EXPERIMENTAL STUDY OF JOURNAL BEARINGS

By Professor Ch. Hanocq *

PART I. COMPLETE BEARINGS

This paper summarizes a series of theoretical and experimental studies on the full journal bearing which were published between 1929 and 1931.†

Theoretical Deductions. Theory indicates that, in a complete bearing, the centre of the shaft is displaced in a direction perpendicular to the

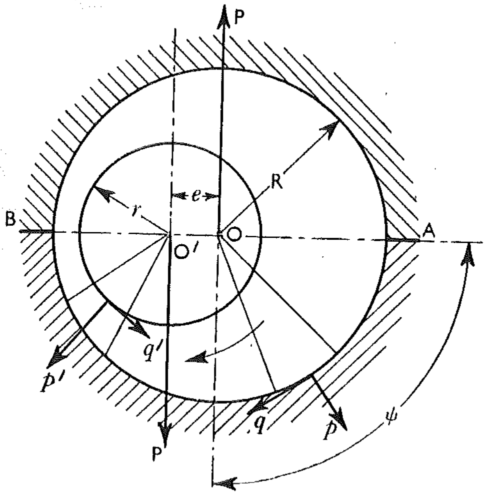

Fig. 1. Displacement of Journal

direction of the load (Fig. 1) as the value of

$$X=\frac{\mu N}{p}\left(\frac{r}{a}\right)^2$$

passes from ∞ to zero. When $X=\infty$, then $e=0$, whereas when $X=0$, then $e=a$. In the above equation μ represents the coefficient of absolute viscosity in kilogrammes, metres, seconds; N is the angular speed in revolutions per second; p is the specific pressure in kilogrammes per square metre; e is the distance between the centres; a the radial clearance $(R-r)$; and c the ratio a/e.

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† See *Revue Universelle des Mines* and Mémoires du Congrès International de Mécanique appliquée, Liége, 1930.

Fig. 2 shows how c varies with X and gives the values of $f_c(r/a)$ and of $f(r/a)$. It can be shown that between f_c and f there is the relation

$$f=f_c+\frac{e}{r} \qquad (1)$$

since, if we neglect the forces q in relation to p (Fig. 1), then

$$M=M_c+P\times e \qquad (2)$$

The theoretical deduction regarding the value of c and thus of $(f-f_c)$ only holds for very low average specific pressures $\bar{p}$, defined by

$$\bar{p}=\frac{P}{l\times d} \qquad (3)$$

This theory is based on the hypothesis that a continuous film

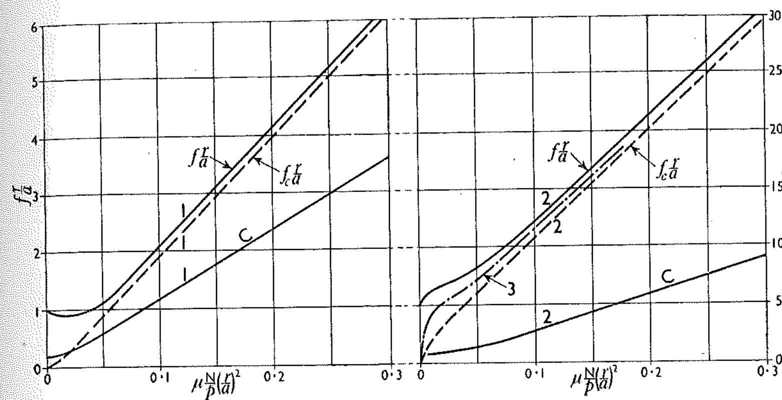

Fig. 2. Variation of c with X

surrounds the whole shaft. If we try, under these conditions, to deduce the value of the pressure at any point, or, preferably, the value of the ratio $p/\bar{p}$ for different values of X, it is found that:—

(1) The distribution depends essentially on the point at which the lubricant is introduced into the bearing;

(2) When the oil is introduced on the upper surface or on the horizontal diameter negative pressures may exist under the upper half-brass, i.e. pressures less than that of the atmosphere. As these pressures cannot be greater than 1 kg. per sq. cm., there is an average limiting value for the mean pressure $\bar{p}$ beyond which the theoretical conditions are no longer fulfilled.

