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LOCOMOTIVE DESIGN

By Geo. L. Fowler and Carl J. Mellin

Part II

VALVE MOTION

Second Edition

CONTENTS

51.

CHAPTER I

THEORY OF VALVE MOTION*†

Next in importance to the boiler in determining the efficiency of the locomotive as a whole, is the valve motion, and too much stress cannot be put upon the value of a proper design for this element in the machine. The Stephenson link motion, which may be said to be the one universally used upon American locomotives, possesses the peculiarity of being exceedingly sensitive to a close adjustment of all of its parts in order that a correct action and proper distribution of the steam may be obtained; while with the roughest kind of haphazard design or no design at all, it will do its work after a fashion and make the wheels go round. It is evident, however, that in order that the steam distribution in the cylinder may be as efficient as possible for all speeds and all points of cut-off, the utmost care must be exercised in the designing of the valves and machinery by which they are driven.

Up to within a few years the flat side valve was in universal use. At first it was the common unbalanced D-valve, which was followed, as the steam pressure was increased, by the balanced valve, while the current practice, where high pressures are used, lies between the flat balanced valve and the piston type. The piston valve possesses some advantages over the flat valve in that it is fully balanced, though the slide valve can be quite satisfactorily designed in this respect. In short, it is a matter of choice and convenience of construction and maintenance as to which shall be used, though the tendency of modern practice is toward an increasing use of the piston valve.

The main difference to be considered in the designing of the two types of valves is that while the flat slide valve is invariably arranged for an outside admission of steam to the cylinder, the piston valve should be arranged for an inside admission. This is not absolutely necessary, but the advantages are that the steam passages can thus be better protected from the cold and radiation, and the steam chest heads and packings relieved from all pressure except that of the exhaust steam, which is but a few pounds above that of the atmosphere and so puts really very little stress upon these parts. If the piston valve is so designed that the admission is on the inside, it should be made hollow with as large a passage through the center as possible.

* The present number of MACHINERY'S Reference Series is the second part of a treatise on complete Locomotive Design, covered by Nos. 27, 28, 29, and 30 of the Series, and originally published in RAILWAY MACHINERY (the railway edition of MACHINERY). Each of the four parts of the complete work treats separately on one, or more, special features of locomotive design; and while the four parts make one homogeneous treatise on the whole subject, each part is complete by itself. In order to give concrete form to the examples and theoretical considerations, it is assumed that a consolidation freight locomotive and an Atlantic type passenger engine are being designed. It is further assumed that these locomotives are designed for a division 150 miles long, laid with rails weighing 75 pounds per yard, and with a ruling grade of one per cent ten miles in length.

† MACHINERY, Railway Edition, April, 1905.

The object of this is to secure a large area for the movement of the exhaust, so that, at the instant of release, the pressure may be reduced to a minimum. The use of the hollow valve facilitates this by permitting an escape through the exhaust passages at each end; and these latter should be carried to the base of the exhaust pipe and made to meet at a very acute angle so that the two currents will combine without forming obstructive eddies and thus aid in the fanning of the fire. Such a combination will be found to permit of the use of a larger exhaust nozzle than is possible with a solid valve.

As for the mechanisms by which the valve is moved, there are four, or as they are sometimes classified, five different systems that are in use upon locomotives, and which are named after their designers, *viz.:* the Stephenson, Gooch, Allan, Walschaerts, and Joy.

The first is known as the shifting link motion, and is that mostly in use in the United States. The second is directly opposite in its

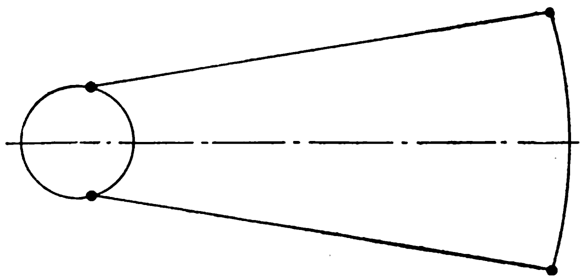

Fig. 1. Open Eccentric-rods

action, in that the link is stationary and the link-block, attached to the valve-rod, is moved up and down. The Allan is a combination of the first two in that both the link itself and the valve-rod are shifted, by which it becomes possible to make the link straight. This, of course, greatly simplifies the construction and maintenance of the link, and the motion is extensively used in Europe. All of these motions are operated with two eccentrics, one for the forward and the other for the backward motion.

In the Stephenson and Allan motion when the eccentric-rods are open, as in Fig. 1, the lead is increased as the link is hooked up and the point of cut-off made earlier. If, however, the rods are crossed as in Fig. 2, the hooking-up of the link reduces the lead, though this reduction is much less than the increase in the former case. Again, with either open or crossed rods, the corresponding increase or reduction of the lead is much less with the Allan than with the Stephenson valve motion. With the Gooch, Walschaerts, and Joy motions the lead is constant for all points of cut-off.

It will be seen, by reference to the diagrams Figs. 1 and 2, that the eccentric-rods are said to be "open" when, with the eccentric centers upon the same side of the axle as the link, the rods are not crossed; and "crossed" when the rods do cross in that position. This holds true whether the motion be transmitted direct to the valve stem or indirectly through a rocker arm. This term of direct or indirect application refers to the relative movement of the valve and the link-block. When they move in the same direction the motion is said to be "direct"; when in the opposite directions, "indirect."

The Walschaerts gear is driven by a combination of an eccentric or short-stroke return crank from the main crank-pin and a connection to the crosshead. The Joy gear is driven from a connection to the connecting-rod. The former has been extensively applied on the continent of Europe and the latter to some extent in England. The Walschaerts gear has recently been applied to several of the largest engines built in the United States with such pronounced success that the use is being extended, and some engineers have expressed the opinion that the prospects are that it will eventually take the lead over the Stephenson gear, as it has abroad, on account of the many advantages which it possesses and which have not been properly demonstrated in American practice until lately.

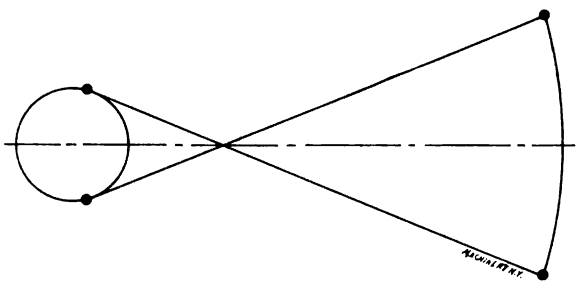

Fig. 2. Crossed Eccentric-rods

As the Stephenson motion is the one mostly used in this country it will be first considered. It is probably the most flexible of any in use, and can be most readily adapted to irregularities in the running and operation of the machine. At the same time it will get out of adjustment very easily and requires the utmost care in its designing in order that it may work properly. With this as an introduction we may now enter upon an examination of the motion and its requirements. To work out the valve motion theoretically and mathematically is a long and tedious operation. As its application to, and development for use upon, a locomotive involves a number of irregularities that are necessarily neglected, when treating the subject from a purely mathematical standpoint, this method will be simplified by use of diagrams by which the steps needed for its development for actual use will be shown.

In the first place, the introduction of a rocker-arm between the eccentrics and the valve serves as a means of increasing or reducing the travel of the latter as compared with the throw of the eccentrics as well as transferring the line of motion to suit convenience in locating valves and valve spindles. The angularity of the connecting- and eccentric-rods introduces irregularities that can, to a great extent, be compensated by the location of the link-saddle pin. The longer these rods can be made the better for the action of the motion, and it is well not to make them too short, when it would become impossible to compensate for the irregularities that would be introduced.

Experience and calculations have shown that, to secure a satisfactory action of the valve motion, the connecting-rod should not be less than six times the radius of the crank, that is, three times the piston stroke, and that the eccentric-rods should not be less than eight times the throw of the eccentric. As a matter of fact, the eccentric-rods are usually of a greater length than this. In the case of the link the radius should be equal to the distance from the center of the link-block, when in its central position, to the center of the axle, and the distance between the eccentric-rod pins should not be less than two and a half times the throw of the eccentric. If it is less than this the angle assumed by the link relative to the block will be such that the slip of the latter will not only be excessive, but it will be apt to stick and put undue stresses on the entire mechanism of the motion.

As already intimated, the irregularities introduced by the angularity of the rods may be compensated for by adjustment of the location of the saddle-pin, which will always be inside the center line of the arc. If, however, it is carried too far, it will give an objectionably long slip to the link, especially if the rod lengths are near their minimum. So, as this offset of the saddle-pin, as it is called, is less with an outside admission valve and an indirect motion, or, what is the same thing, an inside admission and direct motion, than where the contrary conditions exist, it is always best to use one or the other of these two combinations when no other advantage is to be gained by a reversal of the conditions. The adjustment of the valve also requires that particular attention should be paid to the lead that is obtained at full travel, as well as the increase resulting from a linking-up of the engine, so that, when running with an early cut-off the lead and pre-admission may not be excessive.

This can best be studied by a consideration of the effect of the movement of the link on the travel of the valve. With the link in full gear the block is usually so related to the pin of the corresponding eccentric-rod that its motion to and fro is equal to the diameter of the path of the controlling eccentric. Then, if the two rocker-arms are of the same length, the travel of the valve will also be the same. In other words one eccentric controls the valve and the other merely causes the link to oscillate about the block, as far as relative motions are concerned. But, when the link is raised, both eccentrics have an effect on the motion of the valve, and the resultant of this is as though another and controlling eccentric of a shorter throw were to be introduced between the two. The throw of this resultant eccentric decreases as the link is raised until mid-gear is reached, when the throw is at the minimum. Meanwhile its position has shifted from the center of the forward eccentric to a point in line with the crank and midway between the two actual eccentrics. At this point the radius is equal to the sum of the lap and lead in mid-gear. As the link is raised still further, the radius gradually increases and shifts its center until, in full gear back, it coincides with the center of the backing eccentric.

The center of this imaginary or resultant eccentric has then traversed a path that is a parabolic curve connecting the centers of the two eccentrics and passing through a point in line with the crank and distant from the center of the axle by an amount equal to the sum of the lap and lead in mid-gear. The height of this curve or its distance from a straight line connecting the two eccentric centers is equal to the increase of lead between full and mid-gear. For a given maximum cut-off and valve travel, the required angular advance and lap can be calculated, when no lead is allowed in full gear or an excessive pre-admission is to be avoided, by the following formula, based on a radius equal to 1:

$$\sin d = \sqrt{1 - p} \qquad (1)$$

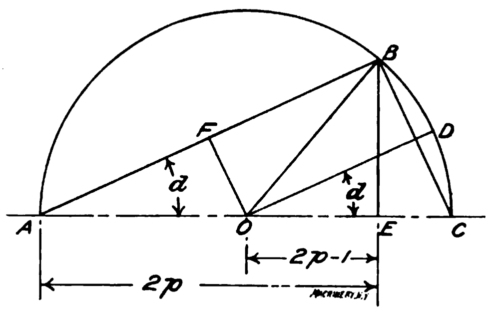

Fig. 3. Diagram for Finding Angular Advance of Eccentric

where $\sin d =$ lap, or lap and lead,
$p =$ cut-off in hundredths of the stroke, and
$d =$ angular advance of eccentrics.

If r equals radius of the throw of the eccentrics, then multiplying the second term of the equation by r the formula gives the value of $\sin d$ as follows:

$$\sin d = r\sqrt{1 - p} \qquad (2)$$

An analysis of this formula is readily made by reference to diagram Fig. 3. Draw the semicircle ABC to represent the path of the center of the eccentric, with a center at O and a radius equal to 1. Lay off the distance $AE = 2p$ and draw EB at right angles to the diameter AOC. Connect the points A and B and B and C, and from O draw OF at right angles to and bisecting the line AB, and also draw OD parallel to AB. With this construction the arc AB will be equal to that swept through by the crank and the eccentric from the beginning of the stroke to the point of cut-off, and the arc DC which is one-half of BC will be equal to the angular advance of the eccentric. Then the angle DOC equals d.

From the diagram

$$AC : BC = BO : EC.$$

By multiplying and substituting the value $AC = 2$ and $EC = 2 - 2p$ we have

$$BC = \sqrt{2\,(2 - 2p)} = 2\sqrt{1 - p}$$

and

$$\frac{BC}{2} = \sin d = \sqrt{1 - p}$$

For example, suppose a maximum cut-off of 0.85 of the stroke is desired with no lead in full gear and a 6-inch throw of the eccentric and travel of valve. The advance of the eccentric and the lap of the valve will be obtained by substituting these values in the formula:

$$\sin\ d = 3\sqrt{1 - p} = 1.16 \text{ inch},$$

or nearly 1 3/16 inch lap without lead. The radius being 3 inches the value obtained based on the radius as a unit must therefore be multiplied by this amount, as above, in order to get the result expressed in inches as provided for in formula (2) and in the example given.

