Link Motion

A THESIS

BY

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as applied to the valve gear of locomotives.

The locomotive engine is perhaps, as good an embodiment as can be found, of the inventive genius and mechanical skill of our modern civilization. In no other machine or engine of parallel importance, is there a more imperative demand for strength, power, durability simplicity and lightness. The three notable things which have given life to the locomotive are, the tubular boiler and inside fire box, the steam blast pipe, and the reversing-link-motion. The first provides a large heating surface in small compass, the second gives a means of urging the fire and of making steam with a rapidity that increases with the demand for power, and the third places

this power within the complete controll of the engine driver.
The link cannot be said to be the invention of any one man or of such and such men; it was rather a growth from the necessities of the case, and its full value was not at first realized.
The first record we have of the employment of such a device was in 1832, when William T. James of New York designed an engine which had a valve gear with a link not differing greatly from the present form. Reliable accounts of the ease and quickness with which this engine could be reversed confirm reports which ascribe a true link motion to this engine, although it is not certain what was its exact form. This unfortunate engine was sent to the southern states and

blew up, carrying its link motion with it so completely, that it had no part in the after introduction of this admirable reversing gear.
We now turn to the so called Stephenson link, since it was as truly the parent of all present forms as though James link had never been invented. To illustrate the growth of this valve gear, a few drawings are introduced from Chapter III of Colburn's Locomotive Engineering from which this short historical sketch has been condensed.
In figure 1 plate I is shown the valve motion of Stephenson's Killingworth engines, used in 1829. The motion of the eccentric was carried through a bell-crank lever to the valves on the cylinders which were placed vertically, over the boiler, and the reversing was accomplished by slip-

ping the eccentric round the shaft, as is indicated by the circular slot in the eccentric disk.

An adaptation of the reversing motion first introduced by Carmicheal in twin ferry boats running on the Frith and Tay, is shown in figure 2 of the same plate. This gear, which is exceedingly simple as a reversing gear only, was long held in favor in both Europe and America. It may be of interest to know that it was long used by Baldwin in America. The reversing lever brings one gab of the eccentric rod into action with one arm of a rocking shaft, or else the other gab into action with the other arm of the same rocker, thereby reversing the motion of the engine. The angle made by a line through the centre of the axle and one end of the rocker arm, with a line through the centre of the axle and the centre of the rocking shaft, corresponds to the angular advance usually given to the eccentric. Figure 3 shows this gear arranged for lap and lead, and it will be seen that long rocker arms and short eccentric rods are required, if much lap is given to the valve. One practical disadvantage of this gear is that the disengagement of one gab does not necessarily engage the other, and therefore the valve must sometimes be moved by a special hand gear, indicated in figure 2 by dotted lines, to the right place to engage with the gab.

In plate II figure 4 is shown one of several devices for reversing with two fixed eccentrics for each valve.

Each eccentric rod is hung by its own suspending link, in such a manner that one is thrown into gear and the other out, by one motion of the reversing lever. The two gabs of the eccentric rods are made to gear with the same arm of the rocking shaft, one of course, being idle while the other one is in gear. This gear was made by Roberts and Sharp in 1842

A reversing gear patented by Robert Stephenson in 1841, is shown in figure 5 of the same plate. The end of the valve spindle carries two gabs, into which may be thrown the end of either eccentric rod, thereby reversing the engine. The eccentric rods are held apart by a straight distance link and are suspended by the same hanging link. It is only necessary to replace the straight distance link by a curved bar and let it slide through a swivelling block on the valve stem and we have a true link-motion. Indeed the resemblance between this gear and the Stephenson link as designed by Howe in 1843, and shown in figure 6, is very striking. Howe's gear was evidently designed as a neater mechanical job than the device shown in figure 5, and was used only to reverse, the link being either quite raised or quite lowered. The use of the link to produce variable expansion was an afterthought, rather discovered than invented; the fact that this design was never patented being an indication that its full value was not at first realized, by the inventors.

The link motion for locomotives satisfies the wants of the place it fills so well that one can hardly believe that it will be supplanted, unless the whole practice of railway engineers should change in future years. It has, it is true, certain defects, but they are less objectionable in locomotives than they would be in other engines, and what may be an evil in a slow moving stationary engine may be at the same time, a positive advantage in a locomotive. In starting and at slow speeds with heavy loads, a long cut-off is used, with a small amount of lead, and a compression that is not at all excessive. The short cut-off is used mainly at high speeds, and the great compression allows of a smooth running of the reciprocating parts; absorbing their momentum, and giving it out again like a spring, on the return stroke. The increase of lead is also an advantage, as it aids in filling the cylinder with steam at high speeds, assuming that the boiler pressure is attained in the cylinder at the beginning of the stroke. That a certain loss accompanies the compression and reëxpansion of steam, aside from the power required to run the machinery, is shown by the indicator diagram, figure 3 plate VIII, taken from the Harris-Corliss engine in the mechanical laboratory of the Institute of Technology. To obtain this diagram, the valves of the engine were set to give a high compression, and then the cut-off claw was tripped so that no new steam was taken when

the diagram was drawn by the indi-
cator. The pencil passed on the upper
curve when the steam was being
compressed, and on the lower one during
reëxpansion. Any loss in this way
is more than compensated by filling
the prejudicial space of the cylinder
and steam passage, with exhaust
steam which has been compressed even
to the boiler pressure. By properly adjust-
ing the link so as to counteract one
irregularity by another, the motion may
be made all that is desired, and the
cut-off may be equalized as perfectly as
is of any practical advantage. The release
of the steam is also early, at short
cut-off, and this, which would be a source
of loss in a stationary engine, is a
positive advantage, since there is more
difficulty in properly exhausting the

steam than in filling the cylinder with steam at boiler pressure, at the speeds at which so short a cut-off is used. There are three distinct types of link motions proper; first the shifting or Stephenson's link, second the stationary or Gooch's link, and third the Allen or straight link. In the first form, as we have seen, the link is raised or lowered so as to bring different points of the link arc in contact with the link block. In the second the link is curved the opposite way and is held at a constant height, while the link block is carried by a radius-arm so as to come in contact with different points of the link arc. In the third form the link is straight and both link block and link are moved simultaneously in opposite directions, that is one up and the other down, or vice-versa, in such a manner that the same object is accomplished as in the first two forms. The form almost universally used in America is the Stephenson link, suspended at the middle and with the eccentric rods set back from the link arc. This form is represented by plate III. S is the axle, usually of the forward driver wheel, K and J are the forward and backward eccentrics; D and C are the eccentric rods, f and e the centre pins of the link, a b is the link, B the link block; h is the rocker shaft. and h l the upper arm of the rocker, the lower one being in part hidden by the link; I d is the saddle plate, c the saddle pin, c e the hanger, e g and g G the reversing shaft arms; g is the reversing shaft, and $G G_1$ is the reversing rod moved by the reversing lever in the driver's cab. This is the form of link worked out by different methods to show their application, in the remainder of this thesis, and this description will be considered as a definition of the parts of the link gear.

Zeuner's Circular Diagram.

The motion of the link itself and consequently, the motion communicated to the valve is so extraordinarialy complicated that an exact solution of the problem is impracticable. An approximate solution however is given by Professor Gustav Zeuner in his Treatise on Valve-gears, Chapter I of the Second Division, for the Stephenson link, following which he discusses other forms of link motions by similar methods. It would be hardly in place to introduce his solution in full here, for I could not hope to give it in a form even approaching the excellence of his without taking it bodily from his work. Instead of attempting that, there will be given a full solution of a problem, both numerically and by a diagram. with no more explanation

than is required to make the applica-
tion of the method intelligible.
The ordinary D slide valve, which is exactly like the valve used with the link, is shown in figure 1 plate IV, in its central position. a a are the ports to and from the cylinder, e is the exhaust port leading to the open air or the condenser, and i is the eduction or inside lap and l is the induction lap or simply lap. The valve is moved by an eccentric so as to put each end of the cylinder in communication alternately with the steam chest and the exhaust pipe.
The whole theory of the plain slide valve is so clearly set forth in all our text books that it is needless to go into it farther than to show the method of the circular diagram, which is extended to the link motion in the solution given.

In order to avoid confusion the terms used are defined as follows:—
Admission, the opening of the valve to admit steam from the valve chest.
Cut-off, the closing of the valve and cutting-off the supply of steam.
Release, the opening of the communication between the cylinder and exhaust.
Compression, the closing of the exhaust port which confines the remainder of the steam in the cylinder.
All four are preferably given in percent of the stroke from the beginning though often given in inches of the stroke.
Lead, the amount the port is open to admit steam when the crank is on a dead point, usually given in fractions of an inch.
Inside lead, the amount the exhaust side of the port is open, when the crank

is on a dead point; given in fractions of an inch.
Angular Advance, the angle which the eccentric arm makes with a perpendicular to its own line of dead points when the crank is on a dead point; given in degrees.
In all properly designed simple slide valves, the ratio of the eccentric arm to the eccentric rod is such that the motion may be considered harmonic without appreciable error. That is to say, the displacement of the valve from middle position is equal to the eccentric arm multiplied by the cosine of the angle which it makes with its line of dead points. Thus in figure 2 plate IV, the displacement is $OS = OC \cos \gamma$. The thin lines OR_0 and OC_0 represent the crank and eccentric arm when the piston is at the end of its stroke and the crank is

on a dead point, and the heavy lines OR and OC represent the same when the crank and eccentric arm have moved through the angle ω. The lines of dead points of eccentric and crank coincide in the figure at R.OY and δ is the angular advance. The eccentric is supposed to act directly on the valve, for if it acted though a rocker, OC would be diametrically opposed to its present position.

It will readily be seen that when the crank has moved through any angle ω, the displacement of the valve will be equal to $\xi = r \cos \gamma = r \sin(\delta + \omega)$, where r is the eccentric arm or eccentricity. By use of the trignometrical formula for the sine of the sum of two angles we get equation 1.

1. $\xi = r \sin \delta \cos \omega + r \cos \delta \sin \omega$

This equation of course, entirely neglects the slight error introduced by the angularity of the eccentric rod. In equation 2 is given a general form which connects the motion produced by a link with the motion of a simple slide valve.

2. $\xi = A \cos \omega + B \sin \omega$

This equation is proved by Zeuner on page 17 of his work, to be the polar equation of a circle, the rectangular coördinates of whose centre are

$a = \frac{A}{2}$ and $b = \frac{B}{2}$

In the case of the simple valve motion $a = \frac{v \sin \delta}{2}$ and $b = \frac{v \cos \delta}{2}$. Now in plate IV figure 3 we take $\underline{XX}$, to be the line of centres of both the crank and the eccentric. O is the origin of coördinates and OY is perpendicular to $\underline{XX}$.

Lay-off $OB = a$ and $BC = b$ and C will be the centre of the circle referred to. With C as a centre draw a circle passing through O. To get the displacement of the valve for the angular movement ω of the crank in right handed rotation as indicated in the preceding figure, draw OR, making the angle $XOR_{,} = \omega$ as shown. Then OP the portion of this line OR, included by the circle is the displacement of the valve from its central position.