(3) If it is assumed that atmospheric pressure prevails over the whole of the upper half-brass, the distribution of the pressures $p/\bar{p}$ can be

expressed mathematically. For $X=0{\cdot}013$, the values of $p/\bar{p}$ corresponding to this assumption are given in Fig. 3. Here again the pressures are negative and for $X=0{\cdot}013$, $p/\bar{p}$ attains a minimum value of 4, i.e. with $\bar{p}=0{\cdot}25$ kg. per sq. cm., p will be 1 kg. per sq. cm. This equilibrium in turn becomes impossible beyond a certain value of the average pressure $\bar{p}$; atmospheric pressure tends to become established on the arc BC and equilibrium between the load P and the elementary forces $prd\alpha$ is set up only on the arc AC.

The variation of c and of ψ in terms of X depends thus on the way in which atmospheric pressure is established on part of the periphery of the upper half brass; in other words it depends finally on X and on $\bar{p}$.

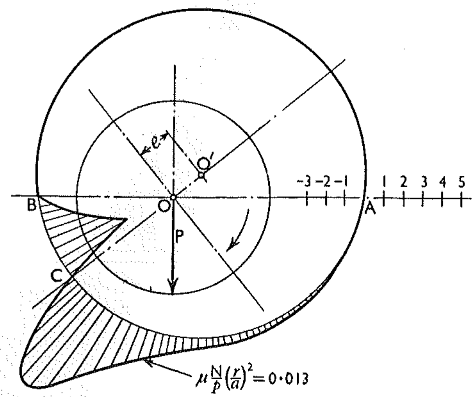

Fig. 3. Values of $p/\bar{p}$

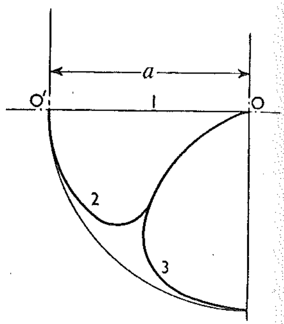

Fig. 4. The Three Limiting Cases

Fig. 4 summarizes the three limiting cases of the displacement of the centre of the shaft in relation to the centre of the bearing:—

Case 1. Complete bearing, continuous film, low pressure:

$$\psi=\frac{\pi}{2}$$

Case 2. Complete bearing, but with a continuous film only on the lower half brass.

Case 3. Complete bearing, with a continuous film on the arc AC only.

As the value of f is always

$$f=f_c+\frac{e}{r}\sin\psi \quad . \; . \; . \; . \; . \; . \; . \quad (4)$$

the difference between f and f_c will, other things being equal, always be less in cases 2 and 3 than in case 1, since $e\sin\psi$ tends finally to become zero in case 3 whereas in case 2 $e\sin\psi$ tends towards the value a as in case 1.

We can thus anticipate that conditions will become unstable when $\bar{p}$ is varied or, even when p and N are constant, X decreases, owing to the decrease in viscosity μ with the temperature. Fig. 2 gives the values of c and of $f(r/a)$ and $f_c(r/a)$ according to case (1), in comparison with the same values deduced according to case (2).

Experimental Study. The method employed was designed to enable the values of M and M_c to be determined simultaneously, so that f, f_c, and $e\sin\psi$ could be determined.

To determine M a decelerating method was employed; a shaft supported by two similar bearings and loaded with two flywheels turns at a speed N; by determining N at different times t, the deceleration curve can be deduced, giving N or ω in terms of t. From this curve we can then deduce $d\omega/dt$ for different values of N and therefore M, since

$$2M+M_1=-I\frac{d\omega}{dt} \quad . \; . \; . \; . \; . \; . \; . \quad (5)$$

where M_1 represents the moment of resistance due to atmospheric friction on the flywheels (which can be calculated as a function of their dimensions and speed ω).

The direct method was used for M_c, the bearing being carried on two ball bearings so that it was free to rotate about its axis. A counterweight p applied at a distance λ from this axis, enabled the bearing to be kept in its initial position, whence $M_c=p\times\lambda$.

To determine f and f_c at high loads, ball bearing mountings were employed and the coefficient of friction of the ball bearings was investigated in order to calculate the couple M_r due to them, as this couple affects equation (5). In this way f and f_c were determined for values of $\bar{p}$ ranging from 2·7 to 20 kg. per sq. cm.

