

NOTE ON THE TAPER OF CONNECTING-RODS.

By WILLIAM D. MARKS,Whitney Professor of Dynamical Engineering, University of Penn'a.

In a recent work on the steam engine,* the writer made the following remark :

“ It is customary to make round connecting-rods with a taper of about one-eighth of an inch per foot, from the centre to the necks, which should be of the calculated diameter. Experiment does not show an increased strength from a tapering form.”

Passed Assistant Engineer C. H. Manning,† in a pleasant correspondence with the writer, differed from him, deeming it the better method to taper connecting-rods from the crank-pin end to the cross-head end, as “ experience had shown that connecting-rods usually failed at the crank-pin.”

Led by this remark to make a more thorough investigation into the stress upon connecting-rods due to their own inertia, than he had before deemed necessary, the writer submits the results, hoping they may be of interest to engineers engaged in the designing of mechanism.

The connecting-rod, if of *wrought-iron*, must be considered either, 1st, as a short column, tending to rupture by crushing, or 2d, as a long column, tending to rupture by buckling. The tendency to fail in tension, can be neglected as being much less than in the two cases mentioned.

In the first case it is obvious that tapering will not add to its strength, if we neglect the stress due to its inertia and weight.

In the second case, theoretically, if we disregard the stress in flexure due to the inertia and weight of the connecting-rod, the increase in its diameter will be a maximum at its centre. (See “ Weisbach’s Mechanics of Engineering,” Sec. iv, Art. 267.)

The connecting-rod, if of *steel*, may be considered, 1st, as a tension-rod, tending to fail in tension, or 2d, as a long column, tending to rupture by buckling. The ability of steel to withstand a much

* The “ Relative Proportions of the Steam Engine.”

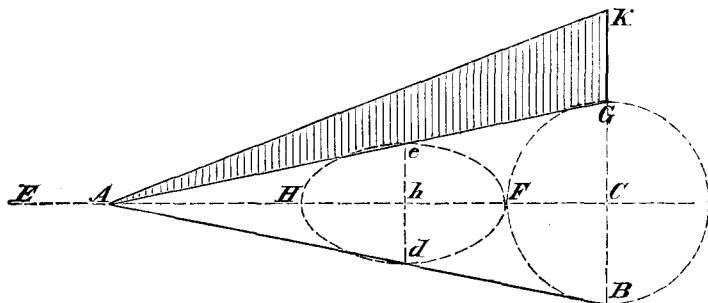
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greater stress in compression than in tension, avoids the necessity of considering it as a short column, failing by crushing. In both cases the inertia and weight of the rod are disregarded.

These are the general conditions which have controlled the mechanical engineer in the consideration of the proportions of wrought-iron or steel connecting-rods. In addition, it has been customary in order to meet an unknown stress in flexure, due to the inertia of the rod, to increase the diameter of the rod at the middle, or latterly to give an increased diameter at the crank-pin, and taper from this to the smallest dimension of the rod at the necks or neck, this increase being purely empirical.

Prof. R. H. Thurston, of the Stevens Institute of Technology, in a personal letter, October 31st, 1878, says: "Where iron and steel

FIG. 1.



are used, the figures adopted as constants must vary greatly, especially with steel. The value of E (modulus of elasticity) varies *enormously*," with which opinion the writer agrees perfectly, only regretting that the very basis of all correct calculations is thus taken from us, and we are forced to be contented with results which have a reasonable probability of correctness, if we assume average values for the safe stresses per square inch, and the moduli of elasticity of the material with which we are dealing.

If a straight line, AB , Fig. 1, have its extremities A and B respectively caused to move reciprocally upon the straight line $A E H$ and in the perimeter of the circle $B F G$, any point upon the line AB as d will trace an ellipse as $d H e F$.

If, now, we let the length of the line $AB = l$; the radius CB of the circle $= r$; the variable distance of the point d from the point

Substituting this value of m in equation (3), and integrating between the limits $l_1 = 0$ and $= l$, we have, denoting the whole resistance of the rod AB by P ,

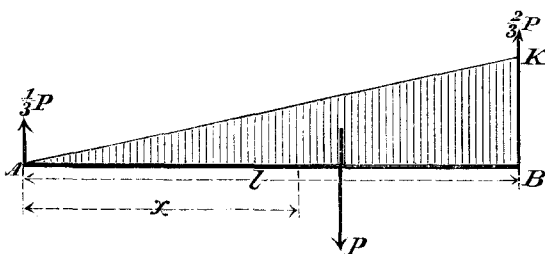
$$P = \int_0^l \frac{F \gamma v^2}{g l r} l_1 dl_1 = \frac{F \gamma v^2}{2 g r} l,$$

or since the weight of the rod $= G = F \gamma l$,

$$P = \frac{G v^2}{2 g r}. \quad (5)$$

And since this load upon the rod due to the resistance of its own inertia increases uniformly from the end A to the end B , the rod can, with sufficient approximation, be supposed in the condition of

FIG. 2.



a horizontal beam loaded with a triangularly-shaped load, as shown, Fig. 2, in the cross-hatched portion AKB .