To prove this we have

$a^2 + b^2 = (\frac{1}{2} r \sin \delta)^2 + (\frac{1}{2} r \cos \delta)^2 = \overline{OC}^2$

$\overline{OC}^2 = (\frac{1}{2} r)^2 (\sin^2 \delta + \cos^2 \delta) = (\frac{1}{2} r)^2$

$OC = \frac{1}{2} r$

And since OP_0 is the diameter of the circle of which OC is the radius and C the centre, then

$OP_0 = 2\,OC = r$

Now $OP = OP_0 \cos P_0OP$

for if we drop a perpendicular from P_0 upon OP, the angle $P_0 P O$ is a right angle since it is inscribed in a semi circle. But $OP_0 = r$ and $P_0 O P = 90° - (\delta + \omega)$ hence $OP = r \sin(\delta + \omega)$

$OP = r \sin\delta \cos\omega + r \cos\delta \sin\omega$

which is the equation for the displacement of the valve given on page 18.

The diagram gives the motion of the valve for half a revolution only, but since the motion of the valve is harmonic, the motion for the other half revolution is the exact counterpart of the motion in the first half, and is given by a similar circle whose centre is given by the coördinates, $OB_1 = -a$ and $OC_1 = -b$, and whose centre is at C_1. If now from O we draw one circle $V_1 V_5$ with a radius equal to the inside lap, and another $W_1 W_6$ with a radius equal to the outside lap, and still a third $R_1 R_6$ with a radius equal to the length of the crank on a reduced scale, we are ready to locate the different interesting points of the valve motion.

The admission begins when the valve is displaced from its central position by an amount equal to the lap. This is found to occur before the crank has reached the dead point; at the intersection W_{12} of the valve circle and lap circle, for at this point the valve circle gives the displacement equal to the lap. and the valve is moving toward the right to open the port. OR_{12} is drawn through this point from O. The angle $R\ OR_{12}$ gives the angle at which the crank stands before the dead point at admission, on the forward stroke, and if R and R_{12} be projected at I_1 and H_{12}

then $I.H_{,2}$ gives the distance that the piston lacks of being at the end of the stroke when admission commences, if the motion of the piston be assumed to be harmonic also. Cut-off occurs when the valve is returning to shut the port and the displacement is again equal to the lap. This point is at W_2 at the second intersection of the valve and lap circles. OR_2 drawn through this point gives $R_2 OR$ for the crank angle and $I.H_2$ for the distance the piston has moved forward in its stroke. It will now be seen that this diagram can be discussed more readily if we admit that this line OR, which we have drawn in each case and distinguished by the subscript figure, represents the crank, and that the projection of R upon $I.I_1$ or $I_0 I_2$

represents the piston position, measuring the angles from OX and the piston positions from I_1 or I_2.
It may be a little confusing to do so, for it seems to make the crank and piston move the wrong way, but a little care will prevent serious difficulty, and the absolute measurements of angles and distances independent of direction, are correct. Rankine on page 308 of Machinery and Millwork gives substantially the same diagram in a manner that avoids the difficulty. He lays off the angular advance backward from YO and then starting with his crank at OX, , he obtains, in the same manner, the valve displacement, and at the same time his radius vector which gives the valve displacement, represents the crank in its true position.

The motion of the valve itself which is of more importance, is shown better by Zeuner's diagram both in direction and amount. Even if we make a correction for the connecting rod, as will be done in case of the cut-off of the link motion, it is only necessary to remember which way the error is to be laid off to get the correct results. Moreover considerations which enter into the solution of the link motion render this diagram preferable.

We have seen that the admission and cut-off come at the intersections of the lap circle with the valve circle, and a similar consideration will show that release and compression come at the intersections of the inside lap circle V_{11} V_5 with the valve circle.

Beginning with the crank at R_0 of figure 2 plate IV and with right handed rotation as indicated, we have a summary of the important valve and piston positions in the following account of a half-revolution. Admission on the left hand side of the piston began when the crank was at the angle $X\,OR_{,2}$ before the dead-point, and when the piston lacked the distance $H_{,2}I$ of being at the end of the stroke. When the crank was on the dead point and the piston position was represented by I, the valve had already opened the port by an amount We equal to the lead. When the crank position was given by R_0 the valve had reached its farthest displacement to the right and was on the point of returning, the eccentric then being on one of its dead points. R_2 gives the crank position and H_2 the piston position at cut-off. R_3 gives the crank position and H_3 the piston position when the exhaust port was closed on the right hand side of the piston, and from this point the steam was compressed before the piston till the valve opens for admission on the return stroke. The valve is at its central position when the crank position is represented by R_4, OR_4 being at right angles to R_0R_7. The release of steam on the left hand of the piston occurs when the crank position is given by R_5 and the piston position by H_5. Admission on the right hand side of the piston occurs at the crank position R_6 and the piston position H_6, and when we get to OX, the crank has arrived at its other dead point. The valve displacement is given in each case by the portion of the radius vector included by the valve circle; the induction port opening is given by the distance measured along the radius vector from the lap circle to the valve circle beyond, shown at $V_1 P$ on OR_1; and the eduction port opening is shown by the distance, measured along the same line, from the inside lap circle, as $V_1 P$ on OR_1. The crank and piston positions are shown in exactly the same way for the return stroke by the lower valve circle, the piston positions being projected upon $I_0 I_2$. There is no real need of drawing more than one valve circle, since the second is a duplicate of the first, and in the diagram for the link motion, only one valve circle for each grade will be drawn. By imposing certain conditions which might be fulfilled practically, for the forward gear of the link at least, Professor Zeuner gives in his Valve-Gears, an algebraic solution with very simple results, and an accompanying diagram of the same form as the one for the simple slide valve.

Let r be the eccentricity; let l be the length of the eccentric rod measured from the centre of the eccentric to the link arc; let c be the length of the link along the link arc, which is approximately equal to the distance between the link centres, shown at f and e on plate III; and let u be the distance that the link block is from the dead point of the link. Then the displacement of the valve for any crank angle ω is,

3. $\xi = r(\sin\delta + \frac{c^2 - u^2}{cl}\cos\delta)\cos\omega + \frac{ur}{c}\cos\delta\sin\omega$

It is assumed that the connection of the eccentric rods to the link is such that the effect is appreciably the same as though the link centre-pins were on the link arc, where they are

often placed in European practice. The link block is assumed to be directly attached to the valve spindle, and both are assumed to move on a straight line passing through the driver axle, the line taken to be horizontal. The link in one case, which corresponds to the problem to which the method will be applied in the solution to be given, is supposed to be suspended by the middle point of the chord subtending the link arc, and the bodily motion of the link is assumed to be such that there is no appreciable slip of the block in the link.
The valve is set with equal lead, and it is proved that the lead will stay equal at all grades of the link, even if the inequalities of the motion be taken into account. It is shown that the link

arc ought to be curved to a radius equal to the length of the eccentric-rods, and there is good reason to believe that any wide departure from this rule is disadvantageous. It must be strictly followed to make this solution applicable. Zeuner also shows that the suspension of the link which best accomplishes the requirement that there shall be no rising and falling of the link, will be given if the upper end of the hanger be moved in a parabola, which may be substituted by a circular arc of radius equal to the length of the eccentric rods, and whose centre is at a height above the centre of the axle equal to the length of the link hanger, and at a distance behind the centre of the axle equal to the height of the link arc above its subtending chord. This suspension brings the upper end of the hanger vertically over the mid position of the saddle pin when the link is in midgear, and keeps it so for all gears, forward or backward. The method of suspension is a little different if the link hanger be attached to one of the link centres.

Of course it would be impracticable to use so long reversingshaft arms, but on plate V is given a method of suspension, taken from Colburn's Locomotive Engineering, being there attributed to D. K. Clark, and is disapproved of since a different mode of suspension gives more desirable results. The dimensions used in the diagram of this plate are the same as those used in the problem following, and it is drawn to scale, one eighth size. XX is the centre line of the gear, and O is the centre of the driving axle. The link is first put into mid gear, and the crank is set on its dead point, so that the link is upright. Taking first the heavy lines which represent the link at the forward end of the motion, we have D and D_1 for the centres of the forward and backward eccentrics. From these points as centres strike indefinite arcs on which the link centres must fall. Now prepare a templet to represent the link, with one edge cut to represent the link arc, and with three points to represent the link centres and saddle pin. Apply this templet to the drawing so that the forward centre c will fall on the arc struck from D, and the backward centre c_1 will fall on the arc struck from D_1, while the saddle pin is on the centre line XX. From E and E_1, which are the other positions of the eccentrics when the crank is again on a dead point, strike arcs as before, and apply the templet in the same manner, and the link centres will be found at f and f, while the saddle-pin is at b. If now we lower the link into full forward gear, and apply the templet as before, but bring the upper link-centre on the line XX, instead of the saddle pin, we get g and g, for the link centres and d for the saddle-pin, when the crank is on one dead point; and with the crank on the other dead point these points will fall at h and h, and c respectively. With the length of the link hanger as radius and with a and b as centres, strike arcs intersecting at l, and also from c and d as centres strike arcs intersecting at i. With the length of the reversing shaft arm as a radius and with i and l as centres, strike arcs intersecting at m; then m is the desired location of the reversing shaft. This method is inserted merely to show that the conditions imposed can be followed nearly enough if desired.

Returning to equation 3

$$\xi = v\left(\sin\delta + \frac{c^2 - n^2}{cl}\cos\delta\right)\cos\omega + \frac{nv}{c}\cos\delta\sin\omega$$

we see that if we let $v\left(\sin\delta + \dfrac{c^2-n^2}{cl}\cos\delta\right) = A$ and $\dfrac{nv}{c}\cos\delta = B$ it assumes the form

$$\xi = A\cos\omega + B\sin\omega$$

which is identical with equation 2.

We saw that this equation gave, in the case of the simple slide valve, the polar equation to a circle which we called the valve circle, and used in drawing the diagram to give the valve positions. As has been stated, this is a general equation to a circle whose centre is given by the coördinates $\frac{A}{2} = a$ and $\frac{B}{2} = b$, and it is proved in Zeuner's work, that if the value of u be substituted in these equations, i.e. $a = \frac{1}{2} r (\sin \delta + \frac{c^2 - u^2}{c l} \cos \delta)$ $b = \frac{1}{2} \frac{r u}{c} \cos \delta$, we get for that grade of the link, a circle which may be used in finding the valve displacements in exactly the same way as with the valve circle for a slide valve already discussed.

