Research into the Laws of Heat Dissipation in Journal Bearings

By Ch. Hanocq. (From *Revue Universelle Des Mines*, Vol. 3, No. 7, 1947, pp. 245-258, 32 illustrations.)

Knowledge of the physical laws relating to heat dissipation from transmission bearings is as important as a knowledge of the coefficient of friction, yet no systematic research has hitherto been undertaken. The purely empirical formulæ used, at present, for the design and dimensioning of bearings do not correspond to physical reality. They are only valid within the limits of the experimental circumstances from which they were derived, and generalized deductions are apt to be misleading. The following tests lead to new, and more accurate, formulæ having a strictly physical background and meaning.

TESTS AND TEST RESULTS.

A test bearing was arranged inside an insulated chamber (Fig. 1) and loaded by a dynamometer acting,

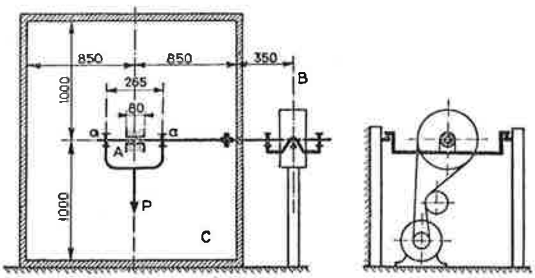


Fig. 1.

via a lever and bell crank, on two ball bearings at either side of the test bearing. The test temperature of the air in the chamber could be kept constant within $\pm$ 1·5 deg. C. and was chosen to be between 25 and 50 deg. C., with most tests being run at 28 deg. C. (82·5 deg. F.). The transmission shaft was driven from outside the chamber to avoid air disturbance inside. Conditions for heat exchange with the ambient air were thus as unfavourable as possible and a minimum value of the coefficient of heat transmission would therefore result. Direct measurement of the friction couple would lead only to an apparent coefficient of friction f_c different from the true coefficient f which was measured from the slowing down of a free running flywheel which was arranged outside the chamber on hinged bearings so that no additional reaction could reach the test bearing. One or two friction tests were taken after each test run. During these runs the air temperature was kept strictly constant and the oil bath temperature was measured at 15 minute intervals and plotted against time until, after four to five hours, the resulting curve was practically horizontal and the oil temperature was stable. When such equilibrium is reached, the heat generated by friction is equal to the heat dissipated.

Different types of test bearings were arranged in the following groups :—

Group I. Two short rigid bearings with bronze bushes and a central oilring fixed to the shaft. Diameter of shaft 40 and 60 mm. ($1\frac{9}{16}$ and $2\frac{3}{8}$ in.). See Fig. 2.

Group II. One specially designed bearing with a limited degree of self-alignment, with zamak bushes, and an oilring fixed to shaft on one side of the bush, 40 mm. dia. See Fig. 3.

Group III. One specially designed rigid bearing with a bronze bush, with an oilring fixed to shaft on one side of bush and length equal to the diameter of 40 mm. Fig. 4.

Group IV. Two very long self-aligning bearings with white metal bushes, central oilring fixed to 40 mm. dia. shaft and external swivel pivot. The two bearings differ in amount of play only. See Fig. 5.

Group V. Two very long bearings with white metal bushes, floating oilrings and internal swivel pivot. 40 and 60 mm. shaft diameters. See Fig. 6.

Group VI. Two bearings, one as group IV, one as group V, 40 mm. diameter, bearing surfaces reduced to half by counter-boring bearing ends.

Earlier tests by the same author* had led to the equation :

$$pfV = k(t_r - t_a) + k'(t_r - t_a)^4 \text{ cm.kg./cm}^2 \text{ sec.} \quad (1)$$

where p = kg. load per cm² projected bearing surface, f = the true coefficient of friction, V = rubbing speed in cm/sec., t_r = oil bath temperature, t_a = ambient air temperature, both in deg. C., and

$$k = 0{\cdot}33,\ k' = 4{\cdot}10^{-6}.$$

Eqn. (1), however, does not take account of the influence of the outer bearing surfaces, and the test results now given by Figs. 2, 3, 4, 5 and 6 show that k and k' are not constant but vary from group to group as well as within the groups. Equating, more strictly, the total heat generated to the total heat dissipated, we obtain :—