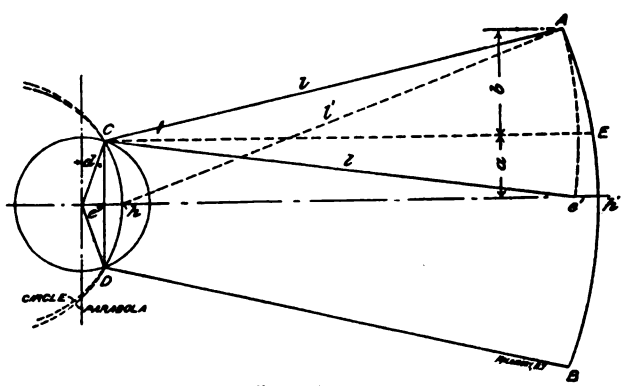

Fig. 4. Link Motion Diagram Illustrating Increase of Lead with Open Rods

Having thus determined the angular advance and lap of the eccentric it remains to obtain the positions of the virtual eccentric as the engine is hooked up towards mid-gear. It is now in order to examine the curve that the center of the virtual or resultant eccentric traverses in passing from the point of full gear forward to full gear back.

Let the diagram, Fig. 4, represent the eccentrics and the link, in which the circle CD is the path followed by the actual centers about the axle. C will then be the point occupied by the forward eccentric when the crank is on the center, and D that of the backing eccentric. The first step in the problem is to determine the height eh or distance of the resultant eccentric center from the chord CD, when the link is in mid-gear. This height eh is equal to the displacement of the valve caused by moving the link from full to mid-gear. In Fig. 4

$d =$ the angular advance of the eccentric,

$a = \dfrac{CD}{2}$,

b = one-half the distance between the pivot points of the link, minus a,

$2(a+b)$ = distance between pivot points of the link,

l = length of eccentric rods,

l' = radius of the link.

When the link is in full gear forward the eccentric-rod occupies the position Ce', and when the link is in mid-gear the rod occupies the position CA. It will be seen that, in moving from e' to A, the pivot point of the link moves away from the vertical center line, passing through the center of the axle until it reaches the line CE and after that approaches the same until it reaches A. The amount of separation may be expressed as equal to

$$l - \sqrt{l^2 - a^2}$$

and the approach as

$$l - \sqrt{l^2 - b^2}$$

which would make the total approach

$$[l - \sqrt{l^2 - b^2}] - [l - \sqrt{l^2 - a^2}] = \sqrt{l^2 - a^2} - \sqrt{l^2 - b^2}$$

At the same time the center line of the link that is occupied by the link block has moved to the point h', which is that of the center when the pivot points are at A and B. This distance, $e'h'$, through which the block is moved, is equal to the versed sine of the angle Ahh', less the distance by which the pivot point A actually approached the vertical passing through the center of the axle. If the radius of the link be taken as equal to the length of the eccentric-rods, the center of the link in mid-gear will be at the point h.

The versed sine of the angle Ahh' will then be

$$l' - \sqrt{l'^2 - (a+b)^2}$$

Hence

$$e'h' = l' - \sqrt{l'^2 - (a+b)^2} - (\sqrt{l^2 - a^2} - \sqrt{l^2 - b^2})$$

Then

$$hE = hh' = Ce' = l = l'$$

$$\text{and } eh = e'h'$$

In this case the radius of the link l' is assumed to be equal to the working length of the eccentric-rod l. An inequality in these dimensions makes no difference in the accuracy of the formula.

According to Prof. Zeuner in his analysis of the Stephenson valve motion, the curve passing through the points ChD is a parabola, but between the limits C and D in which it is used, it coincides so closely with a circle that it may be regarded as one whose radius is expressed by the formula:

$$r = \frac{B^2 + a^2}{2B} \tag{3}$$

in which

$$B = eh.$$

Before taking up the working out of the details of the valve motion mechanism, a few diagrams will be presented illustrative of the various conditions that will be encountered in practice.

Fig. 5 is a diagram showing the action in full gear conditions. To construct it, draw the circle *CDGH* with the diameter equal to the travel of the valve. From the same center and with *N* equal to the lap of the valve, draw the lap circle. At the extremity of the horizontal diameter with the lead in full gear as a radius, draw the small lead circle *C*. Draw the line *EF* tangent to the lead and lap-circles; then parallel to it the dotted line *CD* which defines the point *D*; and also the line *GH*, defining these two points as well. By drawing *FP* at right angles to *AB*, the point of cut-off *P* is located and can be measured from *C*, in the percentage of the diameter of the valve travel circle to which it will bear the same relation as the piston position, at that instant, bears to the stroke of the engine.

If the point of maximum cut-off has been decided upon, the process may be reversed by laying off the desired cut-off point *P* on the line *AB* and drawing line *PF* to intersect the travel circle. Draw the lead circle as before, about *C*, and then lay down the line *EF* tangent to the same. The lap is the perpendicular distance from the line *EF* to the center *O* and its circle may be drawn tangent to *EF*. The line *GH* is drawn as before parallel to *EF*.

These few lines practically define all of the points that will be needed for a study of the valve action at full gear when the exhaust lap is zero or line and line. The line *OE* indicates the crank position when the valve opens and *OF* when it closes at the cut-off. The exhaust opens at *G* and compression begins at *H*, and by projecting these points to the diagram below, it is possible to obtain the prominent points of an indicator diagram for full gear action.

When the center of the axle lies in the axis of the cylinder and valve motion the crank will coincide with the line *KL* at the instant that the centers of the two eccentrics are at *C* and *D* respectively. The curved line connecting *C* and *D* indicates the locus of the virtual or resultant eccentric, that operates the valve under the combined influence of the two eccentrics, and its distance from *EF* at any point represents the lead that the engine is working with, when hooked up so that the center of the virtual eccentric coincides with the given point.

Fig. 6 is a diagram showing the effect of raising the link and thus shortening the period of steam admission. In a general way it closely resembles that given in Fig. 5. A change has, however, been made by the introduction of a small exhaust lap, in order that the effect of delaying the exhaust opening at *G* and advancing the compression at *H* may be seen. By raising the link so that the virtual center of the eccentric is at *T*, the valve travel becomes equal to the diameter of a circle drawn through *T* with *O* as a center. The lead at this point, being greater than that used for the describing of the circles at *C* and *D*, is taken as the radius for drawing a circle at *C′* where the circle passing through *T* intersects the horizontal diameter. The line *E′F′* is then drawn tangent to this new lead circle as well as to the lap circle, intersecting the new valve travel circle at *F′*. The radius *OF′* is then drawn through this point, *F′*, which is, in turn,

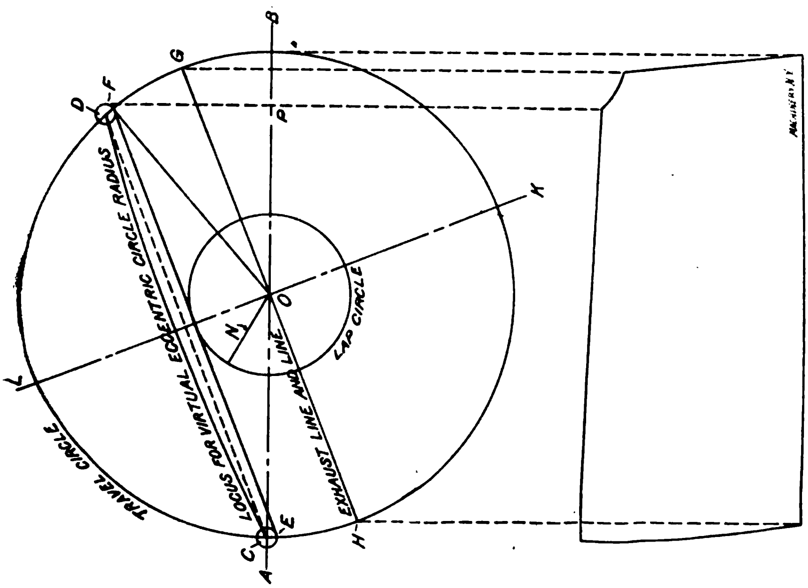

Fig. 5. Diagram showing Conditions when in Full Gear

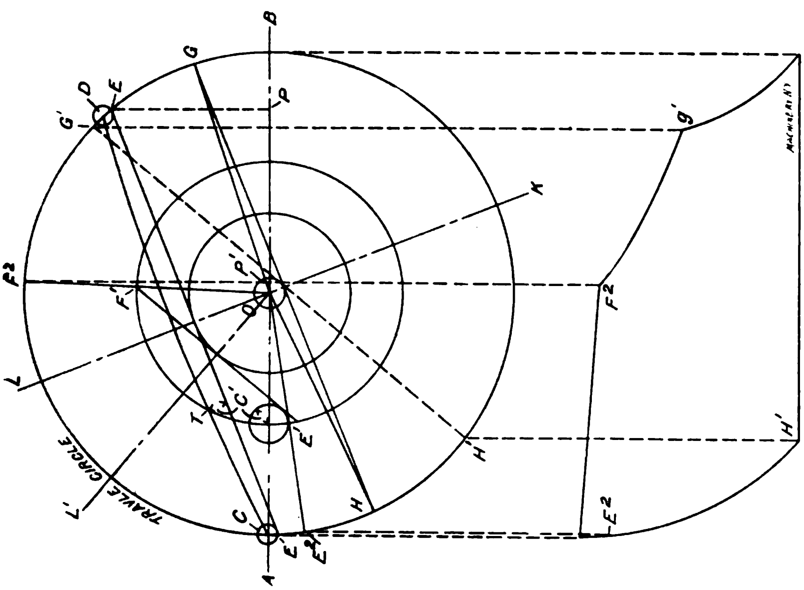

Fig. 6. Diagram showing Effect of Raising the Link, thus shortening the Period of Steam Admission

projected upon the line *AB* which it intersects at *P'*. This latter point indicates the point of cut-off or valve closure which, in this instance, is at about half-stroke.

With the change in the angle between the center line *AB* and the cut-off line *E' F'*, there is a corresponding change in the exhaust and compression line, the two remaining parallel, so that the point of exhaust opening *G* is advanced to *G'* and the commencement of compression *H* is advanced to *H'*, while pre-admission is moved ahead to *E'*. It thus appears that, with the advancement of the point of cut-off, the same thing is done with the opening and closing of the exhaust and the lead opening. If these points are then projected upon the line *AB*, their location with reference to the stroke of the piston

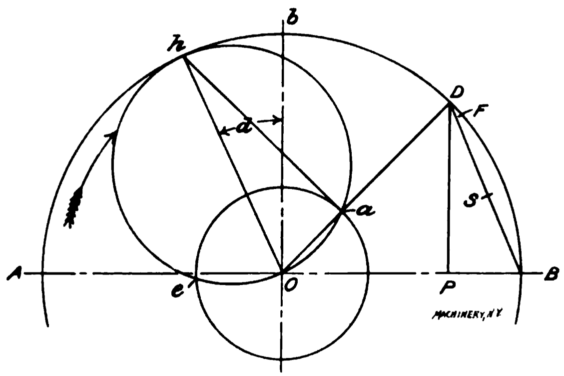

Fig. 7. Zeuner's Diagram

is at once fixed; while by extending them below, the outline of an indicator card is obtained.

Fig. 7 is the well-known Zeuner diagram. It is a method of graphically determining the valve events and was devised by Prof. Zeuner, of the Polytechnic Institute of Zürich, Switzerland, and is, therefore, known by his name. It offers a simple and reliable means of ascertaining not only the main valve events, but the amount of port opening both for steam admission and exhaust, for any piston or crank position throughout the entire stroke, and for any desired link position of the gear.

The angle of advance is obtained by Formula (2) when the lap in full gear is equal to sin *d*, or, in other words, when the valve is set with no lead, or by this diagram, where the lead can be introduced without much complication. It will be noted, however, that the angle of advance *d* must be laid off towards the crank, or 90 degrees — *d* in advance of the same, in order to obtain a correct reading.

In the construction of the diagram, let *AOB* be the diameter of a circle upon which the semi-circle *ADB* is described. The desired point of cut-off is laid down at *P*, the period of admission extending from *A* to *P*. From *P* a line *PD* is drawn at right angles to *AOB* to *D*, which is then connected with *B* by the line *DB*. On this line *DB* lay off *DF* from *D* equal to the amount of lead that is desired; and divide *BF* into two equal parts at *S*. Then $BS = FS$ will be the required lap.

With *BS* as a radius draw a circle about *O*, as a lap circle, and from the point *a* where it intersects the line *OD* draw *ah* at right angles to *OD*. Draw the radius *Oh* and with this as a diameter draw the valve circle *Oeha*, and the angle of advance will be found to be equal to *hOb*. The application of this diagram will be set forth in Figs. 8 and 9.

