. The locomotive is usually charged to a pressure of about 800 pounds per square inch. Charging pressures below 200 pounds per square inch are generally impracticable and 200 pounds is only satisfactory for very light loads and for distances of a few hundred feet. The entire operation of charging, from the time the locomotive first stops until it starts again with a full charge, seldom requires more than two minutes. The pressure of the air entering the high-pressure cylinder is ordinarily 250 pounds per square inch and the corresponding pressure entering the low-pressure cylinder is 50 pounds.

On the surface, within a radius of two or three miles, compressed air locomotives are the equivalent in power and flexibility of steam locomotives of the same weight, and will do exactly the same work.

From time to time efforts have been made to increase the efficiency of the compressed air locomotive by use of more elaborate valve gears and by the use of compound cylinders, but without effective and substantial preheating, or interheating, all such efforts are predestined to failure.

The H. K. Porter Company, of Pittsburgh, Pa., construct the greater number of this class of locomotive.

GASOLINE LOCOMOTIVES.

Internal combustion locomotives, since their introduction, several years ago, have been changed and improved to such an extent that they are now demonstrating their particular fitness for use in lumber mills, brick yards, contracting operations, and even in light switching in railroad yards. In fact, wherever loads are to be hauled at moderate speeds, and within the range of available motor powers, the various sizes and types of gasoline locomotive have proven successful.

There are several concerns now constructing and marketing the gasoline locomotive, but the variety of models manufactured by the Baldwin Locomotive Works will suffice to give the reader an idea of the general sizes and capacities of the locomotives in general use.

The standard sizes, weighing 3½, 5, 7 and 9 tons, cover a range sufficient to meet the requirements of ordinary industrial service, while the 23-ton locomotive is especially fitted for heavier duty in industrial service, as well as for the lighter switching in yards and terminals.

General Construction.

The general construction of the gasoline locomotive, so far as is practicable, is similar in design to the modern steam locomotive, making for an efficient locomotive, simple in construction and operation. As may be seen from Fig. 334, the engine is vertical, and, through a multiple disc clutch, encased in the fly-

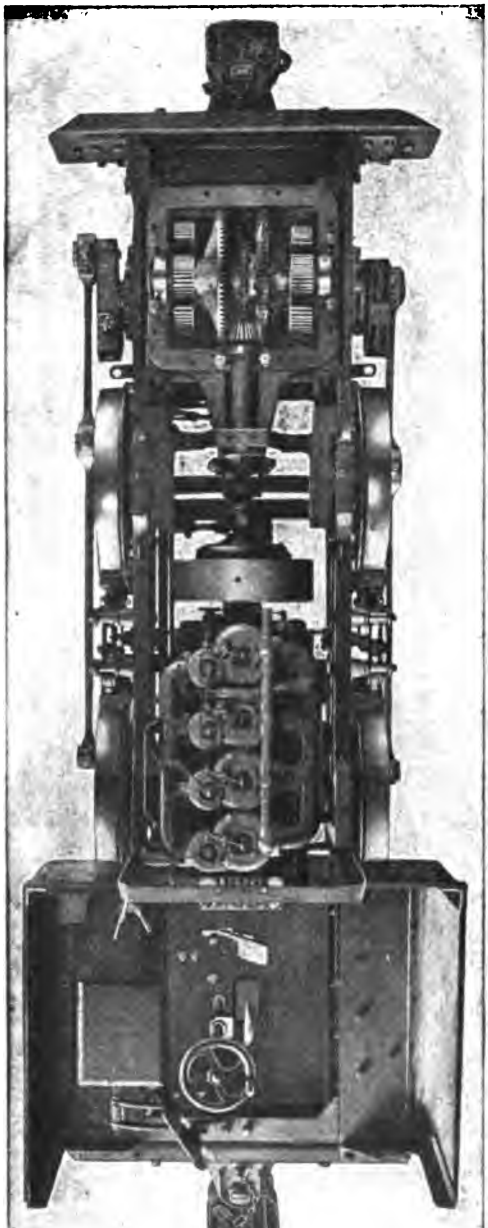


Fig. 334.

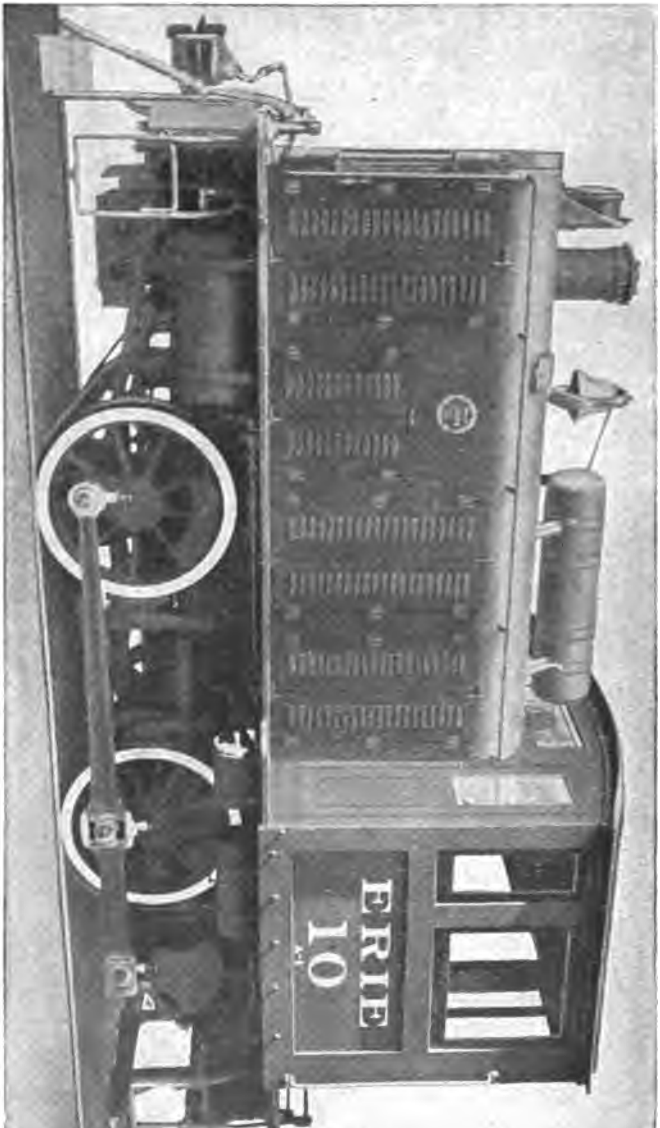


FIG. 335.

wheel, delivers its power through a system of bevel gears, to the wheels, by means of patented side rods of special design. This arrangement provides for two speeds forward, and two reverse.

The engines used, on the Baldwin locomotives, are of the four-cycle, four cylinder (six cylinders for use in the 23-ton) type, are water cooled, and are specially designed to withstand extreme service conditions. Lubrication is automatic, and an electric self-starter is usually applied, as it is practically indispensable.

The frames are of the cast steel bar type, exceptionally strong, and are generally similar to those used in steam locomotive practice. Except on extremely narrow gauge locomotives, the frames are placed inside the wheels, as shown in the illustrations.

An efficient hand or foot operated brake is provided, with brake shoes operating on the wheel tire, as may be seen in Fig. 335. Application of air brakes may be had, if desired. The arrangement for the use of sand, running in either direction, may also be seen in the illustration. The fuel tanks are of seamless drawn steel, located, when possible, over the hood, as is illustrated in Fig. 335. The radiator is especially large and strong, and is placed at the front of the locomotive, while, at the rear, a completely enclosed cab is provided.

The fuel consumption, under normal conditions, will be about one-tenth of a gallon of gasoline per horse power per hour. The locomotive will run equally well in either direction, and has a speed of five miles per hour in low gear, and ten in high gear. Slightly higher speeds, for special conditions of service, may be provided for. Following is a table giving some interesting data concerning gasoline locomotives, of the Baldwin type:

Performance, Rating and Dimensions.

Weight of locomotive in pounds	7000	10000	14000	18000	46000
Number of cylinders	4	4	4	4	6
Dimensions of cylinders	4¼x5½	5x6	5½x7	6½x8	7¾x12
Draw bar pull in pounds, high gear	750	1100	1600	2100	4700
Same, low gear	1700	2400	3500	4400	10000
Diameter of driving wheels, in inches	24	26	30	36	42
Wheel base	3′ 0″	3′ 6″	4′ 6″ 4′ 0″	4′ 9″ 4′ 0″	6′ 6″
Length over frames	10′ 1″	10′ 5″	13′ 0″	14′ 2″	18′ 8″
Minimum gauge in inches for outside frames	24″	24″	24″	30″	

The 23-ton type is of standard gauge, with inside frames only.

SHOES AND WEDGES.

The proper fitting, or adjustment, of the shoes and wedges is an involved mechanical problem, and one of the most important tasks connected with the repair work of a locomotive; for the performance of the engine will depend, to a great extent, upon the accuracy with which the shoes and wedges are aligned.

The work of laying out and fitting shoes and wedges may, at first, appear mysterious or confusing to the young or inexperienced mechanic, but the operation is comparatively easy of accomplishment, being simply a matter of taking, and laying off, certain measurements accurately, as we shall endeavor to make clear.

While the pedestal jaws are a part of the locomotive frame, they are built at right angles to the main frame member, and are so formed as to receive the boxes for the driving wheel axles, holding them in their proper relation to each other, and to the frame and cylinders of the engine. However, were the driving boxes fitted directly to the pedestal jaws, without the use of shoes and wedges, or some other device for taking up wear, and consequent lost motion, the engine's performance would be very unsatisfactory.

For this reason, shoes and wedges are used to form a bearing between the pedestal jaws and the driving box holding the driving wheel axle. These shoes and wedges are generally made of cast iron, and are planed to fit the pedestal jaws, and to fit into the grooves on the faces of the driving box. The object of these castings is to hold the box containing the driving wheel in its correct position, at right angles to the cylinder center lines, and to permit of adjustment for wear. It is common practice to fit the shoe to the face of the forward pedestal jaw, and the wedge to the rearward jaw, although at times this arrangement is reversed, and occasionally both shoes and wedges are located between the driving box and one pedestal jaw. But, as pointed out, the first is the most common practice, and the one which we shall consider here.

The wedge must be so adjusted as to take up all lost motion between the pedestal jaws and the driving box, while the two flanges on the driving box prevent any lateral motion, but the wedge must not fit so tightly as to cause the driving box to stick: the fit should be such that the driving box is free to move up and down in the jaws.

It must, therefore, be apparent that the fitting of shoes and wedges is very important. Their proper adjustment will prevent much rod brass and driving brass trouble, as well as tire cutting, and even broken crank pins or side rods. In addition, improper alignment of the driving axles will make it difficult, if not impossible, to couple the connecting-rods on their pins, if the solid-end type rods are used.

Before explaining the methods of laying out and fitting shoes and wedges, we think it would be well to make clear the correct setting in place of their foundation—the frames, and, as a result, the pedestal jaws.

Fitting Pedestal Braces and Setting Frames.

When frames are new, or have been removed for extensive boiler repairs, or other purposes, it is important that when they be put in place again, they be properly lined and squared up. In what follows, we shall endeavor to show how this can be accomplished.

The last operation, when taking down old frames, should be to place them bottom side up, on thick blocks or horses; if you do this they will be in convenient position for fitting the pedestal braces.

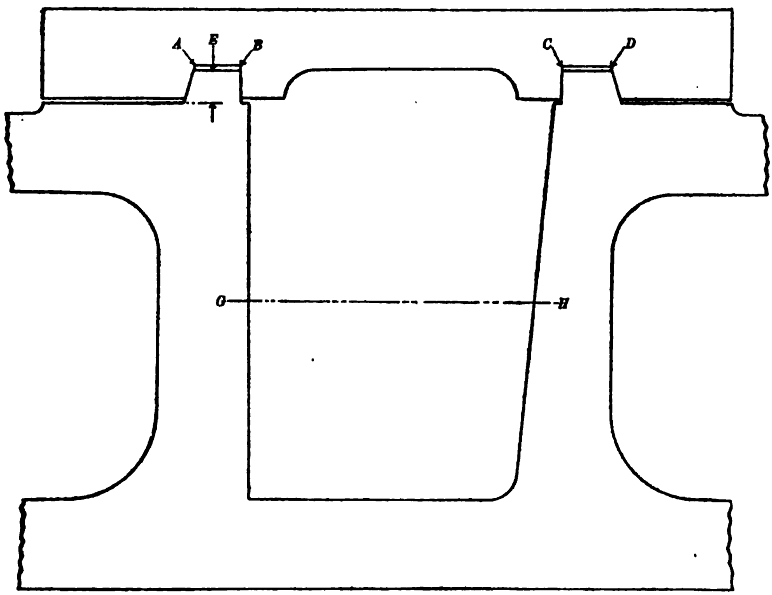


FIG. 336

We will suppose that new pedestal binders are necessary, and the first thing to be done in fitting them is to prepare the frame, or bottom ends of the pedestal jaws, for receiving them.

The fit A B, C D, Fig. 336, should be filed perfectly at right angles to the frame's length, and all to the same level, which should be about 7 degrees from a right angle to the top of frame. B and C should be ⅛ inch below the face of the pedestal jaw, as shown in Fig. 336, to allow the jaws to be refaced without destroying the fit. Before the pedestal binder is laid out, it should be planed on one side.

Now lay the binder on the pedestal jaws, planed side down, and with a small straight edge, held against the fit and against the bottom of the binder, scribe a line on the binder in four places. Scribe lines on the edge of binder next to frame to show the right bevel to plane to.

Make the depth of recess ¼ inch more than the distance E, Fig. 336. After planing off the recess to these lines, if the work has been accurately done, the binder will drop to within ⅛ inch of the bottom of frame. It should now be fitted down 1/16 inch farther by filing, when the holes may be laid out. If possible, the

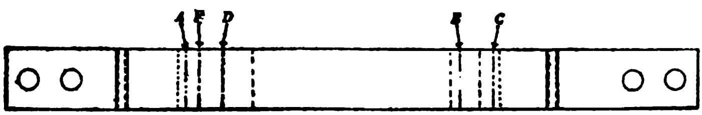


FIG. 337.

bolt holes should be laid out and drilled, so that no reaming will be necessary, since reaming the holes weakens the frame. It is important that the hole for the wedge-adjusting bolt be in the proper position. If it is too far from the face of pedestal jaw, it will interfere with the driving box. If the hole is too far the other way, it will come in contact with the face of pedestal jaw.

To find its proper position, proceed as follows: Fig. 337 is the pedestal binder. The line A represents the face of the back pedestal jaw, and C the face of the front jaw. The line G H, Fig. 336, is parallel to the top of the frame, and passes through the center of the pedestal. If the distance between A and C, Fig. 337, be 12¾ ins., and the driving box be 11¼ ins., it is evident that the thickness of driving shoe will be $\frac{12\frac{3}{4} - 11\frac{1}{4}}{2} = \frac{3}{4}$ inch.

Hence, scribe the line E, Fig. 337, ¾ inch from C. The distance between E and D equals the size of driving box, 11¼ inches. It is now plain that the space D A represents the thickness of the bottom of the wedge. Suppose the diameter of that part of the bolt that enters the wedge to be 1¼ inches, then the center of bolt hole in binder should be 11/16 inches back of the line D, or on the line F. This will allow 1/16 inch clearance between the bolt and driving box.

Having finished this part of the work, we will proceed to put the frames in place. But, before putting them up, the expansion plate studs should be examined carefully, and if any of them show signs of leaking, they should be replaced with new ones.

Any studs that prevent the frame from sliding out and in, when the frame is in place, should be taken out, and the new ones not put in until after liners are fitted. A die nut should be run over the old studs that are good, to straighten the threads.

Now place blocks across the pit, directly under where the pedestals will come, when frames are in place, to support them while fitting the liners and buckles. Set the frame on the blocks, and raise, or lower, it to the proper height, which may be determined by using the buckles as a gauge. When the frame is at the right height, the buckle will slide on the studs.

Now put in the splice bolts, then fasten the deck in place. Set inside calipers to the distance between the frames at the deck. Then, by means of rods and plates of iron, which should be made of at least ¾-inch iron, placed at K, at M, and at N, Fig. 338, set the frames the same distance apart at these points as at the deck, using the calipers as a gauge.

Now run lines through the centers of the cylinders, letting them extend to the back end of the frames. Then measure the distance from the outside of pedestal jaws to the lines. This distance should be the same at all the pedestals, but very likely it will not be.

Suppose the distance from the left front pedestal to the line to be 11½ inches, and from the right front 12 inches. This indicates that the frames are ¼ inch too far to the left in front. To draw them over, insert iron wedges between the boiler and the frame, at L and J, Fig. 338, on the right side, and drive them down until the frame has been drawn over the required

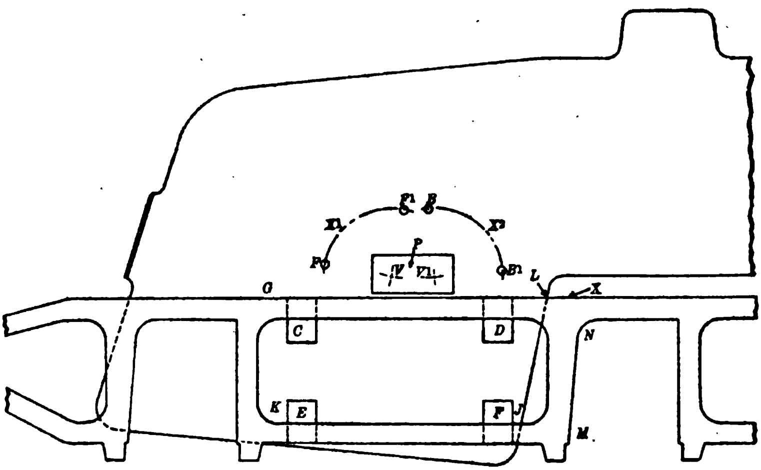


FIG. 338.

distance, which, in the present case, is ¼ inch. As the right frame is drawn out, the left will be drawn in, by the rods previously mentioned, which bind the frames together.

Next insert iron wedges between the frame and the boiler at L and J on the left side of the engine, but do not drive them down any, as these are merely to fill up space between the frame and boiler, to hold the frames in place after they have been drawn over.

We will now go to the back pedestals. Suppose the distance from the left pedestal to the line through the cylinders to be

$11\frac{5}{8}$ inches, and from right pedestal to line $11\frac{7}{8}$ inches; then the proper distance from pedestal to line, on both sides, is

$$\frac{11\frac{5}{8} + 11\frac{7}{8}}{2} = 11\frac{3}{4}$$ inches. Hence the frames are $\frac{1}{8}$ inch too far to the right. Draw them to the left by means of wedges at G and K, on left side, then insert wedges on right side, to hold them firmly in place. Now try the front pedestals again, as setting the back ones will be liable to throw them out of line. If such is the case, they can be put in line again by driving the wedges farther down on the side that is farthest from the line, being careful to first raise the wedges on the opposite side.

We now have the frames the same distance from the lines at all four pedestals, but something more is necessary. They must be at right angles to a line drawn across their tops. We will try them at the front pedestals first.

Place a straight edge across the frames at X, Fig. 338, and then place the short side of a 2-foot square against the straight edge; the side of the pedestal should be parallel to the long side of the square. Suppose it is found to be $\frac{1}{8}$ inch away from the square, at the bottom end of the left side; then the right pedestal will be that distance from square at top end, since the frames are held parallel by the rods. They could be squared up by raising the bottom wedge (in front) on the right side, and by driving the one on the left side down, but that would throw both frames about 1/16 inch too far to the left. To square them up, and at the same time to keep them in line, proceed as follows:

Raise the bottom wedge on the right side enough to allow the bottom of the pedestal to go toward the boiler 1/16 inch, and drive the bottom wedge on the left side down, sufficient to draw the bottom of the pedestal out 1/16 inch. This will leave them out of square, the same as they were, but only half as much, and they have been drawn 1/32 inch too far to the left. Now raise the top wedge on the left side enough to allow the top of the pedestal to move toward the boiler 1/16 inch, and drive the top wedge on the right side the same amount.

We now have the frames square, and have drawn them back into line. Proceed in the same manner with the back pedestals. The frames now have the proper positions, and, in order to determine whether or not they move, and to know when the liners are of the right thickness, the positions of the frames should be marked in some way. A very good method is to use an ordinary tram. Make a center punch mark on the side of the boiler, near G, L, J and K, Fig. 338; then with the point of the tram in these marks, scribe arcs on the frames at the four places.

Do the same on the other side of engine. It is plain that these arcs must come to the same position, when the liners are fitted and buckles on. We are now ready to fit the liners C, D, E, F, Fig. 338. Generally the old liners can be used again. If they are too thin to fill the space between boiler and frame, a piece of boiler plate can be riveted onto the side next to the frame. If a piece of the exact thickness cannot be had, rivet on one slightly thicker than is required, then plane it down to the exact thickness.

After the liners are in place, the studs can be screwed in, through the holes in them, by using a stud nut.

The buckles should be loose enough on the frame to allow them to slide on it without binding, when the boiler is expanding or contracting.

The lateral, or cross braces can now be put on. If they are not of the correct length, have the blacksmith lengthen or shorten them to suit.

To lay out the holes in a new expansion plate, when the studs are in the boiler, with any degree of accuracy, is generally not very easily done. The following method has been found to give satisfaction: Make a small center punch mark in the center of each stud that is to pass through the plate. Then set dividers to any convenient radius—say, 10 inches—and, with centers of studs as centers, scribe the arcs X^1 X^2, Fig. 338, on the side of the boiler, and make four center punch marks, F, F^1, and B and B^1, on the arcs, two on each arc, about 90 degrees apart if possible; more or less will answer the purpose, but 90 degrees will give the best results. Now lay the expansion plate P, Fig. 338, on top of the frame, as shown (this is not the position usually occupied by an expansion plate, but will serve our purpose), with the part that goes next to the boiler against the ends of studs. If we now use the points F, F^1 and B, B^1 as centers, and with the same radius used to scribe X^1 X^2, scribe arcs V V^1 on the expansion plate, their points of intersection will not lie in a line with the center of stud, but will be to the side of this line nearest the arc X^1 or X^2; or, in other words, the radius used was too short. The correct radius with which to draw the arcs V and V^1 can be found thus:

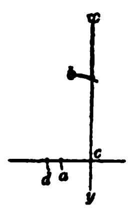


FIG. 339.

On a board, or other plain surface, draw two indefinite lines at right angles to each other, Fig. 339; then lay off on one line the distance *a c*, equal to the length of the studs, and *a d*, equal to the thickness of the expansion plate. With *a* as a center, and with the same radius as the arcs X^1 X^2, Fig. 338, scribe the arc *b*, Fig. 339, across the line *x y*. Now set the dividers to the distance *b d*, which is the correct radius with which to scribe the arcs V V^1, Fig. 338, on the expansion plate, using F and B^1 as the centers, to have their point of intersection in line with the center of the stud. Hence this point will be the center of the hole in the expansion plate.

It is hardly necessary to say that an arc must be scribed on the side of the boiler from the center of each stud that passes through the plate.

Assuming that the frames, boiler, cylinders, and pedestal braces are all in their proper places, our first task will be to square up the engine.

Object of "Squaring Up" an Engine.

Our object, now, is to see that the main driving axle, or shaft, is set at right angles to the cylinder centers, and that the centers in each of the other pairs of drivers (on each side) be set an equal distance from the centers of the two main drivers.

It must not be understood that every pair of driving wheels must be the same longitudinal distance apart, but that corresponding wheels should tram perfectly on each side.

Various methods are employed for locating a "square" line on each frame of an engine, from which to lay off the shoes and wedges and properly secure these centers. Some use a fish-tail tram, which is a long rod (3/8" iron pipe is suitable, of about 12 foot length), one end being formed into a rather blunt point. The other end is an ordinary tram point, provided with the usual set screw for clamping it in position. This tram is used from the center casting, and locates a center on the inside of each main pedestal jaw. Others use a three-pointed tram from a center located midway between the frames, on the back end of the cylinder saddle. The rocker box centers are also used for this purpose; others line one cylinder, or both. But, in any event, the object is the same—to secure a "square" center upon each main pedestal jaw, from which to lay off the shoes and wedges. The preferred method, perhaps on account of its saving in time, although it is not the most accurate, is to use lines running centrally through each cylinder, and extending back along the frame a sufficient distance to adjust, from them, the square centers on the pedestal jaws. Care must be taken, in using this method, to have the center casting set centrally.

Setting the Center Casting.

As the truck leads the engine, it is very important that the center casting should be properly located. To secure the center casting in its proper position, set it exactly midway between the frames, and central with the cylinder saddle, front and back, taking the measurements from the turned surface of the casting, or from its dead center. When the center casting is properly set, ream or rosebit the holes, and use "driving fit" bolts to hold it permanently in this position.

Lining the Cylinders.

When lining the cylinders, fine lines or wires should be set central with the counterbores of each cylinder. These lines are generally fastened at the forward end of the cylinder, to adjustable pieces held by the cylinder studs, but they can be attached at either end of the cylinder; these lines should be of sufficient length to extend a little back of the main jaw, which embraces the main pair of driving wheels, and weights should be attached to the rear ends of the lines, to keep them taut, and to guard against changes in their lengths due to variations in the temperature. If the lines are properly and accurately strung, they will show precisely where the axle center should be, but the center lines of the cylinders occasionally vary from the center line of the frames and, as previously stated, such measurements as derived from these lines are not always reliable.

Locating Square Centers.

Assuming now that lines have been strung through the center of each cylinder, and are properly fastened, our first step will be

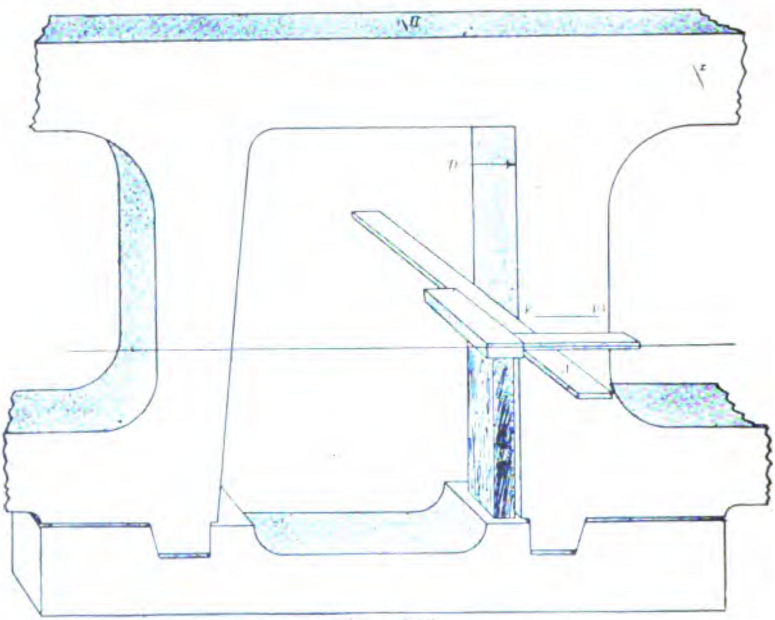


FIG. 340.

to measure the distance between the two lines just back of the cylinder saddle, and also near the backward ends of the lines, to ascertain whether or not they are parallel; also measure between each line and frame, front and back, to see whether the frames are parallel with the cylinders. If they are not, they may nearly always be moved at the back end enough to make them line centrally.

It is considered advisable to have first painted the jaws of the frames with a thin coat of white lead. This makes an excellent background on which to scribe the necessary lines, and does not consume much time, as it will dry while you are completing the preliminary work to setting the shoes and wedges.

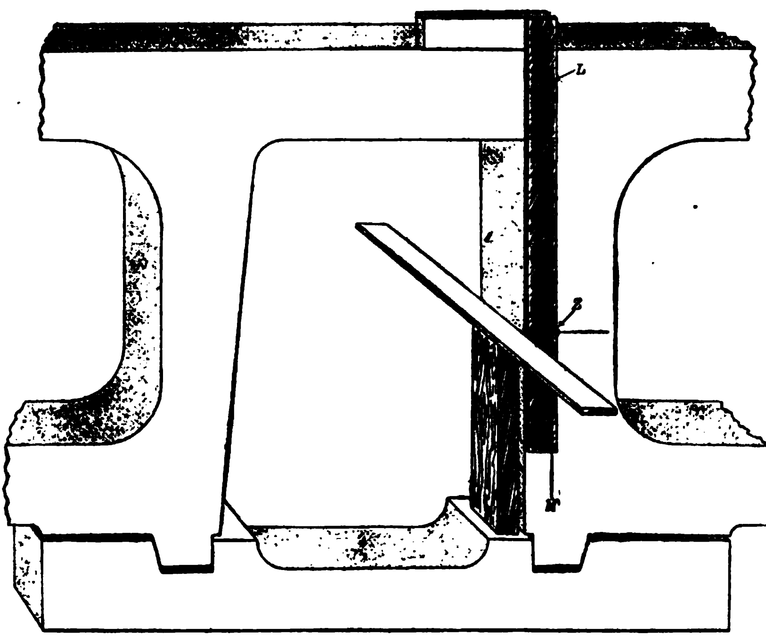


FIG. 341.

Place a straightedge, A, on the support, B, as shown in Fig. 340, or clamp it to the front face of the forward jaws, just below, and at perfect right angles to, the lines extending back from the cylinder center. Place a two-foot square against the straightedge and try it, on each side of the engine, with the lines drawn through the cylinders, and ascertain if the straightedge is at perfect right angles to both lines. If the square does not show a perfect right angle between the straightedge and the lines, it will be necessary to place liners, or shims, between the straightedge and one of the jaws, to secure the right angle desired between the straightedge and the cylinder center lines.

When the desired position has been secured, use a pair of hermaphrodite calipers and scribe the short horizontal line V V^1 on the inside and outside of each forward main jaw, far enough forward to clear the flange of each main shoe, keeping the distance from the top of the frame to these lines equal.

Now place a square on the top of the frame, as shown in Fig. 341, allowing the blade of the square to touch the straightedge, and draw the vertical line L M perpendicular to the top of the frame. The point where the vertical and horizontal lines intersect will be the *square center* Z, so scribe a small circle around this point to distinguish it from other marks. This vertical line is known as the "square line," and, from this line as a basis, we locate the centers of the pedestal jaws; so similar vertical lines should be drawn on each forward jaw, on both sides of the engine.

Locating Jaw Centers.

Now that we have located the square center Z, and the square line L M, Fig. 341, it is necessary to locate the center of the pedestal jaw, showing exactly where the center of the axle should be.

Assuming that both pedestal jaws taper, as shown in Fig. 342, we will extend the vertical lines A and B true with the inner faces of the jaws, on the side of the frame, and then, above the jaws, scribe the line D E parallel to the top of the frame, then, with a pair of dividers, locate the center C, midway between the intersections of the lines A and B, on the line D E, above the main jaw, and the point C will represent the center of the pedestal jaw. Now use the dividers again, and ascertain whether or not the distance from the pedestal jaw center C to the square line L M is equal on both sides of the engine. If it is not, move one center forward and the other backward, a distance equal to one-half the difference between these distances, thus making the distance equal on both sides.