$$ApfSV = Ak_0 S'(t_r - t_a) + Ak_0' S'(t_r - t_a)^4 \quad .. \quad (2)$$

and

$$pfV = k_0 (S'/S)(t_r - t_a) + k_0' (S'/S)(t_r - t_a)^4 \quad (3)$$

where A = the projected bearing surface, S = the total bearing surface, S' = the total outer surface of bearing block except for that part of the bottom face which is covered by the bearing support. k_0 and k_0' now lead to certain constant values irrespective of type and size of bearing, and general test results relating to the value of k_0 are :—

(1) k_0 increases with increasing diameter in groups I and V ;

(2) k_0 is much smaller for group V than for group I. The long bushings of group V have less metallic contact with the housing and heat transmission is therefore smaller.

(3) k_0 is larger for group IV than for V. The fixed oilring of group IV provides a more active oil circulation and the better metallic connexion between bush and housing favours heat transmission not only by oil convection but also by metallic conduction.

(4) Bearings in groups II and III show very high k_0 values because their bushes are short and metallic connexion with housings is ample.

(5) Within expected experimental errors, the shortened bearings of group VI show no improvement over their respective originals of groups IV and V. This important result indicates that k_0 does not change with bearing surface S and that the influence of the outside surface S' is decisive.

INFLUENCE OF BEARING METAL, SELF-ALIGNMENT, AND FILM LUBRICATION.

Fig. 7 shows the measured friction values from which the diagram Fig. 2 was drawn. Points 1, 2, 6 and 9 of Figs. 7 and 2 stand out and are clearly due to imperfect lubrication and metallic contact. All the other points are grouped characteristically about the theoretical friction curve, which is drawn in Fig. 7. In fact, if the straying values of these points are replaced by the corresponding theoretical friction values the curve of Fig. 2 still remains correct in shape but the stray field of the experimental points is narrowed considerably.

* *Rev. Univ. des Mines*, August 1, Sept. 1 and 15, 1929.

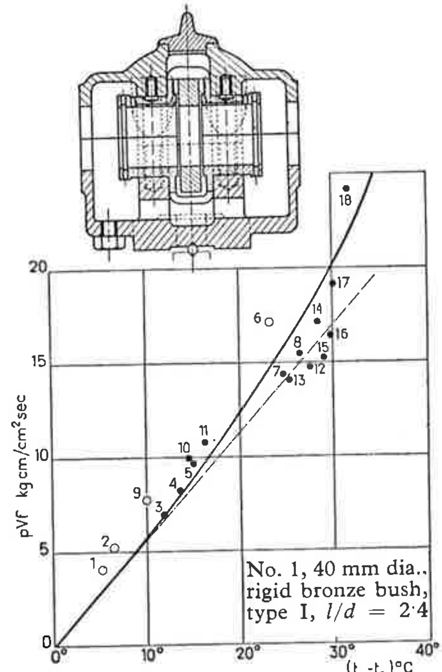
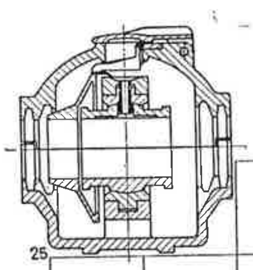


Fig. 2.

These findings were confirmed by further tests when the bronze bush of bearing Fig. 2 was replaced by a zamak bush of identical size. Fig. 8 summarizes the results of the tests. The values for imperfect lubrication are separated in curves B and Z, and the values for hydro-dynamic film lubrication, with theoretically correct friction coefficients, are given by curves B_n and Z_n for bronze and zamak bushes respectively. Heat dissipation is thus shown to be influenced by the way heat is generated, whether



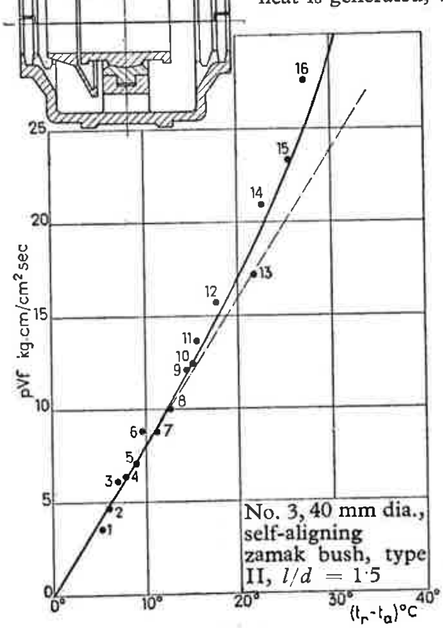


Fig. 3.