In Fig. 8, which is a combination diagram, the lap and lead circles as well as the positions *C* and *D* of the eccentrics are the same as in

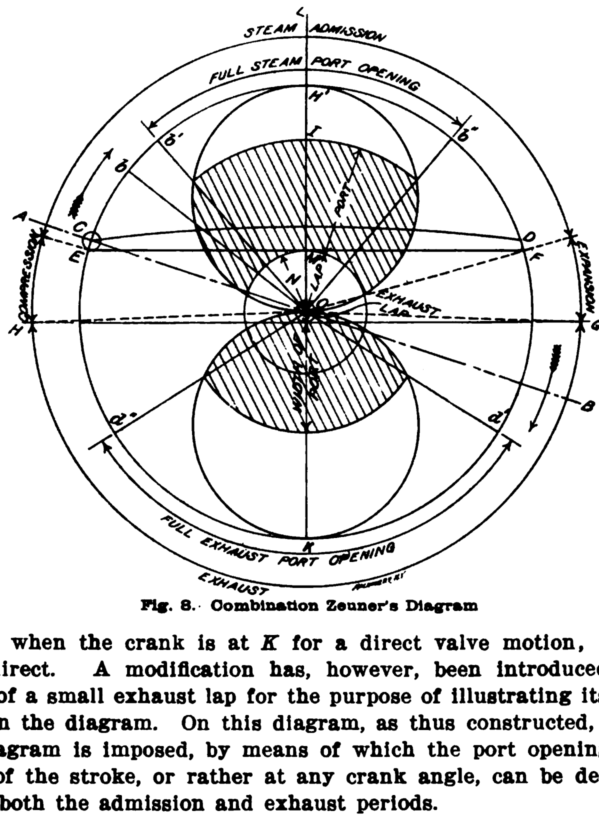

Fig. 8. Combination Zeuner's Diagram

Fig. 5, when the crank is at *K* for a direct valve motion, at *L* for an indirect. A modification has, however, been introduced in the shape of a small exhaust lap for the purpose of illustrating its appearance on the diagram. On this diagram, as thus constructed, the Zeuner diagram is imposed, by means of which the port opening at any point of the stroke, or rather at any crank angle, can be determined under both the admission and exhaust periods.

By locating this diagram relatively to the crank, the center line of the diagram will make an angle equal to that of the angle of advance, but on the opposite side of the 90 degrees to that of the actual location of the eccentric, or at a point where the eccentrics would be when the crank had reached *B*, and was turning in the opposite direction. With this location of the diagram it will be found that its centers will fall on the line *OL* of the original diagram, Fig. 5, which is one at right angles to *EF*. As the diameter of the valve circle is equal to the radius of the eccentric circle, it will be seen that the former intersects the lap circle at the same points as *OE* and *OF* by which the opening and closing of the valve is indicated respectively.

On the line *OL* lay off the distance *IM* outside the lap circle equal to the width of the port and draw the port circle through *I* with *O* as a center. By cross hatching that portion of the area of the valve circle lying between the lap and port circles, a graphical representation of the port opening is obtained. It is now possible to find the actual port opening for any position of the crank throughout the admission period, wherever a radius is drawn. For example, when the crank is on the line *Ob*, the opening will be about three-quarters the full width of the port, whereas at any point between b' and b'' there is a full width of port opening. At the latter point the valve commences to close, an act that is finally accomplished when the crank reaches *OF*.

The exhaust valve circle is located on the opposite side of the center and is of the same diameter as the steam circle. With the small exhaust lap as a radius, a circle is drawn about *O* as a center, from which the width of the steam port is laid out along *OK* and the port circle drawn just as on the steam side. The lines *OG* and *OH* indicate the exhaust opening and commencement of compression respectively, while the lines Od' and Od'' indicate the limits of the crank positions where there is a full port opening. A study of this combination diagram will show that all of the valve events coincide exactly in the two that have been thus superimposed, throughout the entire revolution of the crank.

Fig. 9 is a Zeuner diagram laid out to show its adaptability to the determination of the several valve events for the different points of cut-off. From the diagrams already discussed the full gear location of the eccentric was found, which is here indicated as r^5. It also appeared in the discussion of the diagram given in Fig. 6, that the line *OL* representing the center line of motion moves more and more towards the center line of the crank as the link is raised and a shortened cut-off obtained, until, in mid-gear, the two coincide; and, further, that the intersection of these lines and the circles forms the radius of the virtual eccentric and indicates the positions of the latter when the crank is at *A*, or at the beginning of the stroke.

At the various points of cut-off indicated by r', r^2, r^3, etc., the center of the virtual or resultant eccentric at R', R^2, R^3, etc., falls on the line *CD*, which represents the locus of these radii, and the diameters of the circles passing through these points are equal to the travel of the valve at the corresponding link positions. For the sake of simplicity no exhaust lap or clearance is shown in the diagram, Fig. 9, but the valve edge is considered to be line and line. Under these conditions the exhaust and compression points fall at right angles to the radial line of these several positions. Thus, the exhaust opening of the position corresponding to R^3 is at e^3, while its closing and the commencement of compression occurs at f^3. The same, of course, holds true of the other positions. If R^3 is located, then r^4, or the point in the revolution of the crank where the valve closes is found by means of a radius drawn through the point of intersection of the lap and valve circles, the latter having the line OR^3 as a diameter.

If, on the other hand, the location of R^3 or the center of the virtual eccentric is to be obtained from the predetermined point of valve closure as at r^3, the radius Or^3 is drawn; and, through the center O and the point of the intersection of the radius with the lap circle, a circle is drawn with a diameter of such a length that, when a circle is drawn about O as a center with this diameter as a radius, the larger

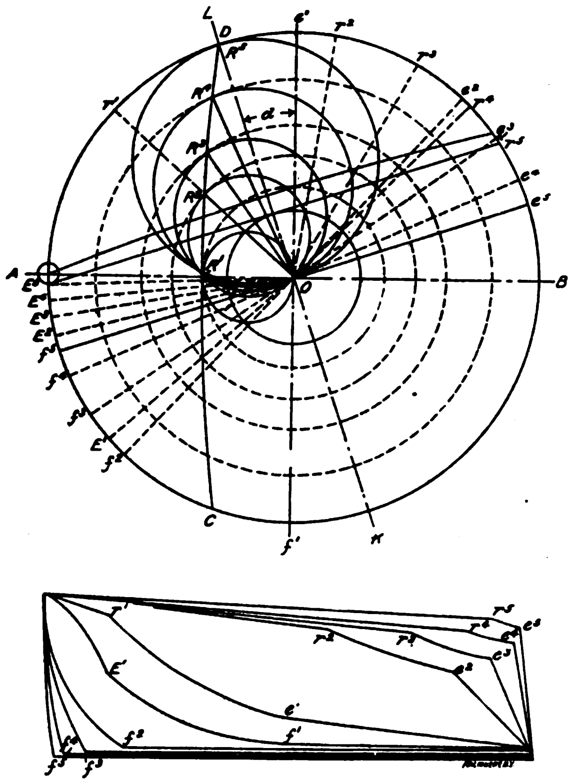

Fig. 9. Zeuner's Diagram, Illustrating the Change of Valve Events for Different Points of Cut-off

and smaller circles shall be tangent to each other at the point where they intersect the locus CD of the virtual eccentric. The method was fully explained in connection with Fig. 7. An examination of the diagram will show this to be the case with all of the points of intersection R', R^2, R^3, R^4 and R^5 that have been laid down.

The several points of pre-admission E', E^2, E^3, etc., are located in the same manner by drawing radii through the other intersections of the valve and lap circles. By projecting the several events it becomes possible to construct the outlines of the indicator diagrams that are shown at the lower part of the figure, thus facilitating the study of the action of the valve by the clearer detail so obtained.

Thus far attention has been directed solely to the forward motion. In the consideration of the back gear, it will, of course, be found that exactly the same conditions will obtain, but they will appear in a reversed position on the diagram. That is to say, the valve circles should be drawn below the line *AB* while the exhaust can be obtained by projecting each eccentric position line to the opposite side of *O*.

With the principles as set forth in these diagrams well in mind, it becomes possible to make an analysis of the consecutive events of the motion of the valve at various rates of admission and to enter intelligently upon the task of working out the details of the mechanism by which the valve is to be operated, and the theoretical studies can thus be put into a tangible and practical form.

CHAPTER II

CALCULATION OF VALVE DETAILS*

With the study of the movement of the valve and the eccentric thoroughly in mind, the next step is to so proportion the various parts of the gear that an efficient distribution of the steam will be obtained. It must be borne in mind, however, in this work, that many of the ratios and proportions that will be given are outlines only, and that variations from the figures are allowable and even required in order to meet the exigencies of design and construction.

As an example of this, in starting at the cylinder, the ports in the valve face should be made with an area of from 1/10 to 1/12 that of the cross-sectional area of the cylinder. This is, in itself, quite a range, but is necessary on account of the great differences in the amount of steam that has to be admitted to the cylinder by the different classes of engines, such as fast express, road freight, and switching. With 1/10 or 1/12 the cylinder area as the port area, the next step is the proportioning of the length of the latter to the width. In this the width should be, or rather may well be, made about 1/12 the length, subject to necessary variations. And finally the average travel of the valve may be put at about three and one-half times the port opening. These are the rough starting figures, and the lap of the valve may be determined either from Formula (2) or fixed arbitrarily at from one-fifth to one-sixth the travel of the valve in full gear. The use of the formula is to be preferred as the latter method leaves much uncertainty as to what the point of maximum cut-off will be and may involve irregularities in the equalization of the same on the two strokes.

In the days when 16 inches represented the standard or maximum diameter of cylinder, the valves were small and light, and the pressure of steam upon their backs did not cause enough frictional resistance to necessitate balancing. But with the increase in cylinder diameters and steam pressures, the larger valves that are used call for a balancing so as to relieve the rods of the tremendous stress that would otherwise be imposed upon them in the work that they are called upon to do. This balancing may be accomplished by any one of the accepted methods.

It should remove the steam pressure from the back of the valve over an area amounting to the sum of the areas of one steam port, the exhaust port and the two bridges, plus 8 per cent of this sum for plain valves and plus 5 per cent for Allan-ported valves. The reason for the use of a smaller amount with the Allan-ported valve is that steam cuts under and through the port and balances its own area when one of the openings is covered by the seat.

* MACHINERY, Railway Edition, May, 1905.

This balancing is introduced in order to lessen the resistance of the valve under ordinary working conditions; but, in calculating the dimensions of stems and rods, it is necessary to consider the work that would have to be performed in case the balancing strips were broken and the full pressure were to be put upon the back of the valve. The frictional resistance of the valve on its face may be taken as 20 per cent of the total load or 20 per cent of the valve area multiplied by the boiler pressure. Of course such a high coefficient of friction would only obtain if the valve face were very dry, but this is exactly what has to be provided for in case a breakdown under these adverse conditions is to be avoided. For this reason the valve stem and other parts of the motion must be made heavy enough to sustain the stress thus imposed in case of an accident to the balancing strips and lubricating apparatus. This may be expressed by the formula:

$$R = 0.2\,Plw \qquad (4)$$

in which

R = the frictional resistance of the valve,
P = boiler pressure in pounds per square inch,
l = length of valve in inches,
w = width of valve in inches.

The area through the keyway of the valve stem should, therefore, not be less than $\frac{0.2\,Plw}{10{,}000}$, which would put a stress of 10,000 pounds per square inch of section on the metal when in tension. In the matter of diameters the valve stem and eccentric rods must be made strong enough to carry this load in compression and for that purpose should be calculated accordingly. It must be borne in mind that this stress has to be carried back through each of the working and sustaining parts of the eccentrics, increasing somewhat as it advances on account of the added resistance of the motion of the several pieces.

Leaving the valve stem, the stress is next sustained by the rocker and its shaft, two sections of the same part that must be considered independently. The size of the arms may be calculated from the formula:

$$P = \frac{Sbh^2}{6l} \qquad (5)$$

in which

P = the maximum load or resistance to be overcome,
S = the maximum fiber stress permissible in the metal used,
b = thickness of the rocker arm,
h = width of the rocker arm at any desired point,
l = length of the rocker arm from the valve stem or link block connection to any desired point (see Fig. 10).

It will be noted that Formula (5) is that used for calculating the stresses imposed on a beam that is fixed at one end and loaded at the other, the expression $\frac{bh^2}{6}$ being that of the moment of resistance.

This involves the final assumption of one of the two dimensions b or h.

