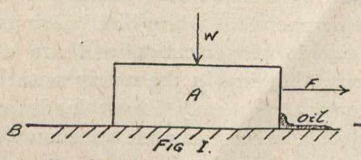
of his special brand of engine oil. Now, all of these have their uses in reducing friction, and it is worth while to endeavor to see clearly where and when the saving takes place in the use of any of the remedies. Let us briefly consider the theory of the matter.

Friction is of two kinds: (1) Static Friction, or the friction between two bodies at rest; and (2) Kinetic Friction, or the friction between bodies in motion.

Static friction is the force required to produce motion between two bodies; it is directly proportional to the normal pressure between them and independent of the areas of the surfaces in contact. It depends almost entirely upon the nature of the surfaces.

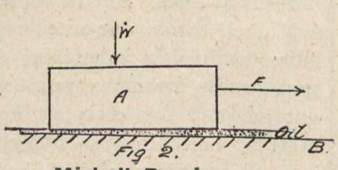
To have a low starting friction in a bearing the surfaces require to be highly finished and to be of such metals as have naturally a low co-efficient of friction, and it is here



Ordinary Collar Bearing.

Oil squeezed out and surfaces in metallic contact.

$$\text{Co-efficient of friction } \frac{F}{W} = \text{about } .003.$$



Michell Bearing.

Wedge of oil maintained between surfaces and no metallic contact.

$$\text{Co-efficient of friction } \frac{F}{W} = \text{about } .03.$$

that the use of an anti-friction metal comes in. The best eliminator of friction, however, is the roller ball.

Referring to the diagram, if all the curves had been continued until they intersected the vertical line, the intersecting points would indicate the co-efficients of static friction for each of the four bearings, and you will notice that these would all be about the same, and very high compared to that of the ball bearing. One advantage, then, of the ball or roller bearing is its very low starting friction, and in machines which are constantly being started and stopped, such as travelling cranes, this is a most valuable feature, and a saving of about 50 per cent of power can be effected. We see that this advantage is lost when the speed of the latter has reached about 10 feet per minute. The blue line on the large diagram, which shows the co-efficients of friction in a theoretically perfect journal bearing (taken crom Archbutt & Deely on "Lubrication"), shows a sudden drop in the friction when the speed approaches about 10 feet per minute. On the other hand, if you look at the red line, which shows the co-efficient of friction of a flat thrust bearing, you will see that while there is a slight fall as the speed increases there is no sudden drop and the friction remains high. There is, therefore, some change which takes place in a properly lubricated journal bearing, and not in an ordinary thrust bearing. This change is the foundation of the oil wedge, as pointed out by Beauchamp Tower. It forces the surfaces apart and brings a new condition of things into operation. The bearing then no longer follows the laws of static and low-speed friction, but those of the flow of viscous fluids, with which Professor Osborne Reynolds' name is intimately associated. According to this theory, the co-efficient of friction increases as the area of the surfaces in contact, as the speed and as the viscosity of the oil. In other words, the friction then becomes merely the force necessary to shear a certain sectional area of oil. It also depends upon the proportion of length to breadth of surfaces in contact; but for any given bearing under these conditions the work done in shearing the oil film increases as the spread increases, consequently you will notice that the blue line gradually rises, as does also the yellow line of the ball bearing.

It is not necessary to deal with the laws of low-speed kinetic friction(which are similar to those of static friction), as my principle object is to point out:—

(1) The change that takes place as soon as the oil wedge is formed.

(2) That an ordinary thrust bearing never has a co-efficient of friction anywhere near as low as a properly lubricated journal bearing.

Having shown a reason for differences in the behaviour of a journal and ordinary flat thrust bearing is the absence of any point of "no pressure" at which the oil can insinuate itself, I will now deal with a new form of thrust bearing which gives similar results to a properly lubricated journal bearing. The black line on the diagram shows the result of a test of one of these bearings and the close agreement which it has to a journal bearing as shown by the blue line.

In 1902 Mr. A. G. M. Michell—an Austrailan engineer—published a paper in which he shows mathematically, on Reynolds' principle for the lubrication of plane surfaces, how the position of the centre of oil pressure in such surface could be calculated, and that if the external pressure were applied at the calculated point the block would then be free to lift at its leading edge and admit oil in the same way as a journal bearing (Figs. 1 and 2). He also demonstrated this by means of a model. Although of scientific interest, this was not of much practical value, as slides moving in one direction only are never used in practice. Three years later, however, he took a patent for the application of his theory to thrust bearings, and showed by practical tests that his calculations as applied to a block moving in a straight line were very closely approximated to a number of blocks moving in a circular path. Since then several of these bearings have been made and applied with complete success to centrifugal pumps and steam turbines. They are specially adapted to continuous running machinery, and enable similar pressures to be supported on a thrust bearing as on the journal type, and with equally low co-efficients of friction.

Ordinary marine thrust bearings are loaded up to about 60 or 70 pounds per square inch, and doubtless many of you have experienced trouble in keeping them cool without the water service even at this low pressure. You will, therefore, probably be interested in a thrust bearing which will work at 500 or 600 pounds per square inch and with about one-tenth of the friction. To give two actual cases, a Michell

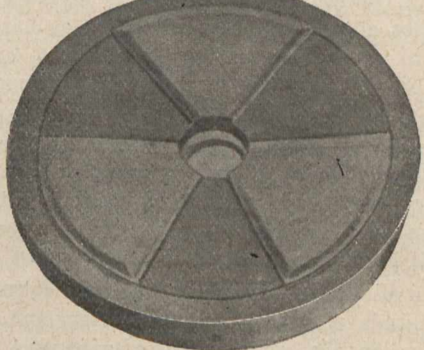


Fig. 3.—Michell Thrust Bearing.

thrust bearing on a vertical type centrifugal pump running at 300 revolutions per minute and loaded to 220 pounds per square inch was examined after running night and day without stop for four months and the marks of the scraper tool found to be still on as when started. Another fitted to a steam turbine has been running at a speed of 1,800 revolutions per minute with a load of 375 pounds per square inch for the last two months without requiring attention. Blue prints are shown giving the details of these two bearings (Figs. 4 and 5).

The essential feature of Michell thrust bearings consists in dividing the working face into a number of segmental blocks, each of which is pivoted at a point about one-third the width of the block from the trailing edge, and