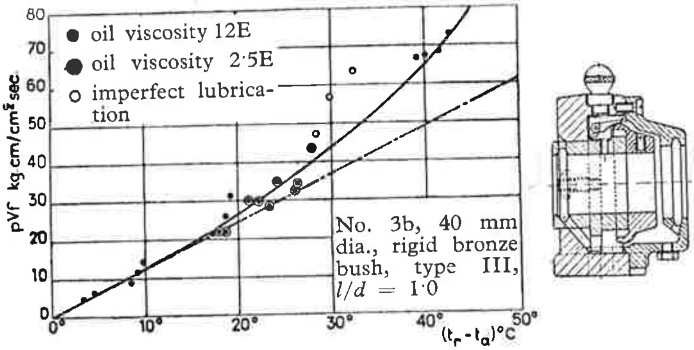


Fig. 4.

by perfect film or imperfect lubrication (compare curve B with B_n, and Z with Z_n). Curve Z_n, running constantly above B_n, indicates the higher conductivity of zamak. Moreover, the curves for zamak bushes show a wider stray field for imperfect lubrication, i.e., imperfect lubrication and metallic contact and abrasion is more serious in the case of zamak bushes than for bronze bearings. The tests with self-aligning bearings (Figs. 3, 5 and 6) show hardly any scatter of results. Measured values of f also coincide well with theoretical values whether film lubrication is perfect or not. Imperfect lubrication is therefore much less detrimental if the bearing is self-aligning. Finally, bearings of type I with bronze bushings were run with special low-viscosity lubricating oil so that no perfect film could be formed and large metal contact was maintained. Curve Y, Fig. 8, gives the results. Curves Y and B, taken with imperfect lubrication, show a greater heat dissipation than does the corresponding curve B_n representing tests with perfect film lubrication.

When equilibrium is reached during tests, the amount of heat dissipated and imparted to the ambient air must be proportional to the difference between the temperature t_e at the outside surface of the bearing block and the ambient air temperature t_a. During the tests, the oil bath temperature t_r only was measured and allowance must be made for the difference between t_r and t_e. If e.g. for test points on curve Y or B, the

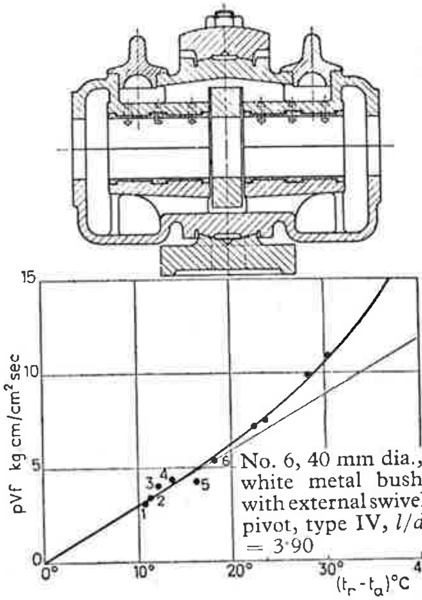


Fig. 5.

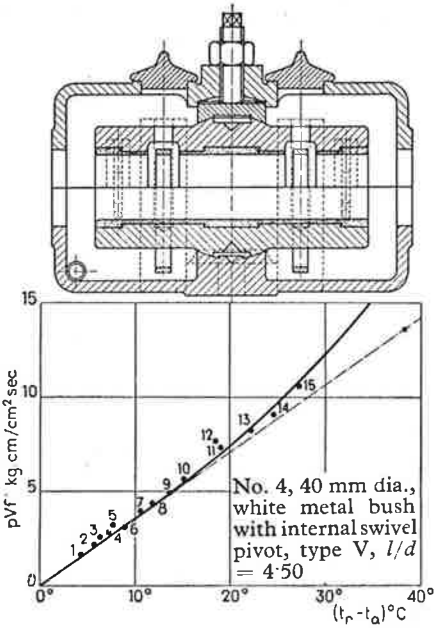


Fig. 6.

