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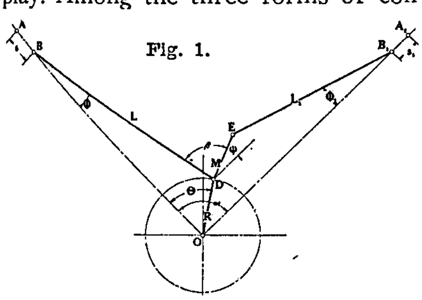
Positioning of Link Rod Wrist Pins

In Articulated Connecting Rods

By Glenn D. Angle

THE method or construction employed in connecting two or more rods to a single crankpin of an engine has always appeared complex to the lay mind. One of the stock questions asked by visitors at aircraft shows is with regard to such method, and it is one which is not simple to explain satisfactorily, unless the exhibitor has a cutaway model of his engine or a connecting rod assembly on display. Among the three forms of construction that may be employed namely, the forked or straddle type, the slipper type, and the articulated type, the latter seems to be the least understood. This is probably because of the irregular movement of the link rods due to the inner ends traveling in a path having no simple geometric form nor generally conforming to any fixed relationship to the circular path of the crankpin.

Fig. 1.



Piston Position With Articulated Connecting Rods. V-type engine.

One engine manufacturer cleverly solved the problem by exhibiting a specially built model that was driven at slow speed by an electric motor. This model represented a transverse section through a radial engine, having a crank with the big end of the master rod hinged to the crankpin and a guide for the small end of each connecting rod along the axis of its respective cylinder. The movements of all rods during their slow motion could be easily followed, and it is unlikely that any observer departed before satisfying his curiosity concerning this type of construction. Judging from the crowd usually gathered before this exhibit, it unquestionably proved to be interesting and instructive. At any rate, the manufacturer was repaid for his efforts by the added attractiveness of his exhibit and by saving the time of his representatives for more important matters at hand.

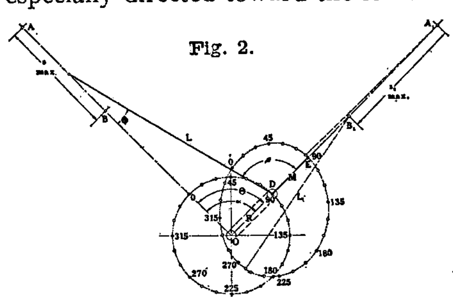
Although the above references to the confusion commonly entertained concerning the movements of the connecting rods of an articulated system are of no engineering consequence, they do bear on the hesitancy on the part of certain engineers to adopt this form of construction, especially when everything pointed to the advisability of them doing so. Numerous instances of this nature have been observed over a period of years, and there is only one logical explanation to be advanced.

Conditions Where Articulated Rods Are Essential

Let it be first explained that the forked and slipper type connecting rods are quite satisfactory forms of construction when selected after due consideration is given to the conditions to be encountered and when the design has been developed without disregarding any of these conditions. However, there are combinations of elements, whether it be the number of rods to be accommodated, the existing loads, or both, where the only safe connecting rod construction that may be employed is the articulated form. Attempts to employ the forked or slipper types under such circumstances indicate an element of fear or lack of knowledge on the part of the designer relative to the kinematics of the articulated connecting rod movements.

It is proposed in this article to briefly discuss certain phases of these motions in a simple manner with a view toward clarifying the work of the designer. This discussion will be especially directed toward the location of the link rod wrist pins as it affects the length of stroke and the compression ratio.

Fig. 2.

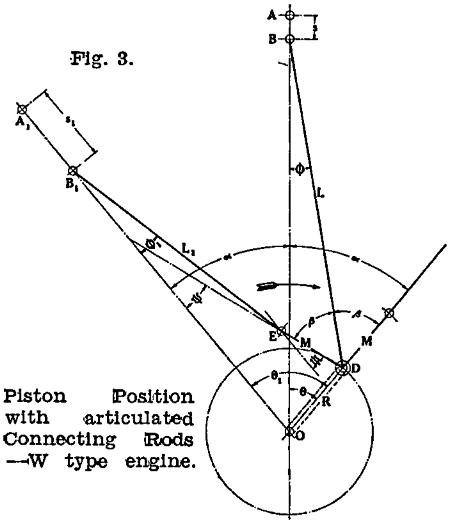


Path of Link Rod Wrist Pin when $\beta=\alpha+\phi$.