If, for example, b is taken to be 1½ inches and the resistance is put at 10,000 pounds, and the size of the arm is desired at 8 inches from the outer center, with a fiber stress of 10,000 pounds per square inch, Formula (5) becomes

$$h = \sqrt{\frac{6 \times 10{,}000 \times 8}{10{,}000 \times 1.5}} = 5\tfrac{5}{8} \text{ inches.}$$

As the rocker shaft is usually supported by the rocker box for its entire length, it is subjected to torsional stresses only, and these are covered by the general formula:

$$P = \frac{S \pi D^3}{16 R} \qquad (6)$$

in which

P = maximum load on, or resistance to the motion of, the rocker arm,

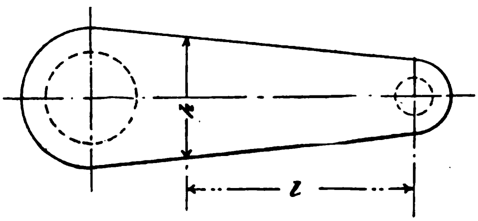

Fig. 10. Rocker Arm

S = allowable fiber stress in the metal,
D = diameter of rocker shaft,
R = length of rocker arm,

the torsional moment of resistance being expressed by $\frac{\pi D^3}{16}$ for a round solid section.

As it is the diameter of the shaft that is to be found, the formula should be transformed into

$$D = \sqrt[3]{\frac{16 P R}{\pi S}}$$

and then substituting $P = 10{,}000$; $S = 10{,}000$; and $R = 10$ inches, we have

$$D = \sqrt[3]{\frac{1{,}600{,}000}{31{,}416}} = \sqrt[3]{51} = 3\tfrac{3}{4} \text{ inches, approximately.}$$

The link should be strong enough to sustain the thrust of the link block when unsupported by the saddle. The formula used is a modification of the general formula for a beam fixed at the ends and loaded in the middle and is

$$P = \frac{4Sbh^2}{6L} \qquad (7)$$

in which

P = the stress imposed by the valve,

b = width of link across the face,

h = thickness of metal in link,

L = length of slot.

Let $S = 10,000$; $L = 15$ inches; and fix the width at $3\frac{1}{2}$ inches; then the formula for the thickness h becomes

$$h = \sqrt{\frac{6PL}{4Sb}} = \sqrt{\frac{900,000}{140,000}} = 2\frac{1}{2} \text{ inches.}$$

The eccentric rods have already been referred to and they can be calculated as indicated with the understanding that, owing to their position and the liability to cramping and the imposition of excessive stresses due to looseness of the parts, it is well to give them a strength capable of resisting a stress 25 per cent in excess of the calculated resistance of the valve.

Finally as to the eccentrics, the only point to be covered is the width of the face. The diameter is usually fixed by the diameter of the axle, the throw, and the constructional requirements of the type of eccentric that is to be used. The width of the face, therefore, is determined solely by the amount of pressure that it is decided to put upon it per square inch of area. As in the case of all other bearings, an ample surface is a good investment and will repay in immunity from hot and seizing straps in the future operation of the machine. It is well, therefore, to limit the pressure to 250 pounds per square inch measured by a multiplication of the diameter by the width of the face.

This, then, closes the outline of the work to be done in the calculation of the dimensions of the several parts of the valve motion and it remains to examine the methods of application and the modifications that will have to be made in order to adapt the formulas to the two locomotives under consideration.

CHAPTER III

DESIGNING THE VALVE MOTION*

The principles upon which the several parts of the valve motion are designed having now been set forth, the next step will be the study of the application of these principles to the two engines that we have under consideration. In this work the start is made at the engine cylinders where the point affecting the whole of the valve motion is to be found in the width of the steam ports. It has been stated that the area of these should be from 1/10 to 1/12 that of the cross section of the cylinder. By referring to the cylinder drawings it may be assumed that we find that for the consolidation engine, with a diameter of 21 inches, the port measures 18 inches by 1½ inch, giving it a ratio of 1 to 12.8 to the cylinder section. As the piston speeds on this

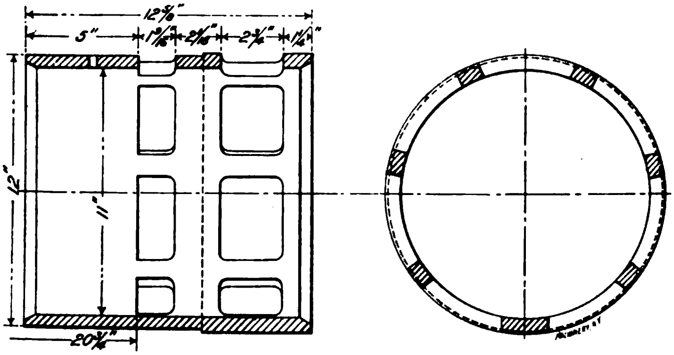

Fig. 11. Valve Chest Bushing for Atlantic Type Locomotive

engine are to be comparatively low, and as it is always desirable for constructional reasons to keep even dimensions at all times, this variation is allowable.

In the case of the Atlantic type engine, we assume that the drawing shows that the port opening in the cylinder casting is 2¼ inches wide. This is evidently too wide and is designed for the use of a bushing to be pressed into the interior. The use of a bushing for piston valves has the three-fold advantage of making the castings of the cylinder ports easier, of facilitating the cutting of the admission ports to the exact size, and of making it possible to renew the steam chest without reboring it. Accordingly bushings like that shown in Fig. 11 are pressed into the space for the valve. In this the width allowed for the port is 1 9/16 inch, and it extends entirely around the bushing except for six bridges 1 inch wide and one 2 inches wide at the bottom. There is also a space surrounding the whole port opening in the cylinder

* MACHINERY, Railway Edition, July and August, 1905.

casting by which steam can flow around the bushing into the cylinder. The diameter of this space is about 16 inches and it is 2¼ inches wide. The outside diameter of the bushing is 12 inches. The available opening for the flow of steam with this bushing in position is, at a maximum $(1\,9/16 \times 12) + (16 - 12) \times 2\frac{1}{4} = 27\frac{3}{4}$ square inches.

In the case of this engine a bushing is used in the cylinder as well as the valve case, and the inside diameter of this bushing is 19½ inches. This makes the ratio of port opening to cylinder area very nearly as 1 to 10.75, from which it will appear that due allowance has been made for the difference in the maximum piston speed of the two engines. The ratio of length of port to width is approximately 12 to 1.

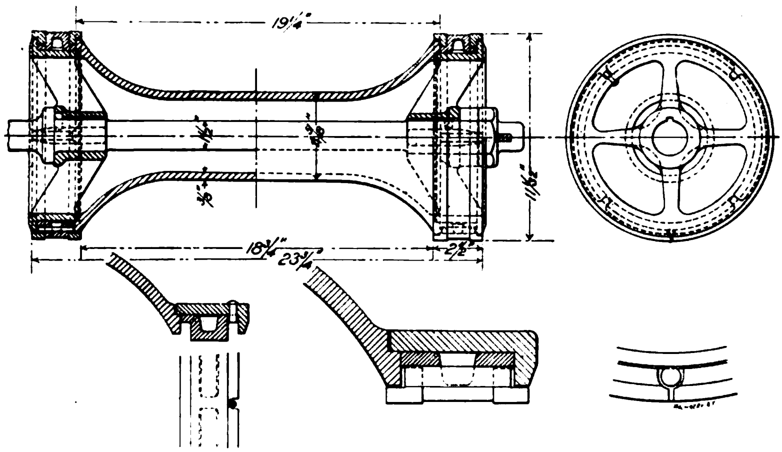

Fig. 12. Piston Valve for Express Passenger Locomotive, Atlantic Type

In the case of both of these engines the width of the ports is about 1½ inch, so that the travel of the valve should be 5¼ inches. Taking this figure for the Atlantic type engine, the valve diagram, Fig. 13, can be constructed according to the directions already given.

Before starting on this it may be stated that, in the case of high-speed engines, it is frequently more troublesome to get the exhaust steam out of the cylinder than it is to get the live steam into it. It is, therefore, common and approved practice to give the valve a negative exhaust lap. That is to say, the valve, when in its central position, leaves both ports uncovered to a small extent on the exhaust side. This makes the exhaust opening a little earlier than would otherwise occur and so hastens the outflow of steam and cuts down the back pressure. For this reason, the valve on this Atlantic type engine is given 1/16 inch negative exhaust lap.

As certain points in the designing of the valve motion must be decided arbitrarily, we will assume a negative exhaust lap of 1/16 inch, a lead in full gear of 1/16 inch and a maximum point of cut-off of 0.83 of the stroke. In Fig. 13 draw the circle *ABC* with a diameter of 5¼ inches, the assumed travel of the valve, and at *A* describe the lead circle 1/16 inch in radius. Draw the diameter *AC* and lay off *D*, making *AD* 0.83 of *AC*. From *D* erect the perpendicular *DE* and from *E* draw a line tangent to the lead circle. From the center *O* draw a circle tangent to the line *AE*, and its radius will be equal to the lap of the valve which will be found to be 1 inch in the present instance. By drawing *OF* at right angles to *AE* and the arc of the port opening *IH*, the angle of full port opening is obtained. In like manner the other elements of the motion of the valve may be studied. We have determined from this that the steam lap of the valve should be 1 inch.

In designing the valve, which is intended for inside admission, we take the distance between the ports on the bushings (Fig. 11) which is 20¾ inches, allowing 1 inch for lap on each side, making 18¾

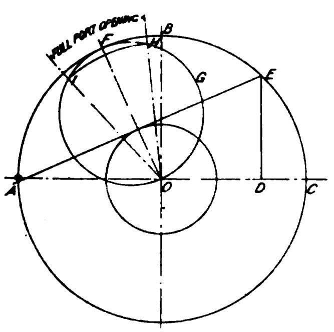

Fig. 13. Valve Diagram for Atlantic Type Locomotive

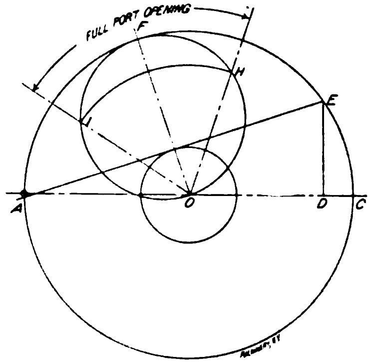

Fig. 14. Valve Diagram for Consolidation Locomotive

inches as the distance between the lips of the valve. The distance between the outside edges of the ports is 23⅞ inches, and allowing 1/16 inch for negative lap at each end makes the over-all length of the valve 23¾ inches.

With these dimensions it is possible to construct the valve shown in Fig. 12. In this the body is of cast iron with spring packing rings, the former being hollow so as to secure a perfect balance and permit the exhaust steam to escape through the center and flow out of the passages at each end of the steam chest. The packing can be of any desired type, that here shown being sprung in and turned 1/32 inch larger than the bore of the steam chest.

In the case of the flat valve of the consolidation freight locomotive, similar assumptions must be made, but they must be based upon different conditions of service. In the first place the engine is to be worked more slowly, and it must be able to exert its maximum tractive effort. For these reasons it is desirable that the maximum point of cut-off should be later and the period of full port opening longer. Hence it will be found to be advisable to increase the travel of the valve which may well be brought up to 6 inches, and by putting the maximum point of cut-off at 9/10 the stroke, the lap of the valve, with

1/16 inch lead in full gear, becomes ⅞ inch. The diagram, Fig. 14, which corresponds to Fig. 13 for the express engine, shows the features desired. It is given in order to illustrate the difference that will be found in the diagrams of high- and low-speed engines. From this the lap will be found to be ⅞ inch, and this with the dimensions we assume as taken from the cylinder drawing makes the outside dimensions of the flat valve 21 inches. On engines of this character negative exhaust or inside lap is unnecessary, and the valve is made line and line. It may be balanced in any desired manner in accordance with the general proportions already laid down. A valve proportioned to meet these requirements is shown in Fig. 15.

It may be noted here that the slide valves should be made of hard cast iron and of a size suited to meet the conditions of the steam

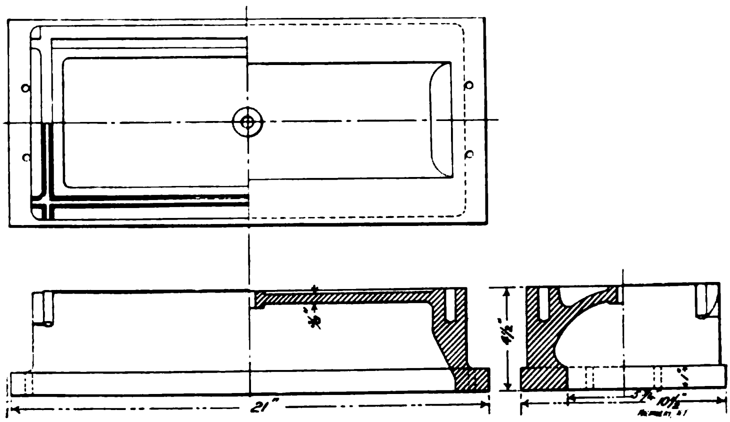

Fig. 15. Balanced D-valve for Consolidation Locomotive

ports. It will be found that on ordinary engine service the outside or steam lap will vary from ⅞ inch to 1¼ inch, dependent upon the service required and the capacity of the cylinder. The lead allowed does not ordinarily exceed 1/16 inch in full gear, and is often made line and line especially when the construction is such that the eccentric rods are less than 4 feet long. The negative exhaust lap varies from line and line to ⅛ inch for high-speed engines.