On page 86 will be found the dimensions of a link motion designed by aid of a model, at the Hinkley Locomotive Works, and the following are such dimensions as are needed for this method rearranged in a convenient form:—

r = eccentricity = 2.75 inches

l = length of eccentric rods = 54 inches.
$c = \frac{1}{2}$ distance between link centres = 6 inches.
lap = 7/8 of an inch   full gear lead = $\frac{1}{16}$ of an inch.
The angular advance is obtained from the property that the valve displacement in full gear, when the crank is on a dead point, is equal to the lap plus the full gear lead. The valve displacement at this point, is equal to the eccentricity multiplied by the sine of the angular advance, if we consider the action of the link in full gear to be equivalent to a simple eccentric. Hence

$$\sin \delta = \frac{7/8 \text{ of an inch} + \frac{1}{16} \text{ of an inch}}{2\ 3/4 \text{ inches}} = \frac{.9375}{2.75} \text{ inches}$$

$$\therefore \delta = \sin^{-1} \frac{.9375}{2.75} = \sin^{-1} .340909 = 19^\circ\ 56'$$

where δ is the angular advance.

u is taken in tenths of c so that there are ten grades besides the dead point. Substituting the value of u in the equations for the coördinates of the valve circles, that is, making $u = .1c$, $.2c$ &c. we obtain a and b as given in columns 1 and 2 of the table of results on page 53. These points thus found for the centres of the valve circles, lie on a parabola, as may be seen from the fact that the equation for a given on page 36 contains the second power of the variable u, while the equation for b contains the first power only.

On plate VI the line XX, is taken to represent the line of dead points of both eccentric and crank. O is the origin of coördinates, and XX, and OY perpendicular to each other, are the axes of coördinates. From O the abscissae of the centres of the valve circles, Oa_0 Oa_1 to Oa_{10}, taken from column 1 of the table of results, are laid off, the subscript figure being used to indicate the grade of the link.

From $a_0\ a_1\ a_2$ ---- a_{10} are laid off the coördinates $a_1 b_1\ a_2 b_2$ ---- $a_{10} b_{10}$ taken from column 2 of the table. The centre of the valve circle for the dead point of the link, falls on $\overline{XX_1}$ since b for this grade is zero. The coördinates given in the table were calculated by the formulae given and were tested by the properties of the parabola to the fourth decimal place. Rankine on page 498 of the Steam Engine and on page 257 of Machinery and Millwork, gives a construction that is equivalent to substituting for the parabola, a circular arc drawn through a_0 and b_{10} with its centre on $\overline{XX_1}$, in which case the centres would be found approximately in each grade, by making the centre fall on the arc, at a perpendicular height from $\overline{XX_1}$ equal to $\frac{u}{c}\ a_{10} b_{10}$ From the centres $a_0\ b_1\ b_2$ ---- b_{10} the valve circles are drawn through O. With a radius equal to the lap, the lap circle is drawn through $M M_1$, from O as the centre. The semicircle $R_1 C_3 P_{10}$ is drawn to represent the crank circle on a reduced scale. The length $XX_1 = I I_1$ is taken $\frac{5.00}{60}$ of an inch, so that if the distance from I_1 to any point of the stroke be read from an engineer's scale divided to sixtieths of an inch, and multiplied by two, the fraction of the stroke is obtained in decimals. The valve circles, and all accompanying circles, lines, and dimensions, are given twice the actual size. The line $I_1 I_1$ which gives the piston positions, is taken below the crank for convenience. This diagram is exactly similar to the diagram for one eccentric, the only difference in its use being that there is a valve circle for each grade of the link, so that we have in effect eleven diagrams like the one already discussed, each one by itself being as simple, although the number of lines makes a seeming confusion.

If we now begin with the crank on a dead point as represented by $O\underline{X}$, we may follow the relative motions of valve and piston through a half revolution and thus learn the whole story told by this diagram.

The displacement of the valve when the crank is on a dead point is given by that portion of the line $O\underline{X}$ included within the valve circle for each grade, being Oe_{10} for the tenth grade and Oe_0 for the dead point. Since the valve has no inside lap, the inside lead is equal to this displacement, which is found by multiplying by 2 the abscissae for the centres of the valve circles, for the several grades, and is recorded in column 4 of the table. That this method is correct is evident from the proportion $Oe_{10} : Os_{10} :: Oa_{10} : Ob_{10}$ The outside lead recorded in column 3 is obtained by subtracting the lap from the valve displacement at a dead point of the crank, or what is equivalent in this case, from the inside lead given in column four of the table. On the diagram it is found by measuring on OX, from M to the valve circle, being Me_{10} in the tenth grade.

The crank angle at the greatest displacement of the valve and therefore at the greatest port opening, is found on the diagram, by drawing the diameters of the valve circles from O, $R_1 R_2 --- R_{10}$ being the crank positions, and $S_1 S_2 --- S_{10}$ being the extremities of the diameters. The angles measured from OX are calculated from the equation $\tan^{-1} XOS = \frac{b}{a}$, and are recorded in column 7 of the table, a and b being taken from the columns of coördinates, 1 and 2. The greatest port opening for the exhaust is equal to the greatest displacement, as there is no inside lap, and is found by measuring on the diameters of the valve circles from O to S_0 S_1 S_2 --- S_{10}. The values given in column 6 are calculated by the equation $OS = 2 \frac{Oa}{\sin XOS}$ and are checked by an independent calculation from the equation $OS = 2\sqrt{Oa^2 + Ob^2}$. When no subscript letter is given in these equations and those following, it is to be understood that they are applicable to all grades. The greatest port openings on the induction are obtained on the diagram by measuring from the lap circle $M M_1$, on the diameters of the

valve circles to $S_0 S_1 S_2 ---- S_{10}$, and is calculated by subtracting the lap from the greatest travel given by column 6, and is recorded in column 5.
The crank angles at cut-off are found on the diagram by drawing the lines $OC_0\ OC_1 ---- OC_{10}$ through the intersections of the valve circles and the lap circle, $C_0\ C_1\ C_2 --- C_{10}$ being the crank positions at cut-off. The angle XOC for each grade, is equal to the sum of the angles XOS and SOC. XOS is known since it is the crank angle at the greatest valve displacement, and SOC is calculated by the formula $\sin^{-1} SOC = \frac{\text{lap}}{\text{greatest displacement}} = \frac{OM}{OS'}$. The sum of SOC and XOS is the crank angle at cut-off recorded in column 8.
When the crank angle is known and also the ratio of the length of the crank to

the length of the connecting rod, the piston position may be calculated exactly by means of the formula $s = r + l - r\cos\delta - \sqrt{l^2 - r^2\sin^2\delta}$, in which s is the distance that the piston has moved forward from the end of the stroke, r is the length of the crank, l is the length of the connecting rod, and δ is the angle that the crank has moved from the dead point. Or if the piston position is given in decimals of the stroke the form which is more convenient for use is

$$\frac{s}{2r} = \frac{1-\cos\delta}{2} + \frac{\frac{l}{r} - \sqrt{\frac{l^2}{r^2} - \sin^2\delta}}{2}.$$

The truth of these equations may be seen from figure 2 plate V, the parts being lettered as in the formulae. S is the length $r+l$, minus the projections of r and l upon the line of centres. The projection of r upon this line is $r\cos\delta$, and that of l is the third side of a right triangle of which the known sides are l and $r \sin \delta$. The projection of l is therefore $\sqrt{l^2 - r^2 \sin^2 \delta}$. The second equation, which is the one used, is made up of two parts $\frac{1-\cos\delta}{2}$ and $\frac{\frac{l}{r} - \sqrt{\frac{l^2}{r^2} - \sin^2\delta}}{2}$. The first part gives the piston position if the motion is assumed to be harmonic, and the second part is the correction for the connecting rod, to be added when the piston is moving toward the crank axis and subtracted when it is moving from the axis; these motions are usually designated the forward and backward strokes.

The piston positions at cut-off are given in column 9 for harmonic motion, and on the diagram the same ~~are~~ measured on the line $I_1 I$, from I_1 to the projections $C_0\ C_1\ C_2 - - - C_{10}$ of the corresponding crank positions. The correction for connecting-rod seven and one half times the length of the crank, as calculated by the second part of the formula for piston positions, is given in column 10. The difference between the piston positions at cut-off on the forward and return stroke is shown in column 11, and is found by taking twice the correction of column 10. The valve has no inside lap, and therefore the exhaust closure or compression occurs when the valve is at a mid-position, at 90° from the crank position at greatest travel. The crank positions at the greatest valve displacements are shown on the diagram at $R_1 R_2 --- R_{10}$ and the corresponding crank angles are given in column 7. If 90° be added to the angles given in column 7, column 12 is obtained, which gives the crank angles at compression. The crank positions $P_1 P_2 P_3 \text{---} P_{10}$ on the diagram are obtained by laying them off 90° from $R_1 R_2 \text{----} R_{10}$, which can be most readily done by aid of a chord of 90° taken from the crank circle. The projections $P_0 P_1 P_2 \text{---} P_{10}$ on $I_1 I_1$ show the piston positions under the supposition that the motion is harmonic, and column 13 gives the same in decimals of the stroke. The correction for connecting rod is omitted, since the extent of the error introduced thereby is plainly shown in the table, for the cut-off, and it would be needless labor to repeat the work for compression. From the diagram we see that the effect of the link is such as to change the travel of the valve and at the same time to change the angular advance, so that the lead is preserved as nearly constant as is of practical advantage. The increase of the lead from full to mid gear is shown both by the diagram and the table. It increases rapidly from the tenth grade, with the cut-off at .89 of the stroke to the sixth grade with a cut-off at .73 of the stroke. From the sixth grade to mid-gear, the increase is slow and of no great importance. If then the mid-gear lead be properly chosen and care be taken that it does not disappear at full gear, no farther trouble need be taken. The width of the port is 1.25 inches and this opening is obtained only at the seventh grade with a cut-off at .78 of the stroke. In Rankine's Steam Engine page 485 it is stated that the exhaust pipe and passages should be double the area of the steam pipe and passages.

If the port will give free exhaust, then half the width of the port or .625 inches should give free admission, a port opening attained at the fourth grade with a cut-off at .57 of the stroke. At shorter cut-off the port opening is contracted, but even when the cut-off is at .35, given at the second grade, the port opening is nearly sufficient, and the increasing lead lessens the evil. The exhaust is already wide open at the first grade with a cut-off at .23 which is quite as short as is used practically.

The percent of cut-off given in column 9 varies quite rapidly near the middle of the link, for a given vertical movement of the link, and quite slowly near the full gear, and the exhaust closure varies in a like manner. Though a matter of interest in studying the link-motion

this rate of variation has no practical importance.
The greatest correction for the connecting-rod is given for the third grade with a cut-off at .46 of the stroke, being there .033 of the stroke, so that the greatest inequality of cut-off is .067 of the stroke. If this be multiplied by the stroke, 24 inches, the greatest inequality is 1.6 inches, an amount that would probably be objected to by locomotive designers, though it is not at all certain that it would make a very appreciable difference in the working of the engine. It is, however, perfectly practicable to get a much better result by use of a model or by graphical construction. We may expect therefore that this method will not be used or its conditions complied with for the present if ever, among practical men. A more

comprehensive view of the nature of the motion given to the valve by a link can be obtained from a study of this method than by working a model or by making a graphical construction. To thoroughly understand the algebraic discussion may require more mathematical training than draughtsmen usually have, but certainly every mechanical engineer should have such a comprehension as is given by this method, of a mechanism so simple and sure in its action and so complex in its theory.

Table of Results.