Methods For Master Rods With One Link Rod

For our first example, let us consider the most simple form, that is, a master rod with one link rod attached thereto as employed in a Vee type engine. The first question confronting the designer is whether to arrange the link rod to lead or follow in the direction of rotation. Common practice is to place the link rod on the leading side, but in every design the choice should be made following a complete analysis of the forces involved and their effects.

Fig. 3.



Piston Position with articulated Connecting Rods —W type engine.

In a twelve-cylinder Vee type engine constructed by Renault in France, the connecting rod assemblies were alternately arranged so that three link rods would lead and three others follow on the six-throw crankshaft. The obvious purpose was to obtain a more perfectly balanced engine, but the advantages derived from such an arrangement are doubtful. This opinion is based mostly upon the fact that the practice was later discontinued and not as a result of any investigations.

The coefficients vary according to whether the link rod leads or follows and thus affect the harmonics, which cover a wider range in the articulated system. Although the torque curve will differ in each case, this is not of any great consequence as it does not affect the power output of the engine to any measurable degree. The more important considerations are the bending strains imposed upon the shank of the master rod as a result of the link rod forces applied to the extension or boss carrying the wrist pin, the side pressure on both cylinder walls, and the velocity of the link rod piston during the expansion stroke.

The travel of the piston attached to the link connecting rod at any point in the cycle can be computed by the following equation:

$$S_1 = OA_1 - OB_1 = OA_1 - [L_1 \cos\phi_1 + M \cos\psi + R(\cos\alpha - \theta)]$$

when L = length of master connecting rod DB
L_1 = length of link connecting rod EB
M = length of link radius DE
R = crank radius OD
ϕ = angle between master rod and cylinder axis
ϕ_1 = angle between link rod and cylinder axis
ψ = angle between link radius and cylinder axis
θ = crank angle relative to master rod cylinder
α = angle between cylinders
β = angle between master rod axis and link radius
and $\psi = \phi + \beta - \alpha$

Referring to Fig. 1, the angle β is fixed according to the desired location of the wrist pin axis (E) relative to the cylinder axis OA_1, when the point B_1 falls on A_1 or $s_1 = O$; or with respect to the axis OA_1 when the crank radius (R) falls on OA_1 or when $\theta = \alpha$. The choice, of course, depends upon whether the link rod leads or follows; but one must also take into account the total length of stroke (s_1), and the position of the wrist pin axis (E) at the point of firing the link rod cylinder, together with the resultant bending loads on the master connecting rod.

The length of the master connecting rod is controlled by conditions outside of our present study; namely, the minimum length by the maximum angle that the rod makes with the cylinder axis (ϕ), since the side pressure loads on the piston are a function of tan ϕ, or by the length of stroke in conjunction with the required dimensions of the piston, cylinder, and crankcase in order to avoid interferences of any sort. In terms of stroke (s) or 2R, the master connecting rod may have an L/R ratio of 3.2 to 4.4 in the conventional engine.

The length of the link connecting rod (L_1) is determined by the required distance OA_1, the angle β, and the link radius DE or M. The latter distance must be held to the minimum possible to avoid undue bending loads on the master connecting rod, with its resultant additional piston side pressure in that cylinder, since the forces are a function of M and tan $(\phi_1 + \psi)$. It follows that both angles ϕ_1 and ψ should be held to the minimum possible for the same reason, to say nothing of the side pressure loads in the link rod cylinder which are a function of tan ϕ_1. In many cases ϕ_1 controls some of the other factors mentioned, either on account of the side pressure loads in the link rod cylinder, or by rod interferences with the cylinder skirt when ϕ_1 is maximum.

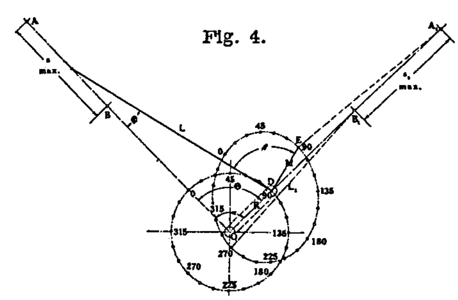
Assuming that we have determined the measurements of OA, R and L to meet the requirements of the design, and that the distance DE has been fixed at the minimum possible with the connecting rod construction to be employed, we must now determine angle β, as well as the length of the link connecting rod L_1, with a view toward obtaining the most favorable conditions.