The balancing of the slide valves is of special importance and the parts should be well and accurately made. Two general methods are in use, of which the oldest is known as the Richardson. It consists of ½-inch by 1½-inch strips set in suitable grooves and resting on springs, thus forming a rectangular enclosure on the top of the valve and bearing against a smooth balance plate above the valve.

A later form is the American balance which consists of a conical ring cut through at one point, and fitted to a taper bearing on top of the valve. A cover piece similar in section to the Dunbar L-section ring is employed to cover the joint. This packing requires no springs since its reaction on the taper bearing due to its elasticity and the steam pressure tend to lift it; hence the upper part of the ring bears against the slides upon the balance plate in the same way as the Richardson valve.

The piston valve is usually made with a hollow cast iron body with followers and packing rings of an L-section at each end, and is worked inside a bushing through which the steam ports are cut as previously described. The L-shaped rings possess an advantage over square rings in that they give a positive opening and closing edge, whereas this is not necessarily the case with the latter, because the projection of the follower which holds them in position interferes somewhat with the free passage of the steam, if not taken into account as a part of the lap; while it is liable to leak, causing wire-drawing and excessive pre-admission if it is so considered. On account of the many bridges

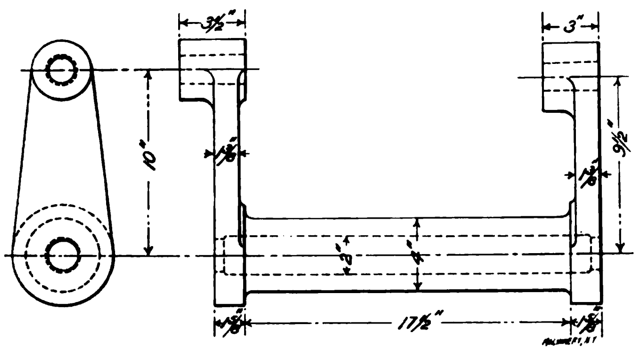

Fig. 16. Direct Motion Rocker Arm for Atlantic Type Locomotive

on the ports of the bushing and the more or less checked flow of steam past the top of the valve towards the cylinder, the diameter of the piston valve should be such that the circumferential ports should be at least 50 per cent longer than those for a flat slide valve. Further, the valve should invariably be made for inside admission, so that the stuffing-boxes are not obliged to sustain boiler pressure, and the valve events can be made the same as with the ordinary slide valve.

With the proportions of the valve thus worked out, the designing of the other ports of the motion becomes a comparatively simple matter. Before this can be done, however, it is usually necessary to locate some of the other parts, and work out the frames and driving mechanism so as to insure proper clearances when the engine is in motion. Such work usually decides the location of the link, rocker-boxes and connections. As would be clearly shown in a side elevation of the engines, the arrangement of the working parts of the two are quite different. In the case of the Atlantic type engine, the valve has a direct motion from the eccentrics obtained by the use of a rocker like that shown in Fig. 16. From this it will be seen that the valve arm is longer than the eccentric arm, so that the throw of the latter is less than the travel of the valve. According to the proportions given, the throw of the eccentric should be 4.98 inches. This can be made 5 inches, which will be done.

In the case of the flat valve of the consolidation engine, an indirect motion rocker like that shown in Fig. 17 is used. Here, too, it has been found to be convenient to make the eccentric arm shorter than the one driving the valve stem. With the proportions chosen, the throw of the eccentric becomes 5.08 inches, and it is made 5 inches as before, thus modifying to a very small extent the several valve events as found from the diagram, but not enough to materially affect the action of the engine.

With the valve and the eccentric proportioned, it becomes possible to work out the details of the whole motion. Starting with the eccen-

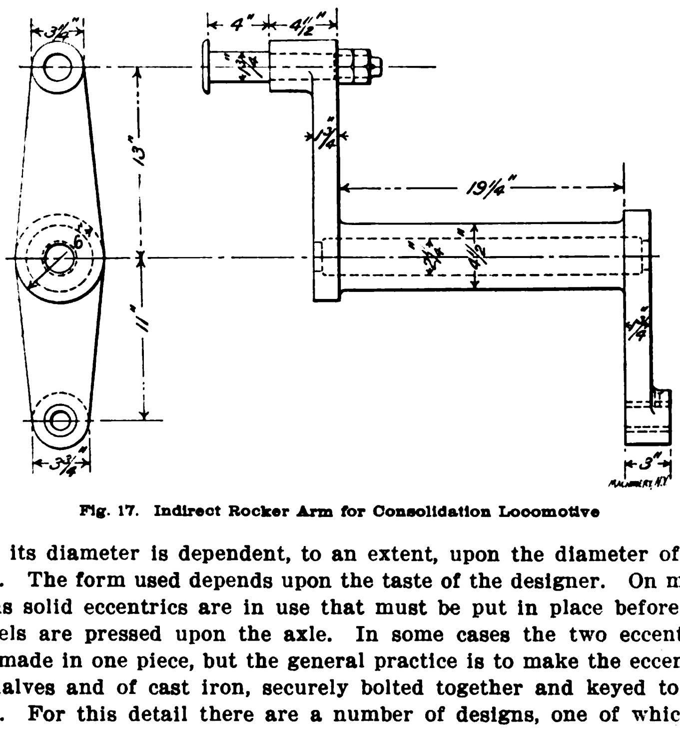

Fig. 17. Indirect Rocker Arm for Consolidation Locomotive

tric, its diameter is dependent, to an extent, upon the diameter of the axle. The form used depends upon the taste of the designer. On many roads solid eccentrics are in use that must be put in place before the wheels are pressed upon the axle. In some cases the two eccentrics are made in one piece, but the general practice is to make the eccentric in halves and of cast iron, securely bolted together and keyed to the axle. For this detail there are a number of designs, one of which is shown in Fig. 18. It is made as small as possible, consistent with strength. If the axle is assumed to be 9 inches in diameter, at least 1 inch of metal should be allowed on the thin side, which would make the outside diameter 16 inches; the allowance for re-turning makes 16¼ inches. The width of the eccentric must be sufficient to permit of keying without tilting, and bolting the two parts together without danger of splitting or cracking.

We have seen that the eccentric must have ample strength to drive the valve under the most adverse conditions when the packing strips are broken and the surface dry; it must also have ample bearing area to prevent heating. Substituting the values of the valve in Formula (4) we have a boiler pressure (P) of 200 pounds; a length (l) of 21 inches, and a width (w) of 10½ inches. Hence the resistance

$$R = 0.2 \times 200 \times 21 \times 10.5 = 8820 \text{ pounds.}$$

As the pressure on the bearing surface should not exceed 250 pounds to the square inch, the area of this surface should not be less than

$$\frac{8820}{250} = 35.3 \text{ square inches.}$$

As the diameter is 16¼ inches, the width should not be less than 2 3/16 inches. As an increase over this is desirable, and as there is

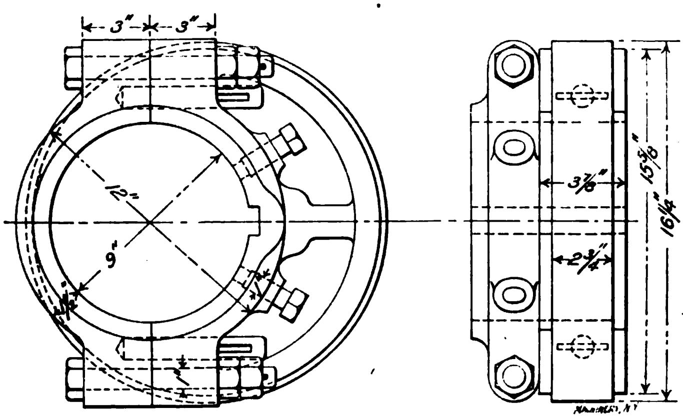

Fig. 18. Eccentric for Consolidation and Atlantic Type Locomotives

plenty of room, it is made 2¾ inches, which cuts the pressure down to about 200 pounds, thus giving good working conditions so far as bearing surface is concerned even under exceptional resistances.

The eccentric is keyed to the axle, as adjustment of this part and fastening by setscrews are things of the past and are entirely obsolete methods of construction. The details of the form shown here are sufficiently distinct in the engraving to require no explanation. Attention is merely called to the fact that every precaution in the way of check nuts and cotters is used to prevent the parts from becoming loose or lost.

Closely allied with the eccentrics are the eccentric straps. They may be of cast iron or bronze and must have a strength sufficient to move the valve under adverse conditions without an appreciable amount of yield. This is necessary in order that they may preserve their full bearing surface in contact with the eccentric at all times and not pinch the latter because of some distortion or yielding. Formula (7), which is the general one for a beam fixed at the ends and loaded in the middle, may be used for the calculation of the body of the strap. It may be modified, however, to

$$P = \frac{4\,Q\,S}{l} \qquad (8)$$

in which

P = the stress imposed by the valve,

S = allowable fiber stress in the metal,

l = distance between fastening bolts in inches,

Q = the section modulus.

The latter must be worked out for all sections other than a rectangle which is $\frac{b\,h^2}{6}$ as already given. In the case of the strap shown in Fig. 19, the width is 3¾ inches and the depth 2⅞ inches. The section is flat on one side and semi-circular on the other, and if the computa-

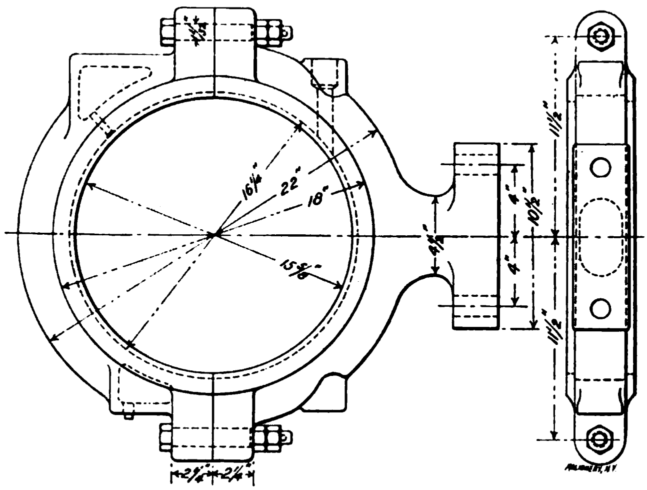

Fig. 19. Eccentric Strap for Consolidation and Atlantic Type Locomotives

tions are made as to its strength it will be found that the maximum fiber stress put on the metal will fall below 2500 pounds per square inch of section. This is low, but it is one of those places where plenty of metal is a good investment as it insures against hot straps and annoying delays upon the road when the engine is in service. The neck of the strap is also made of ample strength and the foot is faced to receive the direct thrust of the rods. The bolts used for the fastenings should be of ample size to hold the parts firmly together, and when this is done their strength will be sufficient to carry the load that is put upon them. The fastenings of the eccentric rods to the straps is made much more secure than it was when engines and valves were lighter. At that time there was a possibility of adjusting the length of the rods, with the result that they frequently slipped in service. Current practice does away with this adjustability with a resultant simplification of the parts.

Fig. 20 shows a substantial form of eccentric rod that is used on the engines under consideration. It will be seen that it is made in "rights" and "lefts" so that the jaws at the forward end line up together to take the link. It is of the utmost importance that these rods should be exceedingly stiff and rigid in order to prevent springing when they are working under compression. If that occurs the action of the valve is not what it should be, and an extra and unnecessary stress is put upon the bolts, pins, links and eccentric straps, all of which tends to a more rapid wear and an increase of the cost of maintenance, to say nothing of the danger of causing delays and breakdowns on the road.

