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“Factors affecting the Evenness of the Turning Moment in Locomotives.”

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AN even turning moment has always been recognized as a very desirable feature in a locomotive, even as far back as 1812 when Blenkinsop first employed two cylinders with cranks set at right-angles to each other with this result in view. The object of this Paper is to examine briefly the factors which govern the variation in turning effort in an ordinary two-cylinder locomotive, and also the effect of using three or four cylinders.

In most modern engines, the cylinders, which are made large enough to develop sufficient power at an economical cut-off when running at normal speeds, are more than sufficiently large for the late cut-offs and low piston speeds that obtain when starting a train. Consequently the tractive effort exceeds the adhesion even when the rails are dry. Should, however, the rails be slippery owing to climatic conditions, the tractive effort developed by an engine will be limited solely by the adhesion, even if sand be used. Furthermore, the starting resistance, per ton, of a train is greater than the resistance per ton at the low speeds attained by freight trains over heavy portions of the road : and consequently for this class of traffic the heaviest train for a given engine is that which it can start from rest, on the worst portion of the road, and under the worst climatic conditions. This, as was seen above, depends entirely on the adhesion.

It may be well to commence by stating a simple and well-known fact. Since the coefficient of friction between wheel and rail diminishes as soon as slipping commences, it follows that if the maximum turning moment exceeds the frictional resistance at any point in a revolution, slipping will at once commence and continue even though the mean turning moment may not be sufficient to cause the wheels to slip. Hence, to prevent slipping, we must limit the maximum turning moment to a value just insufficient to cause slip. The nearer the mean turning moment approaches the maximum turning moment, the greater is the mean tractive effort which can be got from

the same weight on the coupled wheels. In other words, at low speeds, the mean tractive effort for any given adhesive weight is proportional to the percentage which the mean torque bears to the maximum torque. This, as was seen above, is of paramount importance in freight engines, and even in the case of engines designed for high speeds where other considerations, such as boiler capacity, are of greater importance, a more even turning moment may enable the number of wheels, which have to be coupled for starting purposes, to be reduced, and a freer running engine to be obtained.

Figs. 1.

Two-Cylinder Locomotive.—In order to determine the effect of an early or late cut-off, two indicator cards have been taken (Figs. 1). Card "A" corresponds to a low steam chest pressure, such as arises from a small regulator opening, and a late cut-off. Card "B" corresponds to a larger opening of the regulator and an early cut-off. (The area of each card is approximately the same). Below each card is drawn the nett pressure diagram, i.e., the difference between the pressures on the two sides of a piston.

Since it is at very low speeds that an even turning effort is most important, no account has been taken of the inertia of the reciprocating parts.

cating parts. *Fig. 1* also shows a diagram explaining the notation of the crank positions in the turning effort diagrams that follow. In these diagrams it has been assumed that indicator cards for both ends of the cylinder are identical. This is not likely to be strictly true in actual practice, but it is outside the scope of this Paper to consider inequalities in the valve gear which differ in every individual class of engine.

In *Fig. 2a* the torque diagram is drawn for Cards "A" and "B" with a ratio of connecting rod to crank of 6 : 1. The dotted curve is the torque due to the leading crank only, and the ordinate of the full curve at any point is obtained by adding to the dotted curve the ordinate of the same curve at a point 90° behind, i.e., the full line represents the total torque for the two cylinders. No actual scale is specified for the torque, as its numerical value depends on dimensions which are peculiar to each class of engine. All that is here required is to find a ratio which the mean torque bears to the maximum torque. In *Fig. 2b* the diagrams are similar to those in *Fig. 2a*, except that the ratio of connecting-rod to crank is 8 : 1.