	1	2	3	4	5
Grade	Coördinates		Outside Lead	Inside Lead	Greatest Port Opening for Admission
	a	b			
0	.612	0	.350	1.225	.350
1	.611	.129	.347	1.222	.374
2	.606	.259	.338	1.213	.444
3	.599	.388	.324	1.199	.553
4	.589	.517	.304	1.179	.695
5	.576	.646	.278	1.153	.857
6	.561	.776	.246	1.121	1.039
7	.542	.905	.209	1.084	1.234
8	.520	1.034	.166	1.041	1.440
9	.496	1.163	.117	.992	1.655
10	.469	1.293	.063	.938	1.875

Table of Results.

	6	7	8	9
Grade	Greatest Opening of Exhaust Port	Crank Angle at Greatest Port Opening	Crank Angle at Cut-off	Piston Position at Cut-off.
0	1.225	0	44° 24′	.143
1	1.249	11° 57″	57° 28′	.231
2	1.319	23° 5′	72° 11′	.347
3	1.428	32° 54′	85° 7′	.457
4	1.570	41° 20′	97° 27′	.565
5	1.732	48° 16′	107° 56′	.654
6	1.914	54° 8′	116° 56′	.726
7	2.110	59° 5′	124° 34′	.784
8	2.316	63° 17′	131° 5′	.829
9	2.530	66° 52′	136° 38′	.863
10	2.750	70° 4′	141° 31′	.891

Table of Results.

	10	11	12	13
Grade	Correction for Connecting Rod	Inequality of Cut-off	Crank Angle at Compression	Piston Position at Compression.
0	.016	.033	90° 0'	.500
1	.027	.054	101° 57'	.604
2	.030	.061	113° 5'	.696
3	.033	.067	122° 54'	.772
4	.033	.066	131° 20'	.830
5	.030	.061	138° 16'	.878
6	.030	.060	144° 8'	.905
7	.023	.045	149° 5'	.929
8	.019	.038	153° 17'	.947
9	.016	.032	156° 52'	.960
10	.013	.026	160° 4'	.970

Graphical Method.

In the discussion of the link by means of Zeuner's Circular Diagram, which is based on an algebraic solution of the problem, the method was to take the representative type of the link, and, by imposing conditions which could be practically fulfilled, to find equations which gave the positions of the valve for any crank positions, or, vice-versa, the piston positions for the important valve positions. The piston positions corrected for the influence of the connecting rod, at cut-off, gave the greatest inequality of cut-off at the third grade, where it amounted to .067 of the stroke or 1.6 inches. The ratio of the length of the connecting rod to the length of the crank was a favorable one, being seven and one half to one. Had this ratio been less, the difficulty would have been worse.

The difficulty is not that the valve motion is irregular, but that the piston motion is not harmonic, and it is desirable to introduce into the valve motion just such a departure from harmonic motion, as shall make the valve keep pace with the motion of the piston.
This object is accomplished in the graphical method by setting the valve in the desired position for any crank position, or what is equivalent, by drawing the link in such a position as to bring the valve where it is wanted, and then by suspending the link in such a manner as to insure that the link shall be brought to this position when the crank is at the point chosen.
A method for accomplishing this end is given by W. H. Maw on page 175 of Colburn's Locomotive Engineering and

claimed as original, but as it is substantially the same as that given by William S. Auchincloss in his Link and Valve Motions, our attention will be confined to the application of the method there given, to a case involving the same dimensions as those used in the Circular Diagram and on the model to be described farther on.

The folded tracing following the plates at the end of the thesis, is taken from a carefully made drawing of the link motion in hand, on a scale of one half size. This scale was chosen because it gave a more managable drawing, and was accurate enough for illustration of the method; in practice it is advisable to make the drawing full size.

The only appliance needed in addition to the usual instruments of a mechanical draughts man, is a templet to represent the link. This can be made of white holly veneer as suggested by Auchincloss, but the one used was cut from heavy cardboard. This templet must give readily, the link arc, and three points to represent the link centres and the saddle pin. The link arc was cut by tieing a knife in the place of one point of a pair of beam-compasses, and sweeping an arc with the knife point, on the cardboard, the cut being continued through the card board.

A line drawn from the centre to the arc gives the central line of the link, on which the saddle pin is placed. Two lines parallel to the central line and at a perpendicular distance from it equal to half the distance between the link centres, are the lines on which the centres are located. To find the link centres, measure back on the lines on which they fall, the distance the link centres are set back from the arc; 2 3/8 inches in the present case. Auchincloss locates these points by means of ∧ shaped incisions with the point at the apex. On the templet used, two lines were drawn through the point, a small aperture was cut through the cardboard at the point; and a piece of tracing paper was gummed over the hole, to the under side of the templet, and the lines were continued over the tracing paper till they intersected in the same point as before. If the lines are drawn in India ink, they are as convenient for use as the cross hairs of a telescope.

The templet having been prepared,

the line F, A was drawn for the central line of the link motion, and C was taken for the centre of the shaft. With C as a centre and with a radius equal to the throw of the eccentric, 2 3/4 inches, a circle was drawn to give the path of the eccentric centres.
The length of the eccentric rods from eccentric centre to link arc is 54 inches, and the link centres are set back 2 3/8 inches, so that for any position of an eccentric, as F, the link centre corresponding must be found on an arc whose centre is F' and whose radius is 51 5/8 inches. The principle of this method is to find the eccentric positions for the given crank position and draw arcs from these positions as centres, on which arcs the link centres must, fall and then to apply

the link template so as to accomplish the desired result in each case, while the centres fall on the arcs drawn.

The first eccentric positions required are F and B, and f and b, when the crank is on its dead points. To locate them GH is drawn perpendicular to EA and the angles GCF, HCB, GCb and HCf are laid off with a protractor, equal to 19°56', the angular advance. Or the point F may be found by laying off the lap plus the lead on CA towards A, and then erecting a perpendicular to CA through the point thus found, the intersection of this perpendicular with the circle will be at F. Since the valve is moved through a rocker, the crank is found between the eccentrics as at D between F and B.

The first object is to find the mid gear travel and the position of the rocker shaft. From F and B as centres the arcs F and B are struck with a radius of 51 5/8 inches, on which arcs the link centres must fall. In the tracing the eccentric circle is moved up towards the rocker shaft to condense the drawing. The templet is applied so that the upper centre falls on F and the lower centre on B, while the central line of the link falls on CA. The link arc is now upright and intersects AC in d_1. In a like manner the arcs f and b are struck from the centres f and b, and the link is applied with its centres on the arcs f and b and its central line on AC. The intersection of the link arc and AB is now at d_2, and the point A is found by bisecting $d_1 d_2$. In this construction the arcs which lie above the line AC are struck from the forward eccentric as a centre, and the arcs which lie below AC are struck from the backward eccentric. From A is laid off $Al = Al_1$ = the lap = 7/8 of an inch. The small circles and circular arcs drawn through $d\ l$ and similar points from A are used merely to definitely locate these points. The midgear travel is $d_1 d_2$, the midgear lead is $d_1 l$ and $d_2 l_1$ and the rocker shaft is vertically over A, and can be located by striking arcs from l and l_1 with a radius equal to the length of the rocker arm, which arcs intersect in the centre of the rocker shaft. The next step is to equalize the cut off, which is done at half gear by locating

the saddle pin, and at full gear by locating the reversing shaft.
If the crank is at D and moves in right handed rotation, the piston will arrive at half stroke when the crank has moved over 86 1/4°, the ratio of the length of crank to the length of connecting rod being one to seven and one half. F and B will move with the crank through the same angle, and will appear at ½ f and ½ b. With ½ f and ½ b as centres, the arcs ½ f and ½ b are struck, and the link templet is applied with its centres on these arcs and its arc on the point l. I, I is a portion of the link arc and e s is the central line of the link in this position. The valve must now be at cut-off; since the displacement communicated through the rocker

is Al equal to the lap.
If the crank is on the other dead point E, at the beginning of the return stroke, it must move through $93^{\circ}\frac{3}{4}$ before the piston is at half stroke, and f and b, moving with the crank, appear at $f\frac{1}{2}$ and $b\frac{1}{2}$. With $f\frac{1}{2}$ and $b\frac{1}{2}$ as centres the arcs $f\frac{1}{2}$ and $b\frac{1}{2}$ are struck, and the templet being applied with its centres on these arcs and the link arc on l, , we have $I_1^v I_1^v$ for a position of the link arc and $e_1 s_1$ for the central line of the link. The saddle pin must be on the central line of the link in such a position that s shall be as far from the arc I I as s_1 is from $I_1^v I_1^v$. Moreover the method of suspension is such that the saddle pin will move on a line sensibly parallel to CA for a short distance. s and s_1 are

found by trial, to be the only two points which fulfill both conditions, and they therefore give the location of the saddle pin. This point is located for future use, on the templet, in the same manner as the link centres are located. There now remains the equalization of the cut-off at full gear, to be accomplished. By reference to the table of results obtained by the application of the Circular Diagram, it is found that the maximum cut-off is at about .89 of the stroke, and the corresponding crank angles for forward and return strokes are 138°3/4 and 143°1/2. On the forward stroke, the piston will arrive at .89 of the stroke when the crank has moved 138°3/4 from D, and the eccentrics moving with it from F and B go to .89 f and .89 b, from which as centres the arcs .89 f and .89 b are struck. The templet being applied with its centres on these arcs and with the link arc on l, the saddle pin falls at S_4. On the return stroke the piston will again be at .89 of the stroke when the crank has moved 143°½ from E, and the eccentrics will be at f.89 and b.89 at the same angular distance from f and b. The arcs f.89 and b.89 being struck from these points as centres, and the templet being applied with its centres on the arcs, and the link arc on l, the saddle pin falls at S_5. The points S_2 S_3 S_6 S_7 corresponding to S, S S_4 and S_5 are symmetrical to them with respect to the line AC, and are therefore vertically over them, and as far above AC as they are below, and are consequently

easily located.
The problem now is to locate the reversing shaft so that the saddle pin shall be brought to the eight given positions. With S_4 and S_5 as centres and with a radius equal to the length of the link hanger, arcs are struck intersecting in h, and with S_6 and S_7 as centres, using the same radius, arcs are struck intersecting in h_3. From h_3 and h as centres and with a radius equal to the length of the reversing shaft arms, arcs are struck which intersect in T, which is the desired location of the reversing shaft. From T an arc is struck through h and h_3, on which h_1 and h_2 are found by intersecting it by arcs from S and S_3, with a radius equal to the length of the link

hanger. Arcs struck from h h_1 h_2 and h_3 through s_4 s s_2 and s_6, give the path of the saddle pin in full forward half forward, half backward and full backward gears.

To find the lead for the forward and return strokes in full gear, the templet is applied to the arcs F and B with the saddle pin on the arc through s_4 s_5, in which position the link arc intersects the line AC at d giving bd for the lead. In the same way the templet is applied to the arcs f and b with the saddle pin on the arc s_4 s_5, and the return stroke lead is $b_1 d_3$; the slight inequality is of no practical importance.

To find the extreme travel, the forward eccentric is set at D, bringing the backward eccentric to I, as far from B as D is from F. From D and T the arcs D and T are struck, and the link templet being applied to them, with the saddle pin on the arc $S_x S_5$, the link arc is found at $I_1'' I_1''$. The valve is in the other extreme position when the forward eccentric is at E_1, and the backward eccentric is at U, as far from E as B is from F. The arcs E_1 and U are struck from E and U, and the templet being applied to them with the saddle pin on $S_x S_5$, the link arc is found at $I_1^{IV} I_1^{IV}$, giving the extreme travel as the distance between $I_1'' I_1''$ and $I^{IV} I^{IV}$ measured on the line AB.