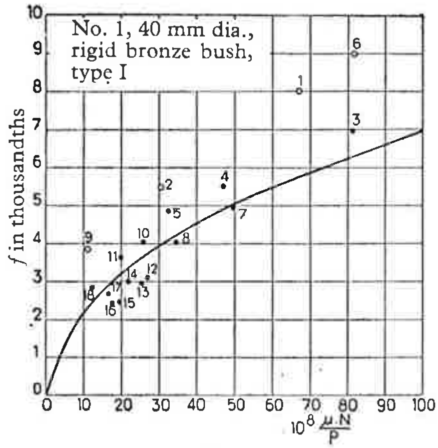


Fig. 7.

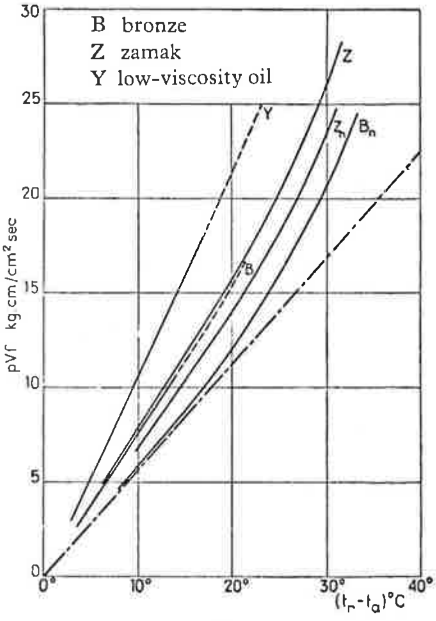


Fig. 8.

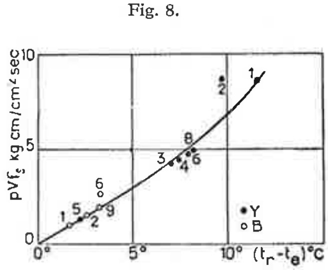


Fig. 9

differences of the abscissæ, $B - B_n$ or $Y - B_n$, are plotted as functions of the corresponding ordinate differences for the same points, a single curve is found to show, for Y as well as for B, the temperature difference $t_r - t_e$ as a function of the additional amount of heat, pVf_s, generated by metallic friction and conducted by direct metallic contact from the bush to the outside wall of the bearing block (Fig. 9). Similarly, the difference between the abscissæ, for equal ordinates, of curves B_n and Z_n (Fig. 8) indicates that the oil temperature t_r of the zamak bearing is constantly 1·5 to 2·5 deg. C. below that of the bronze bearing, which is evidence of the better conductivity of the zamak bearing even under conditions of perfect film lubrication.

DIMENSIONAL INFLUENCES.

The heat generated at the oil film interface is at first equally distributed to the bearing and shaft surfaces ; it then flows through the bearing, the oil, and the shaft cross-sections to the outside, and is dissipated into the ambient air. However, heat only flows freely to the dissipating shaft surface S'' outside the bearing, if, approximately, the shaft cross-section, through which heat must flow, is equal to the heat generating surface of the shaft, i.e., if $r\,d\alpha\,r/2 = r\,d\alpha\,l/2$ or $n = l/d = 1/2$, where $r\,d\alpha$ is a small segment of the shaft circumference, l the bearing length, and d the shaft diameter. If d is small compared with l, i.e., if n is much larger than 1/2, heat cannot flow so easily through the shaft as it can through bearing and oilbath, and the dissipating shaft surface S'' plays only a minor role as compared with the outside S' of the bearing. The exact quantitative law cannot be established because sufficient experimental data is not available, but results from the author's tests on plain transmission bearings and electromotor bearings equally indicate an equation of the form

$$y = \frac{S''}{S'} = \frac{1}{n^2}\left(\frac{d}{d_0}\right)^2 \quad \ldots \quad \ldots \quad (4)$$