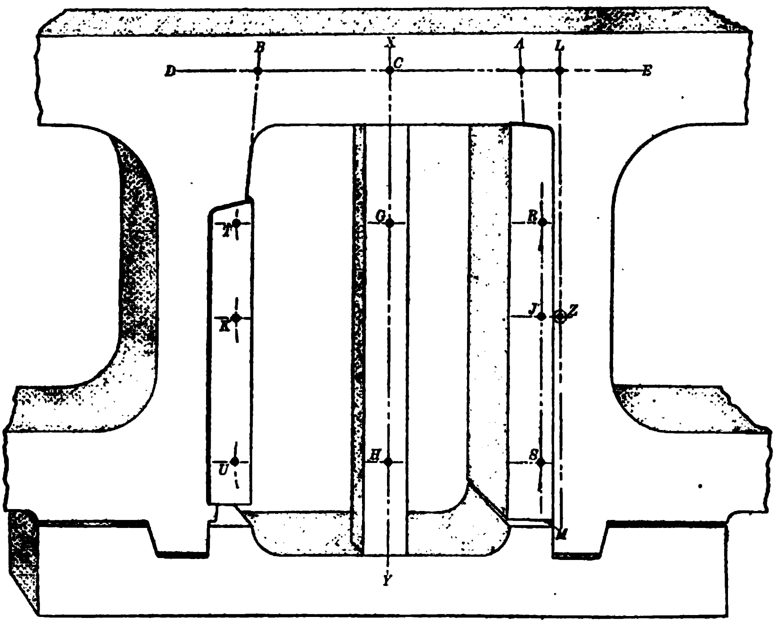


FIG. 342.

If the forward jaw is at right angles to the frame, and the back jaw is tapered, as shown in Fig. 343, a different method should be adopted.

First ascertain the thicknesses of the shoe and wedge at their tops, and see if they are equal. If they are not of the same thickness figure on the liners accordingly; then clamp each shoe and wedge securely in its proper position, allowing about ¼″ clearance space between the bottom of the wedge and the pedestal binder, to pull down the wedge if necessary.

Now use a straightedge on the face of both the shoe and the wedge, and scribe the vertical lines A and B on the frame as in the previous example; locate the center C midway between the lines A and B, on the line D E. If the center obtained is an equal distance from the square line on each side of the engine,

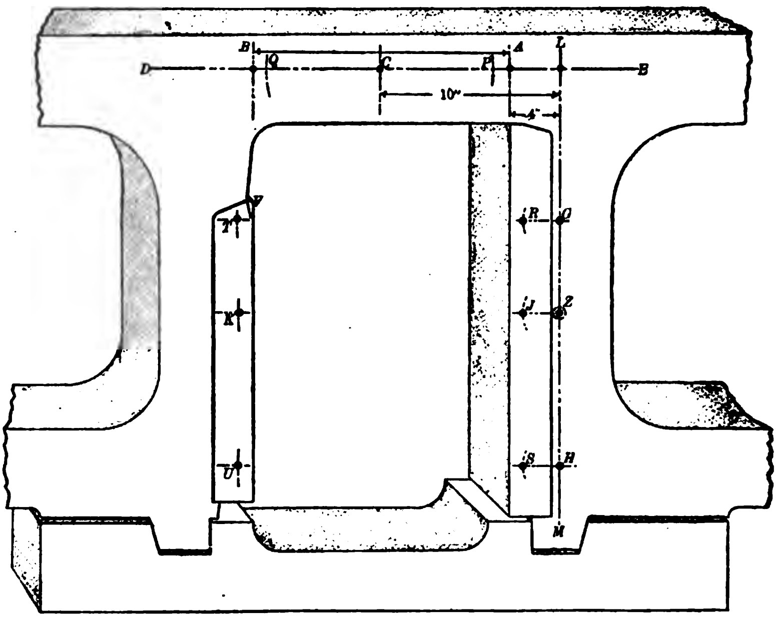


FIG. 343.

the jaw center is correct. If not, make the necessary changes in the manner previously explained.

If greater precision is desired, insert wooden centers in the jaws, as shown in Fig. 342, placing them flush with the outside of the frames. Place a square on top of the frame and scribe the line X Y through the center C, and down through the wooden center, and the point where the line X Y intersects the cylinder center line Z will be the pedestal jaw center. If the distance from the pedestal jaw center to the square line L M is the same on both sides of the engine, the pedestal jaw center is correct. If the distance is unequal on both sides, the differences should be divided the same as previously explained. If the construction of the engine will permit the use of a long tram, wooden centers need not be used in other than the main jaws.

Another Method.

Another method, used occasionally when it is inconvenient to withdraw the pistons from the cylinders, is to locate the points x, Fig. 340, on the *inside* of each frame, equally distant from the tops of the frames, and from the point D, the forward finished face of the pedestal jaw.

Now, with a "fish" tram extending from the point x on each side of the engine, scribe arcs on the back of the cylinder saddle near the center, and the intersection of the arcs will be the center between the frames, so mark the same with a prick punch. With an adjustable tram, extending from the point marked on the cylinder saddle to about the center of the pedestals scribe the arc H on top of the frame on each side of the engine. Place a straightedge across the top of the frames at the points marked H, and if the straightedge shows a perfect right angle, the longitudinal center of the pedestal jaw can be easily located by the use of calipers.

Tramming the Jaw Centers.

Now, having located the center of the main jaw, carefully set an ordinary tram to the length of the side rods, place one end of the tram at the center C, in the main jaw, and with the point of the tram scribe a line intersecting the line D E on the side of the frame over the other jaws, on each side of the engine, and repeat the operation for each of the jaws.

As previously stated, it is not necessary that each pair of wheels be the same distance from any other pair, but wheels on the same axle should tram perfectly on each side, and, therefore, their centers should correspond. If they do not tram perfectly, however, you can move one or both centers forward or backward, in the same manner as explained for mismatched pedestal jaw centers, except where solid type side rods are used, when, in such cases, it will be necessary to line up the shoes and wedges to accommodate the rods. If the boiler sets down between the frames, make the rear jaw and driving box centers 1/32″ shorter than the length of the side rods, to allow for expansion, but make no allowance for expansion in the jaws or boxes ahead of the main jaw.

Laying Off Shoes and Wedges.

Assuming that the square line L M, and the jaw center Z, have been located, we will first ascertain the sizes of all of the boxes, and then add two inches to the size of each box, and divide the total by two. This total will represent one-half the size of the box, plus one inch, so set the small trams to that length, and lay off the amount, in its proper position, from each box center, front and back, which is represented by the letters C and A, and C and B, in Fig. 344.

Now, using A as a center, lay off the distance A F, equal to C A, on the line D E above the main jaw on each frame. Then securely block all the shoes and wedges in their proper positions, and block the wedges up off the binders.

If the shoes and wedges are old, and require liners, place a square on top of the frame, as in Fig. 341, and bring the blade of the square to bear against the face of the shoe, or the wedge, as the case may be, and measure from the near side of the square to the point A. The amount it lacks of being one inch will represent the thickness of the liners required.

Then, with a small tram set to any convenient distance, and with C as a center, scribe the arc *s s* on the shoe, and with the same length, and with F as a center, scribe the arc *o o* on the shoe. Carefully mark the intersection of the two arcs, as X, and

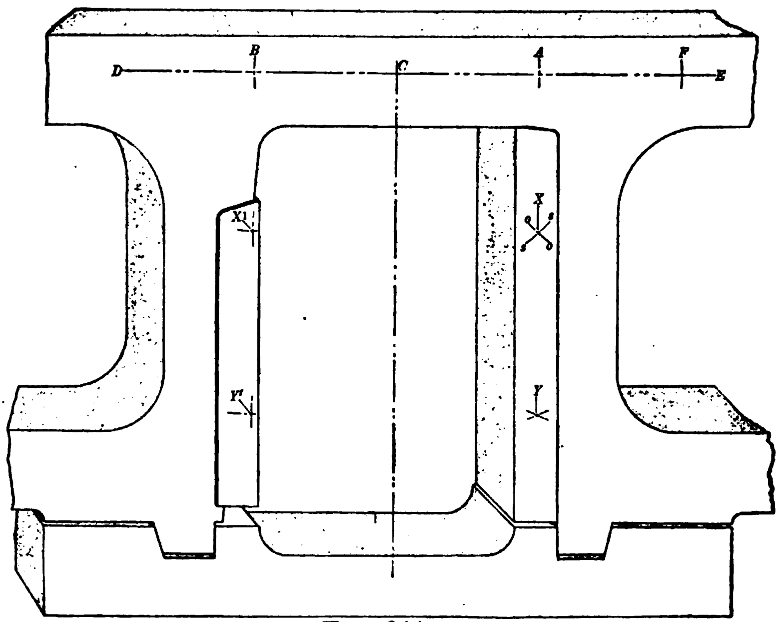


FIG. 344.

prick-punch it for reference; then go to the opposite side of the engine and repeat the operation on the shoe in the main jaw.

Next change the length of the tram to any convenient distance, and, from the same centers, C and F, locate and mark the point Y on each shoe in the same manner, about 3″ from the bottom of the shoe. It is now apparent that the point A on the face of the frame has been accurately transferred to the shoe, and is represented by the two points, X and Y, and that the vertical location of the points was established by the same operation.

It is necessary, however, to locate another point on the line connecting X and Y, on the inside of the shoe, by which to set the shoe on the planer, and this third point may be obtained by placing a straightedge across the faces of the shoes, or by the use of a transfer tram.

If a transfer tram is used, set the point at C, and with the large end extended over the top of the frame, scribe an arc on the inside of the shoe. Then, with F as a center, scribe another arc, and the intersection of the two arcs will be the point desired.

When using the transfer tram it is immaterial whether the boxes be of similar size, or whether they be bored out of center, for, if the correct size of each box has been laid off on the frame, the point located on the inside of the shoe is bound to be correct.

Now set the small tram to the length A X, and transfer it to all of the other shoes, and repeat the operation with the tram set to the length A Y. Locate the vertical position on the inside of the shoes by the use of mephrodites, measuring from the top of the binder.

Next set the long tram to a length from the center C of the main jaw to the next pedestal jaw, and, if the boxes are of standard size, the length will equal the length of the rod, and not otherwise; so, with the length obtained, and with X and Y as centers, scribe similar arcs, top and bottom, and also on the inside, of the other shoes, on both sides of the engine; then set the small trams to the exact size of each box, as A B, and lay off this length on each wedge, as X^1 and Y^1, inside and out, all around the engine, and your job is finished.

It is considered good practice to circle all marks on the shoes and wedges, as a means of identification, and to scribe lines inside and outside of the frames, showing the exact positions of the shoes and wedges, before removing them from the pedestal jaws.

Another Method.

Referring again to Fig. 342, we find a wooden center fit between the binder and the frame of the main jaw with its face flush with the outside of the frame, on which the center of the pedestal jaw has been located. Now, with a pair of hermaphrodites set to any convenient distance, from the top of the frame scribe two horizontal lines across the shoe and wedge and the wooden center piece, and where the two lines intersect the line X Y, on the wooden center piece, locate the centers G and H.

Now, assuming that the calipers show that the driving box measures 12″, use a pair of dividers set to 6″ (which is one-half the thickness of the driving box), and from the points G and H, on the wooden center, scribe four small arcs on the shoe and wedge, and at the intersection of the arcs with the horizontal lines mark the four centers R, S, T and U.

If a shoe and wedge gauge is used, add the distance from the face of the gauge to the point of the gauge, which we will assume to be 1″, to one-half the thickness of the driving box, here assumed to be 6″, and set your dividers to the total, 7″, before scribing the four small arcs. The gauge will prove your work, and that of the planer hand, and, when planed, the gauge point should enter each casting without crowding.

Now you have two points located on the outside of each shoe and wedge on the main jaw, by which to set them on the planer, but it is necessary to locate another point on the inside of both the shoe and the wedge, in order to have them square across their faces. Therefore, scribe a line from the point R to the point S, and set a pair of dividers from the square center Z to the line

R S, and with this length, and from the point Z on the inside of each frame, locate the point J on the "inside" of each shoe. Next set a pair of dividers from the point R to the point T, and, with this length, go to the inside again, and, from the point J, locate point K, keeping the points J and K equal distances from the top of the frame. Sometimes you can set a long tram and lay off the other points from the points just secured, but in case the tram can not be used, the points on the other shoes and wedges must be laid off the same as those for the main jaw.

If, for any reason, the wooden centers are undesirable, or cannot be used, the desired result may be obtained as follows:

Assume that the dimensions of the driving box are the same as those in Fig. 343, which we just examined, and that our shoe and wedge gauge is to be used upon the shoe only.

First locate the points G and H, Fig. 343, any convenient distance from the square center Z, on the square line L M, and, through the points G and H, scribe lines on both the shoe and wedge, as shown in Fig. 343, keeping each line parallel with the top of the frame.

As the distance from the jaw center C, to the square line L M, is 10″, while one-half the thickness of the driving box is 6″, plus 1″ for the gauge, equals but 7″, we have a difference of 3″. Therefore, set a pair of dividers to 3″, and, from the points G and H, locate the two points R and S on the outside face of the shoe. (The illustration is not made to scale.) Now, at the same distance, locate, from the square center Z, the point J on the *inside* of the frame. Remove the main shoes, and have them planed and filed perfectly square across their faces. Now, if the points R, S and J are correct, use the shoe gauge from these points. With a long tram set to the jaw centers (assuming all boxes are of equal thickness), lay off all the other shoes, and have them planed, filed square and trammed perfectly before you lay off any of the wedges.

Now clamp your wedges in position, placing them ¼" above the pedestal brace, and, with a wedge gauge, small tram, or pair of dividers set to an convenient length, from the points R and S locate the points T and U on the outside of each wedge, and with the same length, from the point J locate the point K on the *inside* of the wedge, and you will have three points by which to set the wedge for planing. (The shoes and wedges should all be stenciled so that they may be returned to their correct positions in the various jaws.) Next set a pair of inside calipers to the exact size of the driving box, and chip or file a spot anywhere on the wedge until you locate a point that calipers exactly between the shoe and the wedge, and mark the point with a prick-punch mark, indicated by the letter V. This mark should be just barely scraped by the planer tool, and should be visible when the wedge is finished.

Lining Up Old Shoes and Wedges with Boxes of Different Size and Brasses Bored Out of Center.

Of course, it is a matter of common knowledge that, as a result of a combination of circumstances, all shoes and wedges do not wear the same, the tendency being for the shoe and wedge of the main jaw to wear the most. On the other hand, it will sometimes be found that the wedge will be worn greater than the shoe, so that an equality of lining may prove unsatisfactory in such cases, and there is considerable difference of opinion as to the propriety of lining the shoe and the wedge or both. In some shops it is a rule to line the wedge only, while in others they divide the lining between the shoe and the wedge, and as the latter practice is the most common it will be considered here. It is also considered good practice, when some new shoes and wedges are to be used, to place them in the main jaws.

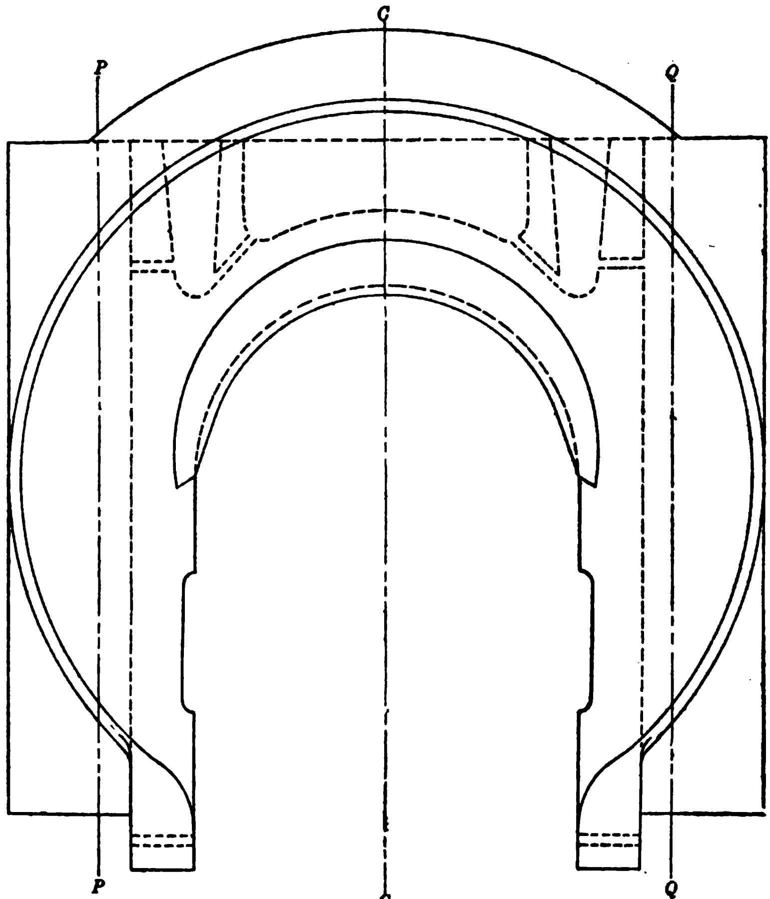


FIG. 345.

Assume that the square center, and the pedestal jaw center, have been properly located by one of the methods explained, and that the shoes and wedges have been planed, or lined, to their proper thicknesses.

First, clamp all shoes and wedges securely in their proper places, making use of spreaders, or jacks, to hold them in place, and setting the wedge about ¼" above the binder, or nut, on the wedge bolt, to allow sufficient space for loosening of the wedge if necessary. Place a straightedge across the faces of the shoes and wedges to ascertain if they are square with the frame; if they are, then scribe the lines A and B on the frame, above the jaw as shown in Fig. 343.

Now place all the driving boxes on the floor in their respective positions, with their tops toward the rail, and the outer sides of each box upward. Use a transfer plate (or gauge for that purpose), and scribe the vertical lines P P and Q Q across the outside flange of each box, true with its bearings for both the shoe and the wedge, as shown in Fig. 345. Then put a center line C C in each driving box, to represent the center line, through the axle. Now set a pair of dividers from the center line C C in the driving box to the line P P and transfer it to the frame, marking the distance in front of the jaw center C, as shown by the letter P, in Fig. 343. Next set the dividers from the center line C of the driving box, to the line Q Q, on the back face of the driving box, and with this length from the jaw center C describe the arc Q, in Fig. 343; do the same on all jaws.

The distance between the lines A and P, on the top of the frame, indicate the thickness of the liner required behind the shoe, and the distance between the lines B and Q show the thickness of the liner required behind the wedge, to keep them central, and liners should be added in accordance with the measurements.

Now rivet your liners to the back of the shoe and the wedge, allowing an extra 1/32" in the thickness for planing, and an additional 1/32" if the liners are to be planed inside.

When all the shoes and wedges have been lined up to the proper thickness, add to the distance C P the amount of the gauge, 1″, and the same amount to the distance C Q.

You can now employ either method, as previously explained, to lay off the shoe and wedge, or with 1″ added to the distance C P and C Q, you can carry down square lines from the top of the frame and locate the four points R, S, T and U, and from these points locate the points J K on the inside of the jaw.

It is not necessary to locate the jaw center on the frame when you are lining up old shoes and wedges, unless you wish to line the shoes and wedges central. You can lay off the main shoes from the square line, and the other shoes from the main shoes, using the length of the rod, and lay off each wedge from its own shoe.

Tramming the Driving Box Centers.

Regardless of the method employed it is considered good practice, if time will permit, to prove your work before the wheels are placed under the engine, so place all of the shoes and wedges in their respective positions in the pedestal jaws when they are planed.

First, use a straightedge across the faces of the shoes and wedges and see if they are true with the frame; if they are, use a pair of inside calipers and see that each wedge is parallel with its shoe, trying four points horizontally, and their width as compared with the driving box.

Next place all of the driving boxes up in the jaws, together with the corresponding binder, shoe and wedge, setting the latter up moderately tight, but not enough to cause the driving box to stick—the latter should be free enough to fall by its own weight.

Now place centers in all of the driving boxes, and see that each pair of boxes trams perfectly with the main boxes; also try the centers of the main boxes with the square lines on the frame, for they should be an equal distance from the lines.

Make any alterations necessary, and scribe a line on the inside, and one on the outside of the frame, at the top of each wedge, so the latter may be set to these lines when the wheels are placed under the engine.

Tramming Wheel Centers.

In some shops it is also considered good practice, after the wheels are placed under the engine, and the shoes and wedges set in place, to tram the wheel centers, before the side rods are put up.

When this is required, go to each wheel and hammer lead into the center holes made by the lathe; then, locate the exact center of the wheel with a pair of dividers, using, for this purpose, the small circle cut in the hub of the wheel by the lathe, and prick-punch the center. If there is no circle use ball end calipers in the hole, and make a new circle for the purpose.

Now, with the long tram set to the length of the side rod, place one point of the tram in a wheel center, and try the opposite end with the centers in the other drivers.

Automatic Adjustable Driving Box Wedge.

This device consists really of two wedges, a floating wedge and an adjustable wedge, held in proper relative positions by a coil spring, as the illustration shows, Fig. 346.

The adjustable wedge is tapered on one side to suit the pedestal jaw taper, and on the other side to accommodate the somewhat lesser taper of the face of the floating wedge. To the adjusting wedge is attached the wedge bolt, which passes down through the pedestal binder and the spring bracket. This bracket is attached, by bolts, to the binder, and serves to hold the spring on the wedge bolt in place. An adjustable spring cap is mounted on the wedge bolt, with the spring between this cap

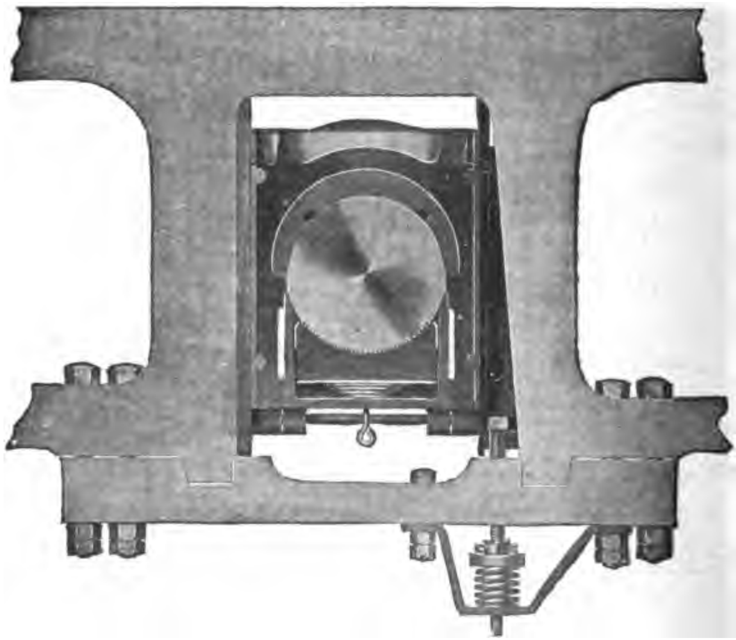


FIG. 346.

and the bracket. The spring, then, serves to hold the adjusting wedge in position, and automatically maintains proper adjustment of the driving box.

Between the double tapered adjusting wedge, and the driving box, is the floating wedge. As already stated, this wedge is also tapered on one face, to suit the adjusting wedge, and, too, it is approximately ¼" shorter than the distance between the binder and the frame rail.

Now, when the box moves up or down, the floating plate is carried with it until it strikes the frame rail or binder. On the upward movement this gives relief between the two wedges, due to their opposite taper. On the downward movement, the two wedges will move together due to the friction of the tapered faces in contact, and the taper on the back of the adjusting wedge will give relief. Thus is prevented the sticking of driving boxes.

The device needs little care, other than ordinary shoe and wedge lubrication, and the setting up of the spring nut when sufficient wear has developed to allow a travel of ¾" of the wedge.

The appliance is handled by the Franklin Railway Supply Co., of New York.

Pedestal Milling Machine.

The portable tool illustrated in Fig. 347, is designed to resurface the pedestal jaws of locomotive frames.

It has a 42 inch bed, flush at back, and has two heavy angle brackets, with extensive adjustment, and will work in a frame from 13 inches to 23 inches width of opening and jaws up to the width of cutter used, when only one cut is taken, or for wider jaws if bed is shifted on angles for two or more cuts.

The machine is driven by powerful enclosed worm gears and has variable power feeds in either direction, as well as cranks for hand feed.

As shown by the illustration, the pully shaft revolves in a swiveling frame, making a universal drive possible for any position of the machine.

The end of the shaft can be fitted with Morse taper for air motor, or other drive if so desired.

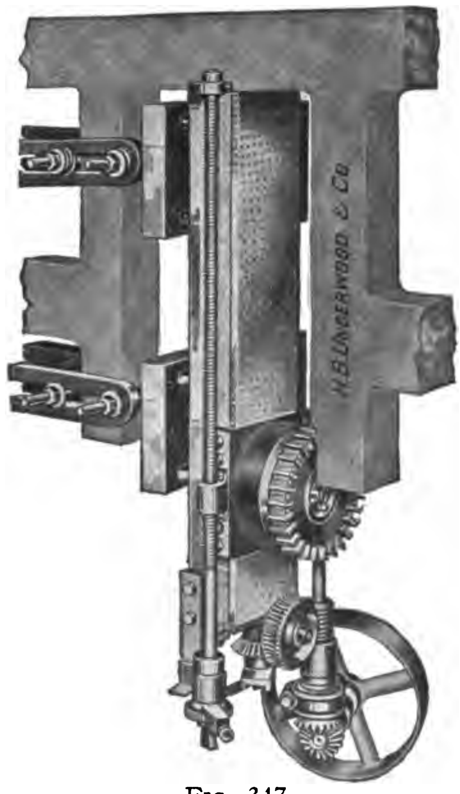


FIG. 347.

This tool is marketed by H. B. Underwood & Company, of Philadelphia, Pa.

GENERAL SHOP WORK.

Counterbalancing.

On the modern heavy locomotive, operating as it does at comparatively high speeds, the correct adjustment of counterbalance weights to the drivers has become a most important necessity. This counterbalance is required because of the effects of the reciprocating parts of the locomotive upon its operative ability. A poorly balanced locomotive will jerk and pound, and this condition is very injurious to the mechanical details, and to the general construction of the engine.

The dynamic augment—more commonly known as the "hammer blow," of the wheel is caused entirely by the action of the counter-weight, when it can do nothing to balance the reciprocating parts of the wheel. This hammer blow is, to a great extent, eliminated by correct balance, but an incorrect balancing of the wheel will make it quite noticeable, and in very poorly balanced wheels, is very severe on the road bed and track. In many cases it has been the cause of a very decided bend, or indentation of the rail at regular intervals, equal to the circumference of the wheel, because of the fact that the error in balancing has, in effect, made the locomotive much heavier at these points than otherwise.

Of course, it is impossible to have the drivers balanced perfectly at all times. For instance, if the balance be perfect under steam, it will be in error when drifting, that is, when steam is shut off. The balancing effect is also influenced by the speed of the locomotive, by the steam pressure on the head of the piston, and even by the point of cut-off, or by the length of the stroke.

Thus it is readily understood that the main pair of drivers will require a heavier counterbalance weight than the other drivers, due to the fact that there are more unbalanced parts working to throw it out of balance than influence the remaining drivers. It will be seen, therefore, by observation, that the main driver always carries a heavier balance weight than do the others.

The weight of the counterbalance necessary on any wheel is materially reduced by the use of lighter alloy metals for the reciprocating parts, for these parts, on a modern heavy locomotive, approximate a total weight of one ton which must be balanced.

It is common practice to place the center of gravity of the balance weight exactly opposite the crank pin center, and the same distance from the center of the wheel as is the center of the crank pin. This center of gravity, or center of weight, of the counterbalance is found by the use of a templet cut exactly to the outline of the counterbalance. A cardboard templet will suffice. The templet method of finding its center of gravity is to balance it, longitudinally and laterally, on a knife edge, marking the point where the two center lines cross. This point is its center of gravity.

Cross-counterbalancing, that is, placing the counter weights at some point varying from the point exactly opposite the crank pin, is often done, to correct the disturbances caused by the fact that the various parts revolve in different planes. This practice, when correctly carried out, is a good one, but it is commonly believed that it is unnecessary to cross-counterbalance in the case of outside cylinders, because of the fact that these disturbing forces are slight when compared to the principal reciprocating and centrifugal forces.

There are many methods of calculating the weight of the counterbalance to be used on each wheel, and a great number of rules governing the problem have been formulated, but we believe that the following method of procedure will be found completely satisfactory, and it is applicable to any locomotive.

Consider first the reciprocating parts, the piston head, rod and nut; crosshead, key, pin and nuts; and the forward (small) end of the connecting rod. These should all be weighed, and the total will be the weight of the reciprocating parts to be balanced.

The small end of the connecting rod may be weighed in the following manner: Place the wrist pin hole over a knife edge support, and place the crank pin hole likewise, except that its support is resting on the weighing machine. Now, if the rod be approximately level, the weight recorded by the scales is that of the small end of the rod.

When this has been done, the side rods—the revolving parts—should be weighed in the above manner, so that each wheel may receive its proportional weight of the revolving parts.

Now, to obtain the weight of the counterbalance to be placed opposite the radius of any one wheel, proceed as follows:

Divide the total weight of the reciprocating parts on one side by the number of drivers on that side. Add to two-thirds of this amount, the proportional weight of the revolving parts (the side rods) found for that wheel. In the case of main drivers, add also the weight of the large end of the connecting rod, ascertained as previously explained. This will be the weight to be balanced, or the weight of the counterbalance to be given the wheel under consideration.

As a check, place the drivers, by pairs, on trestles, with their journals resting on smooth, flat, metal surfaces. Suspend from the crank pin of one wheel, weight sufficient to cause the wheel, after being rotated, to return to such a position that that portion of the wheel from which the weight is suspended will remain horizontal. Then, if the suspended weight equals that calculated for the wheel in question, that wheel is correctly balanced. If not, the balance weight should be adjusted by increasing or decreasing its weight accordingly.

Quartering.

The object of quartering is to set a pair of driving wheels on the axle so that the crank pin centers are exactly at right angles to each other (the right crank pin should lead the left by precisely 90 degrees). There are several methods of doing this, but the following one will give very good results, and is readily understood and accomplished.

Place the axle on horses, or blocks, or on a frame built for this purpose. Then, with heavy wooden blocks, drive the wheels on the axle as far as possible, and so that the right crank pin will lead the left crank pin by approximately 90 degrees (as nearly as the eye can determine).

Locate a center mark in the center of the left wheel, as shown at X (Figs. 348 and 349 show the two wheels in their relative positions on the axle, 348 the left and 349 the right wheel). Set an offset straightedge to this center, as the broken lines show, allowing it to rest on the top of the crank pin collar, and scribe a line of the face of the wheel, at D. Then set the straightedge true with the wheel center, allowing it to bear against the *bottom* of the crank pin collar. Scribe another mark on the wheel face, at A. With a pair of dividers, bisect the distance between these lines, and the line located, C, will lie in line with the centers of the wheel and the crank pin. Clamp this straightedge (or any straightedge) true with the point C and the center of the wheel X, and move the wheel until a spirit level placed on the straightedge shows it to be perfectly horizontal.