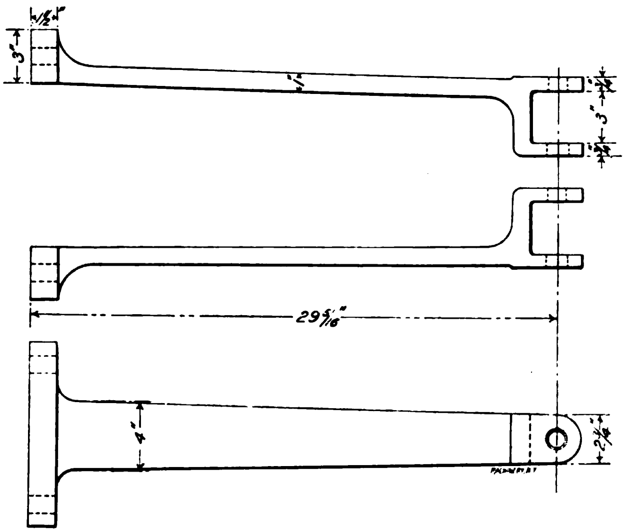

Fig. 20. Eccentric-rod for Consolidation Type Locomotive

The distance from the center of the axle to the rocker arm, or the radius of the link, is 49 inches in the case of the consolidation engine and 60 inches for the Atlantic type. This causes a variation in the length of the eccentric rods of from 29 5/16 inches to 40 5/16 inches. As these rods are flat they must be calculated accordingly. We have seen that the probable maximum load imposed by the valve will be 8820 pounds. In order that there may be an ample margin of strength, it will be well to take this load at 10,000 pounds and proportion the parts accordingly. By using the following formula for this purpose, and assuming the length of the rod to be 30 inches and the thickness 1 inch, we have

$$\frac{P}{A} = \frac{S}{1 + \frac{q\,l^2}{r^2}}$$

in which

P = resistance of the valve = 10,000 pounds,

A = area of rod = width of rod $\times$ 1,
S = allowable fiber stress = 10,000 pounds per square inch of section,
l = length of rod = 30 inches,
$r^2 = 1/12$,
q = 0.00016.

By substitution and transposition this formula becomes

$$A = \frac{(1 + 0.00016 \times 900 \times 12) \times 10,000}{10,000} = 2.73.$$

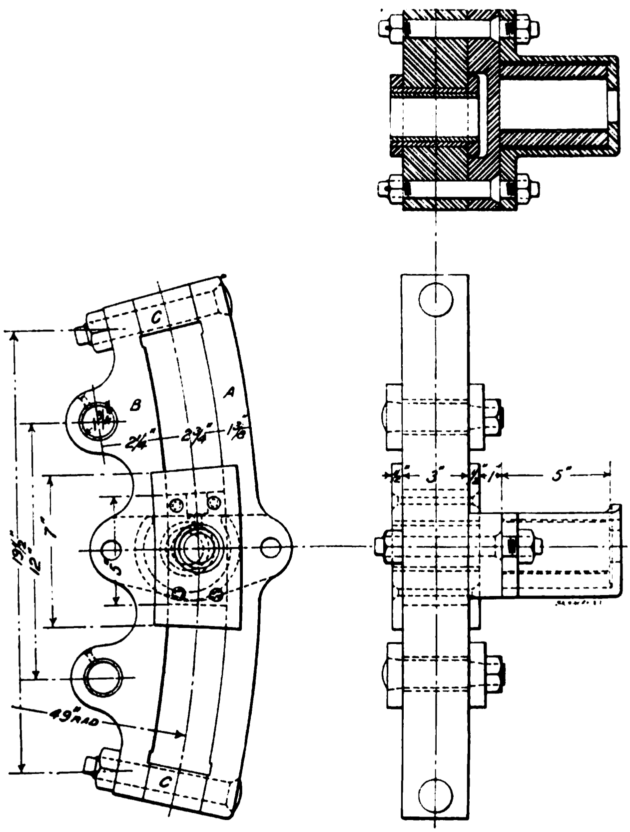

Fig 21. Wrought Iron Case-hardened Link for Consolidation Locomotive

The rod can, therefore, be made $2\frac{3}{4}$ inches wide at the small end and tapered to any desired width (4 inches in this case) at the large.

The diameter of the valve stem can be calculated by the same formula by changing the value of r^2 to $\frac{d^2}{16}$, making $A = \frac{\pi d^2}{4}$, and making l = about 85.

By substitution and transposition, the formula becomes

$$\frac{4P}{\pi d^2} = \frac{10{,}000}{1 + \dfrac{(0.00016 \times 7225) \times 16}{d^2}}, \text{ and finally}$$

$$d = \sqrt{\frac{2 + \sqrt{74\pi + 4}}{\pi}} = 2.35,$$

so that the valve rod may be made about 2⅜ inches in diameter.

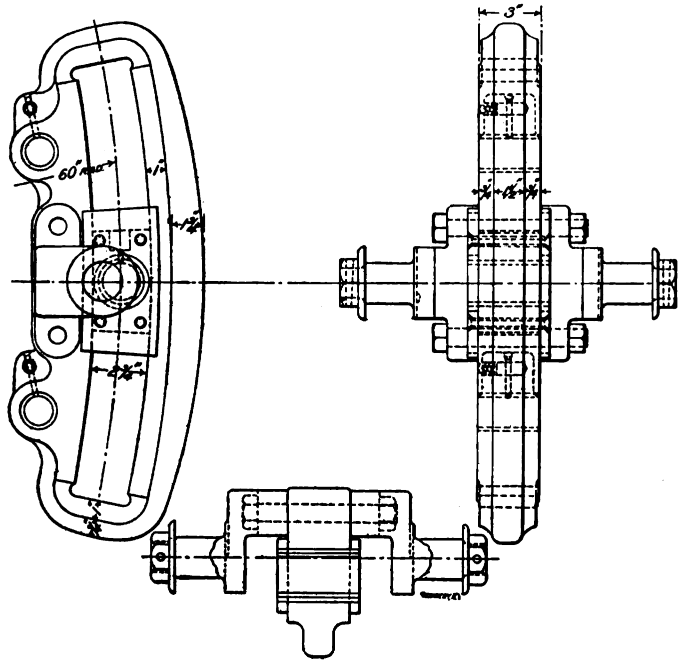

Fig. 22. Cast Steel Link for Atlantic Type Locomotive

The link is also a part that needs most careful attention not only in the designing and proportioning, but in the manufacture. For many years the links on American engines were made of wrought iron, case-hardened and carefully ground to truth. Of late cast steel has been introduced with good results as far as operation is concerned and with a considerable saving in first cost. This metal also possesses the advantage of permitting a better distribution to carry the loads, and can, therefore, be made lighter.

The common form of skeleton link is shown in Fig. 21. It is adapted for use on the consolidation locomotive and owing to limits of space it has been made a trifle shorter than the general proportions given. With an eccentric throw of 5 inches the distance between the eccentric rod pins should be 12½ inches, but in this case it is made 12 inches. The link is built up of four parts, the front *A*, the back *B*, and the two fillers at the ends *CC*. The holes for the eccentric rod pins are protected by case-hardened bushings, and the block is given an ample bearing surface 5 inches long. As in the case of the eccentric rods the link must be of ample strength to do the work without springing. For this we may use Formula (7) though it is common practice to depend upon the supporting of the saddle to prevent springing. Then, again, as the load of the unlubricated, unbalanced valve is excessive, and as it is desirable to make the link as light as possible so that it may be easily handled in reversing, and on account of the support of the link saddle, the stress allowed for may be dropped to 5000 pounds instead of making it 10,000, and the allowable fiber stress may be raised to 12,000 pounds. This leaves the link with ample strength for the ordinary working while in the case of an accident it will not be overstrained on account of the length of the bearing of the link block. The formula as thus modified becomes:

$$P = \frac{4\,S\,b\,h^2}{6\,L}, \text{ in which}$$

$P =$ load imposed by the valve $= 5000$ pounds,
$b =$ width of the link across the face $= 3$ inches,
$h =$ thickness of metal in the link,
$L =$ half the length of the slot $= 9.75$ inches,
$S =$ allowable fiber stress $= 12{,}000$ pounds per square inch section.

$$h = \sqrt{\frac{5000 \times 6 \times 9.75}{4 \times 12{,}000 \times 3}} = \sqrt{2.03} = 1.42$$

from which the thickness may be made 1 7/16 inch.

In case of the cast steel link shown in Fig. 22 for the passenger locomotive, a similar course of reasoning can be followed, except that the fiber stress put upon the metal should be kept down to 10,000 pounds, or even less. It will also be well to work out the section modulus as this has an important bearing on the rigidity of the link.

The method of calculating the size of the rocker has been provided for in Formulas (5) and (6).

Taking the rocker for the consolidation engine as shown in Fig. 17, if the valve resistance is placed at 10,000 pounds, the length of the arm at 13 inches and the thickness at 1¾ inch, then using Formula (5).

$$P = \frac{S\,b\,h^2}{6\,l}, \text{ in which}$$

$P =$ valve resistance $= 10{,}000$ pounds,
$S =$ allowable stress of metal $= 12{,}000$ pounds,
$b =$ thickness of arm $= 1.75$ inch,
$h =$ width of arm,
$l =$ length of arm $= 13$ inches.

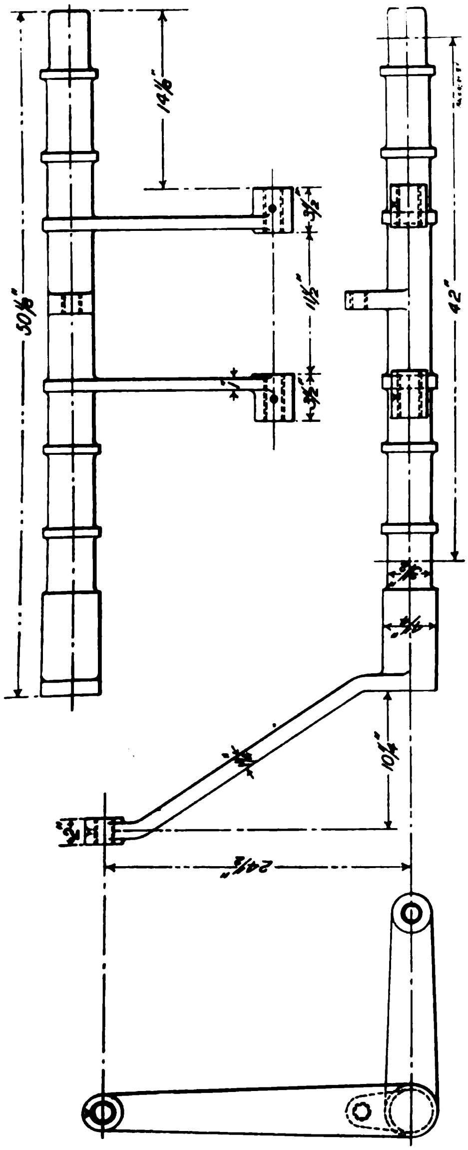

Fig. 23. Lifting Shaft for Consolidation Locomotive

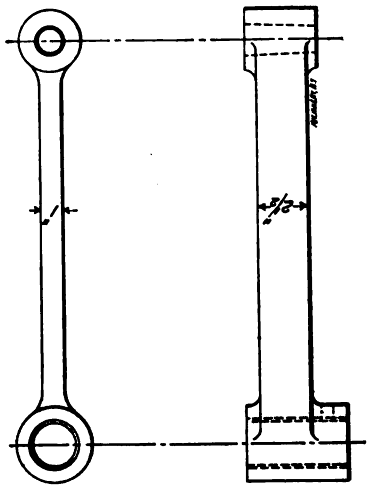

Fig. 24. Link Hanger for Consolidation Locomotive

By transposition and substitution this becomes

$$h = \sqrt{\frac{6 \times 13 \times 10{,}000}{1.75 \times 12{,}000}} = \sqrt{37} = 6 \text{ inches, approx.}$$

As it is always well to have an ample bearing surface for the rocker, and as lightness is a desirable quality in the moving parts, cast steel is used for material and the bearing is made hollow. By assuming a core of 2¼ inches and deducting the value of the metal thus removed from the strength of the shaft, this core would be able to carry a load

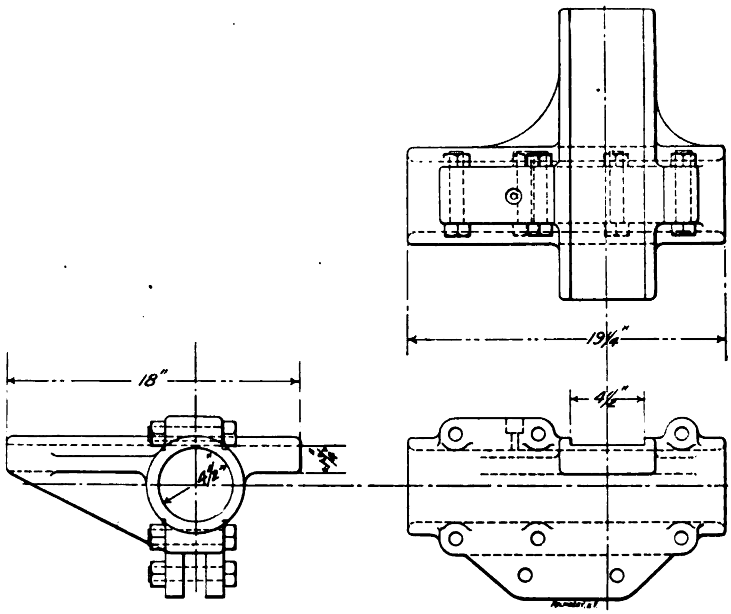

Fig. 25. Rocker-box for Consolidation Locomotive

of about 1750 pounds if calculated by Formula (6). This load is then added to the actual valve pressure, making $P = 11{,}750$ pounds.

$$D = \sqrt[3]{\frac{16\,P\,R}{\pi\,S}}, \text{ in which}$$

$D =$ diameter of rocker shaft,
$P =$ valve resistance $= 10{,}000 + 1750$ pounds,
$R =$ length of rocker arm $= 13$ inches,
$S =$ allowable fiber stress $= 10{,}000$ pounds.
By substitution the formula then becomes

$$D = \sqrt[3]{\frac{16 \times 11{,}750 \times 13}{31{,}416}} = \sqrt[3]{77.8} = 4.27 \text{ inches.}$$

In this case the diameter is made 4.5 inches. These are the outlines of the methods to be followed in the proportioning of the parts of the valve motion.