If the link radius M coincides with the cylinder axis OA_1 when $\theta = \alpha$, then $\psi = O$ and $\beta = \alpha + \phi$, and we have a layout that would apparently be suitable for either direction of rotation. Referring to Fig. 2, it is seen that B_1 does not necessarily fall upon A_1 at this moment under this arrangement, in other words, the piston of the link rod cylinder is not at the extreme outer end of its stroke.

Case of One Master Rod With Two Link Rods

When two link rods are to be attached, one to each side of a master connecting rod as in the case of a W type engine, this arrangement has been adopted for the sake of symmetry, and to avoid the possibility of less favorable conditions in case of incorrect assembly which might easily occur if both sides were not alike. A diagram of this arrangement is shown by Fig. 3. It results in the outer end of the stroke being reached before the crankpin D has crossed the cylinder axis of one row of cylinders and after it has crossed it in the other. The pistons consequently are not at the same relative position with respect to

Fig. 4.

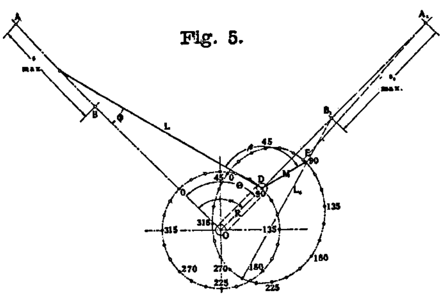


Path of Link Rod Wrist Pin with β is smaller than $\alpha + \phi$.

the cycle, but due possibly to the fact that the velocities are low around this point in the cycle, no undesirable effects have been observed. More important is the fact that the stroke is usually greater in the link rod cylinder, thereby giving these cylinders a greater displacement. However, in some designs the greater velocity of the piston of the link rods during the early part of their stroke somewhat compensates for any increase in the forces that might result from the greater displacement. At any rate, the increased displacement does not appear to affect the smooth operation of an egine as long as the length of the link connecting rod is made slightly less than normal in order to maintain a compression ratio corresponding to the master rod cylinder.

Referring again to Fig. 2, it will be observed that the link rod piston reaches the outer end of its stroke slightly before the wrist pin axis E passes through the cylinder axis OA_1. In this example we have angle $\beta = \alpha + \phi$, and we are assuming that the crankshaft is turning in a clockwise direction. The relative movement of the piston in this region is very slight, as denoted by the slope of the curve representing the path of E; and, for reasons explained, does not appear to give detrimental effects. Account must be taken of the fact that under normal circumstances, the maximum pressures within a cylinder do not occur until around fifteen degrees after the normal outer piston position, that is, when $\theta = \alpha$; therefore so long as point E has passed the cylinder axis OA_1 in either direction before these high pressures occur, there cannot be a sudden reversal of bending loads upon the shank of the master rod resulting from the forces along the axis of the link rod. Some designers give particular attention to this point in connection with the location of the wrist pin axis or axes.

Fig. 5.



Path of Link Rod Wrist Pin when β is greater than $\alpha+\phi$.

For the moment let us transfer our attention to Fig. 4, which is similar to Fig. 2, except that β is smaller, being less than the sum of $\alpha + \phi$. It is interesting to note in this example that the inner and outer positions of the piston are reached at normal points in the cycle, in other words, at the same relative crank positions that the piston of the master rod reaches corresponding points. We will refer to this example again for further discussion.

Let us direct our attention now to Fig. 5, which is a corresponding diagram to Figs. 2 and 4 except that angle β has been made greater than the sum of α and ϕ. The reader should carefully note the position of E at the point where $\theta = \alpha$, and also at other positions of the crank OD or R in comparison with the preceeding examples.