The experimental results (Fig. 5) show that the points for values of $\bar{p}$ less than or equal to 5 kg. per sq. cm. fall approximately on curve 2; the points for higher pressures are more regular and lie on curve 3. As the experimental results give a curve for f_c which corresponds to the theoretical curve, we can deduce f in terms of X. The results are given in Fig. 6, where curve 2 refers to pressures equal to or below 5 kg. per sq. cm., and curve 3 to pressures above 5 kg. per sq. cm.

All these results were obtained with a clearance $a/r=1/170$, using a fixed oil-ring bearing, the oil being admitted at the upper surface.

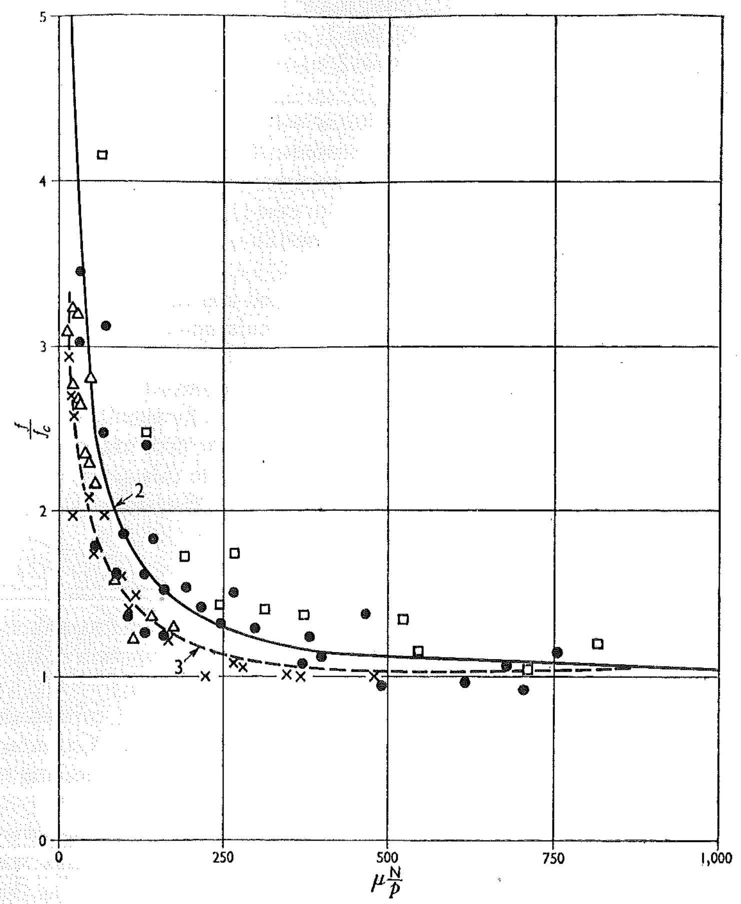

Fig. 5. Values of f and f_c

$2r=40$ mm. $\frac{a}{r}=\frac{1}{170}$

□ $p=$ 2·7 kg. per sq. cm.
● $p=$ 5 ,, ,,
× $p=10$,, ,,
△ $p=20$,, ,,

Conclusions

The author's and certain American investigations show that theory is fulfilled in a remarkable way and that the curves of Fig. 2 suffice to give the value of f. Curve 2 is suitable for low pressures and curve

3 for high pressures, provided a complete journal bearing is used which is lubricated at the upper surface under atmospheric pressure.

Two points should be noted:—

(1) At values of $(\mu N/p)\times 10^8$ less than, say 5, the film ceases to exist, and the equation for f changes completely, as f rises very rapidly with the decrease of X ;

(2) With values of a/r below, say, 1/500, the value of f increases by a constant amount independently of X, varying with a/r in such a way that

$$f_t=f+\Delta f \text{ and } \Delta f=10^{-13}\times 2{\cdot}66\left(\frac{r}{a}\right)^3 \quad . \quad . \quad . \quad . \quad (6)$$

f being derived from the curves of Fig. 2.