Now, on the right wheel, the same method may be used, moving the wheel, however, until the straightedge is absolutely perpendicular. Or, as shown in Fig. 349, scribe on the axle,

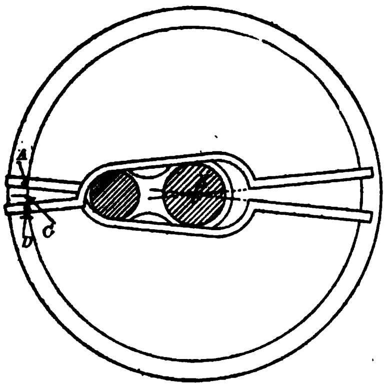


FIG. 348.

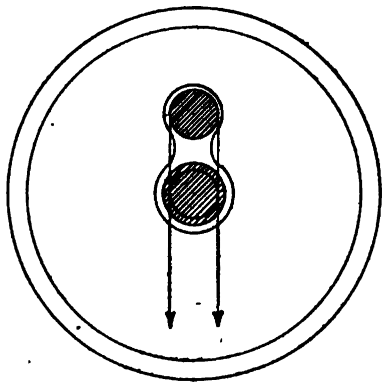


FIG. 349.

with the wheel center as center, a circle equal in diameter to the crank pin collar. About the collar drop a plumb line, weighted at each end, and adjust the wheel until both sides of the plumb line are true with the circle on the axle representing the crank pin collar, when the crank pin on the right wheel will be in an accurately determined perpendicular position. The crank pin centers are now just a quarter turn apart, and the right, Fig. 349, is leading the left, Fig. 348, so scribe the keyways on the axle. This method is particularly applicable when fitting old wheels, with crank pins intact, to new axles. If both wheels and axles are new, locate centers in both axles and both crank pins, clamp an ordinary straightedge to each wheel, true with these center marks, and adjust the wheels until one straightedge is horizontal, and the other perpendicular, then scribe the keyways in both wheels. For, in this case, the axle is first quartered in the lathe, and the keyways planed, or milled out.

Of course, this method will suffice for an old axle and new wheels, scribing the keyways in the new wheels when adjusted as explained.

Or, if the wheels are old, but the crank pins are not intact, and the axle new, do not quarter the axle, but proceed as in the method just explained, and when the wheels are properly set, scribe keyways on the new axle to correspond to those on the old wheels.

Crank Pins.

The limit of wear allowable for crank pins, before renewal, varies greatly, depending upon the regulations of the different roads, and also upon the class of service.

The writer suggests that the pins be renewed if worn below their original size by 3/16", for this limit is regarded as good practice generally.

Should the side rod bushing become loose, while running, the rod would probably interfere with other parts of the engine, unless the crank pin collar had an outside diameter about 5/16" or 3/8" larger than the bushing on the rod.

Wheels and Axles.

FITTING DRIVING BRASSES INTO BOXES.

In fitting new brasses, it is better to place the brasses in the boxes, and bore them, before planing the faces of the shoe and wedge. To do this, the outside of the brass should be turned carefully, to fit the seat in the box. If the planing is done first, then, when the brasses are applied under pressure, causing the boxes to expand at the bottom, and spread quite noticeably, as is usually the case, the shoe and wedge faces of the boxes will be thrown out of parallel, resulting in a very bad pound, which can only be removed by dropping the wheel and refitting the cellar, or planing the box. While the brass is in the box, it should be bored to fit the journal, which has a tendency to cause the boxes to spring inward slightly, due to the removal of the metal in boring. Then, and not before, the planing of the shoe and wedge faces should be done, for the boxes have assumed their final outline, and are in a position to set up properly. The range of pressure required to fit the brass into the box sufficiently tight is from ten to thirty-five tons, depending upon the fit of the brass to its seat in the box, and upon the size of the box.

In regard to boring the brasses, they should be bored a *very little* larger than the journal—only enough to prevent sticking, or seizure, under normal conditions of running. A full half circle bearing should be given the brass, and its sides should extend at least ¼" below the box center line.

As to the boxes, they are reduced three-quarters of an inch below standard size, and brass liners ⅜" thick applied to the shoe and wedge faces of the box, to eliminate the use, and the resultant upkeep, of solid brass shoes and wedges.

LATERAL MOTION BETWEEN HUBS.

When lateral motion between the driving boxes and the hubs of driving wheels becomes excessive, this condition is temporarily remedied by the use of brass plates applied between the box and the hub. These plates are dove-tailed, and are held together by a key. Or, often, they are made with a groove around the outside edge, into which fits a round iron band.

The maximum lateral motion, on any pair of drivers, should be 3/4", or 1" on truck wheels, but it is good practice to allow much less than this amount.

TO LOCATE WHICH PAIR OF WHEELS IS OUT OF TRAM.

If you tram an engine and find wheel centers "out of tram," you should at once locate which pair is out. You may ascertain this by using inside calipers between frame and largest turned face of wheel inside, at front and back of wheel. Each pair of wheels should be square with the frame.

FITTING ROD BRASSES.

File out the top and bottom, and obtain a good side bearing, but do not allow the brasses to bear on the fillet. Key up brass to brass in the strap, and whirl on the pin, but take up all lost motion, to prevent pounding.

File the front end of the main rod brass so as to leave it open about 1/32 of an inch, when fitted on the wrist pin. This allows for driving the key as the brasses wear. On all others key up brass to brass, so that the fit is snug, but not so much so that the rod can not be moved on the pin.

It must be remembered that the tendency of the main rod to increase in length (lengthwise between brass centers) is very great, since the rod length is increased somewhat each time the key is driven for the purpose of tightening the brass.

In regard to side motion, there should not be more than 1/4 of an inch motion of the rod on the pin. This is the maximum —good practice recommends considerable less in this respect.

ECCENTRIC BRASSES.

Some roads use solid brasses in the side rods, which are turned eccentric shape and are capable of adjustment by simply moving a small lever which is secured to a quadrant, which is fastened to the rod. The short levers are adjusted while the crank pins are on center, thereby adjusting the length of the rods to suit the wheel centers on each side.

Lining Guides.

In aligning crosshead guides, it is good practice to fasten the back cylinder head securely in place on the cylinder, with guide blocks plumb with the cylinder center. Then, when this cylinder head is properly applied, see that the guide blocks are firmly and centrally secured to the guide yoke in the rear.

Now a center line should be run through the cylinder, extending to the rear guide yoke. This line is set central with reference to the front end cylinder-bore, and with the stuffing box at the rear end, and is then carried straight back, and held taut behind the guide yoke by suspending a weight on it at the rear.

The guides may now be adjusted to set parallel to this line, and, therefore, to each other. Set the guides a distance from this line equal to the distance from the center of the piston rod hole in the crosshead to its bearing surface on the guides. It is, therefore, necessary to have first centered the piston rod hole of the crosshead. This change in alignment is accomplished by inserting liners between the guides and the guide blocks to which they are bolted.

Now, if there be both top and bottom guides, set the lower guide correctly, as explained, and then set the top guide parallel to it, and a distance from it sufficient to allow the crosshead to move freely, but without an unnecessary amount of play, or lost motion.

However, by this method of hanging the guides, or in hanging them by means of a bar centered through the cylinder bore, it is probable that they will be hung too high. For, with cylinder 23" or more in diameter, on superheated engines, bull rings or solid piston heads are turned at least 1/16" small, so that, with guides hung central, they are at least 1/32" high. Another 1/32" is added to this on the first few trips, by wear on the bottom of the piston heads, so that, on an engine just out of the shop, practically, the guides will be 1/16" too high.

It would be much better to hang the guides 1/16" below the center of the cylinder bore, so as to do away with this error.

On passenger or freight locomotives, which usually run forward, the greatest thrust of the crosshead is on the upper guides. For, when the main crank pin is moving forward through the upper half of its travel circle, pulled by the piston, the connecting rod and piston rod tend to make a straight line, thus forcing the crosshead upward as far as it is permitted by the guides to move. Also, when the piston is moving backward, pushing the crank pin through the lower half of its circle, the angle between main rod and piston rod gives the crosshead an upward thrust.

It is on this account that it is considered good practice to give the upper guide a more broad wearing, or bearing surface, than the lower.

HOW TO LINE UP FOUR BAR GUIDES.

Upend the crosshead upon your bench, and fit a piece of flat copper into the hole for the piston rod, and, with a pair of hermaphrodites, locate the center of the hole, making a very small prick-punch mark in the exact center. Now place two parallel strips or straight-edges against the two wearing surfaces on the bottom side of the crosshead, letting the end of each extend above the end of the crosshead, and clamp them in this position, or have some one hold them while you set the guide gauge, letting the gauge touch both straight-edges, and setting the needle to the center of the hole for the piston rod.

Next bolt up both bottom guides in their respective places, then line the cylinder perfectly true. Now line both guides until the needle of the guide gauge will just touch the cylinder line at both front and back ends. Keep both bottom guides perfectly true across their faces, also lengthwise, and perfectly level with the top of the cylinder, and an equal distance from the cylinder line, and allow 1/32" for lateral motion. Now place the crosshead on the two guides and move it to and fro, and see if there is any rock motion in the crosshead, and notice the points where it bears on the guides. Line the bars to take out all twist and see that the crosshead has a good bearing, then put the inside top guide in place and line it down to the crosshead, keeping it true with the bottom inside guide by using an adjustable square or straight-edge. It should be lined down as close as possible without binding the crosshead, which should move freely by hand. Now put on the outside top guide and line it the same way; remember that you can spring any of the guides up or down in the center (if they are not too thick) by using narrow liners either in front or in back of the bolt. When the guides are properly lined remove one bolt at a time, using clamps in their stead, and rose-bit each hole; fit the bolts and tighten them, then trim off the liners and make a neat job of it.

LINING TWO BAR GUIDES.

Mogul.

First put a center in the crosshead, the same as for four bar guides, then clamp both gibs to the crosshead without any liners, and measure the exact distance from the face of the bottom gib to the center, also try the center and see if it is central sideways.

Now place both guides in their respective places and bolt them securely, then line the cylinder, and with a guide gauge, or straight-edge and scale, try the two guides and see if they are central, and parallel with the cylinder line; line the bottom guide the correct distance from the cylinder center line at both ends, now caliper the crosshead outside of both gibs, and line the top guide close enough to let the crosshead and gibs between them without any liners under the gibs; see that each guide is perfectly square across its face, and level with the cylinder and the sides perpendicular, then put up the crosshead and slip both gibs into place. Put on the outside plate, or plates, and see that the crosshead will move freely, then rose-bit the holes for new guide bolts. If, on account of the thickness of the guide blocks, it becomes necessary to place liners under the top gib, see that the oil holes are cut in the liners before putting them into place.

If the engine is on blocks and there is no support between the fire box and the front end of the engine, it is considered good practice to set the back end of each guide 1/32" low at the back end; when the weight is on the wheels the guides will be central.

When lining old crosshead gibs, place the crosshead at the back end of the guides, and caliper between the bottom guide and the piston; if the piston and guide are parallel place the liners under the top gib; if the piston is low at the back end, place enough liners under the crosshead to make the piston parallel with the bottom guide.

Both Guides Above the Cylinder Center.

Locate a center in the crosshead as previously explained; find the exact distance from the wearing surface of the top gib to the center of the crosshead, then bolt the top guide in its place and line the cylinder; set the top guide the correct distance from the line at both ends, using a twelve-inch scale with an adjustable square head; see that the guide is central with the cylinder line, and that its face is level with the cylinder, then remove the cylinder line and put up the bottom guide; caliper the gib and set the bottom guide that distance from the top guide, keep the bottom guide level across its face and true with the top guide; now slip the solid crosshead gib into place, line close and see that the gib will move freely; then put up the crosshead and you have finished.

LINING ONE BAR DIAMOND GUIDE.

Fasten the guide in its place, put up the crosshead and line the cap closely, so that the crosshead will move freely upon the guide; now line the cylinder, letting the line extend through the hole in the crosshead, then, with a pair of inside calipers, see if the cylinder is central in the hole in all positions; if not, line the guide until it is.

LAYING OFF NEW GUIDE BLOCKS.

Four Bar Guides.

Locate a center in the crosshead, and find the distance this center is above or below the wearing surface on the bottom of the crosshead, tighten all four guide blocks in their proper places, keeping each perfectly level across its top face, then line the cylinder perfectly; now clamp a short straightedge across the face of both front guide blocks, keeping it level with the cylinder and the same distance from the cylinder line as the crosshead center is from the bottom wearing faces of the crosshead, and clamp another straightedge to the faces of the two back guide blocks, setting it the same way; use a spirit level and see that the two straightedges are perfectly true with each other, and level with the top of the cylinder, and each an equal and correct distance from the cylinder line, then scribe a line across all four guide blocks; next caliper the wings of the crosshead; they should both be the same thickness; so lay off another line across the face of each guide block above the first line and parallel with it, and a distance from it equal to the thickness of the wings of the crosshead; work close and have the machine man split your lines, and when the guides are hung it will not be necessary to use a single liner.

Two Bar Guides.

These guide blocks should be laid off in exactly the same manner as for four bar guides, but they can be laid off in many different ways, such as measuring from center of stuffing box to center of guide block hole, etc., but the method we have explained will be found to give good results.

Rod Work.

TO DETERMINE THE LENGTH OF A MAIN ROD.

If the guides are not yet hung, and the engine is still on blocks, proceed as follows: If the main jaw is square with the top of the frame, place a long straightedge across both main jaws, and measure the distance to the face of the back cylinder head. Now add to this length the thickness of the main shoe and one-half the thickness of the driving box, and from the total length, subtract the following: Distance from the center of the crosshead pin to the front face of the crosshead; one-half of the stroke of the crosshead; the clearance, and the length of the front guide block.

If the guides are hung, and the main driving boxes are properly set in the jaws, or if the main wheels are under the engine and wedges set up, proceed as follows: Set the crosshead exactly central in the guides, and use a square from the center of the main driving box, or wheel center; now use a long stick, letting the forward end of it touch the crosshead pin, and mark a line on the back end of the stick true with the wheel center. Add to this length one-half the diameter of the crosshead pin, and you will have the correct length of the main rod.

DIVIDING THE CROSSHEAD CLEARANCE.

On a new locomotive, the points are marked on the guides where the piston touches the cylinder head at each end. This is from ¼ to ½ of an inch more than the piston stroke. Adjust the rod so that this clearance is evenly divided at each end of the stroke. If there be any variation, let it be less at the back end, as the lengthening of the main rod, by wear, will tend to equalize the clearance.

When adjusting a main rod to the proper length, you should always notice whether the key in the back end is in front of or behind the crank pin. When in front you lengthen the rod as the brass wears and you drive the key down, and when behind you shorten it. Divide the clearance accordingly.

Another mechanical point to be considered is the space occupied by the area of the piston rod; when divided centrally the back end should have 1/32" more clearance than the front end, to equalize the exhaust.

In the roundhouse it is sometimes necessary to disconnect a main rod and put it up again. When the engine cannot be pinched onto center, scribe a line on the guides at either end of the crosshead, before you disconnect, and see that the crosshead is true with same line after you have connected.

FITTING PISTON ROD TO CROSSHEAD.

There are two well-known methods of doing this, the one employing the key being the more satisfactory as to results, and it also weighs less than the one using nuts. A very secure fastening is necessary, especially since the use of the piston valve has necessitated the valve rod being fast to the crosshead, and snugly fit. Some designs make the end of the rod butt against the shoulder on the crosshead, in addition to the taper fit, but the rod is very likely to be loose in the taper, and bear altogether against the shoulder, or the taper may be too tight before reaching the shoulder. Better results are obtained by omitting the shoulder mentioned, and by relying upon the key and the taper fit.

TO DETERMINE THE LENGTH OF PARALLEL RODS.

If the wheels are under the engine, obtain the length from the wheel centers. If not, take the length from centers of jaws (if boxes are lined central), and in either case, if engine is cold when length is taken, make the back rods 1/32" too long, to allow for expansion; providing the firebox sets down between the two back pairs of wheels.

TRAMMING CRANK PINS AND PUTTING UP RODS.

Plug up the wheel centers with lead, find the exact center of each wheel, and try them with a tram. If they show out of tram, then try main centers with centers in rocker-arms, and line shoes, and wedges wherever needed, to bring into tram, and also to keep square with rocker boxes (which are supposed to be correct); also see that the wedges are set up moderately tight before tramming main centers, and notice if any of the tires are beginning to cut; if so, that wheel should be lined forward, or the opposite wheel back.

Now place the pins on top and bottom eighth, and jump the wheels until they tram. If the engine has more than two pairs of driving wheels, and the main pin is longer than the others, set tram accordingly. Now, when pins tram, pinch one side of the engine on dead center, and put up rods on that side. Then pinch the other side on dead center, and put up that side. Line snugly between pins.

When up, the front rods should work freely on the pins, but if the boiler is cold, line the back rod tightly between the pins to allow for expansion of the boiler. Each brass should have at least 1/64" lateral. See that rods divide correctly sideways.

MACHINE SHOP PRACTICE.

GEARING THE LATHE.

Simple Geared.

Multiply the number of threads (per inch) you desire to cut by any small number, and put that gear on your screw; then multiply the number of threads (per inch) on your screw by the same number, and place that gear on the spindle. For example: We will use the number 4, although any small number will answer the purpose; to cut 10 threads, multiply 10 by 4, which equals 40; put 40 on your screw. Now multiply the number of threads (per inch) on your screw (we will assume it to be 6) by 4, which equals 24; put 24 on the spindle.

Another way is to take any small gear you may have (put it on the spindle) and multiply it by the number of threads desired, divide the product by the number of threads on your screw (per inch), and put it on the screw. If you haven't both of these gears, try another combination until you have two that will satisfy the requirements.

Many small lathes have a stud geared into the spindle, which stud runs only one-half as fast as the spindle. In finding the gears for these lathes, first multiply the number of threads to be cut, as before, and then multiply the number of threads on the screw as double the number it is. For example: If you want to cut 10 threads, multiply by 4; put 40 on the screw; then, if your screw is 6, call it 12, multiply by 4, and it will give you 48; put that number of teeth gear on the spindle. Many of these lathes have a pin in the end of the spindle, which changes the gear without the necessity of changing gears; simply pull out the pin and the lathe will cut double the thread it is geared to cut.

The following rules for selecting gears to cut threads on lathes, for both single and compound gearing, are satisfactory: if the lathe is simple geared and the stud runs at the same speed as the spindle, select some gear for the screw, multiply its number of threads per inch by the lead screw, and divide this result by the number of threads per inch you wish to cut. This will give you the number of teeth in the gear for the stud. If this result is a fractional number, or a number which is not among the gears which you possess, try some other gear for the screw. But, if you prefer to select the gear for the stud first, multiply its fixed compound gears. In the instance given, if the lead screw had been 2½ threads per inch, its pitch being 4/10 inch, we have the fractions of 4/10 and 25/32, which, reduced to a common denominator, are 64/160 and 125/160, and the gears will be the same as if the lead screw had 125 threads per inch, and the screw to be cut 64 threads per inch.

Compound.

If the lathe is compound, select at random all the driving gears, multiply the number of their teeth together, and this product by the number of threads you wish to cut. Then select at random all the driven gears except one, multiply the number of their teeth together, and this product by the number of threads on the lead screw. Now divide the first result by the second, and you will have the number of teeth in the remaining driven gear. But if you prefer you can select at random all the driven gears, multiply the number of their teeth together, and this product by the number of threads per inch in the lead screw. Then select at random all the driving gears except one. Multiply the number of their teeth together, and this result by the number of threads per inch of the screw you wish to cut. Divide the first result by the last, and you will have the number of teeth in the remaining driver. When the gears on the compounding stud are fast together, and cannot be changed, the driven one has usually twice as many teeth as the other, or driver; in which case you can, in the calculations, consider the lead screw as having twice as many threads per inch as it actually has, and ignore the compounding entirely. Some lathes are so constructed that the stud on which the first driver is placed revolves only half as fast as the spindle. You can ignore this in the calculations by doubling the number of threads of the lead screw. If both the last conditions are present you can ignore them in the calculations by multiplying the number of threads per inch in the lead screw by four. If the thread to be cut is a fractional one, or if the pitch of the lead screw is fractional, or if both are fractional, reduce the fractions to a common denominator, and use the numerators of these fractions as if they equaled the pitch of the screw to be cut, and of the lead screw respectively. Then use that part of the rule given above which applies to the lathe in question. For instance, suppose it is desired to cut a thread 25/32 inch pitch, and the lead screw has 4 threads per inch. Then the pitch of the lead screw will be ¼ inch, which is equal to 8/32 inch. We now have two fractions, 25/32 and 8/32, the two screws will be in the ratio of 25 to 8, and the gears can be figured by the rule above, assuming the number of threads to be cut to be 8 per inch, and those on the lead screw to be 25 per inch. But this latter number may be further modified by conditions named above, such as a reduced speed of the stud, or fixed compound gears. In the instance given, if the lead screw had been of 2½ threads per inch, its pitch being 4/10 inch, we would have the fractions of

4/10 and 25/32, which, reduced to a common denominator, are 64/160 and 125/160, and the gears would be the same as if the lead screw had 125 threads per inch, and the screw to be cut 64 threads per inch.

THREAD CUTTING.

There are three forms of threads now in extensive use in this country. They are, namely, the V-thread, the Whitworth, and the U. S. Standard (sometimes called the Sellers, or the Franklin, as it was designed by William Sellers and recommended by the Franklin Institute of Philadelphia, Pa.).

FIG. 350.

The V-thread and the U. S. Standard have both an angle of 60°. However, while the threads of the V-type are sharp, both at top and root, the U. S. Standard is flattened at both top and root. That is, one-eighth of the height of the thread is cut off the top and one-eighth filled in at the root, as the illustration (Fig. 350) makes clear.

The Whitworth has an angle of 55°, and the edges (both point and root of thread) are rounded. This is shown clearly in Fig. 350. The Whitworth is the English standard.

In cutting threads, always grind the tool to fit the gauge for whatever thread is to be cut.

Screw of V-shaped Threads.

When cutting V-shaped threads, set your tool at right angles with lathe centers, and look at thread carefully on both sides, and see that the threads do not lean like fish scales.

Square Threads.

When cutting square threads it is always necessary to get the depth required with a tool somewhat thinner than one-half the pitch of the thread. After doing this, dress another tool exactly one-half the pitch of the thread and use it to finish with, cutting a slight clip on each side of the groove. Then polish with a pine stick and emery. Square threads for strength should be cut one-half the depth of their pitch, while square threads for wear should be cut three-fourths the depth of their pitch.

Mongrel Threads.

Mongrel, or half V and half square threads, are usually made for great wear, and should be cut the full depth of their pitch, or even more. They are sometimes cut one and one-half the depth of their pitch; the point and the bottom of the grooves should each be in width one-quarter the depth of their pitch.

Double Threads.

The face plate on every lathe has at least four slots in it which carry the dog; so after you have cut one thread change your dog to slot directly opposite on face plate.

Left-hand Threads.

Gear up the lathe the same as for a right-hand thread, and then change the shifter on the end of the lathe. If the lathe has no shifter, put in an extra gear, so as to move the carriage from left to right. Then begin to cut the thread at the left hand, and cut to the right. The same method is used for inside or outside threads.

For convenience, we include the following tables, giving the diameter in inches and number of threads per inch of the various sizes of the three threads previously described.

SHARP "V" THREAD.

Diameter Inches	No. of Threads per Inch	Diameter Inches	Number of Threads per Inch	Diameter Inches	Number of Threads per Inch	Diameter Inches	Number of Threads per Inch
1/4	20	7/8	9	2	4 1/2	3 1/4	3 1/2
5/16	18	15/16	9	2 1/8	4 1/2	3 3/8	3 1/4
3/8	16	1	8	2 1/4	4 1/2	3 1/2	3 1/4
7/16	14	1 1/8	7	2 3/8	4 1/2	3 5/8	3 1/4
1/2	12	1 1/4	7	2 1/2	4	3 3/4	3
9/16	12	1 3/8	6	2 5/8	4	3 7/8	3
5/8	11	1 1/2	6	2 3/4	4	4	3
11/16	11	1 5/8	5	2 7/8	4		
3/4	10	1 3/4	5	3	3 1/2		
13/16	10	1 7/8	4 1/2	3 1/8	3 1/2		

UNITED STATES STANDARD THREAD.

Diameter Inches	No. of Threads per Inch	Diameter Inches	Number of Threads per Inch	Diameter Inches	Number of Threads per Inch	Diameter Inches	Number of Threads per Inch
1/4	20	1	8	2 1/8	4 1/2	3 1/4	3 1/2
5/16	18	1 1/8	7	2 1/4	4 1/2	3 3/8	3 1/4
3/8	16	1 1/4	7	2 3/8	4	3 1/2	3 1/4
7/16	14	1 3/8	6	2 1/2	4	3 5/8	3 1/4
1/2	13	1 1/2	6	2 5/8	4	3 3/4	3
9/16	12	1 5/8	5 1/2	2 3/4	4	3 7/8	3
5/8	11	1 3/4	5	2 7/8	3 1/2	4	3
3/4	10	1 7/8	5	3	3 1/2		
7/8	9	2	4 1/2	3 1/8	3 1/2		

WHITWORTH STANDARD THREAD.

Diameter Inches	No. of Threads per Inch	Diameter Inches	Number of Threads per Inch	Diameter Inches	Number of Threads per Inch	Diameter Inches	Number of Threads per Inch
1/4	20	7/8	9	2	4 1/2	3 1/4	3 1/4
5/16	18	15/16	9	2 1/8	4 1/2	3 3/8	3 1/4
3/8	16	1	8	2 1/4	4	3 1/2	3 1/4
7/16	14	1 1/8	7	2 3/8	4	3 5/8	3 1/4
1/2	12	1 1/4	7	2 1/2	4	3 3/4	3
9/16	12	1 3/8	6	2 5/8	4	3 7/8	3
5/8	11	1 1/2	6	2 3/4	3 1/2	4	3
11/16	11	1 5/8	5	2 7/8	3 1/2		
3/4	10	1 3/4	5	3	3 1/2		
13/16	10	1 7/8	4 1/2	3 1/8	3 1/2		

Turning and Fitting.

HOW TO TURN AND FIT DRIVING AXLES.

Put centers in each end, place on lathe, and drill 1" deep hole in the center in each end. Ascertain the length over all and cut off, and face each end, cutting a small groove on each end. Turn down to the largest finished part, space off the journal bearing fits and eccentric fits, and turn these surfaces to required size. Then cut the space between the eccentric fits down to about 1/16" smaller diameter than the size of the axle at the eccentric fit, and fit the same as crank pins. (See page 759.) Let the calipers drag sufficiently to obtain a 70 to 100 ton fit.

HOW TO QUARTER A DRIVING AXLE IN A LATHE.

Scribe a large circle on the lathe face plate, and prick punch each quarter on this circle, having divided it into four equal parts.

Use the post or a solid tram from the bed of the lathe, and set the axle to one of the four punch marks. Use a sharp-pointed tool in the tool post, run the carriage along, and scribe a fine line, the full length of the wheel fit, on the right end of the axle. Turn the lathe one-quarter turn ahead (by setting to next punch mark) and scribe a similar line on the left end of the axle.

Having these two lines as the center lines of the key-ways, lay off the keyways to the required size.

HOW TO TURN UP AND FIT CRANK PINS.

First see that the pin will true up to required size; put a large center in each end of pin and drill a small ¼" hole about ½" deep in each center; this is done to retain the original centers. Now face off outside end of the pin perfectly square, and cut a small groove around the center so as to find the center later.

Turn collar to the required size. Find the distance the pin should stand out and cut the shoulder, then finish the bearings for the brasses in the proper places and of the required diameter; reverse pin in lathe and fit to hole. To do this (if your calipers are of medium size, say 6", by 1/18" thick), set inside calipers to exact size of hole, set outside pair to them and let outside pair drag 5/16" over fit. This will give you about a 30-ton fit. Do not cut off inside center, but counterbore about ¼" deep and leave a collar to rivet inside.

To find the diameter of a crank pin multiply the diameter of the cylinder by .234.

HOW TO TURN UP AND FIT PISTON RODS.

New Rod.—Put two deep centers in the rod and place it on the lathe, using a square center to get each end to run true. Now replace the square center with a round center, use a diamond point tool, and turn rod off to about 1/16" larger than finished size. Measure the distance the rod should go into the crosshead, and cut a shoulder that distance from the end of the rod. Now caliper taper hole in crosshead at each end, and fit, same as to taper bolts (see page 767); allow 1/16" to grind and drive to shoulder. Now turn rod down to finished size, using very fine feed. Take out tool marks with a lathe file, and polish with emery cloth, or emery paper.

Now reverse in lathe, cut the other shoulder the exact distance from the end of the rod, and fit to piston head; let calipers drag ¼" for a 20-ton pressure, or more in proportion, then counterbore the end to rivet over. The diameter of a piston rod should be about 1/6 the diameter of the cylinder.

TURNING VALVE STEMS.

Drill a small hole in the end of the stem, place on the lathe, using a bolt and nut, or screw jack in yoke, to keep from springing. Cut off the end to the correct length. Now cut shoulder and fit to valve rod (same as fitting taper bolts), then turn down to required size.

TURNING AND FITTING SOLID ROD BRASSES.

On the larger class of locomotives, solid rod brasses are used almost to the exclusion of strap brasses. To fit these, chuck the collar, and bore out 1/32" larger than the pin, then turn the outside 1/32" larger than the hole. Reverse and turn off the collar. If pressed into the rods, they should be bored out in the boring mill, making a 1/64" loose fit.

TURN AND FIT PISTON HEAD, BULL RING, CYLINDER PACKING RINGS, AND FOLLOWER PLATE.

Use Universal Chuck.

Piston Head.—Chuck, bore out the hole for the piston and ream with large reamer, turn outside, or largest part, 1/32" smaller than cylinder, reverse in chuck, turn off lugs and true up both faces to required size, leaving a collar for follower plate.

Bull Ring.—Chuck, bore out about ⅛" larger than outside of the lugs on piston head, turn off 1/32" smaller than the cylinder, turn outside flanges 3" smaller than largest part, face off 1/64" wider than the distance from one face of the piston head to the other, just enough for the follower plate to clamp the bull ring without cracking the plate. If a solid bull ring, chuck and face off one side, reverse in chuck and face off to correct width; bore out 1/16" larger than the lugs on piston head and cut grooves of proper width, and 1/16" deeper than the size of the packing rings.

Packing Rings.—Caliper cylinder. Make outside packing rings 3/16" larger, if the two diameters of bull ring differ 3", then make packing rings about ¼" less; therefore bore out the packing casing 2½" less than the outside of same; use a square nose tool and cut off the rings to the correct width.

Follower Plate.—Chuck and face off true, turn off outside 1/32" smaller than the cylinder, and face off the holes for the follower bolts.

TO FIT UP BULL RING, PACKING RINGS, ETC.

See that the follower plate clamps the bull ring when tightened, put one dowel pin on each side of the bull ring, about 4" apart, and when you place in the cylinders set dowels at the bottom. Saw out the packing rings an amount three and one-seventh times the difference between the size of the cylinder and the outside of the packing rings, and see that they clear the dowel pins and slip through the smallest part of the cylinder. If no dowel pins are used, saw rings so that they will lap. Put in piston head, or spider, bull ring and packing rings, and line up piston head central with the cylinder, placing liners between bull rings and lugs on the piston head, then put on follower plate and tighten up follower bolts.

HOW TO FINISH EXHAUST NOZZLE.