For a vertical engine, we can neglect the weight of the

connecting-rod, considering only the stress due to its inertia.

For the moment of flexure of any cross-section at a distance x from the extremity A , we have, letting $c =$ the load per unit of length at the point K ,

$$\frac{1}{3} P x - \frac{c}{6} x^3 = \frac{1}{3} P \left(x - \frac{x^3}{l^2} \right), \quad (6)$$

which becomes a maximum for $x = l \sqrt{\frac{1}{3}} = 0.578 l$, and at which point, therefore, the maximum cross-section of the connecting-rod should be placed.

In horizontal engines it is necessary to take into consideration the weight of the rod, as well as its inertia; the weight of the rod may be regarded as a uniformly distributed load, always acting in one direction.

For the cross-section, at a distance x from the extremity A , the moment of flexure is letting $G =$ whole weight of rod $= \gamma l$

$$\left(\frac{1}{3} P + \frac{1}{2} G\right) x - \left(\gamma \frac{x^2}{2} + \frac{1}{6} \frac{c}{l} x^3\right) =$$

$$\left(\frac{1}{3} P + \frac{1}{2} G\right) x - \frac{G}{2l} x^2 - \frac{P}{3l^2} x^3 : \quad . \quad . \quad . \quad (7)$$

which is a maximum for

$$x = -\frac{Gl}{2P} \pm l \sqrt{\frac{1}{3} + \frac{G}{2P} + \frac{G^2}{4P^2}} \quad . \quad . \quad . \quad (8)$$

But we have from equation (5),

$\frac{G}{2P} = \frac{gr}{v^2}$, in which g = acceleration of gravity, v = the linear velocity of the crank-pin, B , in feet per second, and r = the radius of the crank in feet. Substituting this value in equation (8), we have,

$$x = l \left\{ -\frac{gr}{v^2} \pm \sqrt{\frac{1}{3} + \frac{gr}{v^2} + \left(\frac{gr}{v^2}\right)^2} \right\} \quad . \quad . \quad . \quad (9)$$

which shows that as the velocity of the crank-pin increases, the value of x approaches more nearly to $l\sqrt{\frac{1}{3}} = 0.578l$, but can never quite equal it.

Referring to equation (7), and taking P = zero, we find the maximum value of $x = \frac{l}{2} = 0.5l$.

We thus see that the greatest moment of flexure of any connecting-rod lies between the limits $0.5l$ and $0.578l$, measured from the point A .

It is of interest further to note, that the crank-pin takes one-half the stress due to the weight of the rod, and two-thirds of the stress due to the inertia of the rod.

The end, A , takes one-third of the stress due to the inertia of the rod, alternately increasing and decreasing the stress upon the guides to this amount and one-half the stress due to its weight.

In engines having a large number of revolutions per minute, P becomes worthy of notice. In slow-moving engines it is very small, and may be neglected.

From these considerations, the writer is of the opinion that the failure of connecting-rods, at the neck nearest the crank-pin, *if they are properly proportioned*, is probably due more to the crank-pins being

out of truth, rather than to the weight or inertia of the rods themselves.

He further takes this opportunity to state, that in a properly proportioned engine of any assumed horse-power, the *only* method of diminishing the weight of the moving parts, is to increase the number of strokes—lengthening the stroke will not do it.

December 7th, 1878.

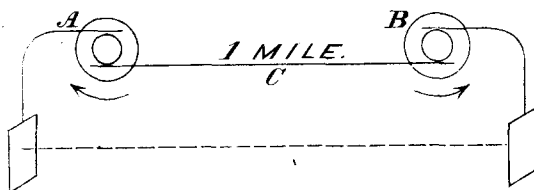
ON THE TRANSMISSION OF POWER BY MEANS OF ELECTRICITY.

By PROFS. ELIHU THOMSON and EDWIN J. HOUSTON.

The statements recently made as to the size and cost of the cable that would be needed to convey the power of Niagara Falls to a distance of several hundred miles by electricity, have induced the authors to write the present paper, in the hope that it may throw light upon this interesting subject.

As an example of some of the statements alluded to, we may cite the following, viz.: That made by a certain electrician, who asserts that the thickness of the cable required to convey the current that could be produced by the power of Niagara, would require more copper than exists in the enormous deposits in the region of Lake

FIG. 1.



Superior. Another statement estimates the cost of the cable at about \$60 per lineal foot.

As a matter of fact, however, the thickness of the cable required to convey such power is of no particular moment. Indeed, it is possible, should it be deemed desirable, to convey the total power of Niagara, a distance of 500 miles or more, by a copper cable not exceeding one-half of an inch in thickness. This, however, is an extreme case, and the exigencies of practical working would not require such restrictions as to size.

The following considerations will elucidate this matter. Suppose two machines connected by a cable, of say 1 mile in length. One of these machines, as, for example, A, Fig. 1, is producing current by