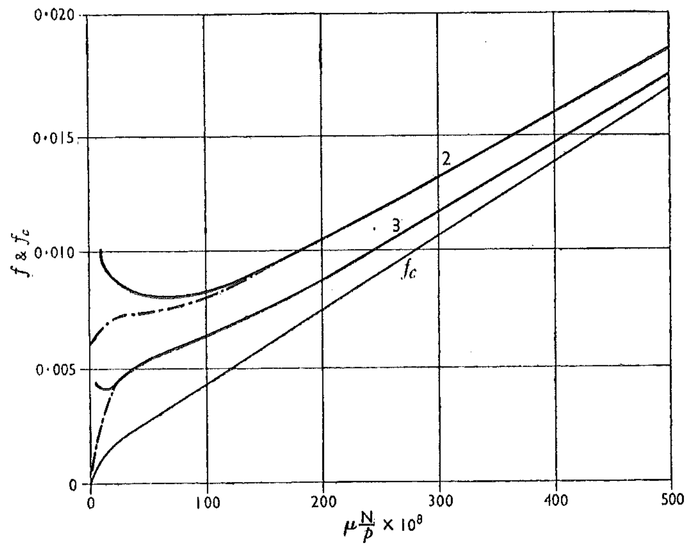

Fig. 6. Values of f in terms of X

Curve 2 Pressures equal to or below 5 kg. per sq. cm.
,, 3 ,, above 5 kg. per sq. cm.

The critical value of $(\mu N/p)\times 10^8$ is dealt with below in connexion with partial brasses and here it suffices to say that the limiting value can fall below 5, particularly when the pressures are not too high, say 10 kg. per sq. cm.

All the curves hold good for ratios of l to d above 0·8 ; at lower values, f increases appreciably, as lateral leakage reduces the thickness of the oil film. So long as hydrodynamic conditions are fulfilled, i.e. for all values of

$$\frac{\mu N}{p} \times 10^8 \geqq 5$$

the coefficient of friction f *depends neither on the chemical nature of the oil nor on the composition of the bearing metal.*

To prove that the nature of the oil has no influence the author used as lubricant a sugar syrup of known viscosity. The values of f_c and therefore of f were given by curves identical with those obtained using oil as lubricant.

Part II. Partial Bearings

The mathematical theory summarized in Part I is not applicable to journal bearings with partial brasses (Fig. 7). For small angles (2β) the bearing can be regarded with fair accuracy as an articulated block, for which the equation for f in terms of the independent variable $\mu N/p$ takes the form $f = A\sqrt{\frac{\mu N}{p}}$.

Boswall has obtained values of A for different angles 2β and has made it possible to measure not only the couple M but also the value of M and therefore

$$f - f_c = e/r \sin \psi = \tan \alpha.$$

With brasses having a relative clearance $a/r = 1/250$, Boswall obtained points falling on different curves according as the bearing angle was between 45 and 90 deg. or less than 30 deg. The approximately parabolic axes in Fig. 9 are reasonable extrapolations of these curves for high pressures.

The author decided to determine the coefficient of friction f for high pressures such as are usual in practice. The first requirement was to determine the equation for f in terms of $\mu N/p$ for brasses with an angle 2β becoming smaller and smaller. To avoid as far as possible the need for measuring $(f - f_c)$ brasses run-in in the cold were employed.

The testing machine employed by the Société Générale Isothermos of Paris, which is based on the principle of the balance applied to the brass (Fig. 8), is designed to work with ordinary journals having a diameter of 140 mm. and an axial width of 300 mm. The load could be increased to 12,000 kg. and the speed to 840 r.p.m. By measuring the reaction R opposed to the force F, applied at M and

resulting from the action of F, the couple $F \times r = R \times L$ can be deduced. Therefore

$$f = \frac{L}{r} \cdot \frac{R}{P}$$

Thanks to the accurate finish of the knife-edges, the coefficient of friction could be measured with an accuracy of roughly 1 in 10,000.