From these two figures it will be seen that the ratio of mean torque to maximum torque is almost the same for both indicator cards. From the dotted line representing the torque due to one crank only, it is seen that Card "A" gives a more even turning moment than Card "B," but on combining the turning moment of the two cranks, the shape of the indicator card is seen to have a very small effect on the evenness of the total turning moment. Practically, it is preferable to use a late cut-off and a small regulator opening when slip is liable to occur, as in this case, as soon as slipping commences the small regulator opening is insufficient for the suddenly-increased piston speed, and tends automatically to check the slipping. Comparing the diagrams in *Fig. 2a* with those in *Fig. 2b* a slight improvement is seen to be effected by the longer rod. The salient point is, that in an ordinary two-cylinder engine, the longer the connecting-rod the more even is the turning moment, but, even with the greatest practicable length, the improvement thus effected is small. In a two-cylinder locomotive the ratio of mean torque to maximum torque is limited, even in the most favourable cases, to about 82 per cent, and in most engines will not be more than about 79 per cent.

In *Fig. 2c* the torque diagram is drawn for both indicator cards as before, but with a connecting rod of infinite length—that is to say the pistons move with simple, harmonic motion. Such a case may appear to have no practical interest, but a little reflection will show that the effect of two pistons driving cranks that are opposite to

each other is the same as that of one piston of double the area and the same stroke, driving its crank through a connecting rod of infinite length, and this figure represents the case of a four-cylinder engine with its cranks equally spaced. It will be seen that in this case the evenness of the turning moment is very much improved.

A geometrical proof of the fact that two pistons acting on cranks

Figs. 4.

180° apart, through connecting rods of any length, are equivalent to one piston of double the area (and the same stroke) and through a connecting rod of infinite length, is given in the Appendix.

Three-Cylinder Locomotive.—Fig. 4a is the torque diagram for a three-cylinder engine, having its three cranks equally spaced. This diagram is derived from Card "B," the dotted line representing the torque due to one cylinder, while the full line is obtained by

adding to the height of the dotted curve at any point the heights of the same curve at points 120° in advance and in arrear. It will be seen that the ratio of mean torque to maximum torque is $89\cdot3$ per cent., that is the same as the four-cylindered engine with a similar indicator card.

If the cranks are spaced so that the two outside cranks are at 90° and the inside crank bisects the obtuse angle between them (an arrangement which has the advantage of keeping the coupling rod crank pins at right angles) a very different turning moment diagram will be obtained.

Conclusion.—The results of the foregoing investigations may be summed up as follows :—

1. The shape of the indicator card has very little effect on the evenness of the turning moment in a two-cylinder engine.

2. The lengthening of a connecting rod has a tendency to improve the evenness of the turning moment, but the improvement thus effected, without making the rod impractically long, is small.

3. The various arrangements of the cylinder and their approximate ratios of mean torque to maximum torque are as follows :—

(a) 4-cylinders with cranks equally spaced	ratio = 89 per cent.
(b) 3-cylinders do. do.	„ = 89 „
(c) 2-cylinders with cranks at 90° . . .	„ = 79 „
(d) 3-cylinders with outside cranks 90° apart, inside cranks 135° from outside cranks	„ = 68 „

Hence, as far as evenness of turning moment is concerned, there is nothing to choose between a 4-cylindered engine, and a 3-cylindered engine with its cranks equally spaced.

It may be added that the remarks in (1) and (2) apply also in the case of a three-cylinder engine with cranks equally spaced.

The Paper is accompanied by seven diagrams, from which the Figures in the text have been prepared.

APPENDIX.

PROBLEM RELATIVE TO THE EQUIVALENCE OF TORQUE DUE TO ONE
AND TWO PISTONS UNDER VARYING CONDITIONS (*See* p. 393).

Let P be the pressure on two pistons, each of area S , which act on cranks OA and OB of similar radius and opposite each other (*Fig. 3*).

Then $\text{total torque} = PS (OG + OH).$

Fig. 3.

From A and B respectively draw AF and BE parallel to the axis of the engine and AL and BK perpendicular to it. Then, if the connecting-rods DB and CA are equal, the triangles DBK and CAL are also equal, from which it follows that EG equals HF .

Now since $OG + OH = OE - EG + OF + HF = 2 (OE),$

it follows that the $\text{total torque} = 2 PS (OE).$

Hence the torque is equivalent to that produced by a piston of twice the area, but with the same stroke and a connecting-rod of infinite length.