The slip is most excessive in full gear, and is found by noting the point in which the arc through U, from the centre of the rocker shaft, intersects the link arc when it is most raised, and again, the point of intersection when the link is most depressed. The link is most raised in the position which gives the return stroke lead; that is when the templet is applied to the arcs f and b with the saddle pin on the arc $S_4 S_5$. On the other hand the link is most depressed in the position $I_1'' I''$, and the distance between the daggers on $I'' I_1''$ marked Slip, gives the maximum slip. By this method we succeed in perfectly equalizing the cut-off at half gear and full gear, which practically equalizes the cut-off for all grades; the exhaust closure which is intimately connected with the cut-off is equalized incidentally, from the nature of the case.

The results given below are taken from the original drawing, and are seen to agree very well with those given by the use of a model see pages 86 and 94. If it be objected that the reversing shaft will interfere with the eccentric rods in backward gear, it is replied that the same objection, though in a less degree may be urged against the results of the model. By certain modifications given by Auchincloss, this and other difficulties may be avoided, but there is not space to discuss them here.

Dimensions Given.

Length of reversing shaft arms	18 inches.
" " link hanger	13 " .
" " rocker arm	10 " .
Radius of link	54 " .
Distance between link centres	12 " .
Centres set back from link arc	2 3/8 " .
Width of exhaust	3 " .

Width of steam port	1 1/4 inches.
" " bridge	1 1/8 " .
Lap	7/8 " .
Full gear lead	1/16 " .
Length of connecting rod	90 " .
Dimensions Found.	
Mid gear travel	2 1/2 " .
Centre of rocker from centre of axle	53 29/32 " .
Saddle pin back of link arc	5/16 " .
Mid gear lead	11/32 " .
Reversing shaft above Axle	6 11/16 " .
" " back of rocker	16 5/8 " .
Extreme travel of valve	5 17/32 " .
Slip at full gear	1 3/8 " .
Lead at full gear from drawing	3/32 " .

Link Model.

We have already discussed the algebraic method and the graphical method of working out a link motion, and there now remains the practical method by use of a model with adjustable parts. It is a purely mechanical method, and is properly called experimenting on a model. It has the advantage of giving certain results, for it is only necessary to give the dimensions to a link that are found to work well on the model, and an exact duplicate of the results given by experiment will be obtained in the machine built. It has however a dis-advantage in that the extreme com-plexity of the link's motions make it very difficult to obtain a clear idea of the influence of the different parts by this method, and there is thus by

introduced an element of luck, certain dimensions chancing to give good results; why, nobody knows.
The model chosen was the one at the Hinkley Locomotive Works, because access could be had to it conveniently and it is known to give good results.
Through the kindness of the superintendent, Mr. Child, in allowing me the examination and use of the model, and through the courtesy of the draughtsman, Mr. Leach, I was enabled to gain a very satisfactory insight into the construction and practical working of the model.
On plate VII will be found a skeleton diagram representing this model, which will be referred to by the letters there given.
The model is placed against the side

wall of the room containing it, and is firmly fixed to the wall. O is the center of an axle, representing the driving axle of the engine, which carries the two eccentrics OD and OD, and also the crank OC, which represents the crank of the driving wheel. The crank in the present instance is set between the eccentrics since the valve is moved through a rocker. The reason is easily understood from figures 2 and 3 of plate VII. In the first the valve is moved directly by the eccentric, and in the second there is a rocker intervening, which causes the eccentric to be placed diametrically opposite to its position in the first. OC is the crank and OE the eccentric arm, and δ the angular advance in each case. The crank OC of the model moves over a fixed circle to which it may be clamped at any position, and on which the dead points are so marked that the crank may readily be set on either. Into holes in OC, for the purpose, a handle representing the crank pin may be screwed, and besides serving to turn the crank and move the model, this handle carries the connecting rod. The stroke may be made 12, 16, 18, 20, 22 or 24 inches by setting this handle in the proper place.

The connecting rod is a bar of wood with a pivot hole in one end to receive the handle which represents the crank pin, and the other end has a long slot in which moves a block by which the length is adjusted. When adjusted, the block is clamped to the connecting rod, and a screw which stands for the cross head pin, is put through a pivot hole in the block and screwed into the long slide I, I, which represents the piston rod, and moves in straight slides on which is marked a fixed scale at I, giving the piston position in inches from the beginning of the stroke. This scale reads from 0 to 24 one way on the upper guide, and from 0 to 24 back on the lower guide, and is placed for convenience near the axle, but the action is the same as though it were brought to its usual place beyond the cross head. The forward and backward eccentrics are represented by OD and OD, ; the details of one eccentric are shown in figure 4 of plate VII. The disk of the eccentric is clamped to a radial arm by a bolt moving in a straight slot to give adjustment of the eccentricity, and the

arm in turn is clamped to the axle by a set screw so as to give the proper angular advance.
The eccentric rods are simple bars of pine and new ones can readily be made if those on hand do not serve the case. The end that is connected with the link centre pin has a brass bushing, and the other end is held firmly by bolts in a cleft iron clamp shown in the detail of the eccentric. The iron clamp is adjustable to a limited extent in length, by a screw and nuts as shown.
The link itself is a straight steel spring whose centre line represents the link arc, and is held in a cast iron frame shown in figure 1 plate VIII. At C and C, the spring bears against two supports in the frame, and at the ends D and D, the spring is held by the

heads of two ~~bolts~~ bolts with nuts at A and A, by aid of which the spring is bent to the required arc. At figure 2 of plate VIII is shown the action of the forces which bend the spring. c and c, are the resistances of the supports C and C, of the preceding figure, and d and d, are the forces exerted by the bolts at D and D,. The spring being bent by the action of two equal and opposite couples near its ends, the curve of flexure is a circular arc. At E and E, are the centre pins of the link which may be set back from the link at any required distance in straight slots as shown, and the pieces in which the slots are cut are adjustable by the screws e and e, to give the proper distance between the eccentric centres. The link is suspended by a saddle plate and saddle pin as in actual service but all the parts are capable of movement to give proper dimensions. The saddle plate M N being clamped to the link frame by screws at f and f, moving in straight slots, has a certain amount of movement possible, lengthwise of the link, and the saddle pin is carried by a block that can be clamped at any position in a dovetail slide cut in M N, to give adjustment across the link. We have therefore a very good control over the dimensions of the link arc, centres and saddle pin.

Referring again to the skeleton diagram of the link on plate VII, z y represents the suspending link, pivoted to the saddle pin at z. The end y of this bar is slid through a box in which it is clamped at the right length by a set screw, and the box in turn is

pivoted to a slide which can be clamped to the reversing shaft arm x m, by which means the proper length is given to this arm.
The end m of the reversing shaft arm. slides over an arc a m to which it may be clamped so as to put the link into any grade of forward or backward gear. Fastened to the wall is a long and broad wooden slide like the slide of a lathe, which carries a wooden frame extending in front of the eccentric rods and link, and to this frame the points x and u are fixed. x represents the reversing shaft, u is the point of support of the arc over which the arm x m moves, and a x is a brace. x and u are adjusted in horizontal distances from the axle O, by moving the supporting frame on

the slide at the back of the model, and are adjustable vertically by clamping them in slots cut in the frame, x and w remaining always at a constant distance apart.

The link block is shown at figure 5 plate VII. It is a brass block with a pivot hole for attachment to the rocker arm, and through it passes the steel spring which represents the link. The front bearing is a movable piece M N which may be set up by the screws e e so as to give a smooth motion to the link in the block, without backlash.

The rocker arm r s t is made of wood, centred at s and slotted at r and t to give adjustment to the length of the arms. The centre s is carried by a double slide, the first moving on the broad slide at the back of the model so as to give horizontal adjustment, and the second moving on the first so as to give vertical adjustment.

Wt represents the valve stem which can be adjusted for length at t, and at W there is a vertical slot that permits it to be kept horizontal.

The valve seat, shown in section is drawn a piece of thin board and clamped in position. The valve, also drawn on thin board, is clamped to a slide over the valve seat, and is moved with the slide, by the valve stem. The valve as well as the scale representing the cylinder are carried back toward the crank so as to be readily seen by the person who works the modle by turning the handle at C. The motion in neither case is any way different from what it would be had they been placed in the usual positions.

Almost all the different adjustments are made by aid of scales marked on the pieces themselves, which gives both ease and certainty.

On my second visit to the Hinkley Locomotive works I found Mr. Leach on the point of working out a link for a locomotive in process of building, and as the clearest way of showing the use of the model, I will give a discription of the experiments made together with the results.

The problem was to find the best motion for the "Mogul" engine No. 2, with cylinders 18 by 24 inches.

Dimensions Given.

Distance from centre of axle to centre of rocker
5 4 inches

" " " " rocker " " of
reversing shaft 15 "

Distance of centre of rocker above axle 7 5/8 "

Distance of reversing shaft above axle	15 3/8 inches.
Length of reversing shaft arms	15 "
" " link hanger	13 3/4 "
" " rocker arms	10 "
Radius of link	54 "
Distances between centres of link	12 "
Centres of link set back from link arc	2 3/8 "
Travel of the valve	5 1/2 "
Width of exhaust port	3 "
" " steam port	1 1/4 "
" " bridge	1 1/8 "
Lap	7/8 "
Inside lap	0
Length of main rod	90 "
Lead, full gear	1/16 "

The eccentrics were taken from the model and set to give the required travel of the valve. The link was bent to the required arc and tested. The other dimensions were set according to the table above. The valve seat was drawn on wood with the required dimensions and clamped to its place. The rocker arm was set perpendicular by a plumb line, and then the valve, also drawn on wood, was set so that the centre of the valve came at the centre of the exhaust with the rocker plumb.

The saddle pin was then set on the link arc, and the link being lowered until the required travel of 5½ inches was given and then the eccentric rod was adjusted to give the same displacement, equal to half the travel, both ways from the centre of the exhaust port, and the position of the reversing shaft arm on the arc m a was marked for reference. Then the link was lifted into full gear backward, and the backward eccentric rod was adjusted in the

same way as the forward eccentric rod, and the position of the reversing shaft arm on the arc m a was marked for reference. To assure that one adjustment should not derange the other, both were repeated and any slight alteration required was made until both remained true.
To give the proper angular advance, the forward eccentric was moved round the axle so that the edge of the valve was on the point marked at one sixteenth of an inch from the edge of the port, thus giving the required lead, when the crank was on the forward dead point, by which is meant the one that brings the piston at the beginning of the forward stroke. The crank was then set on the other dead point and the angular advance was increased and the length of the eccentric rod diminished a

corresponding amount, until the required lead was given to that end. Then this adjustment for angular advance was repeated till both ends were right and remained so. The same adjustment was made on the backward eccentric with the link in full gear backward, and then the whole was repeated forward and backward, though the last readjustment was merely to see that one change did not change some other part.