So as to eliminate the error arising from the difficulty of ensuring that the force P acts precisely along the vertical passing through the

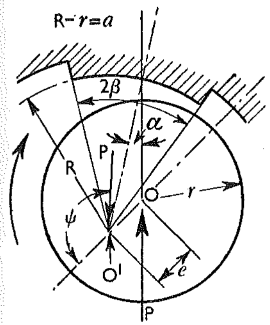

Fig. 7. Journal Bearing with Partial Brass

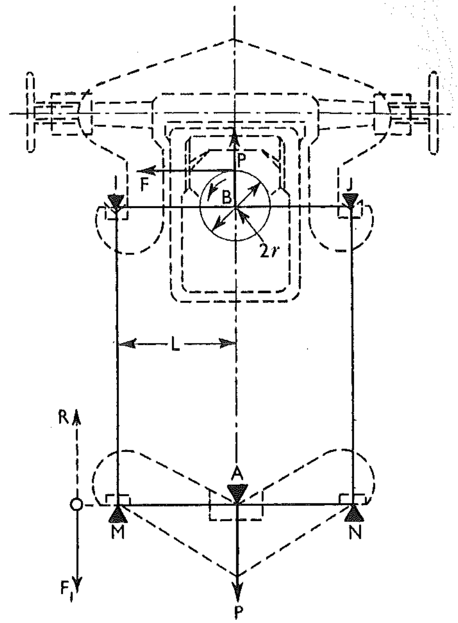

Fig. 8. Diagram of Testing Machine

axis of the shaft, double readings were taken, the working direction being reversed. As the upper rocking lever could be displaced with respect to the brass by hand adjustment of one of the wheels on the right and left of the rocking lever (Fig. 8), the line of action of the load could be moved so as to pass through the centre of the shaft. In this case the readings taken in both directions of running gave practically the same result, since

$$R_1 = F_1 = \frac{F \times r}{L + e^1} \qquad R_2 = F_2 = \frac{F \times r}{L - e^1}$$

Mr. Bastin, director of the Isothermos Laboratory, fitted the rocking lever with a very sensitive water level and a comparator to measure the displacement of the lever with respect to the brass, so that the system could be balanced without adding a counterpoise. Thus if the axis of the applied force is moved to a distance E from the vertical passing through the centre of the shaft so that the rocking lever remains

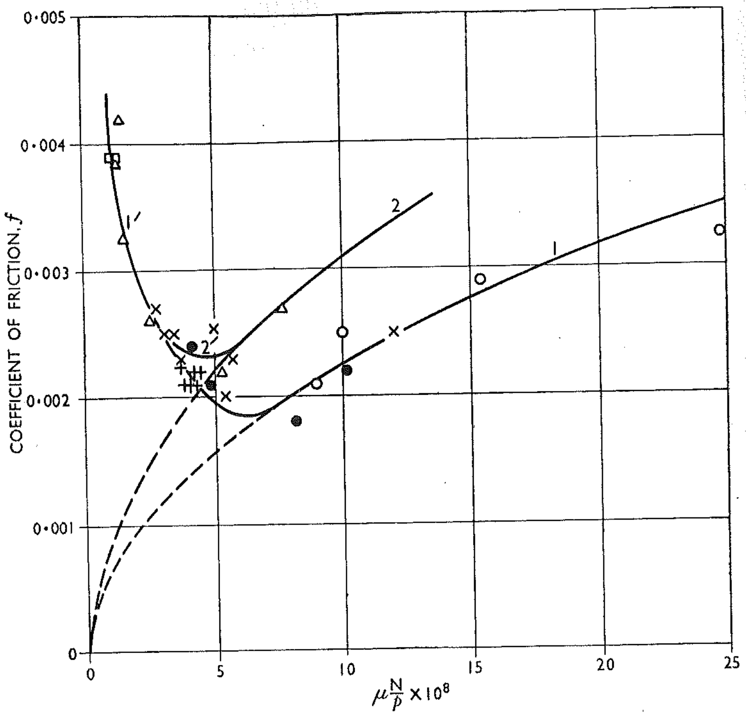

Fig. 9. Curves for Different Bearing Arcs

○	102 deg.	×	41 deg.
●	49 ,,	△	33 ,,
+	45 ,,	□	24·5 ,,

horizontal, the system can be said to be in equilibrium, so that $f=F/P=E/r$.

The results with brasses run-in in the cold are given in Fig. 9. It is noteworthy that curve (1) (for angles above 45 deg.) agrees strictly with Boswall's curve. For angles below 45 deg., the curve (2) traced by the author is the same as that given by Boswall for angles equal to or less than 30 deg.