Chuck in lathe, bore out and turn off to required size, making a close fit in the end of the nozzle box, with grooves for the set-screw.

TURNING AND BORING OUT TIRES.

First ascertain whether or not there are flat spots on the tire to be turned. Caliper each tire and turn the smallest first, so that all of the others may be brought to the same size. When boring out the tires, allow for shrinkage as follows: 1/80" (for 38" wheel) and 1/60" (for 90" wheel) for each foot diameter of the tire; the rate of shrinkage increasing uniformly between these limits. Turn the thread and flanges accurately to a gage.

TURNING AND FITTING CYLINDER BUSHINGS.

Cylinder bushings may be bored out and turned up in a very large lathe, although this job is usually done on a boring mill. However, to accomplish it on the lathe, clamp two large blocks onto the carriage of the lathe, and bore out the wood to the exact size of the outside of the bushing when rough. Then clamp the bushing onto these blocks—use a boring bar and bore out the bushing to the required size, including the counter-bore.

Set the bushing on a large mandrel for the purpose, which has set-screws at each end with which to adjust the bushing. Set to run perfectly true with the inside at each end, and turn off and fit to the cylinder. Make the bushing the exact size of the cylinder, leaving a good finished face at each end. When pressing the bushing into the cylinder, care must be taken not to crack the bushing.

Boring.

HOW TO BORE OUT CRANK PIN HOLES.

Practically all shops have quartering machines for this purpose. The two sides of the bed are set at perfect right angles, with a head and a boring bar on either side, so that it is impossible to bore the holes out of square. The heads are adjustable so that the stroke may be changed if desired.

When there is no quartering machine, the large face plate on the wheel lathe is usually quartered with four large holes drilled for dowel pins, and a small boring bar fastens onto the carriage of the lathe.

BORING OUT CYLINDERS.

New Cylinder in Boring Mill.

Set tool to inside of cylinders, and run tool through to see if there are any low or warped spots; then see if the inside face of the cylinder will plane off to the required size; if it is all right bore out to required size; then bore out counter-bore at each end about 3/8" deep. Get the length between the faces, and face off the ends. If the tool chatters, hang a weight on the end of the bar.

In Lathe.

Clamp two large blocks onto the carriage and bore them out to the same circle as outside diameter of rough cylinder; then clamp the cylinder onto the blocks and proceed to bore them out the same as in a boring mill. This is a very difficult job on a lathe, and one needs a very large lathe to accomplish it.

BORING ROD BRASSES.

When finished a strap rod brass should have a good bearing front and back, but should not bear top and bottom, therefore put a piece of Russian iron between the brasses before boring them out, then key up tight and bore out 1/16″ larger than crank pin. Take out Russian iron when bored. However, solid brasses are used almost entirely on the larger locomotives. They should have a bearing surface all the way around.

Planer Work.

HOW TO PLANE UP A NEW CYLINDER.

Set the inside face square with the bed of the planer. Put a center in each end of the cylinder, and set each center perfectly true with the tool. Plane the inside face to one-half the correct width from face to face of the cylinder. Now plane the top of the cylinder outside of the seat, and turn the cylinder around one-quarter of a turn. Set the inside face square with the bed of the planer, up and down, and set two centers in the cylinder perfectly square across the bed of the planer—use a square from the center to the square line on the bed. Now finish the valve seat and the outside of the cylinder. Turn the cylinder over and plane the bottom square with the inside face, the same distance from the center of the cylinder as the other cylinder, its mate.

When taking a cut down the inside face of a cylinder, or a side cut on any other piece of work, always adjust the circle head of the planer slightly to one side (whichever side you are planing on), so that the tool will not drag.

PLANING UP SHOES AND WEDGES.

New.

Clamp shoe, or wedge, on the bed of the planer, face down, and true up the flange edges. Turn work over and clamp flange down, keeping the sides true with the tool. Take a light cut across the face, just sufficient to true it up. Caliper the driving boxes inside, and finish both outsides, making the shoe or wedge 1/64" smaller than the box. Clamp all shoes down solidly in the chuck, and finish inside. Make each flange of the same thickness, and wide enough to slip over the widest part of the jaw—unless instructed otherwise. Now chuck all wedges and block up the top end of each, so as to set their inside faces true with the tool, and finish the flanges the same as those of the shoes.

Old or New when Laid Off.

Set the shoe or the wedge by the pop marks on the outside, using a surface gage or hermaphrodites, and clamp down solidly to the face of the planer. Line up so that the pop marks show exactly the same height all around. Plane off the face to the required size, leaving plenty of fillet.

HOW TO SET CALIPERS FOR FITTING.

Some men do all their fitting with inside calipers, but the best mechanical method is to set the inside calipers to the exact size of the hole, then set the outside calipers to the exact size of the insides, and make all fits with the outside calipers, letting them drag over the work while in the lathe. Experience will teach the amount of drag to allow for different pressures, suitable to the stiffness of your own calipers. No book can teach you how to make a fit; experience alone can do this, as it is all done by the sense of feeling; yet many valuable points may be obtained by the recorded experience of other men. With an ordinary pair of 6" calipers 1/18" thick, a 5/16" drag of the outside calipers will give you a 30-ton fit, while a ⅝" drag will give a 70-ton fit; much of course will depend upon the softness of the metal and the length of the fit. The amounts above mentioned are suitable for crank pins and driving axles. Expansion must be considered when fitting bushings, but bolts should be made as nearly the exact size of the hole as possible.

HOW TO FIT BUSHINGS.

You must use judgment in doing this work; if your bushing is very heavy, fit almost as you would anything solid, for the hardening process will make it a little larger; but, if a light bushing, allow about 1/64" on a 2" one, and 1/32" on a 4" bushing, and bore out that much larger to allow for closing. A cylinder bushing make the same size as the hole, or a shade smaller, owing to so much bearing surface presented.

HOW TO FIT ALL KINDS OF BOLTS.

First measure the distance through the hole, allow for nuts, thimble point, etc., and cut off to the correct length under the heads. Put in new centers, turn up the end for thread 1/64" under size of tap (to avoid the sharp edge of the thread), and allow plenty of draw. Then cut the threads in the lathe, or have it done in the bolt cutter. Now set your calipers to the exact size of the hole, and, if an ordinary 1" bolt 6" long, make as nearly the exact size as possible. The smaller the bolt and the less the bearing, the tighter you can drive it in, and the larger the bolt, and the more bearing, the less you must allow for fitting and driving in. Do not drive a bolt very tight in a light casting, or you may crack it.

TAPER BOLTS.

Get the size of the hole at each end, and turn to the required size at each end, if there is no taper attachment on your lathe. Then move the back head toward you, and use the scale between the tool and the finished size of the bolt at each end; when the scale shows the same at each end, tighten the back head and turn the bolt down to that size.

HOW TO SET TOOLS IN LATHE.

For Iron.

Set your tool above the center for outside turning, and below the center for boring out.

For Brass.

Set your tool below the center for outside turning, and above the center for boring out.

HOW TO PLANE UP DRIVING BOXES.

Clamp on bed of planer and finish on face, turn over, clamp planed face to bed, and finish the other face, making the box the correct width. Now clamp one side against the angle plate on bed of planer and plane out for shoe, making each flange of the required thickness and depth. Now reverse, and plane the other side the same. Now if you wish to bevel or taper the flanges, place a liner between box and angle plate and finish one end, then change the liner and finish the other end. Boxes are usually slotted for brasses, but if you have a circle feed on planer place all boxes top down on bed, clamp, and plane to required circle. Keep all the boxes the same size when possible.

Bevel Inside of Driving Boxes.

It is a good practice, and a common one, to bevel the insides of the driving boxes (where the shoes and wedges fit) at both ends, leaving about 4" of bearing in the center; this saves flanges of boxes from breaking.

PLANING UP VALVES.

Clamp on planer face down, and true up top by which to set. Now turn and true up the face of the valve. Turn over again and clamp on parallel strips, and find centers at each end, and set true with the tool. Now face off the top to the required height, and plane off the sides where the yoke fits, and the outside edges of the valve, to required size, and keep perfectly central. Keep the bottom flanges of the valve to the required thickness. Now use a square nose tool and cut two grooves for long strips, of proper width and depth, and the correct distance apart, keeping them perfectly central. Now set the valve crosswise on the bed, still using parallel strips, and set perfectly square with the bed. Plane off the outside ends to the correct length, and keep central, cutting end grooves for strips. Have the inside of the valve slotted, or lay off, chip clearance at each end, and plane out.

PLANING VALVE STRIPS.

Clamp in chuck, top faces down, and true up the bottom lugs, turn over in chuck and finish the top side, making the same height as the grooves in the valve. Now chuck and true up one side of each, then turn over and plane off the other side, fitting snugly to the grooves in the valve. You must finish the lugs on short strips in the jumper, then measure the distance between the grooves on the valve, and jump off the ends of the long strips 1/32" shorter, to allow for expansion of the long strips.

PLANING CROSSHEADS.

Clamp on bed of planer and finish the top and bottom and outsides, but leave about ⅛" stock on the wings that bear on the guides, also on the sides that fit between the guides, to be finished when keyed onto the piston. To finish: Place two V blocks in grooves on the planer and clamp piston down, key on crosshead and set perfectly square. Now finish the head to the correct size all around, letting the tool run over both wings without changing, and plane perfectly central.

PLANING ROD STRAPS.

Finish one side, turn over and put a piece of tin under the end that goes on the rod, then finish that side. The tin makes a taper in the strap, so brasses will slip on easy. The inside of the straps are usually finished on a milling machine, where there is one; if not, then on a slotter or planer, or a shaper.

Cylinders.

FITTING TOGETHER.

Use a long straightedge and set the back faces of the cylinder head joints perfectly true with the straightedge at all four points and see that the bottom of the saddle is set true where the center casting fits. Use a spirit level and level each valve seat both ways. Put a center in the end of each cylinder and see that each end trams perfectly, and that each end is the same distance apart. If the cylinders are perfectly true at all the above points, then proceed to rose-bit all the holes and bolt them together with driving fit bolts. If they are not true use liners between them wherever necessary until they are true; then return one cylinder to the planer, calling the planer hand's attention to the thickness and location of each liner.

FACING OLD CYLINDER HEAD JOINTS.

Ordinarily this job is done with a file and scraper, but in some shops a small machine is used for this purpose; it has a cross-feed and can be clamped into the cylinder head studs. It is a great labor-saving device, as it is a very tedious job to file and scrape a cylinder head. The following method, also, has been found to give good results: Shear off a piece of sheet copper the proper thickness and width and drive it into the groove (which is cut into cylinder heads to allow the center of the head to blow out in case of accident to the rod or strap, thereby saving the cylinder). Allow sufficient stock to let the copper extend outside the head a sufficient distance to true it up in the lathe. The joint is turned V shape, and will remain tight while the engine remains in service.

GRINDING CYLINDER HEAD JOINTS.

Short work is made of this job in locomotive works; it being done in a large drill press or boring mill, the head being raised by the spindle at intervals to apply emery; but in railroad shops this is seldom done. Proceed as follows: If possible upend the cylinder, fasten a short pole or board onto the head, and grind it with oil and emery, using No. 2 emery. When you cannot upend the cylinder use a long rod through the cylinder, or a brace and ratchet outside; slack the head frequently to apply emery, and thereby avoid cutting grooves on the joint.

FITTING CYLINDER SADDLE WHEN SMOKE BOX IS RIVETED TO BOILER.

Let the arch rest on the saddle of cylinders in proper place; now level boiler with cylinders, with spirit level used on valve seat and on ring for the dome cap, also on shell of boiler. Now line both cylinders and divide fire box central with cylinder lines. Now drop plumb line over wagon top of the boiler and keep fire box plumb with the lines on both sides.

Now, if the saddle will finish so as to hold the boiler to its proper height up and down, lay off the saddle with hermaphrodites, as follows:

Set hermaphrodites under the center of the smoke arch, with points perpendicular with the center of the smoke arch. Scribe a line all around at both ends and on both sides, and be careful to hold your hermaphrodites in the same perpendicular position all the way around.

Now separate boiler and saddle, and chip saddle down to a good bearing; use a straightedge to chip to lengthwise. When you have finished put some red lead on the saddle and bolt down solid.

If the circle of the saddle is too large for the boiler, and will not true up, put in a sheet iron, or boiler plate liner, and figure on the thickness of the same when laying it off.

FITTING CYLINDER SADDLE WHEN SMOKE BOX IS TO BE FIRST BOLTED TO CYLINDER.

If a large boring bar is not available for this purpose, clamp a board up at the front and back ends of the saddle, to be used for the center of the smoke arch, and tack a piece of tin on each, for a center. Now obtain the outside diameter of the smoke arch, and carry a line up on one board from the lowest point of the saddle, this distance. Then place four centers in the cylinders, and set trams from the cylinder center to a point where both tram marks cross, exactly on the line carried up on the board from the saddle; mark a small center at that point. With the trams set to the same length, scribe two lines from the center of the cylinder at the other end, and make another small center where these lines cross.

You now have both centers, so set the trams to one-half the outside diameter of the smoke arch, and scribe a line on each side of the saddle. Place a straightedge on the outside of the saddle on each side, and set to each line, and scribe a straight line.

It is now laid off, so take down the wooden centers and chip to the line obtained, and then bolt the smoke arch solidly to the saddle. Set the boiler perfectly central between the frames at the back end, and absolutely level both ways. Then you may have the arch riveted to the boiler.

BORING OUT WORN CYLINDER WITH BORING BAR.

Locomotive cylinders become worn more or less out of round, and gradually wear shoulders at each end, when subjected to the reciprocating motion of the piston. It is then necessary to rebore them to a true cylindrical surface, and this is done without necessitating the removal of the cylinder by means of a portable boring bar, which is mounted on brackets attached to the front and back ends of the cylinder. The bar is rotated by a pneumatic motor (or by a belt), which drives through a train of reducing gears, in order to impart a comparatively slow speed to the bar. These tools, as the bar revolves with them, run lengthwise through the cylinder, under the influence of an adjustable feeding mechanism. Thus a cut of any desired amount may be obtained.

One roughing and one finishing cut are usually all that are required, and often, by using two or three tools, one of which will leave a finished surface, one cut will be found sufficient. However, as many cuts should be taken as the proper accomplishment of the work warrants. Often cylinders become so badly scored that several passages of the tools are necessary.

If the counter-bore needs to be trued, which is not often necessary, as it is not subject to wear, re-counterbore it at each end with offset tools. Make this counter-bore of sufficient length to permit of the outer edge of the cylinder packing ring on the piston just traveling over the inner edge of the counter-bore. It should but cut about a quarter of an inch below the inner cylinder diameter. The reason, of course, for counter-boring, is to prevent the piston from wearing a shoulder in the cylinder at each end of its stroke.

When the bar is set up, the brackets are first attached, by the cylinder studs, to the ends of the cylinder, with the bar bushing as nearly central as can be judged. The bar is passed through the brackets, and set central with the counter-bore at each end, as they are more nearly true. Then, when the bar is set, it is but necessary to attach the driving and feeding mechanisms, and the work may commence, in the manner explained.

RELINING CYLINDERS.

When a cylinder has been enlarged by reboring, as explained, until its inside diameter is an inch above the original size, it is common to reduce the working diameter by the insertion of a lining, or bushing, of cast iron. First, rough-turn the bushing on the outside in a lathe, holding one end in a chuck, and supporting the other on a four-armed spider, which receives the tail stock center. After roughing the outer surface, face the bushing 1/16 of an inch longer than the cylinder, and bore the inner diameter to the finished size. Then place in the ends of the bushing center discs, to fit the counter-bore, and turn the outside of the bushing to a diameter slightly greater than that of the inside of the cylinder, which has been previously bored true in the manner explained under the heading "Boring Out Worn Cylinder, etc." The outside of the bushing is first rough-turned, as described above, in order to avoid a probable distortion resulting from this hard outer surface being removed last. Finish these linings, or bushings, before placing in the cylinder (this includes the cutting of ports to match the steam ports in the cylinder). When completed, heat the cylinder with a fuel oil burner, which will expand it sufficiently to allow for the bushing's being inserted by hand. When the cylinder becomes cold, it will contract and grip tightly the bushing which was finished slightly oversize. The cylinder heads will also prevent longitudinal movement of the bushing, and so the extra length allowed in turning should be faced off when the cylinder is cold.

Facing Valve Seats, Etc.

TRUING SLIDE VALVE SEATS.

As the valve's faces and the seats on the cylinders of slide valve locomotives become worn, it is necessary that they be planed true. This planing operation on the cylinder seats is generally done by a small portable rotary planing machine designed for this work. The machine is mounted on, and attached to, the cylinder by means of small studs (four) which screw into the corner holes for the steam chest studs. The planing tool, which has a circular movement, is fed by a star feed. After the seat and the valve are planed, it is good practice to scrape them, to obtain a more nearly perfect bearing. First the valve is scraped to a surface plate, and then black or red lead is applied over the scraped surface of the valve face, which is used as a surface plate in scraping the seat. The black or red marks on the seat, made by rubbing the valve face across the seat, show the high spots on the surface of the seat. These spots should be removed with a flat-end scraper, and the operation continued until the lead marks on the seat show a good general bearing between valve faces and seat. Often the planed surfaces of valve and seat show a sufficiently good bearing, due to the machining, to obviate the necessity of scraping.

APPLYING A FALSE VALVE SEAT.

When the seat of the slide valve has been planed true so often that it is reduced almost to the level of the cylinder surface, a "false seat" may be applied, thus effecting a saving in labor and material.

The old seat is first planed true, either by hand or with a valve facing machine, and then the new seat is fitted to it by scraping, so as to obtain a steam tight fit, especially at the ports. The bolts used for holding the false seat in place vary in location and style. By having projections at each end of the valve seat (or cylinder), any tendency of the seat to move with the valve is resisted, independently of the bolts. The heads of the bolts, which are usually of steel, should be about one-sixteenth of an inch below the surface of the seat, to avoid the uneven wear which would otherwise result between the steel bolt heads and the softer cast iron seat.

Another method of application is by means of lugs, or projections, cast integral with the seat; these lugs are fitted to the inside of the steam chest, and hold the seat in position without the use of bolts. As in the previously explained method, the seat is scraped to a good bearing on the cylinder, and the points at the ports are often finished by grinding with oil and emery. This false seat has projections which fit into slots planed in the steam chest, so that in the event that the seat should break, it will not lift up and interfere with the movement of the valve.

LINING DOWN PRESSURE PLATES.

Regardless of the form of balanced valve used, when the steam chest cover is tightened down the pressure plate should permit the valve to move freely without binding it in any position; in order to line down the pressure plate properly first have the gaskets put on the steam chests, then use two straightedges and a scale, and measure the exact height of the steam chest. We will assume that it measures 8¼"; now place the valve on the valve-seat, place a straightedge on top of the valve and measure the distance from it to the seat on the cylinder; we shall assume this distance to be 7½"; now you must make allowance for tightening down the gaskets and for clearance between the valve and pressure plate, 1-32" being sufficient for clearance; if the gaskets are new, and are made of ¼" copper wire, the gaskets will let the steam chest cover squeeze down 3/32" before the cover is considered tight, while if the gaskets are made of thin flat copper they will not squeeze to exceed 1/32", but we shall assume that we have ¼" copper wire gaskets, therefore add the 1/32" clearance to the 3/32", which the gaskets will squeeze, which amounts to ⅛"; now subtract ⅛" from 8¼", which leaves 8⅛", and as the top of the valve is 7½" high, the difference, which is ⅝", is the distance the pressure plate should extend below the joint on the steam chest cover; so line the pressure plate accordingly, allowing 1/32" stock for the plate to true up on the planer; use turned brass washers for liners between the plate and cover; they should all four be turned to the same thickness. Let the screw bolts pass through them, and tighten the screw bolts as much as possible without breaking them.

REBORING PISTON VALVE CHAMBERS.

When the piston valve chamber has been worn even slightly out of round (1/32"), or if it be cut, scored or shouldered noticeably, it should be bored out. After this has been done, and the valve is replaced, it should be lined absolutely central with reference to the bore of the valve chamber.

POINTERS.

To Remove a Tight Nut.

Heat an open end wrench which fits the nut to a good heat. While hot, place it on the nut, and allow it to remain for a few minutes. The nut will expand under the heat, and may be readily removed with the wrench. If the nut resists this method, it will probably have to be split.

The use of a blow torch would heat not only the nut, but the bolt as well, so that both would expand, and the nut would not be loosened on the bolt.

Making Holes in Glass.

Spread clay, or putty, over and about that portion of the glass at which a hole is desired. Make a hole in this clay, or putty, of a diameter equal to that of the hole desired, and, into this hole, pour a drop or two of molten lead. This will cause a piece of glass the size of the hole desired to drop out.

Comparatively thin glass may be cut with ordinary shears if the operation is carried on under water.

To Harden a Hammer Face.

Heat the hammer to an even bright red, dip the face into the ordinary bath about ¾ inch, moving it about the surface for a half minute; remove from the bath and rub the face bright. Then dip the nose of the hammer about ½ inch in the bath, until the face has drawn to a deep straw color; cool the face, draw the nose down to blue, and cool altogether.

Fire Clay.

For a boiler furnace, common earth mixed with water, into which has been dissolved a little salt, will take the place of fire clay, when the latter cannot be obtained.

To Polish Brass.

Use a fine file on the brass to smooth its surface, and then rub it with a smooth, fine-grain stone, or with charcoal and water. Then, when the brass is smooth, and free from scratches, polish with pumice stone and oil, spirits of turpentine, or alcohol.

Nickle Plating.

Heat a bath of pure granulated tin, argol, and water to boiling, and add a small quantity of red-hot nickel oxide. A brass or copper article immersed in this solution is instantly covered with a light coating of pure nickel.

Tempering Chisels.

Add 3 ounces spirits nitre, 3 ounces white vitriol, 3 ounces sal-ammoniac, 3 ounces alum, 6 ounces salt, with double handful of hoof parings, to 3 gallons of water. Heat the tool to a cherry red, and temper in the above solution. It gives very good results.

Clamping Work to Face Plate.

When it is necessary to bolt or clamp a piece of work to the face plate, and, for any reason, there is a tendency for the work to slip, place a piece of thin paper between the article and the face plate, and it will hold the article in place.

Coloring Metals.

Dissolve ½ ounce of hypo-sulphite of soda in 1 pound of water, and add ½ ounce of acetate of lead dissolved in ½ pound of water. Place the metal article to be colored in this solution, and heat to 190°—200° Fahr. The solution will decompose slowly, precipitating sulphide of lead on the metal to be colored, in brown flakes.

The resulting color of the metal varies with the amount of lead allowed to be deposited on the metal, and even coloring is produced by even heating of the article in the solution. Iron treated in this manner assumes and will retain a steel-blue color. Other colors in various metals vary with the amount of lead deposited, which, in turn, is dependent upon the length of time the article remains in the solution.

The resulting colors have a good lustre, and adhere very firmly if the metal article has been previously well cleansed.

Sharpening Files.

To sharpen a dull or worn file, place it in a dilute solution of sulphuric acid, one part to two parts of water, over night. Then rinse well in clear water. The acid should be kept in an earthenware vessel.

To Prevent Rusting of Machinery.

Dissolve one ounce of camphor in one pound of lard (melted); remove the scum, and add as much black lead, or lamp black, as will give the mixture an iron color.

Clean the machinery carefully, and apply the mixture, smearing it over the entire surface. It may remain indefinitely, or may be wiped off after 24 hours, and will prevent rust for some time. When the mixture is removed, the metal should be polished with a soft cloth.

A mixture of lard oil and kerosene, in equal parts, is also used to prevent rust on bright work. So, too, is a mixture of tallow and white lead, or tallow and lime.

Another method of treating iron or steel which becomes gray and lusterless, is to cleanse it thoroughly with ammonia soapsuds, using a stiff brush. Rinse well and dry by heat, if convenient; then oil plentfully with sweet oil and dust thoroughly with powdered quick lime. Allow the lime to remain for two days; then brush off with a stiff brush, and polish with a softer one, finally rubbing with cloths until the luster returns.

To prevent rust on tools, apply a solution of vaseline heated slowly with a small quantity of powdered gum camphor.

To Remove Rust.

The rusted steel parts may be brushed with a paste of one-half ounce of cyanide of potassium, one-half ounce castile soap, one ounce whiting, and water enough to make the paste of thick cream consistency. The article should first be immersed in a strong solution of potassium cyanide (½ ounce to two ounces of water) if the rust condition is bad.

Iron may often be quickly cleaned by dipping it in, or washing it with a solution of one part of nitric acid, one part of muriatic acid, and twelve parts of water. After using this solution, rinse with clean water.

However, aggravated cases may be treated as follows: Immerse in a nearly saturated solution of chloride of tin, and allow to remain for a day or so. Be sure that the solution does not contain a great excess of acid, or it will act upon the iron itself. When the articles are removed from the solution, they should be rinsed with water, then ammonia, and quickly dried. The resulting dull silver appearance is removed by giving the article an ordinary polish, which restores its normal appearance.

Bright Polish on Iron.

The object of the following process is to caseharden iron so that it will take a bright polish, like steel.

Pulverize and dissolve the following in one quart of hot water: one ounce blue vitriol; one ounce borax; one ounce prussiate of potash; one ounce charcoal; one-half pint salt. Then add one gallon of linseed oil, mix well, bring the iron to a good heat, and cool it in this solution.

To Write on Metals.

Cover the metal surface to be etched with beeswax, or soap, and write the desired inscription in the wax, so that it extends clear through to the metal. Then apply, with a stick of wood, the following mixture: four ounces of nitric acid and one ounce of muriatic acid, mixed and well shaken.

Let the mixture remain in the lettering in the wax for five or ten minutes, then pour on water, to stop the etching process in the metal, and remove the wax.

Casehardening.

After polishing, the article to be casehardened is heated to a bright red, and the surface is rubbed with prussiate of potash. Then allow the article to cool to a dull red, and immerse it in water.

Or, use a mixture of yellow prussiate of potash, 7 parts by weight; bichromate of potash, 1 part; and common salt, 8 parts; pulverize and mix thoroughly. Dip the article in this mixture, or sprinkle the article with the mixture, when it has been brought to a dark red heat. Return the article to the fire and allow it to soak—repeat this process several times, according to the depth of hardening desired. Then plunge into oil or water. This method is practicable for tool steel, soft steel, or iron.

Another method is to place the article in an iron box, together with bonedust, horn, or shreds of hoof, or leather, and subject to a blood red heat. Then submerge the article in cold water.

Still another casehardening mixture is often employed, consisting of three parts of prussiate of potash and one part of sal-ammoniac, or two parts of sal-ammoniac and one part of prussiate of potash.

Cleaning Brass.

The United States Government method is to use a mixture of one part nitric acid and one-half part sulphuric acid, placed in a stone jar. Dip the articles in the acid, immerse and soak them in fresh water, then rub them with sawdust, and they will become quite brilliant. If the brass is greasy, it must first be dipped into a strong solution of potash and soda in water, and rinsed, or boiled in lye or potash.

If a few drops of muriatic acid are included in the nitric and sulphuric acid mixture, the color of the article will be darkened somewhat, depending upon the amount of muriatic acid used.

Brass may also be cleansed quickly by using an ounce of alum in a pint of boiling water. Wash, and rub with a dry cloth. This method will remove stains, as well as tarnish, and will not harm the metal, nor the hands.

Chased brass may be cleaned by washing with soap and hot water, drying thoroughly, and rubbing with a cut lemon: When it appears quite clean, rinse well with warm water, dry and polish with chamois leather. Powder should not be used to clean chased brass.

Drilling Hard Steel.

Mix well one part of spirits of camphor and four parts of turpentine, and apply cold to the hard steel or iron to be drilled. Allow the mixture to stand a few minutes on the metal before commencing to drill, and then run the drill slowly, with fine feed. In this manner, hard steel or iron may be drilled with ordinary drills. This will give much more satisfactory results than would oil.

Also, to prevent the clearance from wearing off the lip of the drill, necessitating frequent grinding, raise a light burr on the face, or cutting edge, by striking it with a hammer as it is held tightly in the vice. This makes the drill cut oversize, sufficient for easy clearance, and is especially satisfactory in drilling small dies.

Drilling Chilled Cast Iron.

First draw the chill. This is done by laying the piece on the forge, covering the spot to be chilled with sulphur, and burning the sulphur off by working the bellows slowly. Then proceed with the drilling.

Taper Holes.

Most of this work is done with a compound rest, the work being held in a chuck or on the face plate, and the outer end supported in a steady rest.

However, many jobs may be handled by driving the work on an arbor or mandrel, setting the arbor over just as though it were the piece to be tapered, with the exception that the tail stock must be set over the other way for boring. With a taper of one-sixteenth to the inch, and an arbor of thirty-two inches, the total taper would be two inches, and the offset half of this, or one inch. Or, with the same taper, and a twenty-four inch arbor, the offset would be three-quarters of an inch, and so on in proportion to the desired taper and the size of the work.

USEFUL INFORMATION.

Doubling the diameter of a pipe increases its capacity four times. Friction of liquids in pipes increases as the square of the velocity.

A gallon of water (U. S. standard) weighs 8-1/3 pounds and contains 231 cubic inches.

A cubic foot of water contains 7½ gallons, 1,728 cubic inches, and weighs 62½ pounds.

To find the pressure in pounds per square inch of a column of water, multiply the height of the column in feet by .434.

One barrel is rated at 31½ gallons, except a barrel of oil, which is 42 gallons.

To find the number of gallons in a circular tank, multiply the diameter in feet by itself, then multiply by the depth in feet, then by 6, and from this sum deduct 2 per cent. Example: A tank 14 feet diameter and 9 feet deep. $14 \times 14 = 196 \times 9 = 1{,}764 \times 6 = 10{,}584$ less 2 per cent $(= 212) = 10{,}372$ gallons. (This is very nearly exact.)

To ascertain heating surface in tubular boilers multiply two-thirds the circumference of boiler by length of boiler in inches and add to it the area of all the tubes.

In calculating horse-power of tubular or flue boilers consider 15 square feet of heating surface equivalent to one normal horse-power.

Each *nominal* horse-power of boilers requires one cubic foot of feed-water per hour.

Steam rising from water at its boiling point (212 degrees) has a pressure equal to the atmosphere (14.7 pounds to the square inch).

A standard horse-power: The evaporation of 30 pounds of water per hour from a feed water temperature of 100 degrees Fahr. into steam at 70 pounds gauge pressure.

One-sixth tensile strength of plate multiplied by thickness of plate and divided by one-half the diameter of boiler gives safe working pressure for tubular boilers.

Steel boiler plate weighs per square foot approximately 2½ pounds (more exactly 2-55/100 pounds) for each 1/16 inch of thickness.

Copper plate weighs 2-83/100 pounds, and brass plate 2-70/100 pounds per square foot of 1/16 inch thickness.

The rate of combustion in a furnace is computed by the pounds of fuel consumed per square foot of grate per hour.

In general practice the rate for a natural draught is, for anthracite coal, from 7 to 16 pounds; for bituminous, from 10 to 25 pounds; and with artificial or forced draught, as by a blower, exhaust-blast, or steam jet, the rate may be increased from 30 to 120 pounds.

The dimensions or size of coal must be reduced and the depth of the fire increased directly, as the intensity of the draught is increased.