The straight slide I, I, representing the piston rod was then set with its index mark at the zero of the scale, and with the crank on the proper dead point; the connecting rod was slipped onto the handle C at one end, and the screw representing the cross head pin was screwed into the slide at the other.

The object of the experiments was to equalize the cut-off. No attention was paid to the lead farther than to notice that it increased from $\frac{1}{16}$ to $\frac{5}{16}$ of an inch from full gear to a cut-off of six inches. The exhaust closure was entirely neglected, but of course the intimate connection between cut-off and compression, insures that if one is good the other cannot be very bad.

The full gear cut-off was tried by turning the crank until the induction edge of the valve stood on the edge of the port, just closing it; and then the piston position was read from the scale at I. The order observed throughout was to try the forward gear first, and to try the forward stroke of the gear first, as is recorded in the tables of results following.

The intermediate points of cut-off

corresponding to the grades of the link, was obtained by setting the piston at the position where it was desired to cut-off, at 18 inches for instance, and then by raising the link until the valve was brought to cut-off. The crank was then turned until the valve cut-off on the other end and the corresponding piston position was read from the scale at L. When a satisfactory cut-off was obtained for the forward gear, the backward gear was tried in a similar manner.

Table I was obtained with the link saddle-pin on the link arc, and table II with the pin set back 3/8 of an inch, both being introduced to show the great influence of this point on the motion. Table III gives the results with the pin set back 7/16 of an inch, and it will be seen that the greatest difference is at full gear in the forward motion, being there $\frac{5}{16}$ of an inch, but the greatest difference in backward motion was at a cut-off of eight inches where it was 7/8 of an inch. This position of the saddle pin was decided to be as good as could be found, and experiments were made on the method of suspension by changing the position of the reversing shaft and the length of its arms and of the link hanger. Only the final table of results is given since there seemed to be no principle to guide the selection of dimensions, and nothing can be learned from the combinations tried and rejected, except that they were not good. There was an appreciable difference between the forward and backward motions, the former being usually the better.

The dimensions finally adopted introduced the following changes in the table of dimensions on page 86:—

Centre of reversing shaft above axle	7 7/8	inches
Length " " " arms	18	"
" " link hanger	13	"

The cut-off in forward gear was all that could be desired since the difference of one fourth of an inch can have no influence at all on the working of the engine, and the backward gear was not objectionable. The slip of the block in the link was very considerable in some positions of the link, but these makers entirely disregard it, for they say that it may be timed up, and that it is not worth while to let it interfere with a good distribution of steam.

We have, by a practical example introducing the same dimensions so far as they were applicable, illustrated the three ways in which link motion is treated and may briefly sum up the advantages as follows:—

The first gives the most thorough comprehension of the nature of the motion of the link, and the influence of the different parts.

The second gives a direct practical way of designing a link motion which practically equalizes the cut-off at all gears, and at the same time gives a good idea of the nature of the motion, and of the influence of the different parts of the gear, especially of such as produce a deviation from harmonic motion.

The third gives with certainty the

results to be obtained by any arrangement, without tedious calculation or careful drawing.

Cecil H. Peabody.

Apr. 2. 1877

Mass. Inst. Tech..

Piston Positions at Cut-off.

Table I

Forward Motion	
Forward Stroke	Return Stroke
21 1/4 inches.	21 1/4 inches.
14 " .	15 1/4 " .
8 " .	10 1/4 " .

Table II

Forward Motion	
Forward Stroke	Return Stroke
21 3/8 inches.	21 1/8 inches.
14 " .	14 3/16 " .
10 " .	10 1/2 " .
6 " .	6 1/2 " .

Piston Positions at Cut off.

Table III

Forward Motion		Backward Motion	
Forward Stroke	Return Stroke	Forward Stroke	Return Stroke
21 7/16 inches.	21 1/8 inches.	21 5/8 inches.	21 1/2 inches.
18 " .	17 3/4 " .	18 " .	18 1/8 " .
16 " .	15 7/8 " .	16 " .	16 1/4 " .
14 " .	14 " .	14 " .	14 1/2 " .
12 " .	12 1/8 " .	12 " .	12 3/4 " .
10 " .	10 1/8 " .	10 " .	10 5/8 " .
8 " .	8 1/4 " .	8 " .	8 7/8 " .
6 " .	6 1/4 " .	6 " .	6 3/4 " .

Piston Positions at Cut-off
Table IV

Forward Motion		Backward Motion	
Forward Stroke	Return Stroke	Forward Stroke	Return Stroke
21 1/4 inches.	21 inches.	21 5/8 inches.	21 5/8 inches.
18 " .	17 3/4 " .	18 " .	18 1/8 " .
14 " .	13 3/4 " .	14 " .	14 3/8 " .
10 " .	9 3/4 " .	10 " .	10 5/8 " .
		8 " .	8 1/2 " .
6 " .	5 7/8 " .	6 " .	6 3/4 " .

Plate I

Fig. 1.

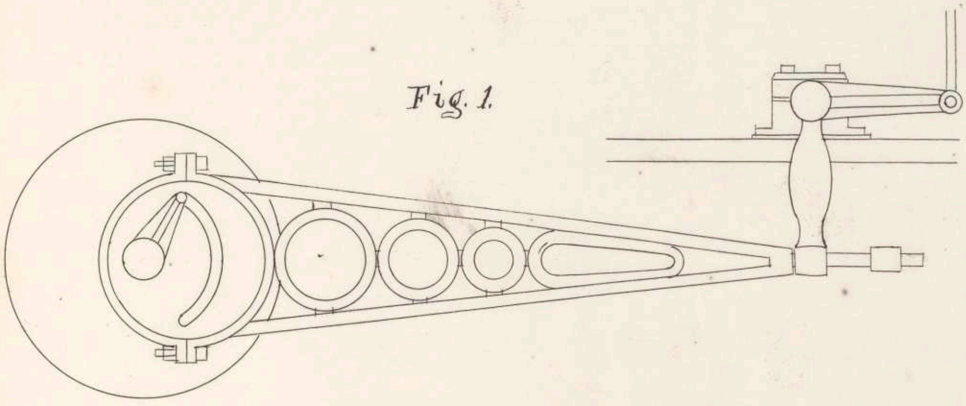


Fig. 2.

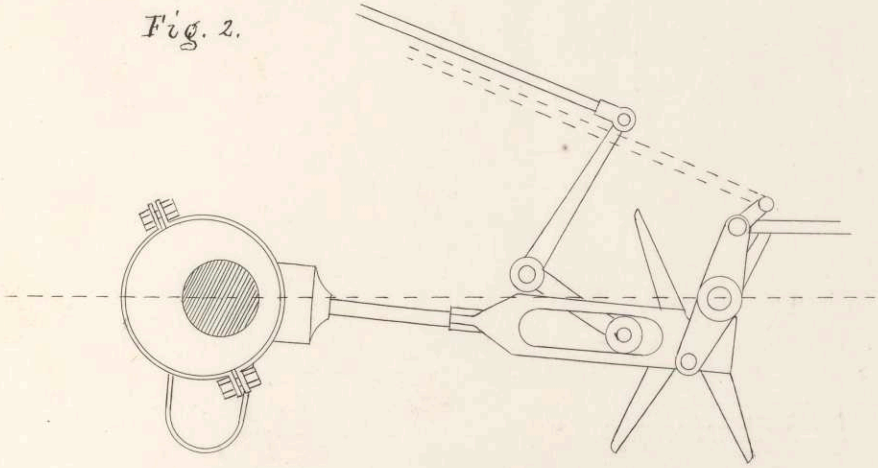


Fig. 3.

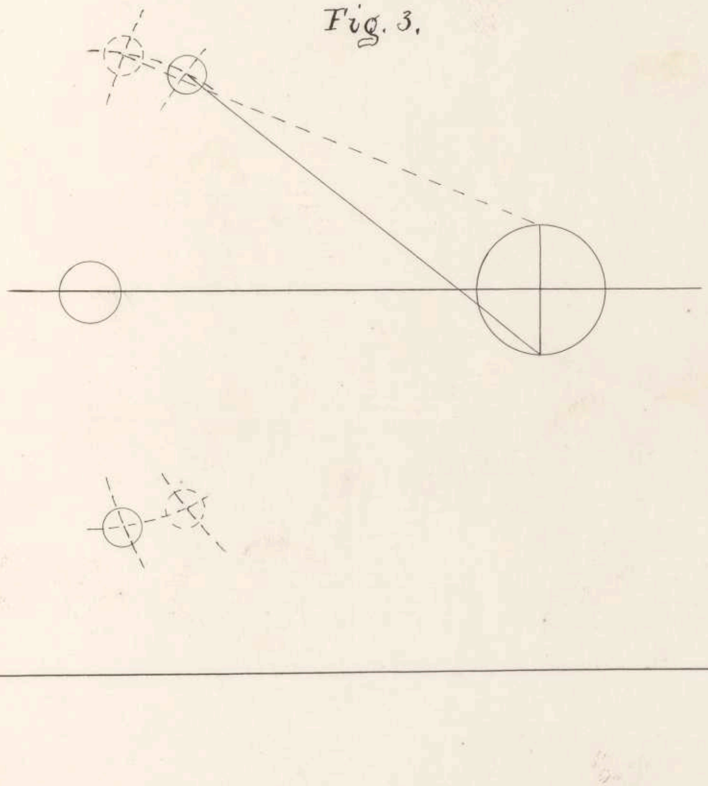


Plate II.

Fig. 4.

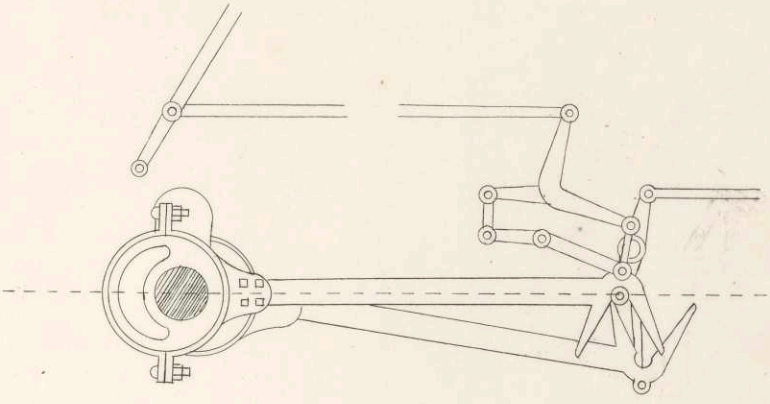


Fig. 5.

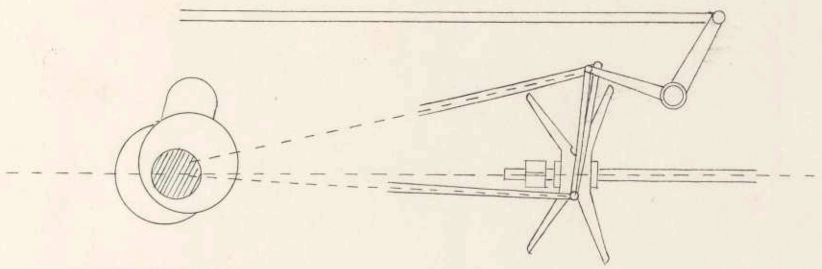


Fig. 6.