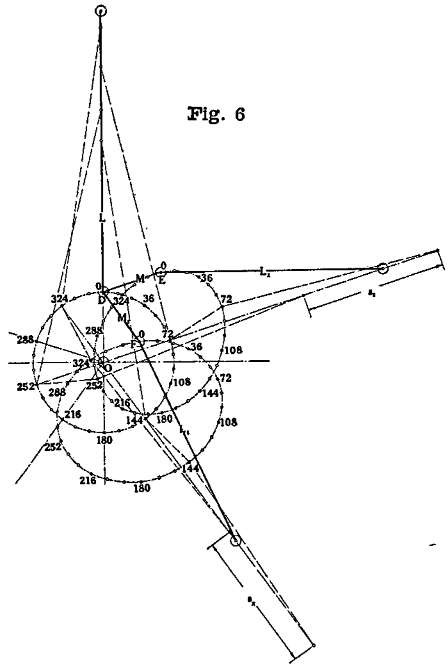
The included angle between cylinders, which is denoted by α, has been made 90 degrees for the present examples. This is the angle between cylinders in the conventional eight-cylinder Vee type engine. Although the relative positions of the wrist pin axis E would be different at other included angles between cylinders, the present examples should suffice to illustrate the general effect of changes in angle β. It should be borne in mind that different lengths of L, L_1, and M will also alter the relative positions of wrist pin axis E, but it is unnecessary for our present study to deal at any length with these effects.

We have in these diagrams a graphical picture of the effects of changes of angle β, yet the most significant point of interest is the variation in the maximum length of stroke (s) for the link rod cylinder. With values for all except angle β held constant in the three diagrams, we find in Fig. 2, in which the angle β is made equal to the sum of α and ϕ, that the length of stroke in the link rod cylinder is approximately seven per cent greater than the normal stroke (s) of the master rod cylinder. In Fig. 5, where β is greater than the sum of α and ϕ, the stroke (s) becomes approximately fourteen per cent greater than normal. These variations are both relatively large and would obviously require several dimensional changes in a design in order to obtain an equal compression ratio in all cylinders.

In Fig. 4, where β is made less than the sum of $\alpha + \phi$, a very desirable condition is obtained since the stroke (s_1) is equal to stroke (s). Moreover, as previously mentioned, the extremities of the stroke occur at normal positions of the crankshaft relative to the link rod cylinder axis. These results may be explained by the fact that in this example we have made the angle β equal to α. It will now be proved that these results can be obtained in any link rod cylinder if the angle between the master rod axis and link radius is held the same as the angle between the master rod cylinder axis and the particular link rod cylinder under consideration.

Before proceeding to prove these facts mathematically, let us study Fig. 6, which shows the path of two wrist pin axes of a five-cylinder radial engine. The principal dimensions of this diagram are the same as Figs. 2, 4,

Fig. 6



Paths of Wrist Pin Axes when $\beta=\alpha$. Five cylinder engine.

and 5 so that a direct comparison can be made with them if desired. The angle between each link radius and the master connecting rod axis is the same as the angle between the respective cylinders. Although the length of the link connecting rods is the same, the exact length of DF or M is slightly different than DE or M, the solution for which follows.

In this example, as in Fig. 4, the strokes of the link rod pistons (s_1 and s_2) are equal to 2R or s, the stroke of the master rod piston. Also, it will be readily observed that the inner and outer positions of these pistons occur at normal crank positions relative to each cylinder. This diagram not only gives us another interesting study of the paths of wrist pins, but it further proves that equal strokes and compression ratios may be obtained in any cylinder employing a link connecting rod regardless of its angular relationship to the master rod cylinder.

Assuming for the moment that the graphical solutions we have just studied are correct and that we can obtain an equal and normal stroke in any link rod cylinder, then it is quite apparent that it is necessary to make OA_1 equal to OA in order to maintain equal maximum compression ratios in all cylinders. In Fig. 7,

Let $OA_1 = OA = R + L$
and $DA_1 = L = M \cos\psi + L_1 \cos\phi_1$

We must now determine a value for either M or L_1 before we can solve for the other. In radial engines having more than one link connecting rod, it is obviously desirable to have all link rods of the same length for sake of interchangeability. This length should be measured from the layout and fixed at some desirable dimension for all cylinders under consideration, but after an approximate minimum length of M has been ascertained to meet the requirements of the design. With a value of L_1 determined, we can now solve for the exact value of M for any cylinder. It will be found that values for M will be different for each link rod of an articulated system according to the values of α, which is the angle between the master connecting rod cylinder axis and the respective cylinder axis of the link rod under consideration.

Since $\alpha = \beta$
and $\psi = \phi + \beta - \alpha$
then $\psi = \phi$

In Fig. 7, let us extend line DE to C where it intersects, at the angle α, a line drawn through A_1. Then $\triangle DCA_1 = \triangle BOD$ because side $DA_1 = DB$ and two of the included angles are equal. With a side and two angles known, the other sides and the angle can be determined as desired.