One square foot of grate will consume on an average 12 pounds of coal per hour.

Consumption of fuel averages 7½ pounds of coal or 15 pounds dry pine wood for every cubic foot of water evaporated.

A bushel of bituminous coal weighs 76 pounds in Pennsylvania; 80 pounds in Kentucky, Illinois and Missouri; 70 pounds in Indiana.

A bushel of hard coke weighs 40 pounds.

A bushel of soft or gas-house coke weighs 32 pounds.

One ton, 2,000 pounds, of bituminous coal requires for storage 40 cubic feet, or one ton of 2,240 pounds 45 cubic feet.

One ton, 2,000 pounds, of anthracite coal requires for storage 33 cubic feet, or one ton of 2,240 pounds 37 cubic feet.

Cast iron weighs about one pound per 4 cubic inches.

Wrought iron weighs about one pound per 3½ cubic inches.

To ascertain the weight in pounds per running foot of square steel, multiply the size in inches (using decimals to express fractions most conveniently) by 4; square this; divide by 5; add 1/16.

To ascertain the weight in pounds per running foot of flat steel, multiply the width by the thickness in inches (using decimals to express fractions most conveniently); multiply by 10; divide by 3; add 2 per cent.

TABLES.

The accompanying table, page 787, furnishes a ready means of obtaining the equivalent travel, in inches, of the piston for various percentages of its travel. It is particularly applicable to usage in connection with our matter on valves and valve gears.

INCH EQUIVALENTS FOR VARIOUS PERCENTAGES OF PISTON STROKES.

Per Cent Cut-Off / Stroke	20"	22"	24"	26"	28"	30"	32"	34"
20	4 "	$4\frac{3}{8}$"	$4\frac{1}{2}$"	$5\frac{3}{16}$"	$5\frac{5}{8}$"	6 "	$6\frac{3}{8}$"	$6\frac{13}{16}$"
25	5 "	$5\frac{1}{2}$"	6 "	$6\frac{1}{2}$"	7 "	$7\frac{1}{2}$"	8 "	$8\frac{1}{2}$"
30	6 "	$6\frac{5}{8}$"	$7\frac{3}{16}$"	$7\frac{13}{16}$"	$8\frac{3}{8}$"	9 "	$9\frac{9}{16}$"	$10\frac{3}{16}$"
33	$6\frac{5}{8}$"	$7\frac{1}{4}$"	8 "	$8\frac{9}{16}$"	$9\frac{1}{4}$"	10 "	$10\frac{9}{16}$"	$11\frac{1}{4}$"
50	10 "	11 "	12 "	13 "	14 "	15 "	16 "	17 "
66	$13\frac{3}{16}$"	$14\frac{1}{2}$"	$15\frac{13}{16}$"	$17\frac{1}{8}$"	$18\frac{1}{2}$"	$19\frac{13}{16}$"	$21\frac{1}{8}$"	$22\frac{7}{16}$"
85	17 "	$18\frac{3}{4}$"	$20\frac{3}{8}$"	$22\frac{1}{8}$"	$23\frac{13}{16}$"	$25\frac{1}{2}$"	$27\frac{1}{4}$"	$28\frac{7}{8}$"
86	$17\frac{3}{16}$"	$18\frac{15}{16}$"	$20\frac{5}{8}$"	$22\frac{3}{8}$"	$24\frac{1}{16}$"	$25\frac{13}{16}$"	$27\frac{1}{2}$"	$29\frac{1}{4}$"
87	$17\frac{3}{8}$"	$19\frac{1}{8}$"	$20\frac{7}{8}$"	$22\frac{5}{8}$"	$24\frac{3}{8}$"	$26\frac{1}{8}$"	$27\frac{7}{8}$"	$29\frac{1}{2}$"
88	$17\frac{5}{8}$"	$19\frac{3}{8}$"	$21\frac{1}{8}$"	$22\frac{7}{8}$"	$24\frac{5}{8}$"	$26\frac{7}{16}$"	$28\frac{1}{8}$"	$29\frac{15}{16}$"
89	$17\frac{13}{16}$"	$19\frac{1}{2}$"	$21\frac{3}{8}$"	$23\frac{1}{8}$"	$24\frac{15}{16}$"	$26\frac{11}{16}$"	$28\frac{1}{2}$"	$30\frac{1}{4}$"
90	18 "	$19\frac{13}{16}$"	$21\frac{5}{8}$"	$23\frac{3}{8}$"	$25\frac{1}{4}$"	27 "	$28\frac{13}{16}$"	$30\frac{5}{8}$"
91	$18\frac{3}{16}$"	20 "	$21\frac{7}{8}$"	$23\frac{5}{8}$"	$25\frac{1}{2}$"	$27\frac{5}{16}$"	$29\frac{1}{8}$"	$30\frac{15}{16}$"

Drill List for Pipe Taps.

Diameter of Tap or Size of Pipe Inches	Diameter of Drill Inches	Diameter of Tap or Size of Pipe Inches	Diameter of Drill Inches
1/8	21/64	1-1/4	1-15/32
1/4	29/64	1-1/2	1-23/32
3/8	19/32	2	2- 3/16
1/2	23/32	2-1/2	2-11/16
3/4	15/16	3	3- 5/16
1	1- 3/16	3-1/2	3-13/16

Letter Sizes of Drills.

Diameter Inches	Decimals of 1 Inch	Diameter Inches	Decimals of 1 Inch
A 15/64	.234	N	.302
B	.238	O 5/16	.316
C	.242	P 21/64	.323
D	.246	Q	.332
E 1/4	.250	R 11/32	.339
F	.257	S	.348
G	.261	T 23/64	.358
H 17/64	.266	U	.368
I	.272	V 3/8	.377
J	.277	W 25/64	.386
K 9/32	.281	X	.397
L	.290	Y 13/32	.404
M 19/64	.295	Z	.413

Lubricants for Cutting Tools.

Material	Turning	Chucking	Drilling Milling	Reaming	Tapping
Tool Steel	Dry or Oil	Oil or Soda Water	Oil	Lard Oil	Oil
Soft Steel	Dry or Soda Water	Soda Water	Oil or Soda Water	Lard Oil	Oil
Wrought Iron	Dry or Soda Water	Soda Water	Oil or Soda Water	Lard Oil	Oil
Cast Iron	Dry	Dry	Dry	Dry	Oil
Brass	Dry	Dry	Dry	Dry	Oil
Copper	Dry	Oil	Oil	Mixture	Oil
Babbitt	Dry	Dry	Dry	Dry or Kerosene	Oil
Glass			Turpentine		

Mixture is 1/3 Crude Petroleum, 2/3 Lard Oil. Oil is Lard. When two lubricants are mentioned the first is preferable.

Table of Decimal Equivalents of 8ths, 16th, 32ds, and 64ths of an Inch.

8ths

1/8 = .125
1/4 = .250
3/8 = .375
1/2 = .500
5/8 = .625
3/4 = .750
7/8 = .875

16ths

1/16 = .0625
3/16 = .1875
5/16 = .3125
7/16 = .4375
9/16 = .5625
11/16 = .6875
13/16 = .8125
15/16 = .9375

32ds

1/32 = .03125
3/32 = .09375
5/32 = .15625
7/32 = .21875
9/32 = .28125
11/32 = .34375
13/32 = .40625
15/32 = .46875
17/32 = .53125
19/32 = .59375
21/32 = .65625
23/32 = .71875
25/32 = .78125
27/32 = .84375
29/32 = .90625
31/32 = .96875

64ths

1/64 = .015625
3/64 = .046875
5/64 = .078125
7/64 = .109375
9/64 = .140625
11/64 = .171875
13/64 = .203125
15/64 = .234375
17/64 = .265625
19/64 = .296875
21/64 = .328125
23/64 = .359375
25/64 = .390625
27/64 = .421875
29/64 = .453125
31/64 = .484375
33/64 = .515625
35/64 = .546875
37/64 = .578125
39/64 = .609375
41/64 = .640625
43/64 = .671875
45/64 = .703125
47/64 = .734375
49/64 = .765625
51/64 = .796875
53/64 = .828125
55/64 = .859375
57/64 = .890625
59/64 = .921875
61/64 = .953125
63/64 = .984375

Table of Decimal Equivalents of Millimeters and Fractions of Millimeters.

1/100 mm. = .0003937 inch

mm.	inches	mm.	inches	mm.	inches
1/50 =	.00079	26/50 =	.02047	2 =	.07874
2/50 =	.00157	27/50 =	.02126	3 =	.11811
3/50 =	.00236	28/50 =	.02205	4 =	.15748
4/50 =	.00315	29/50 =	.02283	5 =	.19685
5/50 =	.00394	30/50 =	.02362	6 =	.23622
6/50 =	.00472	31/50 =	.02441	7 =	.27559
7/50 =	.00551	32/50 =	.02520	8 =	.31496
8/50 =	.00630	33/50 =	.02598	9 =	.35433
9/50 =	.00709	34/50 =	.02677	10 =	.39370
10/50 =	.00787	35/50 =	.02756	11 =	.43307
11/50 =	.00866	36/50 =	.02835	12 =	.47244
12/50 =	.00945	37/50 =	.02913	13 =	.51181
13/50 =	.01024	38/50 =	.02992	14 =	.55118
14/50 =	.01102	39/50 =	.03071	15 =	.59055
15/50 =	.01181	40/50 =	.03150	16 =	.62992
16/50 =	.01260	41/50 =	.03228	17 =	.66929
17/50 =	.01339	42/50 =	.03307	18 =	.70866
18/50 =	.01417	43/50 =	.03386	19 =	.74803
19/50 =	.01496	44/50 =	.03465	20 =	.78740
20/50 =	.01575	45/50 =	.03543	21 =	.82677
21/50 =	.01654	46/50 =	.03622	22 =	.86614
22/50 =	.01732	47/50 =	.03701	23 =	.90551
23/50 =	.01811	48/50 =	.03780	24 =	.94488
24/50 =	.01890	49/50 =	.03858	25 =	.98425
25/50 =	.01969	1 =	.03937	26 =	1.02362

10 mm. = 1 centimeter = 0.3937 inch
10 cm. = 1 decimeter = 3.937 inches
10 dm. = 1 meter = 39.37 inches
25.4 mm. = 1 English inch

WIRE GAUGE STANDARDS.

Dimensions of Sizes in Decimal Parts of an Inch.

No. of Wire Gauge	American or Brown & Sharpe	Birmingham or Stubs' Wire	Washburn & Moen	Trenton Iron Co.	Stubs' Steel Wire	U. S. Std. for Plate
000000				....		.4687
00000				4500		4375
0000	4600	.454	3938	4000		4062
000	4096	425	3625	3600		3750
00	3648	380	3310	3300		3437
0	3249	340	3065	3050		3125
1	2893	300	2830	2850	.227	2812
2	2576	284	2625	2650	219	2656
3	2294	259	2437	2450	212	2500
4	2043	238	2253	2250	207	2344
5	1819	220	2070	2050	204	2187
6	1620	203	1920	1900	201	2031
7	1443	180	1770	1750	199	1875
8	1285	165	1620	1600	197	1719
9	1144	148	1483	1450	194	1562
10	1019	134	1350	1300	191	1406
11	0907	120	1205	1175	188	1250
12	0808	109	1055	1050	185	1094
13	0720	095	0915	0925	182	0937
14	0641	083	0800	0800	180	0781
15	0571	072	0720	0700	178	0703
16	0508	065	0625	0610	175	0625
17	0453	.058	0540	0625	172	0562
18	0403	.049	0475	0450	168	0500
19	0359	.042	0410	0400	164	0437
20	0320	035	0348	0350	161	0375
21	0285	032	0317	0310	157	0344
22	0253	028	0286	0280	155	0312
23	0226	025	0258	0250	153	0281
24	0201	022	0230	0225	151	0250
25	0179	020	0204	0200	148	0219
26	0159	018	0181	0180	146	0187
27	0142	016	0173	0170	143	0172
28	0126	014	0162	0160	139	0156
29	0113	013	0150	0150	134	0141
30	0100	012	0140	0140	127	0125
31	0089	010	.0132	0130	120	0109
32	0079	009	0128	0120	115	0102
33	0071	008	0118	0110	112	0094
34	0063	007	0104	0100	110	0086
35	0056	005	0095	0095	108	0078
36	0050	004	0090	0090	106	0070
37	0045			0085	103	0066
38	0040			0080	101	0062
39	0035			0075	099	
40	0031			0070	097	

Metric Conversion Table.

Millimeters	×	.03937	=	Inches
"	=	25.400	×	"
Meters	×	3.2809	=	Feet
"	=	.3048	×	"
Kilometers	×	.621377	=	Miles
"	=	1.6093	×	"
Square centimeters . . .	×	.15500	=	Square inches
" " . . .	=	6.4515	×	" "
Square meters	×	10.76410	=	Square feet
" "	=	.09290	×	" "
Square kilometers . . .	×	247.1098	=	Acres
" " . . .	=	.00405	×	"
Cubic centimeters . . .	×	.061025	=	Cubic inches
" " . . .	=	16.3866	×	" "
Cubic meters	×	35.3156	=	Cubic feet
" "	=	.02832	×	" "
" "	×	1.308	=	Cubic yards
" "	=	.765	×	" "

MISCELLANEOUS.

Gauge of Track.

The gauge of track of a railroad is always the distance measured in the clear between the rail heads or "ball" of the rails, as the illustration, Fig. 351, shows. The gauge of track is *not* measured between the flanges of the rolling stock wheels, and it is a mistake to increase the track gauge for sake of "clearance," for, in the construction of locomotive and car wheels, the proper amount of clearance, or sideplay, is provided.

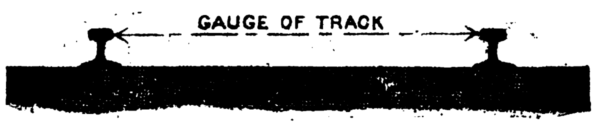


FIG. 351.

However, in order to pass around curves perfectly, theoretically every axle in the train should point to the exact center of the curve, and the outer wheels should be larger than the inner ones. In practice, the difference in wheel sizes is accomplished by coning each wheel tread so that the diameter close to the flange is greater that at the front face. But the radial axle position referred to is impracticable, as most cars and locomotives are built so that two or more axles are always parallel. This will cause a bind between rails and wheels, on curves, and the track gauge, therefore, is slightly increased on curves for the necessary clearance. The amount of increase in the gauge depends upon the degree of curvature.

Elevation of Outer Rail on Curves.

In passing around curves, the centrifugal force tends to tip the rolling stock outward, and to crowd the wheels against the outer rails. This tendency increases with increased speed, and is greater in the case of a sharp curve than an easy one.

To counteract this tendency, it is customary to slightly elevate the outer rail to such an extent that, at a desired speed, there will be no more pressure against one rail than the other. Where the same track is used for both fast and slow trains, this change should be made to accommodate the fast train.

The amount of elevation found advisable is, on standard gauge roads, one-half inch for each degree of curvature for speeds of approximately 35 miles per hour, and for other speeds, and other track gauges, this elevation is accordingly modified. It is common to begin to make a difference in the rail elevation when approaching the curve at some distance, say 50 or 100 feet before the curve is reached, and to keep the center line of the tracks level by dividing the amount of superelevation equally, and by depressing the inner rail below this center line the same amount that the outer rail is elevated above the line.

Adhesion.

The factor of adhesion of a locomotive is the ratio between the working order weight on the drivers and the tractive force (see page 795), and is found by dividing this weight on the driving wheels by the tractive force. The factor of adhesion is usually from four to five; that is, the proportion of the tractive force to the weight is 1 to 4, or 1 to 5, approximately.

If the factor of adhesion be excessive, that is, if the weight is too great in proportion to the power, the locomotive cannot slip its drivers—it is not "smart," but "logy." However, if it be too low, the drivers will slip too readily, and the locomotive will not be able to handle its proper load. The design of the locomotive and the character of the service must be considered in determining the factor of adhesion.

Tractive Force of a Locomotive.

The tractive force of a locomotive, upon which basis the power of a locomotive is usually computed, is arrived at by multiplying the square of the cylinder diameter in inches by the stroke of the piston, in inches; multiplying this by 85 per cent of the boiler pressure, in pounds per inch; and then dividing by the driving wheel diameter, in inches, or:

$$T = \frac{D^2 \times L \times .85\,p}{d}$$

Where T represents the tractive force, in pounds;
D represents the diameter of the cylinder, in inches;
L represents the length of the stroke, in inches;
.85p represents 85% of the boiler pressure, in pounds per square inch;
d represents the diameter of the driving wheels, in inches.

The above formula has been determined from these factors; the tractive force of a locomotive is due to the pressure of steam on the piston as delivered through one revolution of the driving wheels. The tractive force increases in direct proportion to the piston area, length of stroke, and pressure of steam in the cylinders; it decreases in direct proportion as the driving wheel diameter is increased.

Tractive Force of Compound Engines.*

T=Tractive power (maximum).

d=Diameter of H. P. cylinder.

D=Diameter of L. P. cylinder.

S=Stroke of piston.

P=Boiler pressure.

C=Constant (taken from Table on page 797).

W=Diameter of driving wheels.

R=Ratio of L. P. to H. P. cylinder volume.

$$T_1 \text{ (Two-cylinder compound)} = \frac{D^2 S P C}{2 W}$$

$$T_2 \text{ (Four-cylinder compound)} = \frac{D^2 S P C}{W}$$

When C=.52 and R=2.5

T_3 (Two-cylinder compound working simple)

$$= \frac{.85\ d^2SP}{W} = T_1 \times \frac{1.7}{CR} = 1.3T_1$$

T_4 (Four-cylinder compound working simple)

$$= \frac{(2 \times .85)d^2SP}{W} = T_2 \times \frac{1.7}{CR} = 1.3T^2$$

T_3 and T_4 give T when just moving. At slow speeds T working simple will exceed the power working compound by approximately 20 per cent.

For cylinder ratio of approximately 2.5 to 1, as ordinarily used, a constant of 0.52 may be considered as sufficiently accurate for estimates.

*Courtesy of American Locomotive Co.

On superheater compound engines, in order to properly divide the work between the high and low pressure cylinders, the cut-off in the low-pressure cylinder should be approximately 5 per cent. later than the cut-off in the high-pressure cylinder for a cylinder ratio of 2.5 to 1. This difference in cut-off should be reduced for higher cylinder ratios to 0 for a ratio of 2.75 to 1, and increased for lower cylinder ratios to approximately 10 per cent for a ratio of 2.2 to 1.

TABLE OF CONSTANTS ("C").

Per Cent Cut-off H. P. Cylinder	Ratio of L. P. to H. P. Cylinder Volume						
	2.2	2.3	2.4	2.5	2.6	2.7	2.8
90	...	....	.571	.557	.542	.528	.483
89	....	....	.565	.550	.536	.521	.513
88	....	.573	.559	.543	.529	.515	.507
87	....	.567	.552	.537	.523	.509	.500
86	.575	.560	.546	.531	.517	.502	494
85	.570	.555	.540	.526	.511	.497	.489
84	.564	.550	.534	.520	.506	.491	....
83	.559	.544	.529	.515	.500	.486	....
82	.553	.541	.524	.510	.496	....	....
81	.548	.534	.520	.505	.490	....	....
80	.543	.531	.515	.500	.486	....	....

Draw-Bar Pull.

The *tractive force* and the *draw-bar pull* of a locomotive are often looked upon as one and the same thing; however, this is not so. While the tractive force includes that percentage of the power needed to run the locomotive (and tender), as well as to pull the train, the term draw-bar pull is properly applied only to the power available for pulling the train attached to the locomotive (thus the term). Hence, the draw-bar pull is a variable quantity, and is computed by subtracting from the tractive force the amount of resistance the locomotive must overcome in hauling itself (and tender) under given conditions of grade and track.

Horsepower of a Locomotive.

The power required to lift 33,000 pounds one foot, in one minute, is the equivalent of one horsepower.

The power of a locomotive is better expressed in pounds of tractive force (see page 795), since horsepower is dependent upon speed, which, in a locomotive, is variable. However, the horsepower developed by the locomotive may be calculated as follows:

Multiply—

The area of the two pistons, in square inches;

The effective cylinder pressure in pounds per square inch;

Twice the length of the stroke, in feet;

The number of revolutions per minute of the drivers, and

Divide the product by 33,000.

The result is the horsepower of the locomotive.

Again

$$HP = \frac{T\ F \times S}{375}$$

Where T F is tractive force in pounds:

S is speed in miles per hour at which locomotive can haul its heaviest load;

H P is horsepower developed.

That is, the horsepower is found by multiplying the tractive force (page 795) in pounds by the speed at which the locomotive can haul its heaviest load, and dividing by 375.

It has been estimated that one horsepower can be obtained from about 27 lbs. of saturated steam in simple cylinders, with piston speeds of 700 to 1,000 feet per minute, and that, under like conditions, one horsepower is attainable from 21 lbs. of steam superheated to 200 degrees and over. These figures provide steam for all auxiliaries of the locomotive.

Speed and Resistance.

It might be of practical value to the reader to know of the following interesting facts as to speed and resistance of trains.

It requires more power to start a train than is required to keep it in motion after starting. This is because the journal and flange frictional resistances are greater when starting, and rapidly diminish as the train acquires motion. Journal lubrication is more nearly perfect at higher speeds, and, in cold weather when oil congeals and hardens somewhat, this advantage of speed is most noticeable. The decrease in flange friction, and resistance, at higher speeds, is due to the fact that the rapidly moving body has a tendency to move in a straight line and, therefore, the flanges do not bear against the rails as much as at low speeds. For these reasons, a locomotive can be relied upon to haul any train it can start, if the boiler capacity be sufficient, under given conditions as to grade and curvature of the track.

While resistance due to grades is constant at all speeds, the momentum of a swiftly moving train may be utilized in surmounting a short, heavy grade that would stall the same train, moving at a slower rate of speed.

Also, it should be remembered that no locomotive can haul its heaviest train, and make its fastest speed, at the same time. Every locomotive has a definite limit of speed at which it can haul its heaviest load, and beyond this speed the load it can haul decreases quite rapidly. For this reason, a freight locomotive cannot haul, at high speeds, as heavy a train as can be hauled by a passenger locomotive with less tractive force, and less weight on the drivers. This is due to their inherent differences in design, and is demonstrated by the following example:

A passenger locomotive, 18x24" cylinders, 66" drivers, 180 lbs. boiler pressure, 18,020 pounds tractive force, has, at 60 miles per hour, an available tractive force of 4,320 pounds, while a freight locomotive, 18x24" cylinders, 48" drivers, 180 lbs. pressure, 24,785 pounds tractive force, at 60 miles per hour, has only 3,720 pounds available. However, the freight locomotive, at slow speed (about 8 miles per hour) has approximately 35% more tractive force than the passenger locomotive.

The resistance due to head winds, of course, is increased in proportion to the speed of the train—the faster the train is moving, the greater is this resistance. However, head air resistance, when running through still air at 20 miles per hour or less may be neglected, as it amounts to less than 100 pounds, but this resistance for high speed is important, as it is a considerable factor—varying directly as the square of the velocity.

SPEED TABLE.

Showing the speed in miles per hour required to run a certain distance in a given time.

To Use the Table.

Refer to left hand column for the speed you wish to run per hour, then follow that line until it intersects the number of miles you intend to run, and you will find the time required to run it. For example: If you wish to run at the rate of 22 miles per hour to a point 9 miles distant; opposite 22 and under 9 you will find that you have 24 minutes and 33 seconds in which to run the 9 miles.

If you wish to ascertain the speed per hour refer to the time beneath the number of miles traversed, and follow the line to the left hand column. For example: If you run 7 miles in 12 minutes, by following said line to the left you will find in the left hand column that you have run at the rate of 35 miles per hour.

In this table half-seconds and over are shown as one second; less than half-seconds are dropped.

MILES PER HOUR	1 MILE (Hours Minutes Seconds)	2 MILES (Hours Minutes Seconds)	3 MILES (Hours Minutes Seconds)	4 MILES (Hours Minutes Seconds)	5 MILES (Hours Minutes Seconds)	6 MILES (Hours Minutes Seconds)	7 MILES (Hours Minutes Seconds)	8 MILES (Hours Minutes Seconds)	9 MILES (Hours Minutes Seconds)	10 MILES (Hours Minutes Seconds)
4	0.15.00	0.30.00	0.45.00	1.00.00	1.15.00	1.30.00	1.45.00	2.00.00	2.15.00	2.30.00
5	0.12.00	0.24.00	0.36.00	0.48 00	1.00 00	1.12.00	1.24.00	1.36.00	1.48.00	2.00.00
6	0.10 00	0.20 00	0.30.00	0.40 00	0.50.00	1.00.00	1.10.00	1.20.00	1.30.00	1.40.00
7	0.08.34	0.17 09	0.25.43	0.34 17	0.42.51	0.51.26	1.00.00	1.08.34	1.17.09	1.25.43
8	0.07.30	0.15.00	0.22.30	0.30 00	0.37.30	0 45.00	0.52.30	1.00.00	1.07.30	1.15.00
9	0.06 40	0.13 20	0.20.00	0.26 40	0.33.20	0 40.00	0.46.40	0.53.20	1.00.00	1.06.40
10	0.06.00	0.12.00	0.18 00	0.24 00	0 30.00	0.36.00	0.42.00	0.48.00	0.54.00	1.00.00
11	0.05.27	0.10.55	0.16.22	0.21 49	0.27 16	0.32 44	0.38.11	0.43 38	0.49.05	0.54.33
12	0.05.00	0.10.00	0.15.00	0.20 00	0.25.00	0.30 00	0.35.00	0.40.00	0.45.00	0.50.00
13	0.04 37	0.09.14	0.13.51	0.18 28	0.23.05	0.27 42	0.32.18	0.36.55	0.41.32	0.46.09
14	0.04.17	0.08.34	0.12.51	0 17.09	0.21.26	0.25 43	0.30.00	0.34.17	0.38.34	0.42.51
15	0.04.00	0.08 00	0.12.00	0.16.00	0.20.00	0 24 00	0.28.00	0.32.00	0.36.00	0.40.00
16	0.03 45	0.07 30	0.11 15	0.15.00	0.18.45	0.22.30	0 26.15	0.30.00	0 33.45	0.37.30
17	0.03.32	0.07.04	0.10 35	0.14.07	0 17.39	0.21 11	0.24.42	0.28 14	0.31.46	0.35.18
18	0.03.20	0.06 40	0.10 00	0.13.20	0 16.40	0.20.00	0.23.20	0.26.40	0.30.00	0 33 20
19	0.03.09	0.06 19	0.09 28	0.12.38	0 15.47	0.18.57	0.22.06	0.25 16	0.28.25	0.31.35
20	0.03.00	0.06.00	0.09.00	0.12.00	0.15.00	0.18 00	0.21 00	0.24.00	0.27.00	0.30 00
22	0 02.44	0 05 27	0.08.11	0.10 55	0 13 38	0.16 22	0 19.05	0.21 49	0.24.33	0.27 16
24	0.02.30	0.05.00	0.07.30	0.10 00	0 12.30	0.15 00	0.17.30	0.20.00	0.22.30	0.25.00
25	0 02.24	0 04.48	0.07 12	0 09 36	0.12.00	0.14.24	0.16.48	0.19.12	0.21.36	0.24.00
28	0.02.09	0 04.17	0.06.26	0 08 34	0.10.43	0.12.51	0.15.00	0.17.09	0 19.17	0.21.26
30	0.02.00	0 04.00	0.06.00	0 08 00	0 10.00	0.12.00	0.14.00	0.16.00	0.18.00	0.20.00
35	0.01.43	0 03.26	0.05.09	0.06 51	0.08.34	0 10.17	0.12 00	0.13.43	0.15.26	0.17 09
40	0.01.30	0.03.00	0.04.30	0.06 00	0 07 30	0.09.00	0.10.30	0 12.00	0.13 30	0.15.00
45	0.01.20	0.02.40	0.04.00	0.05.20	0 06.40	0 08 00	0 09.20	0.10.40	0.12.00	0.13 20
48	0.01.15	0.02 30	0.03.45	0.05.00	0.06.15	0.07 30	0.08.45	0.10.00	0 11.15	0.12.30
50	0.01.12	0.02.24	0.03 36	0.04 48	0 06.00	0 07 12	0 08.24	0.09.36	0.10.48	0.12.00
55	0.01.05	0.02.11	0.03 16	0.04.22	0.05.27	0.06.33	0.07.38	0.08.44	0.09.49	0.10.55
60	0.01.00	0.02.00	0.03.00	0.04.00	0.05.00	0.06 00	0.07 00	0.08.00	0.09.00	0.10.00
65	0.00.55	0 01.51	0.02 46	0 03 42	0.04.37	0.05.32	0 06.28	0.07.23	0.08.18	0.09.14
70	0.00.51	0.01.43	0.02.34	0.03.26	0.04.17	0.05 09	0.06.00	0.06.51	0.07.43	0.08.34
75	0.00 48	0.01.36	0.02.24	0.03.12	0.04.00	0.04 48	0.05.36	0.06.24	0.07.12	0.08.00
80	0.00.45	0.01.30	0.02.15	0.03.00	0.03.45	0 04.30	0.05.15	0.06.00	0.06.45	0.07 30
85	0.00 42	0.01.25	0.02.07	0.02.49	0.03.32	0.04.14	0.04.56	0.05.39	0.06.21	0.07.04
90	0.00.40	0.01.20	0.02.00	0.02.40	0.03.20	0.04.00	0.04.40	0.05.20	0.06.00	0.06.40
95	0.00.38	0.01.16	0.01.54	0.02.32	0.03.09	0.03.47	0.04.25	0.05.03	0.05.41	0.06.19
100	0.00.36	0.01.12	0.01.48	0.02.24	0.03.00	0.03.36	0.04.12	0.04.48	0.05.24	0.06.00

Speed Recorders.

The speed recorder, as its name implies, is an instrument used in connection with the locomotive to record, at all times, the exact speed at which the locomotive is traveling, and to indicate this speed upon a dial located in the cab, in view of the enginemen. In addition to this, the instrument records, upon a ribbon of paper, the exact speed at which the engine passes any point on the road, the number and location of the stops, and the distance, speed and location of any backward movement of the locomotive.

THE BOYER SPEED RECORDER.

As shown by the illustration, Fig. 352, the recorder contains a pulley, which is driven by a pulley, mounted on the wheel

FIG. 352.

from which the instrument is operated. This is usually the trailer, as application to the driver is undesirable, due to inaccessibility, and error introduced through slippage. The rotation of the recorder pulley operates a rotary pump, which produces a pressure of oil beneath a piston, operating in a cylinder. To this piston, by means of a small wire enclosed in a one-eighth inch pipe, is connected the gauge in the cab. This gauge is of the dial type, graduated from zero to ninety miles per hour, as shown in Fig. 352.