Therefore the conclusions that $A=7{\cdot}10$ for values of 2β between 90 deg. and 45 deg., and $A=8{\cdot}80$ for values of 2β equal to or less than 30 deg. can be extended to small values of $\mu N/p$.

There is, however, a critical value of $\mu N/p$ beyond which the expression changes completely, and this is of prime importance in practice. If it is important to approach the minimum of f more closely, it is still more important not to run the risk of attaining a value of $\mu N/p$ below

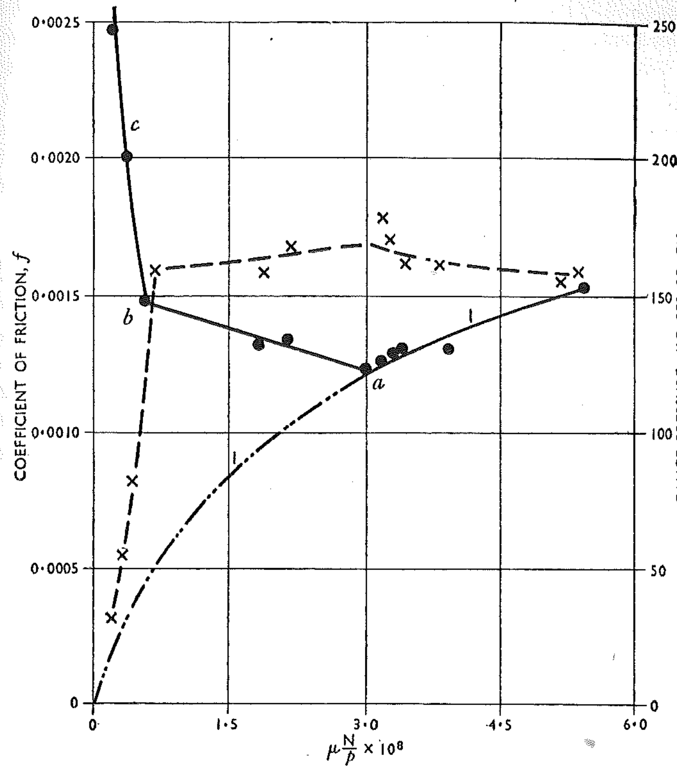

Fig. 10. Critical Value of $\mu N/p$

the critical value as μ becomes less, following on a temperature rise in the bearing. As this question is so important, the Isothermos Laboratory has attempted to ascertain where the danger point lies.

By placing a pressure gauge on the upper surface of the brass, and working with angles $2\beta=60$ deg., such as are common in practice, and clearances of the order 1/50, results were obtained which are reproduced in Fig. 10. The load was kept constant at 7,000 kg. and the speed was varied. The specific pressure p was calculated from

$$p=\frac{P}{2rl\sin\beta}$$

2β being the effective angle subtended by the brass.

The curve of f retains its parabolic form down to the abscissa 3; with lower values of f it slowly increases and only below 0·7 does the phenomenon change completely, greasy friction replacing hydrodynamic friction. This change in conditions is detected by the pressure gauge, which, up to (a) continues to register pressures of 150 to 174 kg. per sq. cm., and after (b) shows pressures falling rapidly towards zero.

This result is remarkable: the film holds under a load of 7,000 kg., while the revolutions fall to 4 per minute. At 1½ r.p.m., the coefficient of friction has not yet become twice the minimum coefficient of friction.

Conclusions

The following conclusions have been arrived at:—

(1) The coefficient of friction f can decrease below 0·002 and may become 0·0015 under certain conditions.

(2) Hydrodynamic conditions are established almost instantaneously thanks to the oil held in the clearance space between brass and shaft. The experiments of the Isothermos Laboratory confirm Goodman's work in this respect and show that at speeds above 3 to 4 r.p.m. greasy friction ceases.

(3) To avoid the danger of falling below the critical point, the oil must be selected so that, at the temperature attained by the bearing, the coefficient of absolute viscosity, expressed in kilogrammes, metres, and seconds, will lead to a value of

$$\mu N/p\times 10^8 \geqq 3$$

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