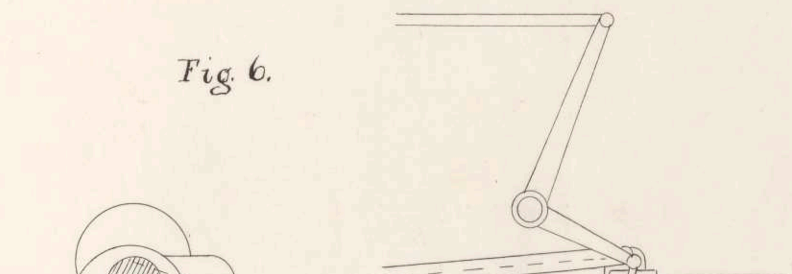


Plate III.

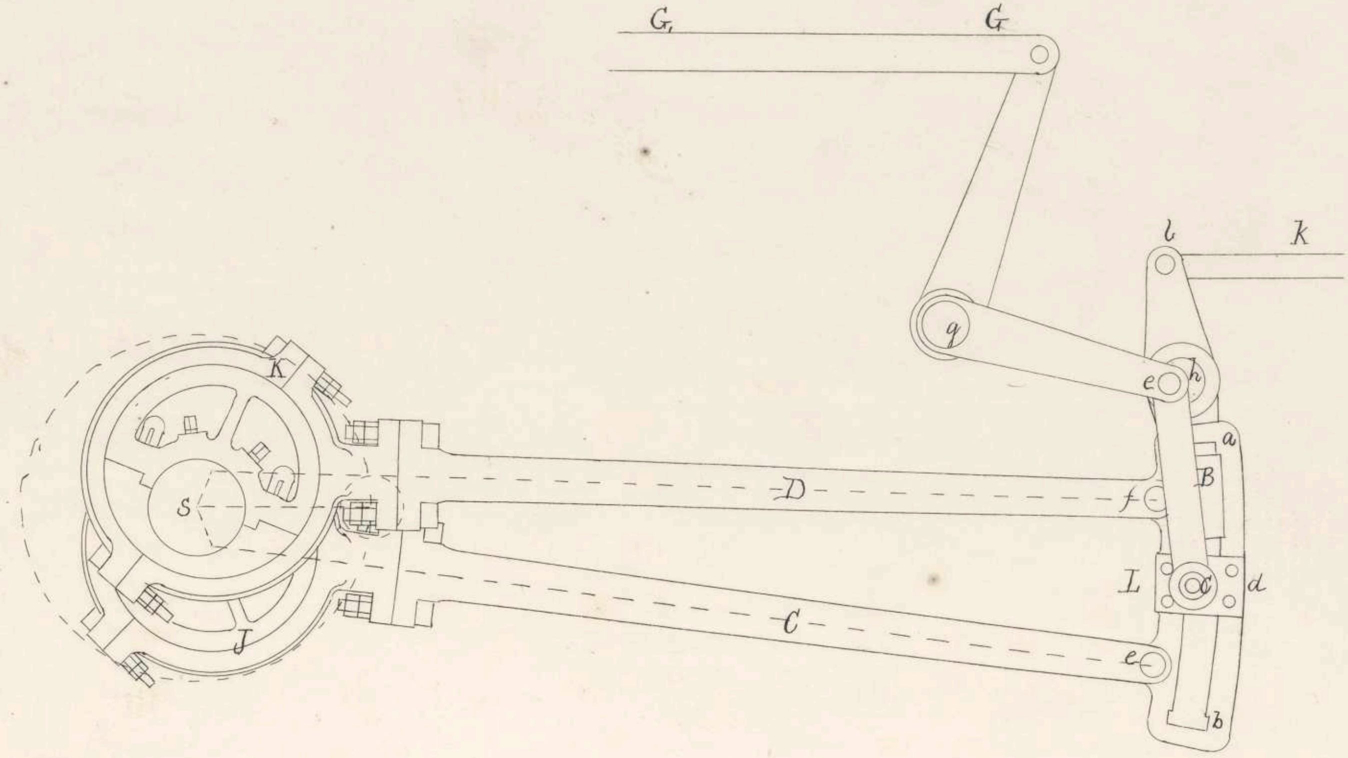


Plate IV.

Fig. 1.

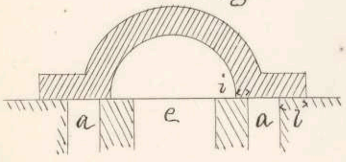


Fig. 2.

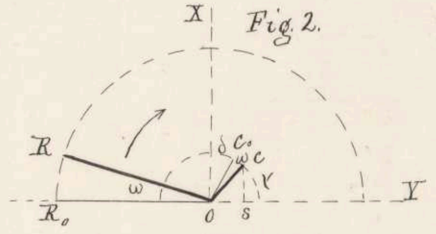


Fig. 3.

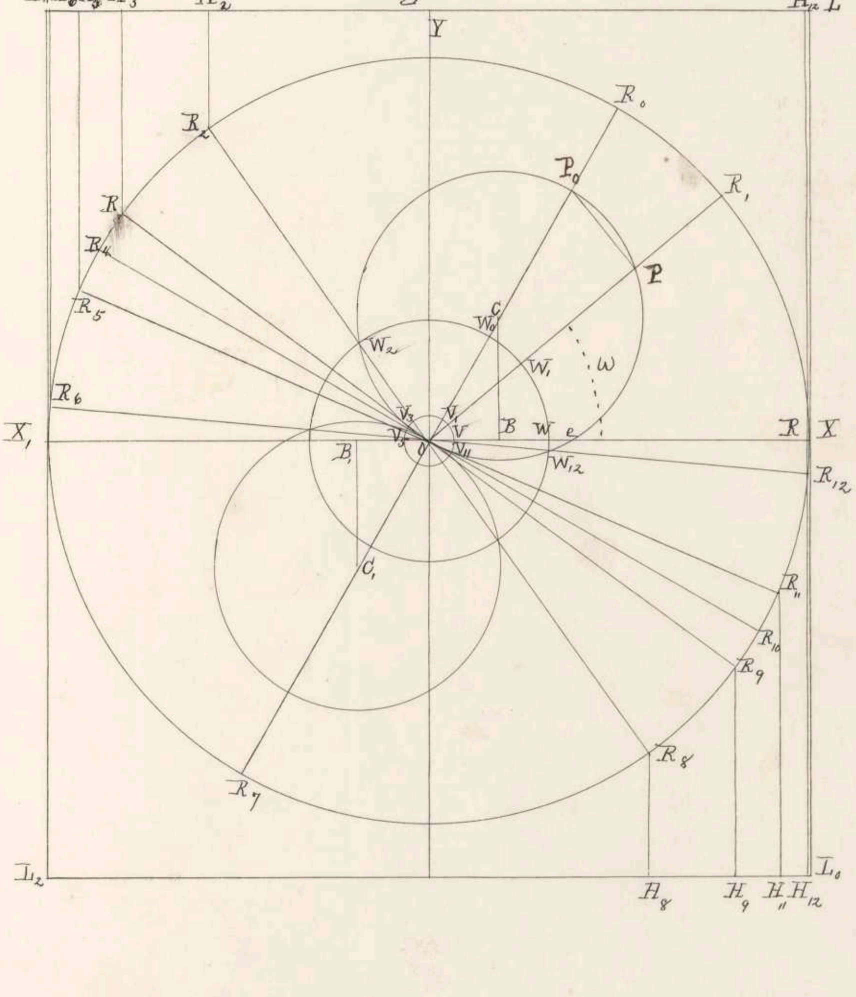


Plate V.

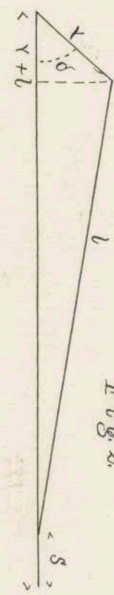
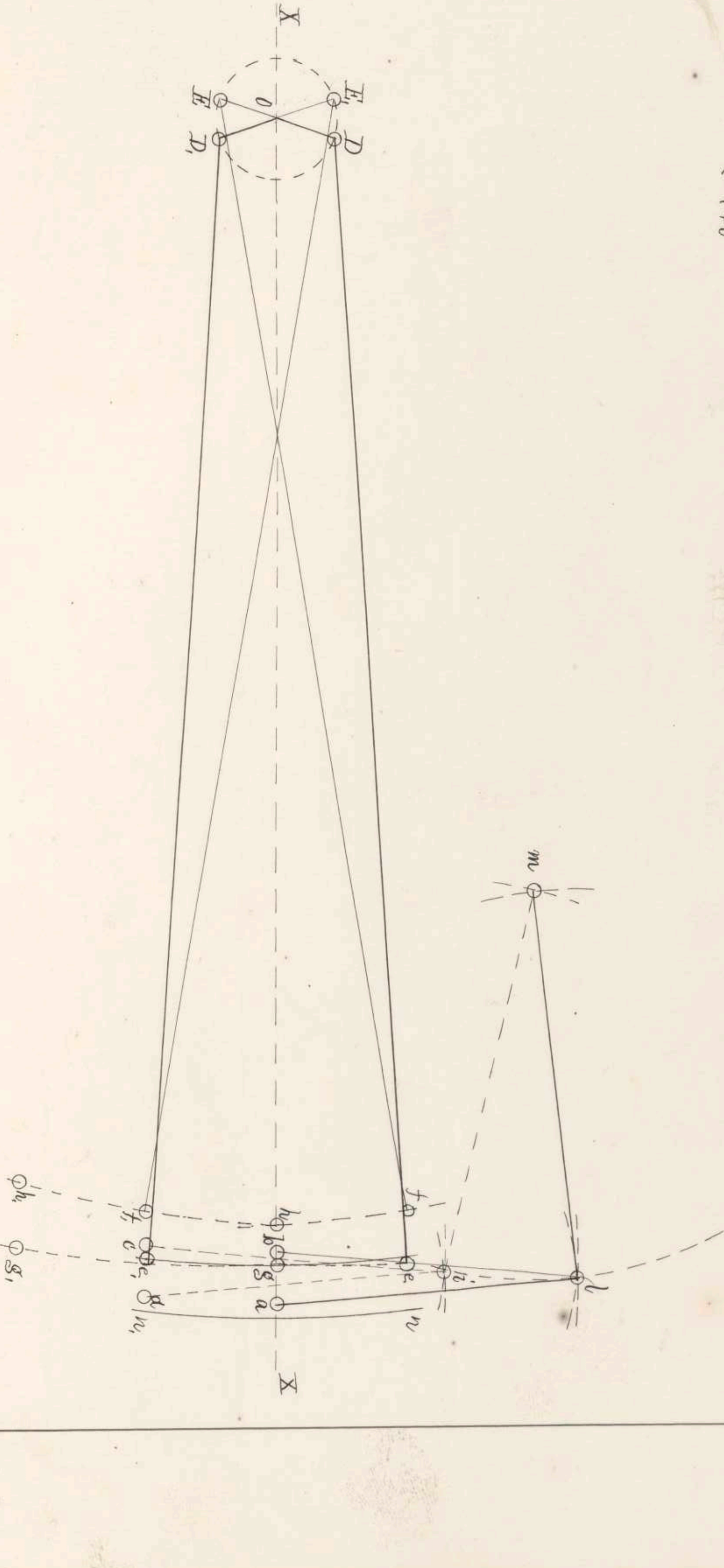


Fig. 2.