In $\triangle ECA_1$ we have obtained values for sides EA_1 or L_1 and CA_1, and angle α, therefore we can solve for the length of EC.

Then $DC - EC = DE = M$

Also, by another method, we can determine a value for $\angle DA_1C$ from $\triangle DCA_1$ and $\angle EA_1C$ from $\triangle ECA_1$ the difference being ϕ_1

Then $M \sin\psi = L_1 \sin\phi_1$

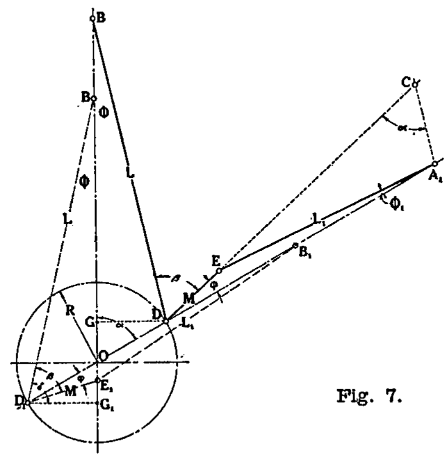


Fig. 7.

Diagram for Reference in Computing Values for Link Radius.

$$\text{or} \quad M = \frac{L_1 \sin\phi_1}{\sin\phi}$$

Likewise $M \cos\psi + L_1 \cos\phi_1 = L$

$$\text{and} \quad M = \frac{L - L_1 \cos\phi_1}{\cos\phi}$$

There are other methods by which exact values of M can be computed, but those given above appear to be about as simple as can be used.

To prove that A_1B_1 or $s_1 = s$ or 2R, let us refer to the diagram in Fig. 7 showing the crank after it has turned 180 degrees from its first position along the axis of the link rod cylinder. If we draw lines DG and D_1G_1 perpendicular to the master rod cylinder axis, then $\triangle BDG = \triangle BD_1C_1$ because two sides and an angle are equal, and the value of angle ϕ is the same as before. In $\triangle BD_1G_1$, $\delta = 90 - \phi$; and in $\triangle OD_1G_1$ $\delta - \beta + \psi = 90 - \alpha$, or $\delta = 90 - \alpha + \beta - \psi$.

But since $\beta = \alpha$

then $\phi = \psi$

and $\triangle DEA_1 = D_1E_1B_1$, because two sides M and L_1 and angle ψ are similar,

Therefore $D_1B_1 = DA_1 = L$

and $D_1A_1 - D_1B_1 = D_1D = 2R = s$

Conclusion

The simple method described for determining an exact location for the link rod wrist pins of articulated connecting rods to obtain equal strokes and compression ratios in all cylinders has been employed by the author for problems of this nature during the past seven or more years. It is not known that any others have developed a similar method for application to their design problems since no treatment has ever appeared in any of the many technical books and publications coming before the author's attention.

Propeller Design

By Frank W. Caldwell
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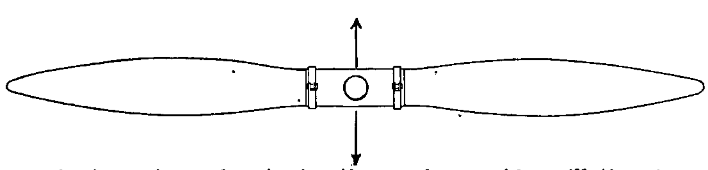

Vibratory stress due to inertia causing rapid oscillation in direction of arrows.

THE form of torsional vibration knows as flutter has been recognized for a long time and a number of propeller blade failures have been classified as torsional failures resulting from flutter. The propeller whirling test has given such a satisfactory means of checking flutter conditions that comparatively little trouble is experienced with this form of vibration at the present time. The vibrations resulting in a weaving action of the blades do not seem to be particularly troublesome as the blades are quite flexible in the direction of the thrust and there is a very powerful damping action from the centrifugal force.

There is, however, a form of vibration occurring in the plane of torque which may be caused either by natural periods or it may be a forced vibration resulting from the impulses of the engine cylinders. The blades are quite stiff in this direction so that vibrations in this plane may lead to comparatively high stresses before the deflection becomes great enough to permit the damping action from the centrifugal force. This appears to be the most troublesome form of vibration encountered at the present time. It leads to a tendency toward failure at the point of attachment of the blades to the hubs and failure of the hub parts.

This type of vibration cannot readily be reproduced on the whirling