The lifting-shaft (Fig. 23) and the link-hanger (Fig. 24) are important parts that must be carefully designed and located. In the matter of strength, if they are made stiff enough to do the work, that is all that is required, and their only load is to carry the weight of the link and resist the forces due to the angularity of the link and the binding action of the block in its slip. They are, however, closely associated with the proper location of the saddle-pin on the link, for on the proper combination of dimensions and positions of the saddle-pin, hanger, lifting-shaft and box depends the smooth and correct action of the valve. All parts of the valve motion may be carefully and accurately laid out and if the bearing of the lifting-shaft is improperly located, the action of the valve will be defective.

An outline of the method to pursue is to take a templet of the link (full size preferred) and locate it in the extreme positions of mid-gear. On transverse lines through the centers of the two positions of the templets locate points equally distant from the center which coincide with the arc swept through by the lower end of the hanger when the lifting shaft is in mid-position. These points will indicate the proper position of the saddle pin. The lifting-shaft box should be so adjusted that the lower end of the hanger will sweep through the corresponding position of the saddle-pin when the link is raised and lowered to full backing and forward gears respectively. The designer should make himself familiar with all of the vagaries and peculiarities of the Stephenson valve motion, and this involves a careful study both of what has been published and of the working out of the problems on the drawing board.

It only remains now to call attention to the form of rocker-box that is used, that intended for the consolidation locomotive being shown in Fig. 25. These boxes are usually bolted to the guide yoke, are made of cast iron and afford a support for the rocker bearing throughout its whole length. All pin holes of the working parts of rockers, link and rods are protected by case-hardened bushings.

With this outline of the course to be followed in working out the Stephenson link motion the reader is cautioned against trusting to any haphazard methods of design and recommended to master its intricacies in every particular before attempting to make a practical application of a design to a locomotive.

CHAPTER IV

THE WALSCHAERTS VALVE MOTION*

Until very recently, whenever an American has considered the designing of a locomotive, the work has invariably been associated with the use of the Stephenson link motion for the operation of the valves. This is true with the exception of a very few instances where interested parties have had some special design of gear to exploit. That the Stephenson gear has held its own for so many years speaks well for its efficiency, and indeed it has been found that in the matter of steam consumption, in special cases it holds this figure down to within a very small percentage of the best that can be obtained with the Corliss gear.

On the continent of Europe, on the other hand, the Walschaerts gear or a modification of it is almost exclusively used; and it is claimed to possess many advantages over the Stephenson motion, one of the more prominent of which is the maintenance of a constant lead for all points of cut-off, an advantage that is not universally acknowledged, however. During the last five or six years the Walschaerts gear has also received a more and more extensive application in this country, hence it is necessary to discuss and analyze it in considering the designing of a locomotive.

The reason for the change of attitude regarding the Walschaerts gear is due to a number of difficulties that have been experienced with the Stephenson valve motion on the large and heavy locomotives of modern construction. Among these are the excessive wear of the heavy eccentrics and straps, and the large amount of space between the frames that is occupied by the eccentrics, rods, links, and hangers, making it exceedingly troublesome or quite impossible to properly brace the frames; the Walschaerts gear has, therefore, been very successfully applied to several thousand engines of recent design.

It is more accessible than the Stephenson motion in that it is applied outside the wheels and requires only a single eccentric return crank and a connection to the crosshead for its operation. The eccentric crank may be a comparatively small pin attached to the main crank, and it does the work of the two heavy eccentrics of the ordinary gear. This arrangement leaves the entire space between the frames clear for the bracing of the same.

Further, it produces a more uniform steam distribution with a lower percentage of pre-admission, to which is added a constant and moderate amount of lead for early cut-off, though on the resultant economy in steam consumption there can be but slight difference when both the gears are in first-class condition. Finally, it is not so likely to get out of

* MACHINERY, Railway Edition, September, 1905.

order as when the driving is done by eccentrics; since, when the parts have been properly fitted, they are not liable to get out of place except when damaged by collision or other accident to which they are more exposed.

The Walschaerts gear is, in reality, much simpler to understand than the Stephenson. Of course, since the results of the two are nearly identical, each must have parts whose functions correspond to those of the other. In the case of the Walschaerts gear the valve receives its motion from two sources, the crosshead and an eccentric crank whose center is located 90 degrees from the center line on the main crank, when the center lines of the cylinder and gear motion coincide and pass through the center of the axle.

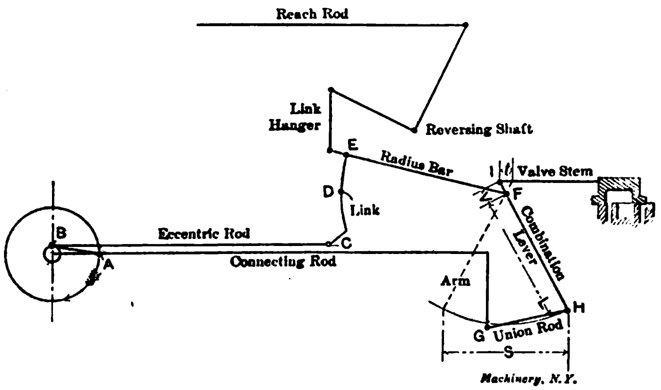

Fig. 26. Diagram of the Walschaerts Valve Gear

By referring to Fig. 4 and the description accompanying it, it will be seen that, as the crank stands on the center, the eccentrics at C and D are given an angular advance which is equal to the sum of the lap and lead of the valve in full gear. In the case of the Walschaerts gear, which is shown diagrammatically in Fig. 26, when the crank is on the forward center at A, the center of the eccentric is at B at right angles to the crank, provided the point C is close to the center line of the cylinder. From B an eccentric rod runs to C, the lower end of a fixed line CDE which is pivoted at D. This link has a groove in which the link block attached to the radius rod slides up and down. The length of this rod from E to F is equal to the radius of the link itself. If now the valve stem were to be connected directly to the end of the radius bar, it is evident that when the crank is on the center, the valve would be in its central position either for forward or backward motion and for any position of the link block of the radius bar in the link. Under these circumstances, there could be no lap or lead to the valve.

The lap and lead is arranged for by dropping a rigid arm down on the crosshead to G and from this a union rod is led out to the lower end of the combination lever at H, this lever being pivoted at the end of the radius bar at F and extending up to the valve stem connecting at I. It is evident that the inclination of the combination lever will be the same at the end of the stroke regardless of the position of the radius bar; and that, therefore, the horizontal displacement of the point I and the valve stem will be the same on either side of a vertical line through F. This horizontal displacement is equal to twice the sum of the lap and the lead, hence the latter is constant for all points of cut-off. These same statements hold true for the opposite end of the stroke, when A is on the back center and B at the bottom, barring a slight variation due to the angularity of the rods.

In the case shown, where the eccentric B follows the crank the engine runs ahead when the radius block is at the top of the link and backwards when it is at the bottom. These conditions would be reversed by having the eccentric lead the crank. The crosshead, then, imparts a motion to the valve equal to the lap and lead when the crank is on either center, just as the angular advance of the eccentrics does in the case of the Stephenson gear.

It will be noticed that the eccentric and the crosshead tend to move the valve in opposite directions during the first half of each stroke and in the same direction during the last half; or, in other words, they work in opposite directions during the first and third quarters of a revolution of the crank starting from either dead point, and together during the second and fourth quarters. The motion derived from the crosshead is constant and is not subjected to reversal in the reversing of the motion of the engine, which is done entirely by a change in the motion imparted by the eccentric, which also controls the variation of the points of cut-off.

In order to accomplish this the motion of the eccentric is transmitted through an oscillating link pivoted at its center and so slotted that a link block attached to the back end of the radius bar can be moved through its whole length, and by placing this above or below the center, a reversal of the engine will be obtained. This motion, either direct or indirect, is taken up by the radius bar and carried out to the combination lever, where it is combined with that obtained from the crosshead and the resultant imparted to the valve. The motion is therefore the same as though it were derived from an eccentric the center of which could be moved on the line of a chord across the axle from one extremity to the other of the throw.

It is evident from this that the several connecting points along the combination lever bear a most important relationship to each other, which must be maintained in order that a proper movement of the valve may be obtained.

The distances between these several points may be found by the formula

$$S : t = L : V, \text{ or } V = \frac{L\,t}{S} \qquad (9)$$

in which

S = stroke of piston,

$t =$ twice the sum of the lap and lead,

$L =$ distance between the crosshead connection H and that of the radius bar F, Fig. 26,

$V =$ distance between connection of the radius bar F and that of the valve stem I, Fig. 26.

For an outside admission of steam, as in the case of the ordinary slide valve, the connection F, Fig. 26, falls below that of the valve stem so that the crosshead increment of the motion is the same as though it were derived from an eccentric or crank opposite the main crank; while the increment controlling the direction of engine rotation is derived from what amounts to an eccentric leading the crank for forward motion, just as in the case of direct-acting eccentrics in the case of the Stephenson gear.

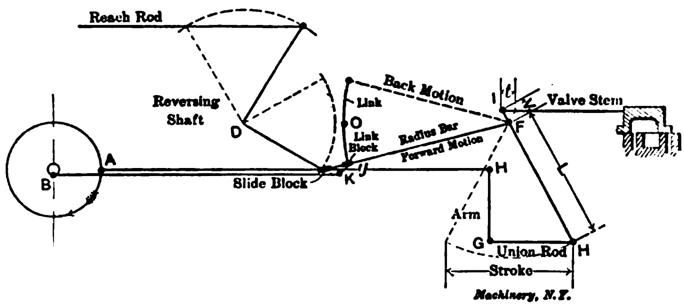

Fig. 27. Modification of the Walschaerts Valve Gear

A modification of the diagram of Fig. 26 is shown in Fig. 27, where it is arranged so that the block shall be at the bottom of the link for forward motion and at the top for backing, which is the reverse of that shown in Fig 26, a condition that is brought about by simply locating the eccentric center 90 degrees ahead of the crank instead of behind it.

When used in connection with a piston valve having an inside admission, the valve stem is usually attached below the connection F of the radius bar on the combination lever, and the eccentric follows the main crank when going ahead with the block at the lower end of the link, which results in the crosshead motion having the same effect as an eccentric on the same side of the axle as the crank.

The reversal of the motion is accompanied by means of a reversing shaft with lever and reach-rod attachments as in common use, and acting upon the radius bar either by a sliding block pivoted directly to the reverse arm, as shown in Fig. 27 or by a hanger from which the bar is suspended. In the first case the bar slides to and fro in the block, and the slip of the link block is equal to the versed sine of one-half of the arc through which the link moves. When the hanger is used it must be of sufficient length to avoid an excessive slip of the block. It is for this reason that the reversing shaft is frequently placed below the radius bar so that the swing of the hanger will tend to compensate for the oscillation of the link and thus reduce the slip to a minimum. In this, too, it is important to ascertain the correct location of the point of suspension of the hanger and to so place the reversing shaft that the hanger does not cause irregularities in the motion of the valve by swinging the link block out of the correct alignment with the radius bar when the former is in any of the several positions that it may occupy.

Having thus reviewed the principles in accordance with which the Walschaerts gear operates it remains to analyze the several parts and determine the relative proportions existing between them. The lap and lead motion having been obtained by means of Formula (9) the radius of the eccentric crank, or the travel of the point F of the link block for any given travel of the valve is found by the formula:

$$b = \frac{R\sqrt{a^2 - c^2}}{R + c} \qquad (10)$$

for outside admission and,

$$b = \frac{R\sqrt{a^2 - c^2}}{R - c} \qquad (11)$$

for inside admission where

a = half travel of the valve,
b = half travel of the link block at point F,
c = lap and lead of the valve,
R = radius of the main crank.

From these formulas one-half of the travel of the point F can be obtained, which for the sake of simplicity and with no appreciable error can be considered as equal to one-half the travel of the link block. In this both are considered to be moving in straight lines, which is not strictly the case, but the error is insignificant and has an effect only at the extremities of the travel, which decreases as the point of cut-off is made earlier, so that for practical reasons the difference may be ignored. In fact the method of supporting the radius bar has an influence upon the horizontal movement of the link block, so that the difference between it and the point F will have to be calculated independently for each individual case and point if a mathematical precision is to be obtained, which has been found to be entirely unnecessary in practice, when the suspension points have been properly laid out.