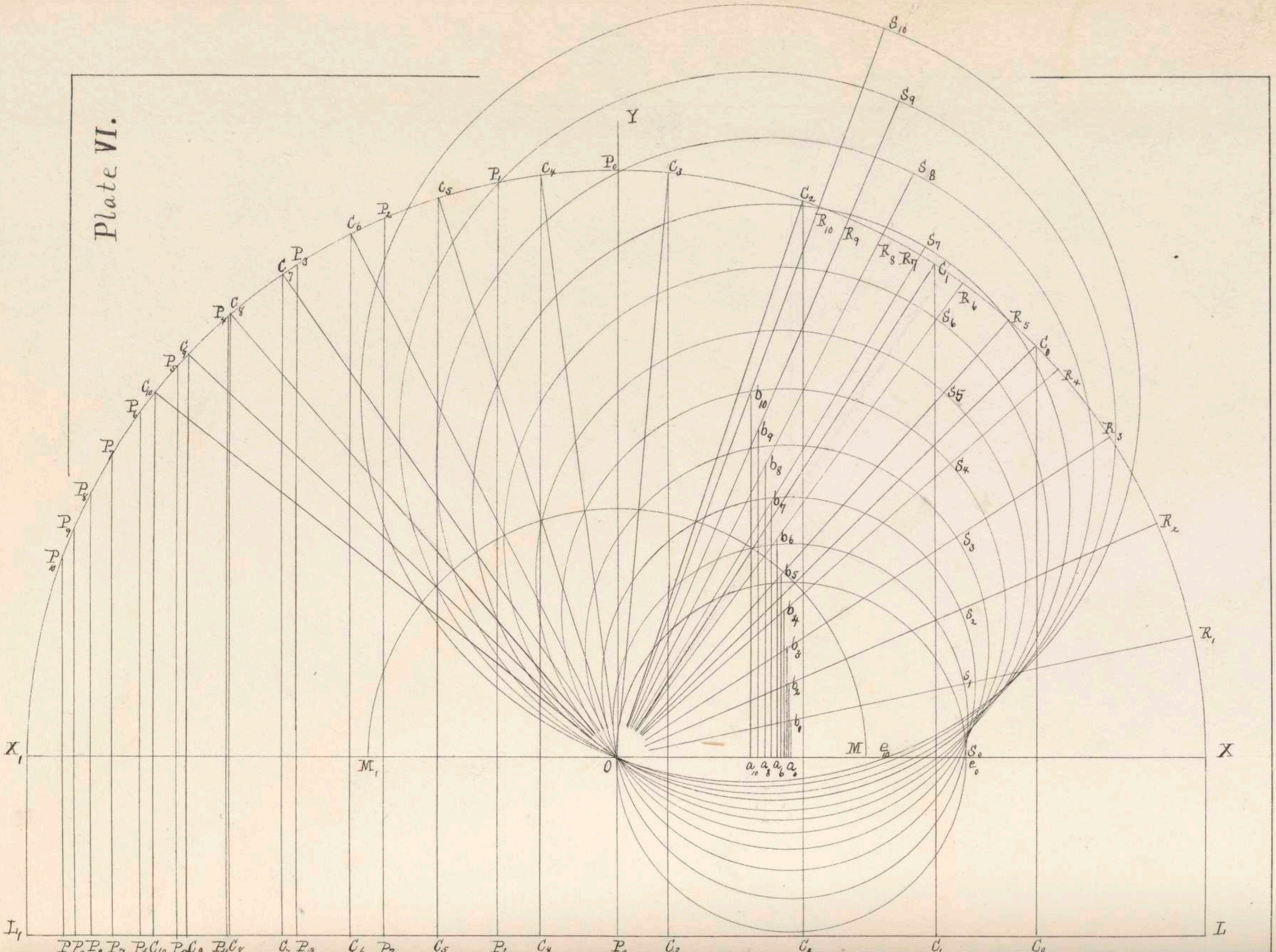


Plate VI.

Plate VII.

Fig. 4.

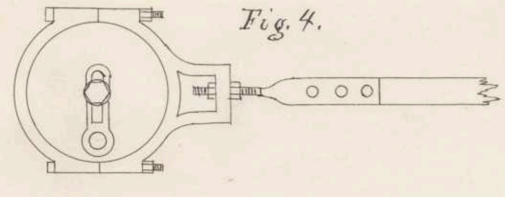


Fig. 1.

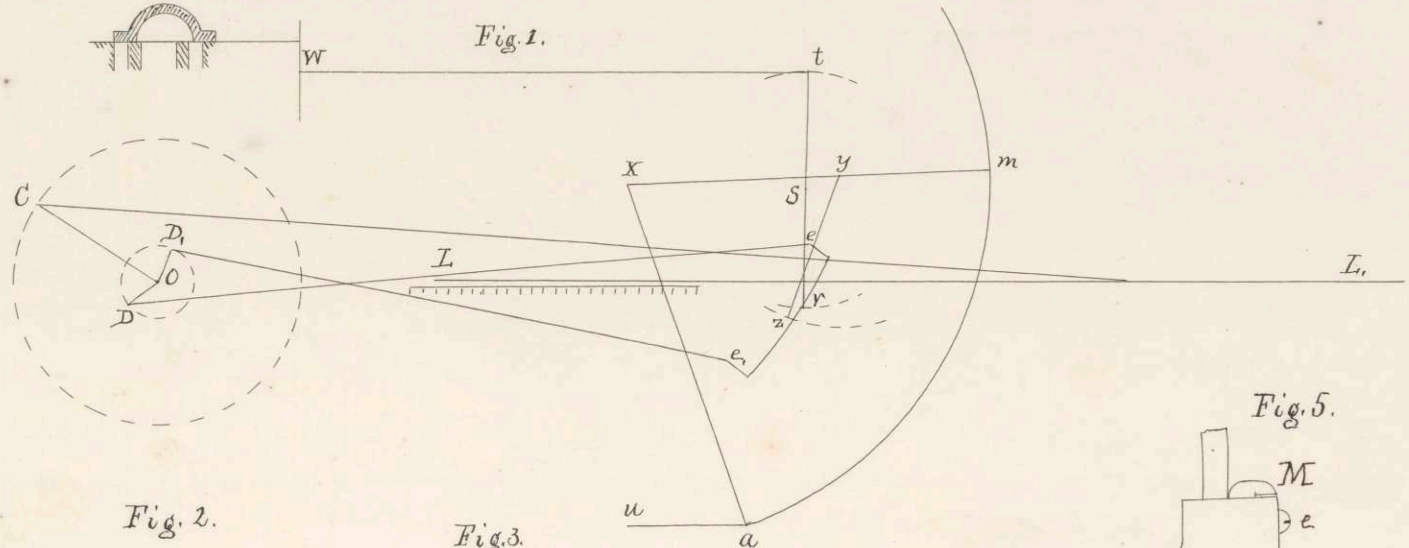


Fig. 5.

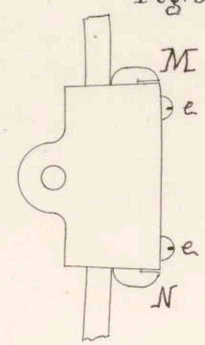


Fig. 2.

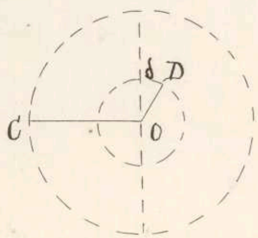


Fig. 3.

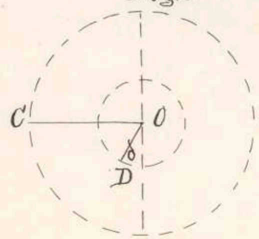
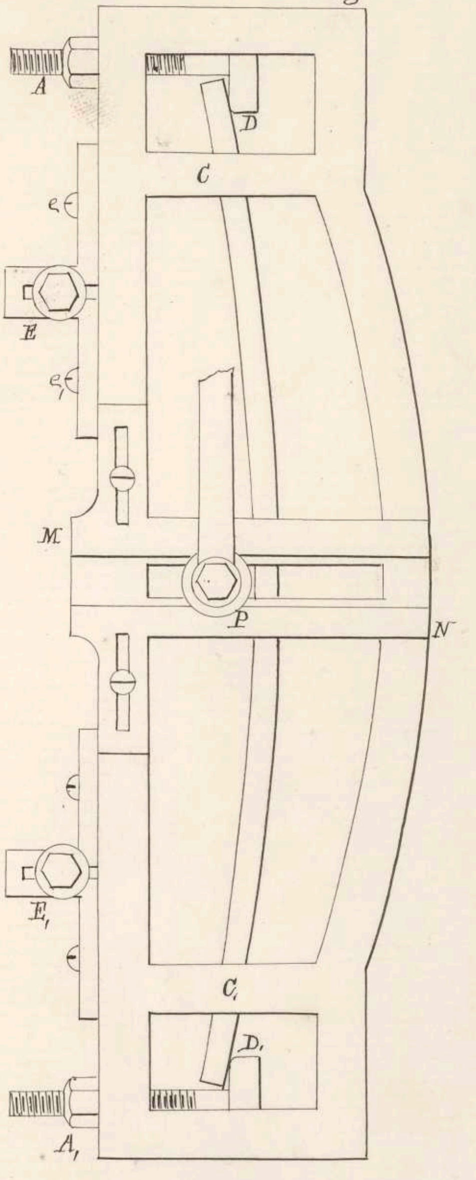


Plate VIII.

Fig. 1.



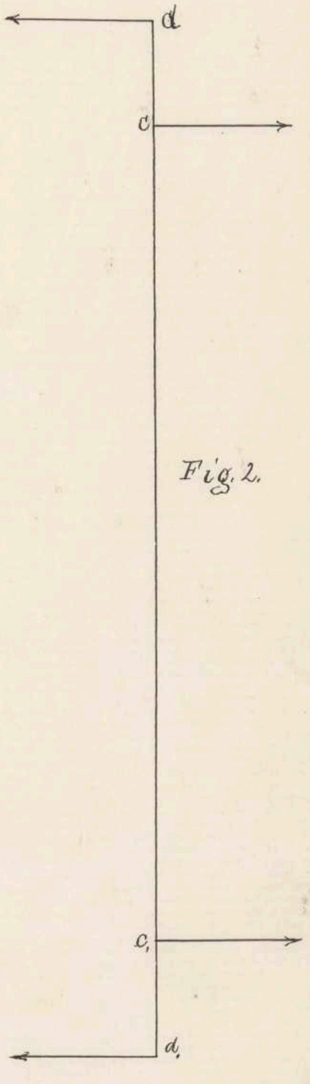


Fig. 2.

Fig. 3.

Link Motion

SCALE ONE HALF SIZE MCH. 18, 1877.

C. H. Peabody.