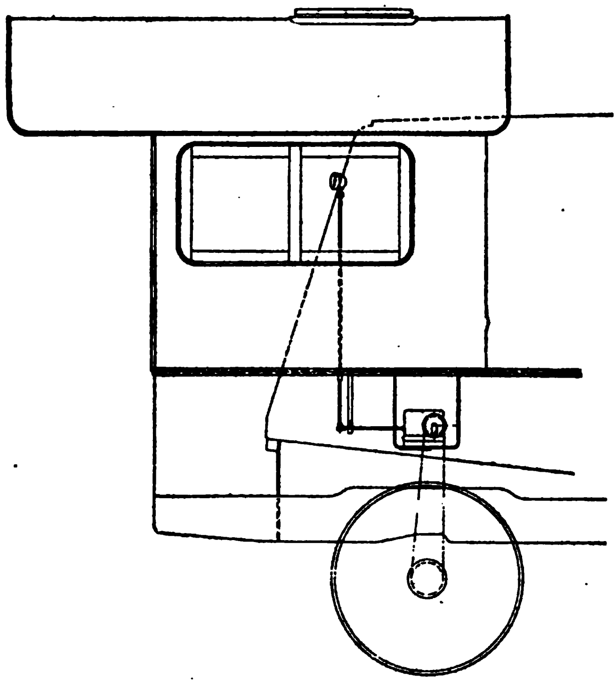


FIG. 353.

A mechanism is provided within the recorder by which a ribbon of paper moves around a drum at the rate of one-half inch to the mile. This paper contains horizontal lines, which, in connection with the line traced by the pencil of the instrument, records the speed, distance traveled, and number and location of the stops made. A typical application of the instrument to a locomotive is shown in Fig. 353. This instrument is handled by the Chicago Pneumatic Tool Co., of Chicago, Ill.

Locomotive Classification.

The most common method of classifying the different types and sizes of locomotives is by means of the Whyte system. In this system, at least three figures are always used, and, with articulated locomotives, from four to six figures. The left-hand figure denotes the number of wheels in the front truck, the right-hand figure the number of wheels in the rear truck, or trailer, and the central figures the number of driving wheels. Thus 2-6-4 indicates a two-wheeled front truck, six driving wheels, and a four-wheeled rear truck. Or, 2-6-6-0 indicates a two-wheeled truck, two sets of driving wheels (six drivers in each set), and no rear truck. When tanks are used in place of a separate tender, the letter T is placed after the classification as to wheel arrangement, and, too, the number of thousands of tons is often added, to indicate the weight of the locomotive. Thus, 2-6-6 T 214 would indicate a two-wheeled front truck, six drivers, six-wheeled rear truck, tank type, weighing 214,000 pounds. The American Locomotive Company also employs the letter S to denote a superheater, and C to determine a compound.

The table on page 805 gives the classification, or type, in figures, a graphical representation of the wheel arrangement, and the general name applied to each particular type. The letter P in the table represents the locomotive pilot.

It will be observed that in the accompanying table, the different types of locomotives have been designated by certain names by which each type is generally known. As it may be of interest to the reader to know of the origin of these general names, we shall explain briefly how some of the more modern types derived their names.

Type.	Wheel Arrangement.	Name.
0-4-0	OO	4-Wheel Switcher.
0-6-0	OOO	6-Wheel Switcher.
0-8-0	OOOO	8-Wheel Switcher.
0-10-0	OOOOO	10-Wheel Switcher.
2-4-0	P oOO	4-Coupled.
2-6-0	P oOOO	Mogul.
2-8-0	P oOOOO	Consolidation.
2-10-0	P oOOOOO	Decapped.
2-12-0	P oOOOOOO	Centipede.
4-4-0	P ooOO	8-Wheel.
4-6-0	P ooOOO	10-Wheel.
4-8-0	P ooOOOO	12-Wheel.
4-10-0	P ooOOOOO	Mastodon.
0-4-2	P OOo	4-Coupled and Trailing.
0-6-2	P OOOo	6-Coupled and Trailing.
0-8-2	P OOOOo	8-Coupled and Trailing.
0-4-4	P OOoo	Forney 4-Coupled.
0-6-4	P OOOoo	Forney 6-Coupled.
0-4-6	P OOooo	Forney 4-Coupled.
0-6-6	P OOOooo	Forney 6-Coupled.
2-4-2	P oOOo	Columbia.
2-6-2	P oOOOo	Prairie.
2-8-2	P oOOOOo	Mikado.
2-10-2	P oOOOOOo	Santa Fe.
2-4-4	P oOOOoo	4-Coupled.
2-6-4	P oOOOoo	6-Coupled.
2-8-4	P oOOOOoo	8-Coupled.
2-4-6	P oOOooo	4-Coupled.
2-6-6	P oOOOooo	6-Coupled.
2-8-6	P oOOOOooo	8-Coupled Double Ender.
4-2-2	P ooOo	Bicycle.
4-4-2	P ooOOo	Atlantic.
4-6-2	P ooOOOo	Pacific.
4-8-2	P ooOOOOo	Mountain.
4-4-4	P ooOOoo	4-Coupled Double Ender.
4-6-4	P ooOOOoo	6-Coupled Double Ender.
4-4-6	P ooOOooo	4-Coupled Double Ender.
0-4-4-0	P OO OO	Articulated.
0-6-6-0	P OOO OOO	Articulated.
0-6-6-2	P OOO OOOo	Articulated.
0-8-8-0	P OOOO OOOO	Articulated.
0-10-10-0	P OOOOO OOOOO	Articulated.
2-4-4-0	P oOO OO	Articulated.
2-6-6-0	P oOOO OOO	Articulated.
2-8-8-0	P oOOOO OOOO	Articulated.
2-4-4-2	P oOO OOo	Articulated.
2-6-6-2	P oOOO OOOo	Articulated.
2-8-8-2	P oOOOO OOOOo	Articulated.
2-10-10-2	P oOOOOO OOOOOo	Articulated.
2-8-8-8-2	P oOOOO OOOO OOOOo	Articulated.
2-8-8-8-8-2	P oOOOO OOOO OOOO OOOOo	Quadruplex.

Santa Fe.

The designation "Santa Fe" has been appropriately applied to the ten-coupled type, with two-wheeled front and rear trucks, because the first locomotives with this arrangement were built for the Atchison, Topeka & Santa Fe Railway system.

Atlantic.

The first road to employ this well-known type of fast passenger locomotive was the Atlantic Coast Line, and thus the design was given the name "Atlantic" in honor of the first road using this type.

Mikado.

Similarly, the type known as the "Mikado" derived its name from the fact that the first locomotives of this design were built for the Japan Railway Company.

Mountain.

This powerful type of passenger locomotive was built to operate on the mountain districts of the Chesapeake & Ohio Railway, and was appropriately christened the "Mountain" type.

THE AIR BRAKE.*

While every railroad man realizes that it is necessary to control the speed of a train, not every railroad man fully realizes that it is as important to stop a train as to start it. In fact, to *stop* is *the* most important when considered from the standpoint of preservation of life and property. It is self-evident that to put a train into motion without adequate means of controlling it would be more dangerous than not to have moved the train at all. There is another sense, however, in which adequate train control is important and that is from the economic and commercial standpoint. Adequate train control permits short, smooth and accurate stops in ordinary service operation. This means maximum traffic carrying capacity by permitting long trains, short headway, least time lost in making stops, high speed, etc., and results in smooth handling of both freight and passenger trains.

The means of securing adequate train control is through the Air Brake and since it is so necessary and important, every railroad man should become familiar with it.

How To Study It.

In commencing the study of the air brake, you should not attempt first, as is often done, to learn the location of the different chambers and passageways of each device and the different port

* This chapter was especially prepared for this book by the Westinghouse Air Brake Company.

connections in the different positions of the moving parts. This is all well enough in its proper place but you should first thoroughly understand the *principles* or *laws* underlying the operation of the different devices and then you can more easily trace the air through them, from the air compressor through the entire equipment to the brake cylinder. A mere knowledge of ports and passages will do little good without thorough understanding of the *principles* or the *why* of operation.

Principle of Operation.

The fundamental principle of operation is that of pressure acting on both sides of a piston which is common to all triple valve devices, such as the distributing valve, etc. When the pressure on one side becomes greater than the pressure on the other side, the piston is moved toward the lower pressure so as to get a balance of pressure. The piston is therefore the movable wall separating the two pressures. The difference in pressure which causes this movement may be produced either by (a) reducing the pressure on one side below that on the other or (b) increasing the pressure on one side above that on the other. The piston has attached to its stem the necessary graduating valves, slide valves, etc., for controlling the flow of compressed air.

Let us refer to Figure 354 and see how this principle applies to the simple triple valve device illustrated, which really represents the elements or fundamental parts of all automatic air brake mechanisms. After you have gained a clear understanding of the operation of this device you can go on to a study of the operation of the distributing valve. The outer face of the piston is always connected to the brake pipe and the inner face to the air storage volume or auxiliary reservoir. With the parts in the position shown, a feed groove connects the chambers on both sides of the piston. Therefore, as brake pipe pressure is built up on the outer face, it also builds up auxiliary reservoir pressure on the opposite face, or as we usually speak of it, "charges the auxiliary reservoir." The air continues to flow in this way through the feed groove until the auxiliary reservoir is charged to brake pipe pressure. The pressures on both sides

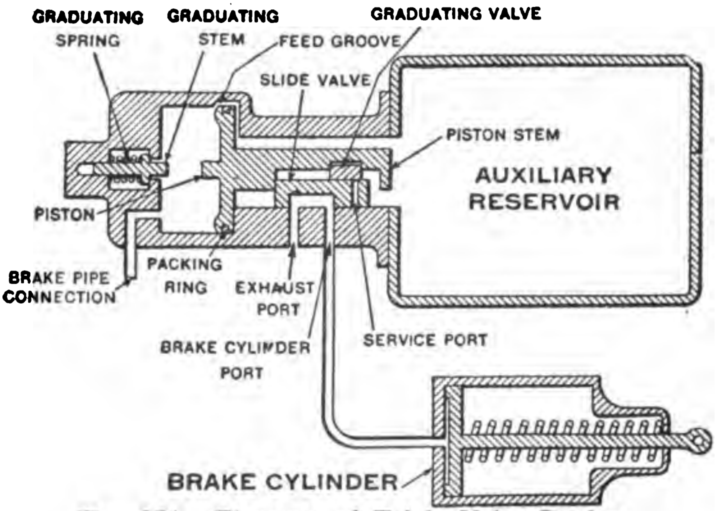


FIG. 354. Elements of Triple Valve Device.

of the piston are now balanced and it cannot move. This is called *Release and Charging* position.

If now you permit air to flow out of the brake pipe, you also reduce the pressure on the outer face of the piston. If you reduce it faster than the air in the auxiliary reservoir can feed back into the brake pipe through the feed groove, the greater pressure in the auxiliary reservoir forces the piston with its attached valves toward the lower brake pipe pressure. This (a) closes the feed groove so that there can be no back-flow from the auxiliary reservoir into the brake pipe; (b) moves the attached graduating valve, thereby opening the service port in the slide valve; and (c) causes the end of the piston stem to butt against the slide valve.

If you still further reduce brake pipe pressure sufficiently, the auxiliary reservoir pressure will move the piston and slide valve until stopped by the graduating stem. This movement of the slide valve (a) cuts off connection between the brake cylinder port and the exhaust port, and (b) causes the service port, which has been already opened, to register with the brake cylinder port. This permits auxiliary reservoir air to flow into the brake cylinder, pushing its piston out and applying the brake, and is called *Service* position.

If you make a continuous reduction in brake pipe pressure, auxiliary reservoir air will continue to flow into the brake cylinder, its pressure reducing at the same rate as the brake pipe, until the auxiliary reservoir and brake cylinder pressures become equal, or "equalize." After the pressures have equalized in this way, air will no longer flow out of the reservoir into the brake cylinder because there is no longer any difference of pressure to cause a flow. Consequently, you cannot increase brake cylinder pressure by further reducing the brake pipe pressure below the "equalizing point" and to do so, therefore, would be a waste of air.

If you make a brake pipe reduction less than sufficient to produce equalization, the auxiliary reservoir air will flow into the brake cylinder only until the pressure on the auxiliary reservoir side of the piston becomes slightly less than that on the brake pipe side. The higher brake pipe pressure will then move the piston and graduating valve back toward the lower auxiliary reservoir pressure until the piston stem butts against the slide valve. The slide valve offers sufficient resistance to stop

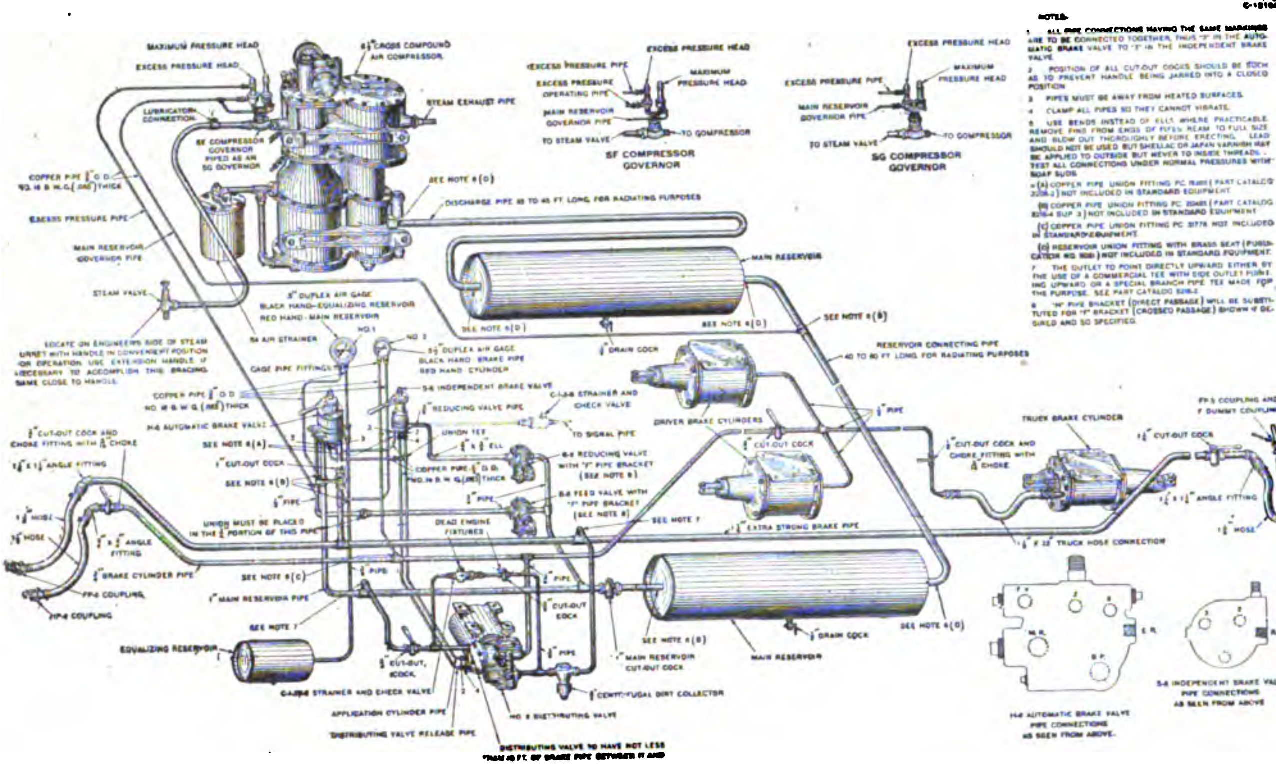


FIG. 355. Piping Diagram of No. 6 ET Locomotive Brake Equipment.

further movement of the piston and graduating valve. This is called *Lap* position. In this position the graduating valve closes the service port in the slide valve and prevents further flow of air from the auxiliary reservoir to the brake cylinder. The parts remain in this position until either the brake pipe or auxiliary reservoir pressure is changed.

If you further reduce brake pipe pressure, the auxiliary reservoir pressure will again move the piston and graduating valve over to *Service* position. This will allow additional air to flow into the brake cylinder and increase the pressure therein. In this way the brake cylinder pressure may be increased in steps or graduations until a full application (equalization) is obtained. This is called a "graduated application."

Suppose that you have not reduced brake pipe pressure below the equalizing point. If now you increase brake pipe pressure slightly, the piston and attached parts will be moved back to *Release and Charging* position. This movement opens the brake cylinder port to the atmosphere through the exhaust port, thus releasing the brakes, and also opens the feed groove, which permits the auxiliary reservoir to be recharged from the brake pipe.

If you have reduced brake pipe pressure below the equalizing point, you must increase brake pipe pressure above the equalizing point or above the pressure remaining in the auxiliary reservoir before the parts will move to *Release* position.

When you reduce the pressure on the brake pipe side of the piston faster than the pressure in the auxiliary reservoir can reduce by flowing into the brake cylinder through the service port, the higher auxiliary reservoir pressure forces the piston with its valves over to *Emergency* position, compressing the graduating spring. In this position auxiliary reservoir air flows past the end of the slide valve and through the brake cylinder port into the brake cylinder. This gives a quicker application of the brakes service port, as in *Service* position. This is called *Emergency* than when the auxiliary reservoir air flows through the small position.

The fundamental functions of the simplest form of triple valve device may then be briefly stated as follows:

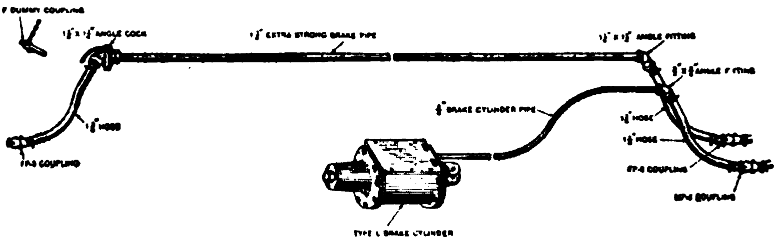


FIG. 356. Piping Diagram of Tender Brake Equipment.

1. When charging and maintaining the pressure in the brake system;
   - a. to permit air to flow from the brake pipe to the auxiliary reservoir.
   - b. to prevent air from flowing from the auxiliary reservoir to the brake cylinder.
   - c. to keep the brake cylinder open to the atmosphere.
2. When applying the brakes;
   - a. to close communication from the brake pipe to the auxiliary reservoir.
   - b. to close communication from the brake cylinder to the atmosphere.
   - c. to permit air to flow from the auxiliary reservoir to brake cylinder.
3. When holding brakes applied;

   to close all communication between the brake pipe, auxiliary reservoir, brake cylinder and atmosphere.

4. When releasing the brakes and recharging the system;
   a. to open communication from the brake cylinder to atmosphere.
   b. to permit flow of air from the brake pipe to the auxiliary reservoir.
   c. to prevent air from flowing from the auxiliary reservoir to the brake cylinder.

The brake pipe reduction may be caused by an opening in the brake pipe or its connections to the atmosphere and may be *intentional,* as through the engineman's brake valve or the conductor's valve, or may be *accidental* as from a burst hose or broken pipe.

All air brake devices having pistons and slide valves operate on the same principle, as just described. However, in some valves used for controlling the application and release of the brakes, as the distributing valve used with the No. 6 ET equipment, instead of the main piston and slide valve directly controlling the admission of air to and exhaust from the brake cylinder, they control the movement of other pistons and valves to accomplish the same purpose.

After you have a clear understanding of the operation of the simplest form of triple valve device you are now ready to go on to your study of the No. 6 ET Locomotive Brake Equipment.

No. 6 Distributing Valve.

The distributing valve is a portion of the ET equipment and has the following features:

(a) The locomotive and tender brakes may be partially or completely applied or released, either with or independent of the train brakes;

(b) The correct locomotive and tender brake cylinder pressure is obtained regardless of piston travel and automatically maintained regardless of leakage;

(c) Gives high locomotive and tender brake cylinder pressure, and therefore short stops with emergency application.

You will recognize the No. 6 distributing valve by referring to Figure 357. This valve consists of two portions bolted together; (a) the valve proper and (b) the cylindrical body or reservoir portion. By referring to Figure 358 you will observe that it has five pipe connections, three on the left of the reservoir portion shown by dotted circles, and two on the right. Of the three on the left, the upper one connects to main reservoir pressure; the middle one is known as the *application cylinder pipe* and leads to the independent and automatic brake valves; and the lower one is known as the *distributing valve release pipe* and leads to the automatic brake valve through the independent brake valve when its handle is in *Running* position. Of the two on the right, the lower leads to the brake pipe; and the upper is the brake cylinder pipe which leads to all the brake cylinders on the engine and tender.

FIG. 357. No. 6 Distributing Valve.

Referring to Figure 359, the upper piston with its attached valves is called the *application portion* and the lower piston with its attached parts, etc., the *equalizing portion.* The reservoir portion contains two chambers, one of which is called the *pres-*

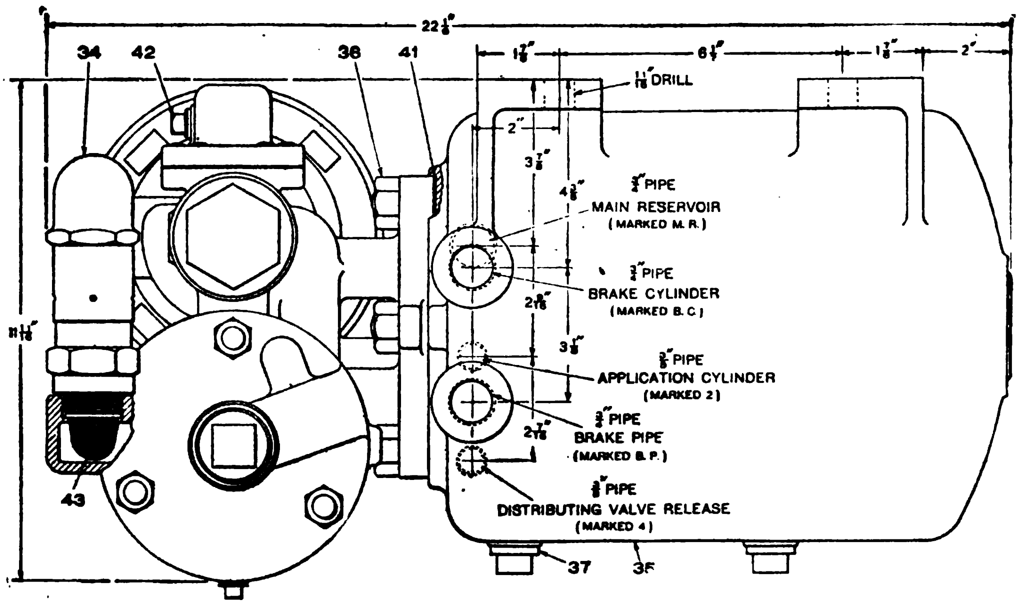


FIG. 358. Outline View of No. 6 Distributing Valve, Showing Pipe Connections.

sure chamber while the other is called the *application chamber.* The equalizing portion corresponds to the simple triple valve device which you have just studied while the pressure chamber corresponds to the auxiliary reservoir. The function of the equalizing portion is to control the pressure in the application chamber

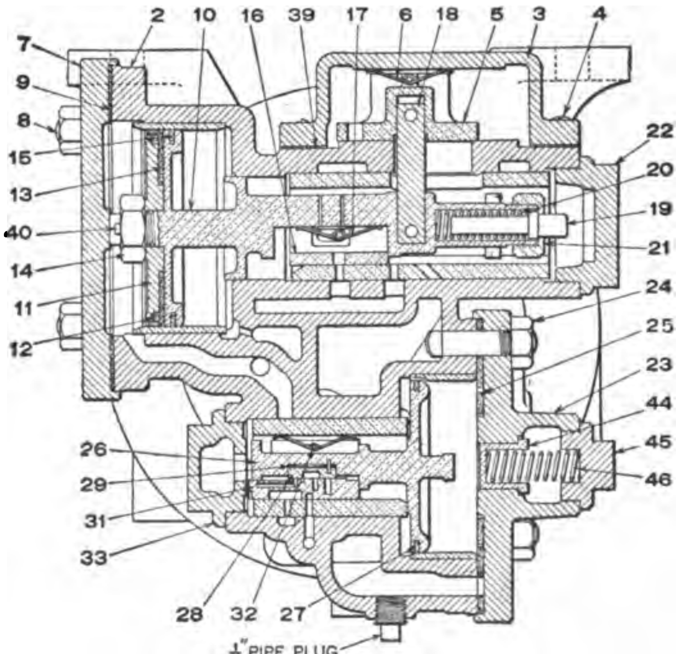


Fig. 359. Sectional View of No. 6 Distributing Valve.

and the connected application cylinder of the application portion and thereby control the flow of main reservoir air into the locomotive and tender brake cylinders and from these cylinders to the atmosphere. The equalizing portion operates only in automatic applications. In independent applications, the locomotive and tender brake pressure is controlled by directly controlling the pressure in the application cylinder by means of the independent brake valve.

Automatic Operation of Distributing Valve.

Figure 361 is what is called a "diagrammatic" view of the distributing valve; that is, a view in which the arrangement of ports and connections is made as simply as possible without regard to the actual shape and design of the distributing valve. While this view shows the parts in *Release* position, you should be able, by carefully studying the following description of operation, to understand what ports are connected in every position.

Figure 362 shows the ports and cavities in the equalizing valve and its seat and in the graduating valve. The reference shown for each port in Figure 362 is the same as for the corresponding port in Figure 361. This has been done in order that in studying the following description of operation you can refer from Figure 361 to Figure 362 and see what the actual location of the ports is. Port *h* leads to the application cylinder, automatic brake valve and independent brake valve; port *w* leads to the application chamber; port *i* leads to the distributing valve release pipe; and port *l* leads to the safety valve.

Charging.

You will observe from Figure 361 that chamber *p* is connected to the brake pipe; therefore, brake pipe air flows through the feed groove *l'* over the top of equalizing piston 26 into the chamber above equalizing valve 31, and through port *o* to the pressure chamber, until the pressures on both sides of the piston are equal.

Service.

When you make a service application, you reduce the pressure in chamber *p*, causing a difference in pressure on the two sides of the equalizing piston 26, which results in the piston moving toward the right.

The first movement of the piston closes the feed groove, and at the same time moves the graduating valve until it uncovers the upper end of port *z* in the equalizing slide valve 31. As the piston continues its movement, the shoulder on the end of its stem engages the equalizing slide valve which is then also moved to the right until the piston strikes graduating sleeve 44. Graduating spring 55 prevents further movement. Port *z* in the equalizing slide valve then registers with port *h* in the seat and

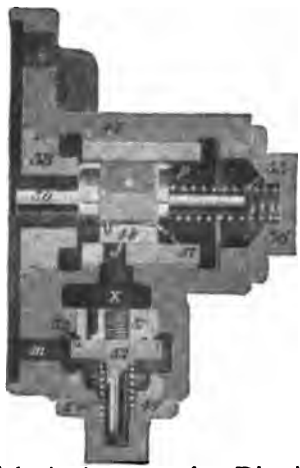


FIG. 360. Quick Action Cap for Distributing Valve.

cavity *n* in the equalizing slide valve connects ports *h* and *w* in the seat. As the equalizing slide valve chamber is always connected to the pressure chamber, air can now flow from the latter to both the application cylinder and application chamber. This pressure forces application piston 10 to the right causing exhaust valve 16 to close brake cylinder exhaust ports *e* and *d*, and to compress application piston graduating spring 20. Application valve 5 is connected to the piston stem through a pin 18 and this movement of the piston therefore causes the application valve to open its port and allow air from the main reservoirs to flow into chambers *b, b* and through passage *c* to the brake cylinders.

During the movement just described, cavity *t* in the graduating valve connects ports *r* and *s* in the equalizing slide valve, and ports *r* and *s* are brought to register with ports *h* and *l* in the seat. This results in connecting the application cylinder to the safety valve. The safety valve is set at 68 pounds and therefore limits the brake cylinder pressure to this amount.

The amount of pressure resulting in the application cylinder for a certain brake pipe service reduction, depends on what the volume of the pressure chamber is as compared with the volume of the application cylinder and its chamber. These volumes are such that with 70 pounds in the pressure chamber and nothing in the application cylinder and chamber, if they are allowed to remain connected by the ports in the equalizing valve, they will equalize at about 50 pounds.

Service Lap.

If the brake pipe reduction you make is not sufficient to cause a full service application, the conditions described above continue until the pressure in the pressure chamber is reduced enough below that in the brake pipe to cause piston 26 to force graduating valve 28 to the left until stopped by the shoulder on the piston stem striking the right-hand end of equalizing slide valve 31. This position is known as *Service Lap*. In this position, graduating valve 28 has closed port *z* so that no more air can flow from the pressure chamber to the application cylinder and chamber. It also has closed port *s*, cutting off the connection to the safety valve. The purpose of this is to prevent any possible leak in the latter from reducing application cylinder pressure for any reduction in this pressure would also result in a reduction of brake cylinder pressure. The flow of air past application valve 5 to the brake cylinders continues until the pressure in them is slightly greater than that in the application cyl-

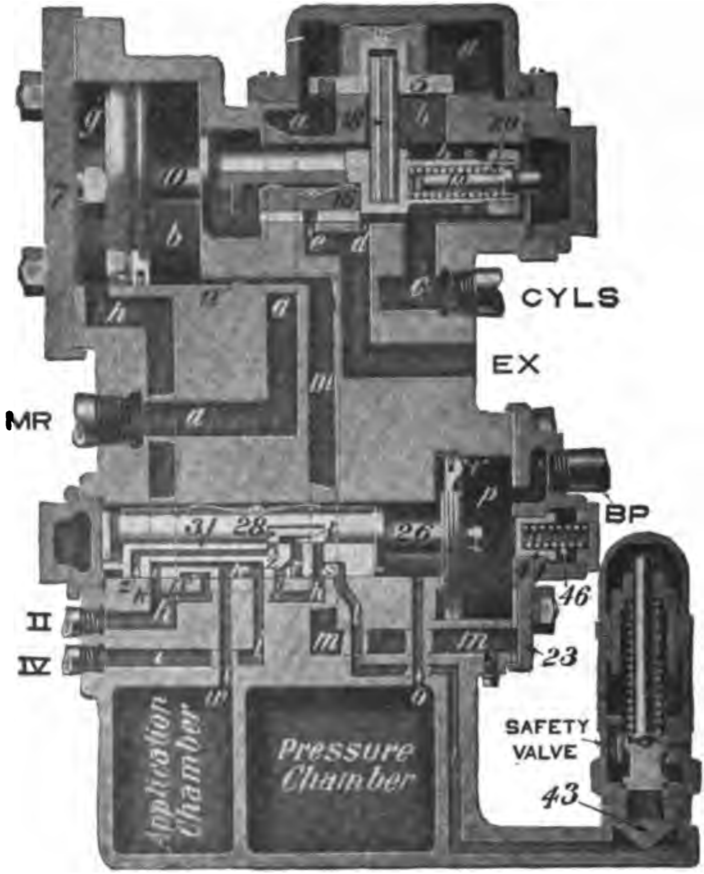


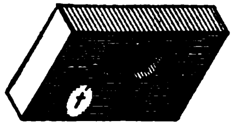
FIG. 361. Diagrammatic Section of Distributing Valve. Release Position.

inder, when the higher pressure and application piston graduating spring together force piston 10 to the left until port *b* is just closed. The piston is prevented from moving farther on account of the resistance of exhaust valve 16, and because the application piston graduating spring has expanded to its normal position. The brake cylinder pressure is then practically the same as that in the application cylinder and chamber.