It will be seen that, in these last two formulas, the proportions of the combination lever are ignored, because the lap and lead as obtained from Formula (9) puts this in the same ratio to the main crank as the two arms of the combination lever are to each other. With an outside admission valve, the locus of the virtual eccentric may be considered to lie in a straight line at right angles to the center line of the crank but upon the opposite side of the axle center to that of the crank itself, or from Formula (9)

$$R : (R + c) = L : (L + V),$$

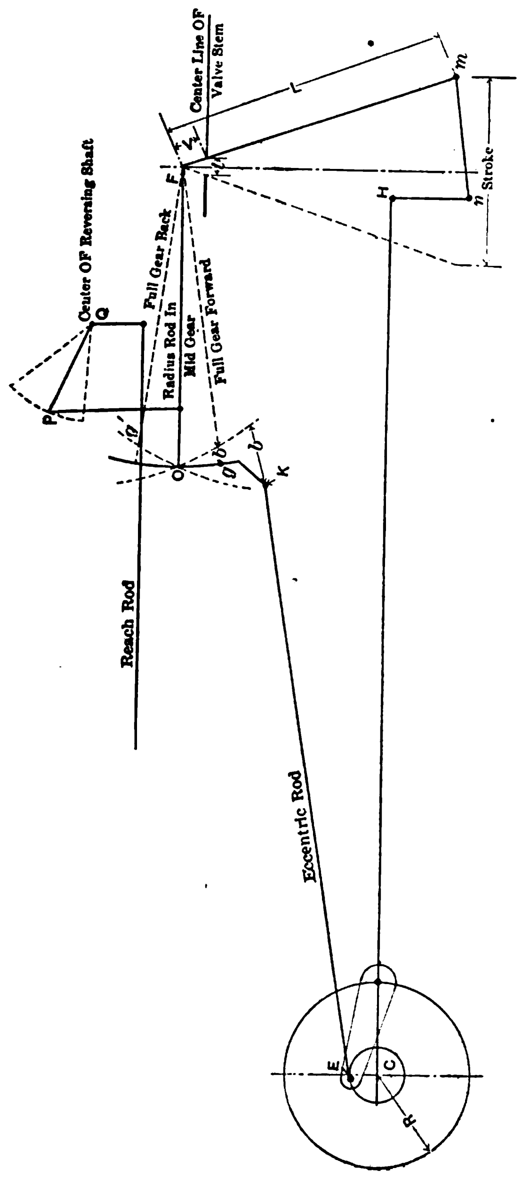

Fig. 28. Method of Reversing the Walschaerts Valve Gear

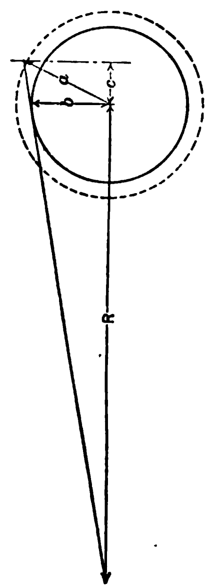

Fig. 29

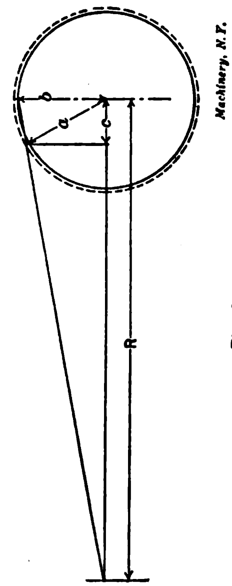

Fig. 30

while with an inside admission valve the sign is changed and we have

$$R : (R - c) = L : (L - V),$$

in which case the valve stem falls below the point F, which must, therefore, have a longer travel than with outside admission in order to maintain the same travel of the valve.

The location of the virtual eccentric at right angles to the center line of motion is equal to $\sqrt{a^2 - c^2}$, from which we may obtain the half travel of the point $F = b$ by the following proportion:

$$\sqrt{a^2 - c^2} : (R + c) = b : R$$

or

$$b = \frac{R\sqrt{a^2 - c^2}}{R + c}$$

for outside admission and

$$\sqrt{a^2 - c^2} : (R - c) = b : R$$

or

$$b = \frac{R\sqrt{a^2 - c^2}}{R - c}$$

for inside admission according to Formulas (10) and (11).

In this case b may be considered as equal to the radius of the eccentric crank, as it would be were it possible to attach the front end of the eccentric rod to the link block. This is laid out graphically in Figs. 29 and 30. If a is the same in both cases, b will be greater in Fig. 30 than in Fig. 29.

With these formulas as a basis it is possible to proceed with the determination of the actual radius of the eccentric crank. Starting with Fig. 27 as a basis, and having settled the travel of the valve and the throw of the point F, the next thing to determine is the distance Og which the link block will have to be moved from its central position to full gear. In this there are two antagonistic requirements to be reconciled. On the side of the angularity of the radius bar, it is desirable that this movement should be as small as possible, while with the angularity of the link in view it should be as large as it can be made. Hence it is necessary to compromise between the two.

Practical experience has shown that it is not well to swing the link through an angle of more than 45 degrees, so that assuming this as that to be used, we have:

$$Og = \frac{b}{\tan 22\tfrac{1}{2}\text{ deg.}} \qquad (12)$$

in which the angle given is that of half the travel on each side of the center.

The connecting point K between the eccentric rod and the link should be as near the center line of the engine as practicable. With inside admission piston valves it frequently happens, however, that the link fulcrum O is rather high, so that a large crank radius would be required in order to secure the requisite amount of motion. Consequently good judgment must be used in order to secure practical results. If, however, it is found necessary to locate this point K at any appreciable distance above the horizontal center line of the engine, the center of the eccentric crank should be set back with the same angularity so that a line drawn through it and the center of the axle will be at right angles to one drawn from the center of the axle to the point K.

The fore and aft position of K is a matter of importance and it should be such that it will swing through the same angle on each side of its central position at the same time compensating for the angularity of the rod.

No absolute formula can be given for the location of the point K, as it must be worked out for each individual case. It will always be back of the tangent to the link drawn through O in the central position, and the distance will depend upon the inverse ratio of the relation existing between the throw of the eccentric and the length of the eccentric rod. It is further influenced by the variation in the angularity between the center line of motion and the tangent to the link as well as its distance from the fulcrum O. Under ordinary conditions the point K will fall from 2 inches to 5 inches in the rear of a tangent to O.

With very short eccentric rods, there may be some difficulty in securing this equal angularity of swing, in which case the distance of K from the tangent to the link can be reduced somewhat and not materially affect the opening and closing points of the valve. The maximum port opening will be affected, it is true, but as this is usually more than that actually required, and as the irregularity gradually disappears as the cut-off is made earlier, it will have very little influence on the working of the engine.

Having obtained the horizontal motion of the link-block at the point g from Formulas (10) and (11) as well as the angular swing of the link, it is evident that the point K must move through the same angle; we thus have by making $k = b'$,

$$Og : OK = b : b'$$

or

$$b' = \frac{OK \times b}{Og} \tag{13}$$

This will also be the radius of the eccentric crank, but owing to the angle made by OK with the tangent to the link, the radius will decrease as this angle increases and must be laid out in each case to meet the conditions involved.

The location of the center Q of the reversing shaft (Fig. 28) must be such that the end of the arm QP at full gear forward will be in such a position that the lower end of the hanger will swing through an arc tangent to the radius bar at its point of attachment. An exact equality in this respect is impossible to attain throughout the whole range of cut-off, so that especial attention should be directed toward securing it over the range in which the engine is to work, which should be between a 30 and 60 per cent cut-off in forward gear for road engines, and half and full gear, front and back, for switching engines.

It will be found, then, that owing to the unavoidable irregularities inherent in the transformation of circular into rectilinear motion (inherent in all types of valve gears) the greater portion of the arc through which the point P passes is in one of the upper quadrants of the circle.

The crosshead connecting point n, Fig. 28, in relation to the point m at the lower end of the combination lever will also be found to have more or less effect upon the regularity of the motion, of which advantage can be taken in the laying out of the gear. Ordinarily the best position will be found by locating the point n on the same horizontal line with m when the combination lever is in a vertical position.

With the slide bearing pivoted directly on the lifting arm as indicated in Fig. 27, the movement of the end of the radius bar is prac-

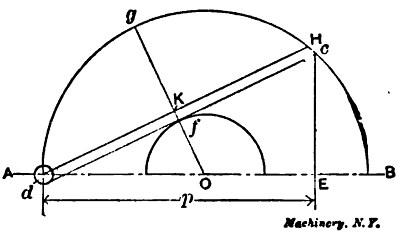

Fig. 31

tically in a straight line throughout the whole range of cut-off in both directions, a fact which greatly simplifies the construction of the gear, and is, for that reason, to be preferred to the use of a swinging hanger.

The valve diagram may be laid out in the same general way as with the Stephenson gear, but as there is no increase of lead for the earlier points of cut-off, a definite amount of lead must be given from the start, by which a complication is added to the formula for finding the lap for a given lead, valve travel and maximum cut-off. It can, however, be done as follows:

In Fig. 31 let AB be the travel of the valve and E the point of maximum cut-off. Draw Ec at right angles to AB and describe the semicircle AcB, on AB as a diameter. With the lead as a radius and A as a center, draw the circle d, and draw cd tangent to circle d. Draw Og at right angles to cd and draw the lap circle tangent to cd, the radius of which, Of, will be the required lap.

Further, draw AH parallel to cd and it will be the locus of the virtual eccentric center and

$$AO = a;\ OK = c$$

in Formulas (10) and (11).

Zeuner's diagram is as applicable here as in the case of the Stephenson gear with the difference that the locus of the virtual eccentric is a straight line instead of a curve, and the points R^1, R^2, R^3, R^4 of

Fig. 9 determine the radii for different travel circles of the cut-off as well as the corresponding diameters of their respective valve circles.

As in the case of the designing of the Stephenson valve motion, it is necessary to make some arbitrary assumptions in order to determine the dimensions of the several parts by the use of the formulas that have been developed. This is done in connection with the diagram of Fig. 28. In this we have certain dimensions already given. The first is the stroke of the piston which is 26 inches. By taking the travel of the valve at 5¼ inches and its maximum point of cut-off at 0.83 of the stroke as in the case of the Stephenson gear and fixing the lead at ⅛ inch, the valve diagram of Fig. 32 can be constructed from which the lap of 1 inch will be obtained. The sum of the lap and lead will then be 1⅛ inch. As the half stroke of the crosshead is 13 inches the ratio of the lap and lead to this motion will be as 1 to 11.55.

By first laying off an outline of the main crank, the center line of the valve stem, the connecting rod and crosshead and a vertical line

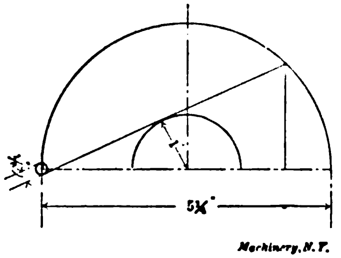

Fig. 32

passing through the center of the connection between the valve stem and the combination lever when it is in the central position, the proportions of the latter are first obtained. The length of the long arm should not be less than 2¼ times the stroke, if too great angularity of motion is to be avoided. In this case the attachment of the valve stem will be below the point F. If we assume the distance between the two to be 3½ inches then the total length of the rod becomes 40.415 inches or a little more than 40 13/32 inches. Laying off the lap and lead on the center line at t, the position of the combination lever at a forward end of the stroke is obtained. The radius of the link should not be less than eight times the travel, but must necessarily be adapted to the engine and varied according to the requirements of construction, assuming it in this case to be 42 inches. The length of the link is determined by the travel of the point F, which is ascertained from Formula (11) which by the substitution of values becomes

$$b = \frac{13\sqrt{2.625^2 - 1.125^2}}{13 - 1.125} = 2.6 \text{ inches.}$$

Then the half length of the link, Og, is obtained by the substitution of values in Formula (12), which then becomes

$$Og = \frac{2.6}{0.41421} = 6.3 \text{ inches.}$$

With the proper allowance for the length of the link block the link should be at least 8½ inches long on each side of O or 17 inches in all. If the point K of the attachment of the eccentric rod is taken at 11½ inches from the link fulcrum, the radius of the eccentric crank, in order to give the link a throw of 45 degrees will be

$$11.5 \times \tan 22\tfrac{1}{2} \text{ deg.} = 11.5 \times 0.4142 = 4.75,$$

or by substitution of the values in Formula (13)

$$b' = \frac{11.5 \times 2.6}{6.3} = 4.75$$

so that it can be made 4¾ inches.

This is, however, subject to slight modifications depending on the angle between the motion center and the radius OK, but in ordinary cases this is so insignificant that it may be left out of consideration.