For every type of bearing with a length $l = nd$, there is a certain limiting bearing diameter d_0 for which y is unity, and y remains unity even if the bearing diameter is further increased. It is more correct, however, to use the heat protecting length L of the bearing block instead of the bearing length l proper and, for bearings of types II and III, to allow for the heat conducting influence of the fixed oilring at the bearing side. With these corrections :

$$y = \frac{1}{{n_1}^2}\left(\frac{d}{d_1}\right)^2 \leqslant 1 \quad \ldots \quad \ldots \quad (5)$$

Experiments discussed here conform to this law if d_1 is chosen as 2·7 cm.

The coefficient of convection k_0 of eqn. (3) can now be written ;

$$k_0 = \frac{K_\infty}{2} \times \frac{S' + S''}{S'} = \frac{k_\infty}{2}(1 + y) =$$
$$= \frac{k_\infty}{2}\left[1 + \frac{1}{n_1^2}\left(\frac{d}{d_1}\right)^2\right] \quad \ldots (6)$$

where k_∞ is the limiting value of k_0 for the case $S' = S''$, or $y = 1$, or $d = d_1$, i.e., for the case when heat is dissipated in equal quantities by the shaft and bearing. This condition is practically obtained by bearing type III, and thus leads to $k_0 = k_\infty = 3{\cdot}60$ as measured. For the hypothetical case $d = 0 = y$, i.e., the shaft takes no part whatever in heat dissipation, and $k_0 = k_\infty/2 = 1{\cdot}8$ m.kg/m²sec. ; a value which corresponds closely to the generally quoted value of the coefficient of natural convection, 15 cal/m² hr.

Eqn. (6) gives the theoretical values which would be obtained if, during experiments, the temperature t_e of the outside wall was equal to the oil temperature t_r. This can be assumed to be practically correct for bearing type III. For the other bearings, a correction factor

$$K = \frac{t_e - t_a}{t_r - t_a} \quad \ldots \quad \ldots (7)$$

must be introduced to bring theory and experiment into agreement. In round figures, $K = 1$ for type III, 0·9 for type I, 0·85 for types II and IV, 0·8 for type V.

Similarly, the radiation coefficient should be corrected to

$$k_0' = k_v' \cdot K^4 \quad \ldots \quad \ldots (8)$$

with $k_v' = 24{\cdot}1 \times 10^{-6}$ as obtained from tests with bearing type III. No correction is made for heat dissipation by radiation from the shaft because the influence of the shaft speed would introduce additional errors. Calculated values of heat dissipated by radiation can therefore be expected to be slightly higher than the theoretical values. Tests, however, agree fairly well with calculations, and radiation plays a very minor role compared with that of convection.

The final formulæ for practical cases can now be derived :

(a) For a non-ventilated end plummer block (Figs. 3 and 4) :

$$pfV = 1{\cdot}80\,\frac{S'}{S}K\left[1 + \frac{1}{2}\,\frac{1}{n_1^2}\left(\frac{d}{d_1}\right)^2\right](t_r - t_a) +$$
$$+ 24{\cdot}1 \times 10^{-6}\,\frac{S'}{S}K^4(t_r - t_a)^4 \quad \ldots (9)$$

the factor 1/2 signifying that the shaft dissipates heat to one side only.

(b) For an end plummer block ventilated from one side only :

$$pfV = 1{\cdot}80\,\frac{S'}{S}K\left[\left(1 + \frac{1}{2}\sqrt{v}\right) + \frac{1}{2}\,\frac{1}{n_1^2}\left(\frac{d}{d_1}\right)^2\right]$$
$$(t_r - t_a) + 24{\cdot}1 \times 10^{-6}\,\frac{S'}{S}K^4(t_r - t_a)^4 \quad \ldots (10)$$

(c) For an intermediate plummer block (Figs. 2, 5 and 6), ventilated from both sides :

$$pfV = 1{\cdot}80\,\frac{S'}{S}K\left[(1 + \sqrt{v}) + \frac{1}{n_1^2}\left(\frac{d}{d_1}\right)^2\right]$$
$$(t_r - t_a) + 24{\cdot}1 \times 10^{-6}\,\frac{S'}{S}K^4(t_r - t_a)^4 \ldots (11)$$

where v = ventilation air speed in m/sec., d in cm., and $d_1 = 2{\cdot}7$.