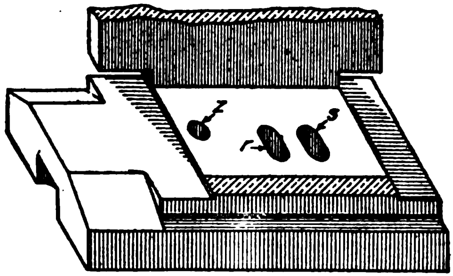
From the above description you will observe that application piston 10 has application cylinder pressure on one side, *g*, and the brake cylinder pressure on the other. When either pressure varies, the piston will move toward the lower. Consequently, if that in chamber *b* is reduced by brake cylinder leakage, the pressure maintained in the application cylinder *g* will force piston 10 to the right. This will open application valve 5 and again admit air from the main reservoirs to the brake cylinders until the pressure in chamber *b* is again slightly above that in the application cylinder *g* when the piston again moves back to lap position. In this way the brake cylinder pressure is always maintained equal with that in the application cylinder. This is the *pressure-maintaining* feature.

Release and Recharge.

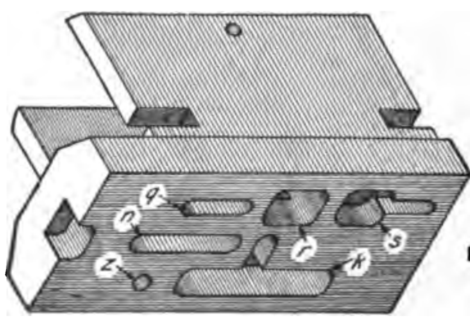
When you place the handle of the automatic brake valve in *Release* position, you increase the brake pipe pressure in chamber *p* above that in the pressure chamber causing the equalizing piston 26 to move to the left, carrying with it equalizing slide valve 31, and graduating valve 28 to *Release position*. The feed groove *l'* is now open, which permits the pressure in the pressure chamber to feed up until it is equal to that in the brake pipe, as before described. This action does not release the locomotive brakes because it does not discharge application cylinder pressure. The release pipe is closed by the rotary valve of the automatic brake valve and the application cylinder pipe is closed by the rotary valves of both brake valves. To release the locomo-



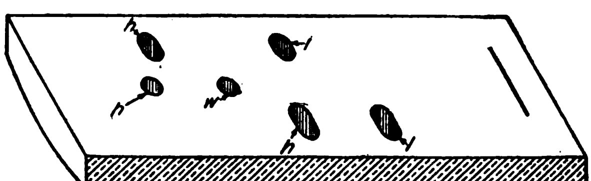
FACE OF GRADUATING VALVE



PLAN OF EQUALIZING SLIDE VALVE



FACE OF EQUALIZING SLIDE VALVE



EQUALIZING SLIDE VALVE SEAT

FIG. 362. Views of the Equalizing Valve, Graduating Valve and

tive brakes, the automatic brake valve must be moved to *Running* position. The release pipe is then connected by the rotary valve to the atmosphere, and as exhaust cavity *k* in the equalizing slide valve 31 connects ports *i*, *w* and *h* in the valve seat, the air in the application cylinder and chamber will escape. As this pressure reduces, the brake cylinder pressure will force application piston 10 to the left until exhaust valve 16 uncovers exhaust ports *d* and *e*, allowing brake cylinder pressure to escape, or in case of graduated release, to reduce in like amount to the reduction in the application cylinder pressure.

Emergency.

When you make a sudden and heavy brake pipe reduction, as by placing the automatic brake valve in Emergency position, the air pressure in the pressure chamber forces equalizing piston 26 to the right with sufficient force to compress graduating spring 46 and to seat against the leather gasket beneath cap 23. This movement causes equalizing slide valve 31 to uncover port *h* in the seat without opening port *w*, making a direct opening from the pressure chamber to the *application cylinder only*, so that they quickly become equalized. This volume, being small, and connected with that of the pressure chamber at 70 pounds pressure, equalizes at about 65 pounds. Also in this position of the automatic brake valve, a small port in the rotary valve allows air from the main reservoir to feed into the application cylinder pipe and thus into the application cylinder. The application cylinder is now connected to the safety valve through port *h* in the seat, cavity *q* and port *r* in the equalizing valve, and port *l* in the seat. Cavity *q* and port *r* in the equalizing valve are connected by a small port of such size that it permits the air in the application cylinder to escape through the safety valve at the same rate that the air from the main reservoirs can supply it by feeding through the rotary valve of the automatic brake valve. This prevents the pressure from rising above the adjustment of the safety valve.

In High-Speed Brake Service, the feed valve is regulated for 110 pounds brake pipe pressure instead of 70, and main reservoir pressure is 130 or 140 pounds. Under these conditions an emergency application raises the application cylinder pressure to about 93 pounds, but the passage between cavity *q* and port *r* is so small that the flow of application cylinder pressure to the safety valve is just enough greater than the supply through the brake valve to decrease that pressure in practically the same time and manner as is done by the high-speed reducing valve until it is approximately 75 pounds. The reason why the pressure in the application cylinder, pressure chamber and brake cylinders does not fall to 68 pounds, to which pressure the safety valve is adjusted, is because the inflow of air through the brake valve with the high main reservoir pressure used in high-speed service is equal, at 75 pounds, to the outflow through the small opening to the safety valve. This is done to get a shorter stop in emergency. The application portion of the distributing valve operates similarly, but more quickly than in service application.

The movable parts of the valve remain in *emergency* position until the brake cylinder pressure slightly exceeds the application cylinder pressure. Then the application piston and application valve move back to the position known as *Emergency Lap* in which the application valve closes port *b*.

The release after an emergency is brought about by the same manipulation of the automatic brake valve as that following service application, but the effect on the distributing valve is somewhat different. When the brake pipe pressure in chamber *p* is increased sufficiently, this pressure, together with the pressure of spring 46, forces the equalizing piston, equalizing valve, and graduating valve to release position. This connects the application chamber to the application cylinder through port *w*, cavity *k*, and port *h*. The pressure in the application chamber at this time is zero; that in the application cylinder is emergency pressure. The pressure in the application cylinder at once expands into the application chamber until these pressures are equal, which results in the release of brake cylinder pressure until it is slightly less than that in application cylinder and chamber. Consequently, in releasing after an emergency (using the *Release* position of the automatic brake valve), the brake cylinder pressure will *automatically* reduce to about 15 pounds, where it will remain until the automatic brake valve handle is moved to *Running* position.

If the brakes are applied by a conductor's valve, a burst hose, or parting of train, the movement of equalizing slide valve 31, breaks the connection between ports *h* and *i* through cavity *k*, so that the brakes will apply and remain applied until the brake-pipe pressure is restored. The handle of the automatic brake valve should be immediately moved to either *Emergency* or *Lap* position, depending upon whether the locomotive is operating in passenger or freight service, to prevent a loss of main reservoir pressure.

Independent Brake Operation.

Independent Application.

When you move the handle of the Independent Brake Valve to either *slow* or *quick application* position, you permit air from the main reservoir, limited by the reducing valve to a maximum of 45 pounds, to flow to the application cylinder, forcing application piston 10 to the right. This movement causes application valve 5 to open its port and allow air from the main reservoirs to flow into chambers *b, b* and through passage *c* to the brake cylinders, as in an automatic application, until the pressure slightly exceeds that in the application cylinder. The application piston graduating spring 20 and higher pressure then force application piston 10 to the left until application valve 5 closes its port. The piston is prevented from moving farther by the resistance of exhaust valve 16, and the application piston graduating spring having expanded to its normal position. This position is known as *Independent Lap.*

It will be seen that whatever pressure exists in the application cylinder will be maintained in the brake cylinders by the "pressure maintaining" feature already described.

Since the supply pressure to the independent brake valve is fixed by the regulation of the reducing valve to 45 pounds, this is the maximum cylinder pressure that can be obtained.

Independent Release.

Automatic Brakes Released.

To release the locomotive brakes, you place the handle of the Independent Brake Valve in *Running* position. This makes a direct opening from the application cylinder to the atmosphere. As the application cylinder pressure escapes, brake cylinder pressure in chamber *b* moves application piston 10 to the left, causing exhaust valve 16 to open exhaust ports *e* and *d,* thereby allowing brake cylinder pressure to discharge to the atmosphere.

If the independent brake valve is returned to *Lap* before all of the application cylinder pressure has escaped, the application piston 10 will return to *Independent Lap* position as soon as the brake cylinder pressure is reduced a little below that remaining in the application cylinder. This movement closes exhaust ports *e* and *d*, and holds the remaining pressure in the brake cylinders. In this way the independent release may be graduated as desired.

You may also release the locomotive brakes after making an automatic application by placing the Independent Brake Valve in *Release* position. This allows the air in the application cylinder to flow through the application cylinder pipe to the atmosphere which results in a release of the locomotive brakes, as just described. You will note that no change is made in the pressure in either the pressure chamber or brake pipe; consequently, the equalizing piston does not move until you make a release through the automatic brake valve.

An independent release of locomotive brakes may also be made in the same manner after an emergency application by the automatic brake valve. However, you should observe that, in this position, the automatic brake valve will be supplying the application cylinder through the maintaining port in the rotary valve; consequently, you must *hold* the handle of the independent brake valve in *Release* position to prevent the locomotive brakes from re-applying, so long as the handle of the automatic brake valve remains in *Emergency* position. The equalizing portion of the distributing valve will remain in the *Emergency* position, while the application portion will go to *Release* position.

Quick Action Cylinder Cap.

The equalizing portion of the distributing valve, as already described, acts on the same principle as the plain triple valve. There are, however, conditions under which it is advisable to have it correspond to the quick-action triple valve; that is, vent brake pipe air into the brake cylinders in an emergency application. To obtain this, the cylinder cap 23, Fig. 361, is replaced by the Quick Action Cylinder Cap illustrated in Fig. 360.

In an emergency application, as equalizing piston 26 moves to the right and seals against the gasket, the knob on the piston strikes the graduating stem 50, Fig. 360, causing it to compress graduating spring 55 and move emergency valve 48 to the right, opening port *j*. Brake pipe air in chamber *p* flows to chamber *x*, pushes down check valve 53, and passes to the brake cylinders through port *m* in the cap and distributing valve body. When the brake cylinders and brake pipe equalize, check valve 53 is forced to its seat by spring 54. This prevents air in the brake cylinders from flowing back into the brake pipe. When a release of the brakes occurs and piston 26 is moved back to its normal position, spring 55 forces graduating stem 50 and emergency valve 48 back to the position shown in Fig. 360.

In all other respects, the operation of a distributing valve having this cap is exactly as described before.

Type H-6 Automatic Brake Valve.

The type H-6 automatic brake valve is used to operate locomotive, tender and train brakes in connection with an S-6 independent brake valve for operating the locomotive and tender brake only. Figure 363 shows the familiar general appearance of the H-6 brake valve. Figures 364 and 365 show two actual sections of the brake valve to illustrate its construction. Figure 364 shows a horizontal section of the brake valve as it would appear if the top case and rotary valve were removed; in other words, this view shows the rotary valve seat. It also shows that this brake valve has six positions, as follows: *Release, Running, Holding, Lap, Service,* and *Emergency.*

Figure 366 is a view showing the ports in the rotary valve. Ports *a*, *j* and *s* extend directly through the valve, port *s* connecting with a groove in the face. *F* is a cavity in the face of the valve. *X* is also a cavity in the valve, but connects with the exhaust opening *o* through a cored passage. Ports *h* and *t* also connect with exhaust opening *o* by means of a cored passage in the interior of the valve. *K* and *n* are cavities connected with each other by a small port. Valves recently manufactured have an additional cavity which does not exist in older valves. This is cavity *w*.

Referring now to Figure 364, *d* leads to the feed valve pipe; *b* and *c* lead to the brake pipe; *g* leads to chamber *d*. *EX* is the exhaust opening leading out at the back of the valve. *E* is the preliminary exhaust port and leads to chamber *D*. *R* is the warning port and leads to the exhaust. Port *p* leads to the compressor governor. *L* leads to the distributing valve release pipe. *U* leads to the application cylinder pipe. Valves recently manufactured have two additional ports which older valves do not have. These are ports *q* and *l′*. Port *q* leads to the side of the valve; it is tapped at this end so that the opening may be closed up by means of a plug, as shown in the small view marked "Section AA," Figure 365. Port *l′* connects with port *l* through a cored passage. This cored passage is tapped so that it may be closed by means of a plug.

We will now study the operation of the brake valve, taking up the handle positions in the order in which you most generally use them:

Charging and Release Position. When you place the handle in this position, you open a large and direct passage from the main reservoir to the brake pipe. This permits a rapid flow of air into the brake pipe to (a) charge the train brake system;

(b) quickly release and recharge the brakes; but (c) *not* release locomotive brakes, if they are applied.

When the handle is placed in this position, air flows from the main reservoirs to the chamber above the rotary valve, thence through port *a* in the rotary valve and port *b* in the seat to the brake pipe. This charges the entire system. At the

FIG. 363. H-6 Automatic Brake Valve.

same time port *j* registers with equalizing port *g*, which permits air at main reservoir pressure to flow to chamber *D* above the equalizing piston in the lower part of the brake valves. Chamber *D* is always connected with the equalizing reservoir; consequently, the equalizing reservoir is charged at the same time. The purpose of the equalizing piston and reservoir will be described later.

The small groove in the face of the rotary valve which connects with port *s* extends to port *p* in the valve seat, allowing main reservoir air to flow to the lower connection of the excess pressure head of the compressor governor. When main reservoir pressure becomes 20 pounds greater than the pressure the feed valve is set for, the compressor is stopped, as you will understand by studying the operation of the governor which is given later.

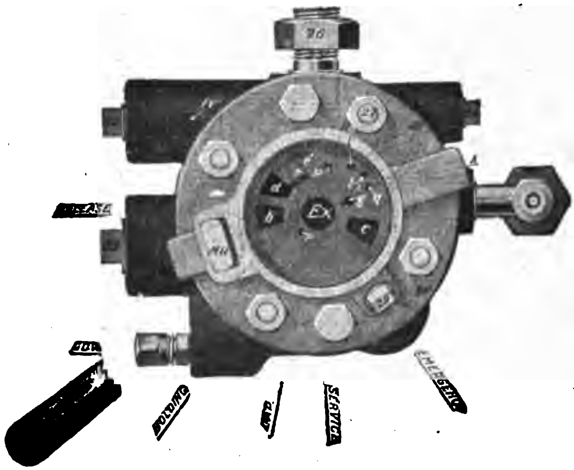


FIG. 364. Showing View of Rotary Valve Seat.

If you leave the handle in *Release* position, the brake system will be charged to main reservoir pressure. To avoid this, the handle must be moved to *Running* or *Holding* position. To prevent you from forgetting this, a small port discharges air from the feed valve pipe to the atmosphere in release position. Cavity *f* in the rotary valve connects port *d* with warning port *r* in the seat and allows a small quantity of air to escape into exhaust cavity EX, which makes sufficient noise to attract your attention to the position in which you have left the handle.

Running Position. This is the proper position of the handle (a) when the brakes are charged and ready for use; (b) when the brakes are not being operated; and (c) to release the locomotive brakes.

Cavity *f* in the rotary valve connects ports *b* and *d* in the seat, which gives a large direct passage from the feed valve pipe to the brake pipe. The brake pipe will then charge up as rapidly as the feed valve can supply the air, but cannot attain

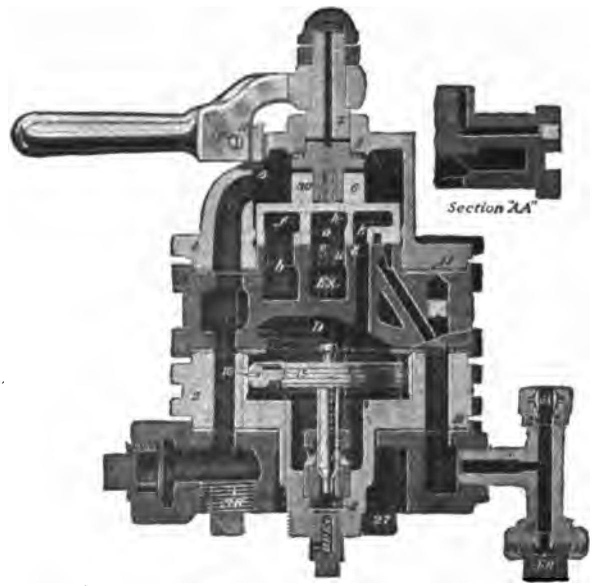


FIG. 365. Section of H-6 Automatic Brake Valve.

Connections: FV—Feed Valve Pipe; MR—Main Reservoir Pipe; Gov.—To Governor; III—Distributing Valve Release Pipe; EX—Emergency Exhaust; II—Application Cylinder Pipe; BP—Brake Pipe; GA—Large Duplex Air Gage; ER—Equalizing Reservoir; BP EX—Service Exhaust.

a pressure above that for which the feed valve is adjusted. Cavity *k* in the rotary valve connects ports *c* and *g* in the seat so that chamber *D* and the equalizing reservoir charge at the same rate as the brake pipe. The pressure on the two sides of the equalizing piston are thereby kept equal. Port *s* in the rotary valve registers with port *p* in the seat, permitting air at main reservoir pressure, which is present at all times above the rotary valve, to pass to the lower connection of the excess pressure head of the governor. The distributing valve release pipe is connected to the atmosphere through the independent brake valve and ports *l, h, o* and EX in the automatic brake valve.

If you have the brake valve in *Running* position when cutting in uncharged cars, or if, after a heavy brake application and release you return the handle of the automatic brake valve to *Running* position too soon, the governor will stop the compressors until the difference between the hands on the large duplex gage is less than 20 pounds. The compressors stopping from this cause calls your attention to the seriously wrong operation on your part, as *Running* position results in delay in charging and is liable to cause some brakes to stick. You should use *Release* position until all brakes are released and nearly charged.

Service Position. When you use this position you produce a reduction of brake pipe pressure to cause a service application.

Port *h* in the rotary valve registers with preliminary exhaust port *e* in the seat, which allows air from chamber D and the equalizing reservoir to escape to the atmosphere through cavities *o* in the rotary valve and EX in the seat. Port *e* is made small so as to make the pressure in the equalizing reservoir and chamber D fall gradually. This at once reduces the pressure of the air on the top of the equalizing piston below that in the brake pipe under the piston. The higher pressure then forces the piston upward, raising the equalizing discharge valve, as the end of the piston stem is called, from its seat and thereby permitting air from the brake pipe to flow to the atmosphere gradually through the service exhaust fitting marked B.P. EX in Figure 365. You will remember that the equalizing reservoir is permanently connected with chamber D so that the reduction in pressure in chamber D also means a corresponding reduction in equalizing reservoir pressure. It will now be clear to you that the purpose of the equalizing reservoir is to add volume to chamber D. Without the equalizing reservoir this volume is so small that, when you move the brake valve handle to *Service* position, its pressure would drop to zero almost in-

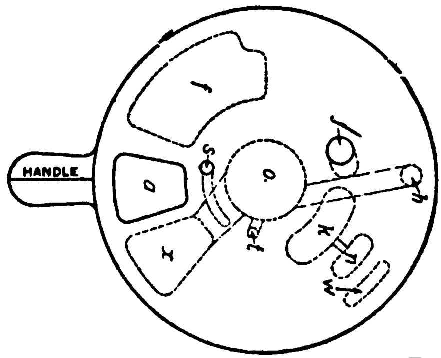


FIG. 366. Rotary Valve of H-6 Automatic Brake Valve.

stantly. You would therefore find it very difficult to make a moderate brake pipe reduction and practically impossible to obtain the exact amount of reduction you might desire.

Service Lap. When you have reduced the pressure in chamber D the desired amount, the handle should be moved to *Lap* position, thus stopping any further reduction in that chamber. Whether the flow of air from the brake pipe ceases at once or continues for a period of time after the handle is placed in *Lap* position depends upon whether the train is a long or a short one. With a short train the total volume of air in the brake pipe is not very great, so that it can escape through the service exhaust nearly as fast as the air in chamber D and the equalizing reservoir is flowing out through the preliminary exhaust port *c*. Thus the pressure below the equalizing piston is falling at about the same rate as that above. In such a case, as soon as the pressure in chamber D ceases to fall, the brake pipe pressure below the equalizing piston becomes slightly less than that above the piston. The higher pressure then forces the piston downward, seating the discharge valve and preventing farther discharge of brake pipe air.

On a long train, however, the total volume of air in the brake pipe is large, and therefore the pressure below the equalizing piston falls at a much slower rate than that above it. In such a case, air continues to escape from the brake pipe after the handle has been placed in *Lap* position for a period of time depending upon the length of train, until the brake pipe pressure has been reduced slightly below that in chamber D. The equalizing piston is then forced downward and the service exhaust opening closed, as already explained. You will observe from this description that the amount of reduction in the equalizing reservoir determines that in the brake pipe, regardless of the length of the train.

The gradual reduction of brake pipe pressure is to prevent quick action, and the gradual stopping of this discharge is to prevent the pressure at the head end of the brake pipe being built up by the air flowing from the rear, which might cause some of the head brakes to "kick off."

Release Position. This position, which is used for releasing the train brakes after an application without releasing the locomotive brakes, has already been described under "Charging and Release Position" on page 830. When you have increased the brake pipe pressure sufficiently to release the brakes, you should move the handle to either *Running* or *Holding* position—to *Running* position when you desire to release the locomotive brakes and to *Holding* position when you still desire to hold locomotive brakes applied.

Holding Position. This position is so named because the locomotive brakes are held applied while the train brakes are being released and their auxiliary reservoirs recharged to feed valve pressure.

All ports register as in *Running* position with the exception that port *l* which leads to the distributing valve release pipe is closed. Any air pressure which may be in the application cylinder of the distributing valve from a previous service application is then held bottled there, preventing the air from releasing from the locomotive brake cylinders. Therefore, the only difference between *Running* and *Holding* positions is that in *Running* position the locomotive brakes are released, while in *Holding* position they are held applied.

Emergency Position. This position is used (a) when the most prompt and heavy application of the brakes is required, and (b) to prevent loss of main reservoir air and insure that the brakes remain applied in the event of a burst hose, a break-in-two, or the opening of a conductor's valve.

In this position port *x* in the rotary valve registers with port *c* in the seat, making a large and direct opening between the brake pipe and the atmosphere through cavity *o* and EX. This causes a sudden and heavy discharge of brake pipe air. At the same time port *t* registers with port *g*, allowing the air in the equalizing reservoir to flow through the ports named to the exhaust, *o*, and the atmosphere, thus reducing the pressure in the equalizing reservoir to zero during an emergency application of the brakes. Also in this position port *j* registers with a groove connecting with cavity *k*, permitting main reservoir air to pass through ports *n* and *u* to the application cylinder pipe.

This results in maintaining locomotive and tender brake cylinder pressure against leakage, as described under "Emergency" operation of the distributing valve on page 824.

As previously stated, type H-6 automatic brake valves recently manufactured have the addition of ports *l* and *q* in the rotary valve seat and small cavity *w* in the rotary valve. The purpose of these additional ports and cavity is to provide for releasing the locomotive and tender brakes in either or both *Lap* or *Holding* positions if it is thought desirable, instead of having them applied, which is the usual operation.

The standard operation in making the first release of a *two application* stop is to move the brake valve handle to *Release* position, then quickly back to *Running,* then to *Lap* and thence to *Service* position. The purpose of moving to *Running* position is to partially release the locomotive brakes so as to prevent a much higher pressure accumulating in the locomotive brake cylinders on the second application than in the car brake cylinders; also to permit the brake pipe and equalizing reservoir pressures to equalize. The purpose of moving to *Lap* position following *Running* is to cut off the supply of air to the brake pipe and to permit the brake pipe and auxiliary reservoir pressures to equalize so that the triple valves will respond to the brake pipe reduction practically as soon as the automatic brake valve handle is placed in *Service* position for the second application. In order to get sufficient time for this equalization of pressures and *consequent* prompt response of the brakes to the second application, some enginemen do not stop the brake valve in *Running* position but move direct from *Release* to *Lap* position between applications, with the result that the locomotive brakes, by not being released, develop a high braking force on the second application, with consequent liability of shocks and sliding wheels. To avoid this high braking force it is only necessary to remove the plug at the outer end of port *q* and then when you place the handle in *Lap* position the distributing valve release pipe will be connected to the atmosphere through ports *l* and *q* and cavity *w*. If it is not desired to hold locomotive and tender brakes applied in *Holding* position, it is only necessary to remove the plug from the cored passage connecting ports *l* and *l'*. Then when the handle is placed in *Holding* position, the distributing valve release pipe is connected to the brake valve exhaust through ports *l* and *l'* and EX.

Leather washer 8 prevents air in the rotary valve chamber from leaking past the rotary valve key to the atmosphere. Spring 30 keeps the rotary valve key firmly pressed against washer 8 when no main reservoir pressure is present. The handle contains latch 11, which fits into notches in the quadrant of the top case, so located as to indicate the different positions of the brake valve handle. A latch spring 10 within the handle forces the latch against the quadrant with sufficient pressure to indicate each position.

To remove the brake valve, close the double-heading cock, place the brake valve handle in *Release* position and then close the main reservoir cut-out cock. (This is to prevent the slide valve of the feed valve and the rotary valve of the brake valve being lifted from their seats.) Then take off the nuts from the four long bolts. To take the valve proper apart, remove the two cap screws from the top case.

S-6 Independent Brake Valve.

By referring to Figure 367 you will recognize the S-6 independent brake valve for operating locomotive and tender brakes only. Figure 368 shows that this brake valve has five positions, as follows: *Release, Running, Lap, Slow Application* and *Quick Application.* You will note that this valve has no equalizing piston or reservoir.

Referring to Figure 368, port *b* in the seat leads to the Reducing Valve Pipe. Port *a* leads to the Distributing Valve Release Pipe which connects to the distributing valve. Port *c* leads to the other portion of the release pipe which connects

FIG. 367. S-6 Independent Brake Valve.

to the automatic brake valve at III. (Fig. 365.) Port *d* leads to the application cylinder pipe which connects to the distributing valve and the H-6 brake valve. Port *h* in the center is the exhaust port leading directly down to the atmosphere. Port *k* is the warning port connecting with the atmosphere.

Figure 369 is a view showing the ports in the rotary valve. Exhaust cavity *g* in the rotary valve is always in communication at one end with exhaust port *h*. Groove *e* in the face of the valve communicates at one end with a port which runs through the valve. This groove is always in communication with a groove in the seat connecting with supply port *b*, and through the opening just mentioned air is admitted to the chamber above the rotary valve, thus keeping it to its seat. Port *m* connects by a small hole with groove *e*; *f* is a groove in the

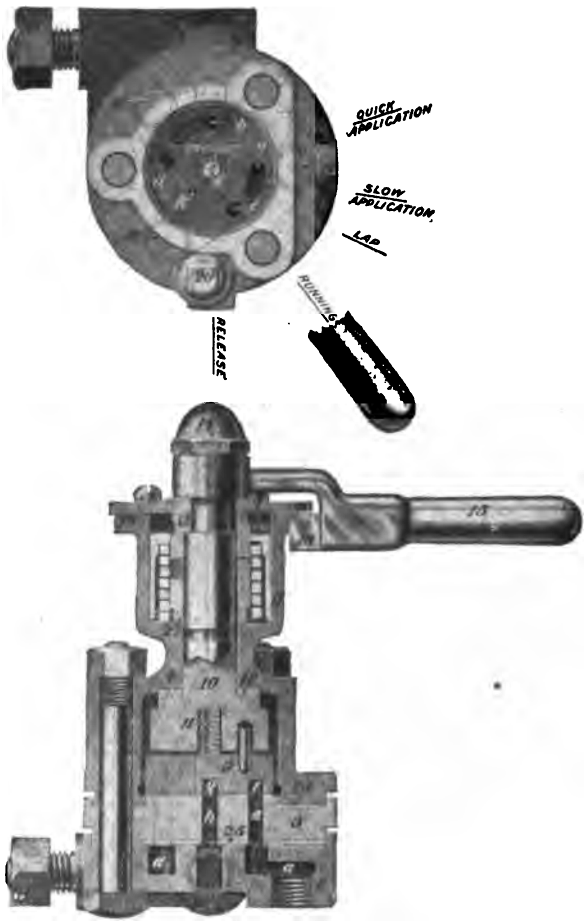


FIG. 368. S-6 Independent Brake Valve.

Connections: RV—Reducing Valve Pipe; EX—Exhaust; IV—Distributing Valve Release Pipe to the Distributing Valve; III—Distributing Valve Release Pipe to the Automatic Brake Valve; II—Application Cylinder Pipe.

face of the rotary valve; *l* is the warning port extending through the rotary valve.

We will now take up the handle positions and study the flow of air that takes place.

Running Position. This is the position that you should carry the independent brake valve in at all times when the independent brake is not in use. Groove *f* in the rotary valve connects ports *a* and *c* in the seat, thus connecting the application cylinder of the distributing valve to port *l* of the automatic brake valve, through the distributing valve release pipe. The distributing valve can therefore be released by the automatic brake valve.

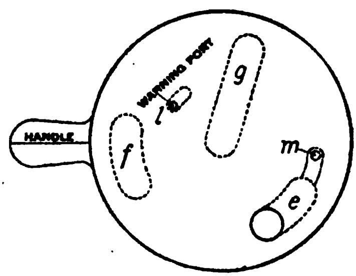


FIG. 369. Rotary Valve of S-6 Independent Brake Valve.

If the automatic brake valve handle is in *Running* position and the independent brakes are being operated, they can be released by simply returning the independent valve to *Running* position, as the application cylinder pressure can then escape through the release pipe and the automatic brake valve.

Slow Application Position. To apply the independent brake lightly or gradually, move the brake valve handle to the *Slow Application* position. In this position port *e* registers with port *b,* allowing main reservoir air that has been reduced to 45 pounds by passing through the reducing valve, to flow through ports *b, e,* and the restricted tail-port *m* to port *d.* It then passes through the application cylinder pipe to the application cylinder of the distributing valve, which causes the brakes to apply in the manner which you have already studied under the operation of the distributing valve.

Quick Application Position. To obtain a quick application of the independent brake, move the brake valve handle to *Quick Application* position. Groove *e* then connects ports *b* and *d* directly, making a large opening between them and allowing supply air (at 45 pounds pressure) to flow rapidly from the reducing valve pipe to the application cylinder of the distributing valve. Since the supply pressure to the independent brake valve is fixed by the adjustment of the reducing valve to 45 pounds, this is the maximum cylinder pressure that can be obtained.

Lap Position. You should use this position to hold the independent brake applied after the desired pressure is obtained in the brake cylinders. In this position port *d* leading to the application cylinder pipe is blanked by the rotary valve. This causes the application piston of the distributing valve to move to *Lap* position, as previously described. The brakes are thus held applied until a further application or a release is made.

Release Position. If the brakes have been applied throughout the train by means of the automatic brake valve and it is desired to release the locomotive and tender brakes only, the handle of the independent brake valve should be placed in *Release* position. In this position, cavity *g* registers with port *d,* which allows the air in the application cylinder of the distributing valve to flow through the application cylinder pipe, ports *d, g* and *h,* to the atmosphere, thereby releasing the locomotive and tender brakes.