Tests with electromotor bearings made by other workers have, in all cases, established a remarkable coincidence of observed values with limit values calculated from the above formulæ.

CONCLUSIONS.

From test results by Lasche, Falz, an authority on these problems, had established an empirical formula for the heat dissipated :

$$Q = a(t_r - t_a)^{1{\cdot}3} \text{ cal/m}^2\text{, hr.} \quad \ldots (12)$$

where $a = 2{\cdot}83$ for a transmission bearing in still air without forced oil circulation. According to the tests as described, however, the curves for pfV as a function of $(t_r - t_a)$ are definitely straight lines near the origin, and there is no justification for the coefficient 1·3. Neither is it justified from a physical standpoint since convection currents in the ambient air cannot noticeably influence heat dissipation from a bearing of relatively very small vertical extension. The factor 2·83 ($= a$) is much too small since Lasche had erroneously determined the "apparent" friction factor f_c from direct friction couple measurements. This could be proved by the author from a repetition of Lasche's experiments. The correct formula, according to the author's tests, should be $Q = 15{\cdot}2\,(t_r - t_a) \times 2$. This agrees satisfactorily with observed values, and the factor $k_0 = 15{\cdot}2$ for strictly natural convection corresponds to the generally known physical convection coefficient.

A comparison of the Stefan-Boltzmann law with the approximate formula for radiation adopted by the author ;

$$Q_r = 3600\,A\,k_0' \times 10^{-6}(T_r - T_a)^4 = K(T_r^4 - T_a^4),$$

results, when $t_r = 58$ deg. C., $t_a = 28$ deg. C., and $k_0' = 24{\cdot}1$, in a value $K = 4{\cdot}4$ in Stefan-Boltzmann's law. This is only slightly higher than would be expected for a rough cast surface. ($K = 4{\cdot}95$ for a black body). The term $(T_r - T_a)^4$ has been adopted by the author because, with mounting temperature, the heat radiation of the shaft and pedestal is no longer negligible and, therefore, the total radiated heat increases more quickly than the theoretical difference $T_r^4 - T_a^4$ would suggest. Moreover, the chosen approximate term allows for quicker evaluation.

A comparison of different test bearings of 40 mm. diameter under identical conditions of equilibrium (1000 r.p.m., 500 kg. load, 28 deg. C. ambient temperature) shows the following oil temperatures : Type I (Fig. 2) 56 deg. C., type IV (Fig. 5) 56·5 deg. C., type V (Fig. 6) 52 deg. C., type III (Fig. 4) 48 deg. C. Type III thus gives a lower oil temperature, a better safety factor, and an energy absorption reduced to less than 50 per cent. The generally accepted empirical notion that the value of the product pV must be limited, i.e., that with higher speeds V lower pressures p must be chosen, is very misleading. The tests show that *a short self-aligning bearing gives the best results.*

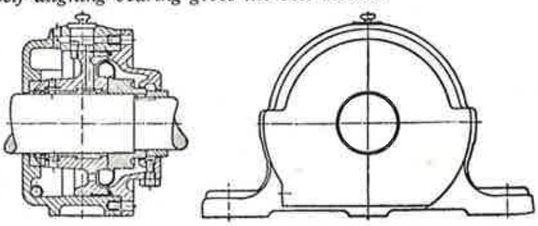


Fig. 10.

The conclusions from these tests led to the development of a bearing as illustrated in Fig. 10. The bearing bush is short, and self-aligning, with abundant lubrication, but without oil splash, from a dished extension of the thrust collar acting as an oil pump. The service temperature under conditions as before is 51 deg. C., i.e., not higher than for a similar ball bearing, the type factor is $K = 0{\cdot}9$. The shape is simple and the bearing block can be equipped equally well with plain bushes as with ball races.

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