By referring to Figure 368, you will note that this brake valve has what is called a "return spring" (reference No. 6). The purpose of this return spring is to automatically move the handle 15 from the *Release* to the *Running* position, or from the *Quick Application* to the *Slow Application* position, as soon as you let go of it. The automatic return from *Release* to *Running* position is to prevent your leaving the handle in *Release* position, and thereby make it impossible to operate the locomotive brake with the automatic brake valve. The action of the spring between the *Quick Application* and *Slow Application* position serves to make the latter more prominent, so that in rapid movement of the valve you will be less likely to unintentionally pass over to the *Quick Application* position, thereby obtaining a heavy application of the locomotive brake when you desire only a light one. As a warning in case of a broken return spring, air is allowed to escape from the reducing valve pipe to the atmosphere through warning port in the rotary valve and port *k* in the seat with sufficient noise to attract your attention.

B-6 Feed Valve.

The feed valve regulates the pressure in the feed valve pipe, and in the brake pipe when the handle of the automatic brake valve is in *Running* or in *Holding* positions, these two pipes being then connected through the brake valve. It is connected to a pipe bracket located in the piping between the main reservoir and the automatic brake valve. The B-6 feed valve is illustrated in Figures 370, 371 and 372, the latter two being diagrammatic views.

This device consists of two sets of parts, the supply and regulating. The supply parts, which control the flow of air through the valve, consist of the supply valve 7 and its spring 8; the supply valve piston 6 and its spring 9. The regulating parts consist of the regulating valve 12, regulating valve spring 13, diaphragm 17, diaphragm spindle 18, regulating spring 19, and regulating hand wheel 20.

Main reservoir air enters through port *a*, *a* to the supply valve chamber B, forces supply valve piston 6 to the left, compresses piston spring 9 and causes the port in supply valve 7 to register with port *c* (See Fig. 372). This permits air to pass through ports *c* and *d* to the feed valve pipe at FVP, and through port *e* to diaphragm chamber L.

Regulating valve 12 is then open and port K connects chamber G, on the left of piston 6, to the feed valve pipe through passage *h*, chamber L, and passages *e*, *d*, *d*. While the regulating valve is open, air feeding by the piston 6 cannot accumulate in chamber G above feed valve pipe pressure, but when regulating valve 12 is closed, the pressure on the left of piston 6 quickly rises to the main reservoir pressure on the right and piston spring 9 forces piston 6 and supply valve 7 to the right, closes port *c*, and stops the flow to the feed valve pipe.

The regulating valve is operated by diaphragm 17. When the pressure of regulating spring 19 on its right is greater than the feed valve pressure in chamber L on its left, it holds regulating valve 12 open. This causes the supply valve to admit air to the feed valve pipe. When the feed valve pipe pressure in chamber L is greater than that of the regulating spring 19, the diaphragm allows regulating valve 12 to close. This causes the supply valve to stop admitting air to the feed pipe.

As explained on page under *Release* position of the H-6 Automatic Brake Valve, in *Release* position the warning port is supplied from the feed valve pipe. This insures that the excess pressure governor head will regulate the brake pipe pressure in *Release* position even though the feed valve is leaking slightly, but not enough to be otherwise detrimental.

The distinguishing feature of this type of feed valve is the duplex adjusting arrangement by which it eliminates the necessity of the two feed valves in high and low pressure service. The spring box 15 has two rings encircling it, which are split through the lugs marked 21 and 22 in the diaphragm, and which

FIG. 370. B-6 Feed Valve and Bracket, Complete.

may be secured in any position by the screw 23. The pin forming part of adjusting handle 20 limits the movement of the handle to the distance between stops 21 and 22. When testing the valve, stop 21 is located so that the compression of spring 19 will give the desired high brake pipe pressure, and stop 22 so that the spring compression is enough less to give the low brake pipe pressure. Thereafter, by simply turning handle 20 until its pin strikes either one of these stops, you may change the regulation of the feed valve from one brake pipe pressure to the other.

To adjust this valve, slacken screws 23, which allows stops 21 and 22 to turn around spring box 15. Turn adjusting handle 20 until the valve closes at the lower brake pipe pressure desired, when stop 22 should be brought into contact with the handle pin. At this point it should be securely fastened by tightening screw 23. Then turn adjusting handle 20 until the higher adjustment is obtained, when stop 21 should be brought in contact with the handle pin and securely fastened.

When replacing this feed valve on its pipe bracket after removal, the gasket must always be in place between the valve and bracket, to insure a tight joint.

C-6 Reducing Valve.

This valve, illustrated in Fig. 373, is the well known feed valve that was used for many years in connection with the G-6 brake valve, but in this equipment is attached to a pipe bracket. The only difference between it and the B-6 feed valve just described is in the adjustment, it being designed to reduce main reservoir pressure to a single fixed pressure, which in this equipment is, as already stated, 45 pounds. To adjust this valve, remove the cap nut on the end of the spring box; this will expose the adjusting nut, by which the adjustment is made. It is called a "Reducing Valve" when used with the independent brake and air-signal systems, simply to distinguish it from the feed valve supplying the automatic brake valve.

Air Gages.

Two duplex gages form a part of the equipment. The two gages are connected as follows: Large Gage (No. 1)—Red Hand, to main reservoir pipe under the automatic brake valve; Black Hand, to equalizing reservoir tee of the automatic brake valve. Small Gage (No. 2)—Red Hand, to the brake cylinder pipe; Black Hand, to the brake pipe below the double heading cock.

The amount of reduction made during an automatic application is indicated by the black hand of the large gage. The

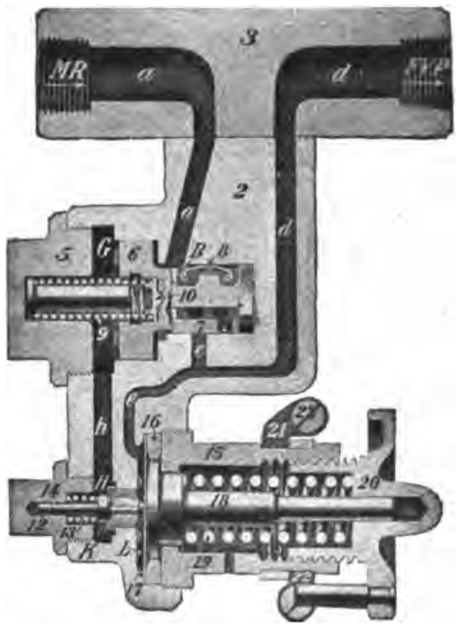


FIG. 371. Diagram of B-6 Feed Valve, Closed.

black hand of the small gage is to show the brake pipe pressure when the engine is second in double heading, or when a helper.

Type SF Compressor Governor.

The governor commonly used with the No. 6 ET Equipment is known as the *duplex*, or double top, "Type SF" governor. Its duty is to automatically control the steam supply that operates the air compressor so that the pressure in the main reservoir will not exceed a given figure. During most of the time on a trip, you carry the automatic brake valve in *Running* position, keeping the brakes charged. But little excess pressure is then needed and the governor regulates the main reser-

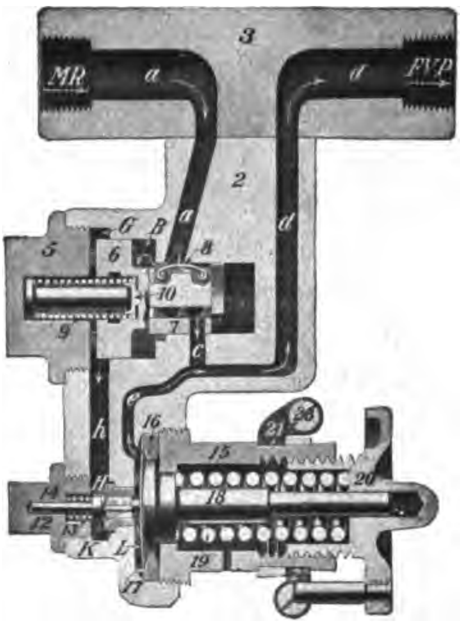


FIG. 372. Diagram of B-6 Feed Valve, Open.

voir pressure to about 20 pounds only above the brake pipe pressure, thus making the work of the compressor easier. On the other hand, when the brakes are applied (*Lap* position of the automatic brake valve, following the use of its *Service* position) a high main reservoir pressure is needed to insure their prompt release and recharge. Therefore, as soon as you place the handle in Lap, Service or Emergency positions, the governor allows the compressor to work freely until the maximum main reservoir pressure is obtained. Again, when the brake pipe pressure is changed from one amount to another by the feed valve, as where a locomotive is used alternately in high speed brake and ordinary service, the governor automatically changes the main reservoir pressure to suit, and at the same time maintains the other features just described.

Another important feature is that the governor connections to the brake valve permit you to raise and maintain the brake pipe pressure about 20 pounds above the feed valve regulation

FIG. 373. C-6 Reducing Valve.

before commencing and during the descent of steep grades, merely by the use of *Release* position of the automatic brake valve, which is the position that you should use during such braking.

The following will explain the construction and operation of the SF Governor. Fig. 374 shows a sectional view of this governor with steam valve 5 open. By reference to the piping diagram in Fig. 355, it will be seen that one steam connection leads to the boiler and the other to the compressor; MR leads to the main reservoir; ABV to the automatic brake valve; FVP to the feed valve pipe; W is the waste pipe connection. Steam enters at the connection marked "From Boiler" and passes by steam valve 5 to the connection to the compressor. The governor regulating head on the left is called the "excess pressure head," and the one on the right the "maximum pressure head." Air from the main reservoir flows

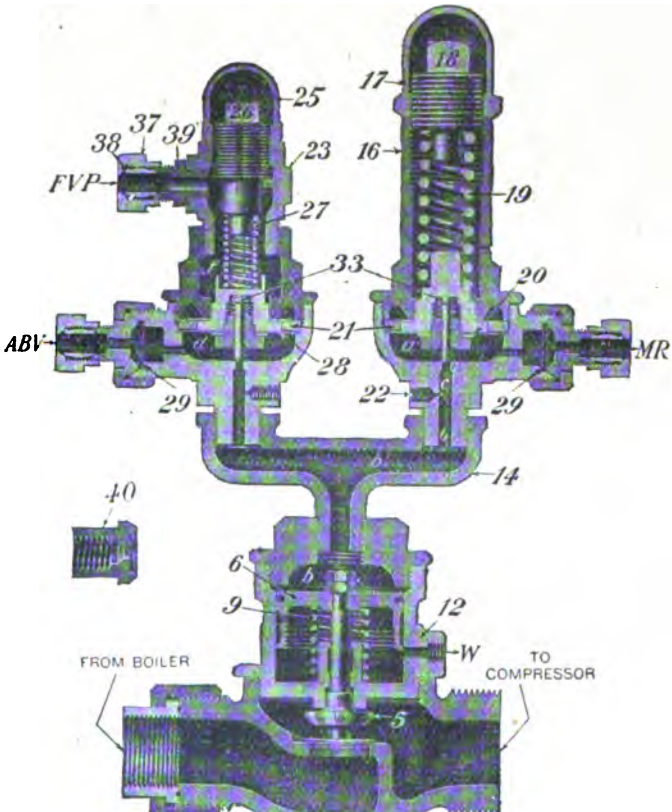


FIG. 374. Type SF Compressor Governor.

through the automatic brake valve (when the latter is in *Release, Running* or *Holding* position) to the connection marked ABV and into chamber *d* below diaphragm 28. Air from the feed valve pipe enters at the connection FVP to chamber *f* above diaphragm 28, adding to the pressure of regulating spring 27 in holding it down. As this spring is adjusted to about 20 pounds, this diaphragm will be held down until the main reservoir pressure in chamber *d* becomes slightly greater than the combined air and spring pressure acting on the top of the diaphragm. At such time diaphragm 28 will rise, unseat its pin valve, and allow air to flow to chamber *b* above the governor piston. This piston will thereby be forced downward, compressing its spring and restricting the flow of steam past steam valve 5 to the point where the compressor will just supply the leakage in the brake system. When main reservoir pressure in chamber *d* becomes reduced, the combined spring and air pressure above the diaphragm forces it down, seating its pin valve. As chamber *b* is always open to the atmosphere through the small vent port *c*, the air in chamber *b* above the governor piston will then escape to the atmosphere and allow the piston spring and steam pressure below valve 5 to raise it and the governor piston to the position shown. Since the connection from the main reservoir to chamber *d* is open only when you have the handle of the automatic brake valve in *Release, Running* or *Holding* positions, in the other positions this governor head is cut out. The connection marked MR in the maximum pressure head should be connected either to the main reservoir cut-out cock, to the main reservoir pipe between the reservoir and the cut-out cock, or to the reservoir connecting pipe to the forward end of the first main reservoir, or to the pipe connecting the two main reservoirs, so as to be always in communication with the main reservoir. Then when the excess pressure head is cut out by the brake valve, or by the main reservoir cut-out cock, the maximum pressure head will control the compressor. When main reservoir pressure in chamber *a* exceeds the adjustment of spring 19 in the maximum pressure head, diaphragm 20 will raise its pin valve and allow air to flow into chamber *b* above the governor piston, controlling the compressor as above described. The adjustment of spring 19 thus fixes the maximum limit of main reservoir pressure during such time as the automatic brake valve handle is in *Lap*, *Service* or *Emergency* positions.

As each governor head has a vent port *c*, from which a small amount of air escapes whenever pressure is present in port *b*, to avoid an unnecessary waste of air, one of these should be plugged.

To adjust the excess pressure head of this governor, remove cap nut 25 and turn adjusting nut 26 until the compression of spring 27 gives the desired difference between main reservoir and brake pipe pressures. While adjusting the excess pressure head, you should have the handle of the automatic brake valve in *Running* position. To adjust the maximum pressure head, remove cap nut 17 and turn adjusting nut 18 until the compression of spring 19 causes the compressor to stop at the maximum main reservoir pressure required. While adjusting the maximum pressure head you should have the handle of the automatic brake valve on *Lap*. Spring 27 is usually adjusted for 20 pounds excess pressure, and spring 19 for a pressure ranging from 120 to 140 pounds, depending on the service.

While the SF governor is standard on most roads in connection with the No. 6 ET Equipment, the SG governor is used on some roads where it is desired to have a simpler arrangement of piping. When this governor is used, the long copper pipe (excess pressure operating pipe) leading from the automatic brake valve to the diaphragm portion of the excess pressure top is done away with. The same simple arrangement of piping may be accomplished with the SF governor by connecting the two diaphragm portions together, as illustrated in Fig 355.

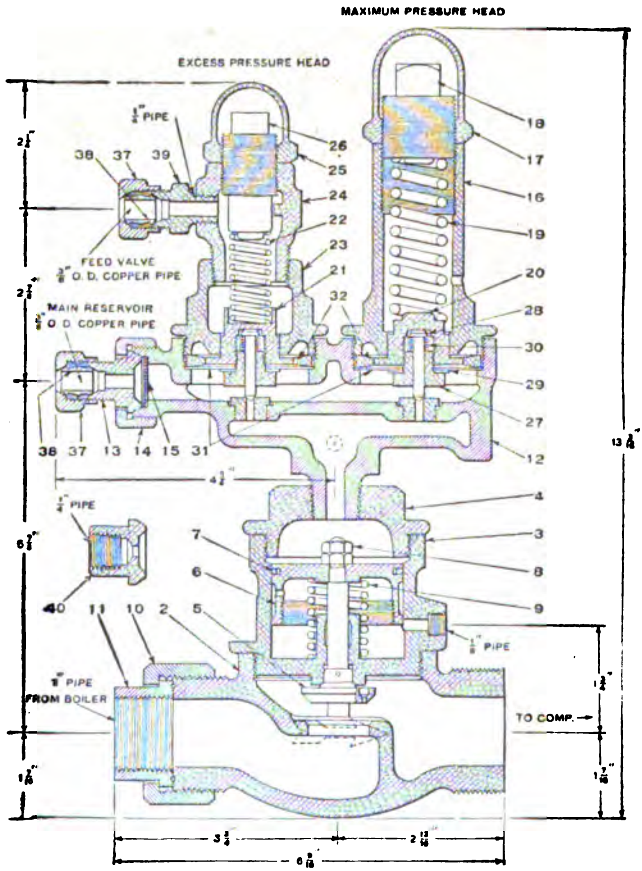


FIG. 375. Type SG Compressor Governor.

The SG governor is illustrated in Fig. 375. The Siamese fitting 12 is so designed that the underside of both diaphragms 31 are permanently connected through a single pipe to the main reservoir. The functions of the governor are not changed in any way, but the cutting out of the "excess pressure head," as in *Lap, Service* and *Emergency* positions of the brake valve handle, is accomplished in a slightly different manner than with the Standard SF governor. As previously explained with the SF governor, when the brake valve handle is in any one of the above positions, the connection from the main reservoir to chamber *d* is cut off by the brake valve, while chamber *f* above the diaphragm is open to main reservoir pressure through the excess pressure pipe and the feed valve pipe. The combined action of spring and air pressure above the diaphragm forces it down, seating its pin valve. However, with the SG governor and the SF type piped as in SG governor, this is accomplished by balancing the diaphragm with main reservoir pressure above and below, permitting the regulating spring 22 to hold the pin valve seated as with the SF governor.

The "Dead Engine" Feature.

The "Dead Engine" feature shown in Fig. 355 is for the operation of the locomotive brakes when the compressor on a locomotive in a train fails to operate from any cause. Fig. 376 shows the combined strainer, check valve, and choke. As these parts are not required at other times, a cut-out cock is provided. This cock should be kept closed except under the conditions just mentioned. The air for operating the brakes on such a locomotive must then be supplied through the brake pipe from the locomotive operating the train brakes.

With the cut-out cock open, air from the brake pipe enters at BP, Figure 376, passes through the curled hair strainer, lifts check valve 4, held to its seat by a strong spring 2, passes through the choke bushing, and out at MR to the main reservoir, thus providing pressure for operating the brakes on this locomotive. The double-heading cock should be closed, and the handle of each brake valve should be in *Running* position. Where absence of water in the boiler, or other reason, justifies keeping the maximum braking power of such a locomotive lower than the standard, this can be accomplished by reducing the adjustment of the safety valve on the distributing valve. It can also be reduced at will by the independent brake valve.

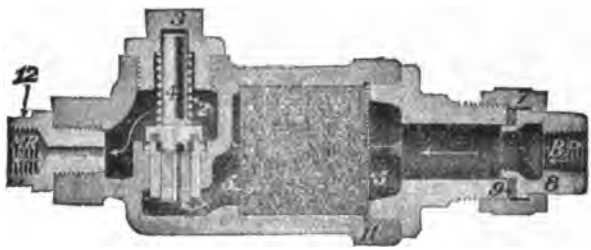


FIG. 376. Combined Air Strainer and Check Valve.

The strainer protects the check valve and choke from dirt. Spring 2 over the check valve insures that this valve will seat and, while assuring an ample pressure to operate the locomotive brakes, keeps the main reservoir pressure somewhat lower than the brake pipe pressure, thereby reducing any leakage from the former. The choke prevents a sudden drop in brake pipe pressure and the application of the train brakes, which would otherwise occur with an uncharged main reservoir cut into a charged brake pipe.

Air Brake and Air Signal Instructions.

See instructions under this heading on pages 534 to 543 of the 1918 Proceedings of the Master Car Builders' Association.

Broken Pipes.

Main Reservoir Pipes.

If the main reservoir pipe breaks between the reservoir and the branch to the distributing valve in such a way that it cannot be repaired, you cannot apply the locomotive brake by either brake valve except when a quick action cap is used on the distributing valve and then only in emergency. In the latter event, the pressure obtained in the locomotive brake cylinders is equal to that vented through the quick action cap from the brake pipe. If the break is between the brake valve and the branch pipe leading to the distributing valve, plug both sides of the main reservoir pipe. The locomotive brakes can then be operated in the usual manner with the independent brake valve.

Main Reservoir Branch Pipes.

If the branch pipe from the *main reservoir pipe to the distributing valve* breaks between the main reservoir pipe and the cut-out cock, plug the main reservoir side of the break and close the branch pipe cut-out cock. The locomotive brakes are then inoperative. You can operate the train brakes in the usual manner.

If the *branch pipe leading to the feed valve and reducing valve* breaks, plug both sides. This cuts out the independent brake valve, the signal system and the use of *Running* (for releasing and recharging the train brakes) and *Holding* positions of the automatic brake valve; also the excess pressure head of the governor. As there would be no pressure on top of the independent rotary valve to hold the valve to its seat, it would be impossible to secure an automatic application of the locomotive brakes. To remedy this, move the independent brake valve handle to *Slow Application* position before applying the brakes and leave it there until it is desired to again release the locomotive brakes. When the automatic brakes are released, return the independent brake valve handle to *Running* position. The train brakes are released and recharged with the automatic brake valve handle in *Release* position. The locomotive brakes can be released by moving the handle of the automatic brake valve to *Running* position or by *Release* position of the independent brake valve. Since the feed valve is inoperative, blank the excess pressure operating pipe by a blind gasket in the union at the governor (marked ABV, Fig. 374*). This cuts out the excess pressure head of the governor so that the maximum pressure head controls the operation of the compressor. To prevent too high a brake pipe pressure with the automatic brake valve handle in *Release* position throttle the compressor by hand.

If the break occurs *between the reducing valve and the branch pipe leading to the feed valve,* plug both sides of the pipe. This cuts out the independent brake valve and the signal system but does not interfere with the handling of the locomotive and train brakes with the automatic brake valve except that the independent brake valve must be manipulated as described in the preceding paragraph.

When the pipe is broken *beyond the feed valve or reducing valve,* it is not necessary to plug the pipe leading from these valves as you can accomplish the same result by turning the adjusting nut sufficiently to loosen the regulating spring and cause the blow of main reservoir air to cease.

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* NOTE: If the SF governor is piped as an SG governor, blank the main reservoir pipe to the excess pressure head of the governor by placing a blind gasket in the union at the governor, which cuts this head out and permits the maximum pressure head to control the operation of the compressor.

If the SG governor is used proceed by either of the two following methods:

(1) Remove cap nut 25, unscrew the adjusting nut 26, place a block of some kind about ½" in diameter and 1½" long within the spring so that one end will rest upon the diaphragm spindle 21, Fig. 375, screw the adjusting nut into place and the diaphragm and its pin valve will then be held seated, thereby cutting out the excess pressure head.

(2) Place a blind gasket in the main reservoir pipe union at the governor and control the compressor by regulating the steam valve.

Another remedy in case the pipe is broken beyond the reducing valve is to slack off the reducing valve adjusting nut, as just described, plug the broken pipe toward the independent brake valve and plug the exhaust port in the bottom of this brake valve. The independent brake valve handle should then be kept in *Running* position. You can operate the locomotive brakes by the automatic brake valve.

Brake Pipe.

The pipe most frequently broken with the No. 6 ET Equipment is the brake pipe branch to the distributing valve. When this happens, plug the end leading from the brake pipe. You can then operate the train brakes in the usual manner, but you cannot operate the locomotive brakes by the automatic brake valve. You can operate the locomotive brake by the independent brake valve in the ordinary way except that *Release* position must always be used to release it.

If the break is *ahead of the branch pipe to the distributing valve,* the distributing valve side of the break may be plugged without affecting brake operation.

If the break occurs *between the branch pipe to the distributing valve and the branch to the automatic brake valve,* plug the pipe as above. It will be impossible to apply and release the brakes by the automatic brake valve but they may be applied and released by the independent brake valve.

If the *pilot section* of the brake pipe is broken ahead of the cut-out cock and it is necessary to couple to a train ahead of the locomotive, there being no cars behind, use a "combination hose" to connect the brake hose to the signal hose at the rear of the tender, open the angle and cut-out cocks, and close the cut-out cock in the supply line to the signal system. Use another "combination hose" to connect the signal hose at the pilot end of the locomotive to the brake hose of the car and open the angle and cut-out cocks. You can then operate the locomotive and train brakes in the usual manner. A similar method can be used if the break is on the engine back of the branch to the brake valve.

If the brake pipe *under the tender* breaks, use the signal pipe as the brake pipe by using the "combination hose" as above described.

Brake Cylinder Pipe.

A broken brake cylinder pipe permits escape of main reservoir air when the brake is applied and may cause the release of one or more of the locomotive brake cylinders, depending upon where the break occurs. If the break cannot be repaired, close the cut-out cock in the pipe leading to the broken pipe. If the break occurs next to the distributing valve reservoir, close the cut-out cock in the main reservoir supply pipe to the distributing valve.

Application Cylinder Pipe.

If the application cylinder pipe breaks, plug the pipe on the distributing valve side of the break. If the break occurs *between the distributing valve and the tee to the independent and automatic brake valves,* you cannot apply the locomotive brakes with the independent brake valve and the emergency maintaining feature is lost; you can, however, apply the locomotive brakes as usual by the automatic brake valve and you can release them by that valve in *Running* position. If the break is *between the automatic valve and the tee,* you can apply and release the independent brake in the usual way but the emergency maintaining feature is lost. If the break occurs *between the tee and the independent brake valve,* you cannot apply the locomotive brakes by the independent brake valve, but the emergency maintaining feature is retained.

Distributing Valve Release Pipe.

A broken release pipe need not cause any delay as it merely cuts out the holding feature of the automatic brake valve. If this pipe breaks *between the two brake valves,* you can hold the locomotive brakes applied while the train brakes are being released and recharging by placing the independent brake valve handle in *Lap* position. You can then release the locomotive brakes by returning the independent brake valve handle to *Running* position. Or the broken release pipe may be plugged on the distributing valve side of the break and you can then release the locomotive brake with the independent brake valve in *Release* position. If the break is *between the distributing valve and the independent brake valve,* plug the distributing valve side of the break. You can then hold the locomotive brakes applied as above, but to release them, the independent brake valve handle must be placed in *Release* position.

Equalizing Reservoir Pipe.

In case of breakage of the equalizing reservoir pipe, plug this pipe at the brake valve union and also plug the brake pipe service exhaust. Then to apply the brakes, move the handle of the automatic brake valve gradually toward *Emergency* position, making the desired brake pipe service reduction gradual and direct, then return the handle gradually to *Lap* position.

Excess Pressure Operating Pipe.

If the excess pressure operating pipe breaks, place the automatic brake valve handle in *Lap* position and plug the pipe toward the brake valve. The brakes can be operated as usual but the maximum pressure head of the governor will control the compressor.

Excess Pressure Pipe.

If this pipe breaks, the compressor will not operate when main pressure is over about 20 lbs. To remedy this, plug the pipe on the feed valve side of the break and place a blink gasket in the excess pressure operating pipe.* This cuts out the excess pressure head of the governor and permits the maximum pressure head of the governor to control the compressor.

Pipe to Maximum Pressure Head of Governor.

If this pipe breaks, plug the pipe on the main reservoir side of the break. The excess pressure head will still control main reservoir pressure when the automatic brake valve is in *Release, Running* or *Holding* position. However, as the maximum pressure head is cut out, you must throttle the compressor by hand when the automatic brake valve handle is in *Lap, Service* or *Emergency* position in order to prevent excessive main reservoir pressure.

Lubrication.

Brake Cylinder.

In cleaning the brake cylinders and pistons give special attention to removing lint, freeing the leakage grooves of any deposit,

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*NOTE: If the SF governor is piped as an SG governor, plug the excess pressure pipe on the feed valve side of the break, then either (1) place a blind gasket in the main reservoir pipe union in the excess pressure head of the governor which cuts out this head and permits the maximum pressure head to control the compressor; or (2) place a blind gasket in each main reservoir connection on the governor and then control the compressor by regulating the steam valve.

If the governor is of the SG type, plug the excess pressure pipe on the feed valve side of the break then either (1) block the excess pressure head as directed in the note on page 858, which cuts out the excess pressure head and permits the maximum pressure head to control the compressor, or (2) place a blind gasket in the main reservoir pipe union at the governor and control the compressor by regulating the steam valve.

and thorough cleansing of the expander rings, packing leathers and pistons. In lubricating the cylinders give special attention to the thorough lubrication of the top of the cylinders and the inside of the packing leathers where the expander rings rest. Use any good lubricant specially prepared for the purpose. Observe particularly that the follower nuts are tight, since they frequently work loose in service.

Brake Valves.

Use a good grade of graphite grease on the brake valve rotary valves whenever it can be conveniently applied, as when assembling the device after overhauling, repairs, etc. However, as graphite grease cannot be used conveniently for lubricating the valve after it is assembled, use a good grade of oil in such cases. Whatever lubricant you use should be applied very sparingly.

Lubricate the equalizing piston by pushing it to its normal position and applying a drop or two of oil to the circumference of the piston bushing, spreading it over the surface as evenly as possible. Then move the piston in and out several times to insure that this oil will be properly distributed on the wall of the cylinder. There should be no free oil left on the parts. Care should be taken not to let any oil get upon the gaskets.

Distributing Valve.

Never remove movable parts of the distributing valve while it is on the locomotive. If the valve portion is not working properly, or needs cleaning and oiling, remove it from the reservoir portion and replace it with a valve portion in good condition. All cleaning and oiling should be done at a bench, by a competent man, where the liability of damage to the internal parts of the valve is least. Any attempt to take the valve portion apart while it is still on the locomotive is almost sure to result in a large precentage of valves being injured by careless handling or dirt getting inside the valve.

The proper specified cleaning period of the distributing valve is best determined for each particular case by careful inspection and trial. Where conditions are severe and the distributing valve exposed to extremes of weather, dirt and so on, the cleaning, oiling and testing will require shorter intervals than where conditions are more favorable, but under the most severe conditions this interval should not be shorter than three months.

The following is the method of lubricating the distributing valve:

Equalizing Portion.

The best lubricant for the equalizing valve and graduating valve and their seats is dry graphite. After the bearing surfaces have been properly rubbed in by a free use of oil wipe them clean with a soft cloth or some soft material. Remove all oil, gum or grease from the valves and their seats.

Lubricate the face of the graduating valve, both upper and lower faces of the equalizing valve, the equalizing valve seat, and the upper portion of the bushing where the equalizing valve spring bears with a high grade of very fine, dry, pure graphite. Rub it in until the parts show a dark copper color.

To apply the graphite, use a stick in the shape of a paddle about 8 inches long and having a small piece of chamois glued to one end. Put a small amount of graphite on the chamois skin and rub on the surfaces specified. Leave no free graphite on these surfaces. When the work is completed, the graduating and equalizing valves and seats must be entirely free from oil or grease. Care should be taken when handling the parts after lubricating that the hands do not come in contact with the lubricated parts as the thin coating of graphite is easily removed. After the piston and graduating and equalizing valves have been replaced in the equalizing portion, move them to *Release* position and apply a drop or two of good oil to the circumference of the piston bushing, spreading it over the surface as evenly as possible. Then move the piston back and forth several times to insure proper distribution of this oil on the wall of the cylinder. There should be no free oil left on the parts. Care should be taken not to permit any oil to get upon the gaskets.

The emergency valve of the Quick-Action Cylinder Cap should be lubricated by the use of a good grade of dry graphite sparingly applied, as directed for the equalizing slide valve.

Application Portion.

The exhaust valve and seat and application valve and seat of the application portion should be cleaned, rubbed in and sparingly lubricated with graphite grease.

Before applying the piston to the application portion, clean the application cylinder and piston. Lubricate the walls of the cylinder and piston ring, using a good grade of lubricant specially prepared for the purpose.

Feed Valve and Reducing Valve.

The only part of the feed valve and reducing valve requiring lubrication is the slide valve which should be lubricated with dry graphite.

Governor.

The governor receives the necessary lubrication on account of its location in the steam supply pipe between the lubricator and the compressor. The governor requires no further oiling.

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