EXPERIMENTAL STUDY OF BALL AND ROLLER BEARINGS

By Professor Ch. Hanocq*

Tests were undertaken by the Société Générale Isothermos, of Paris, in order to find the coefficient of friction for roller bearings at the same loads and speeds as those usual in practice for smooth bearings.

It has been found that, provided lubrication was abundant enough, a coefficient of friction could be obtained under hydrodynamic conditions with smooth bearings which was of the order of 0·002 and even 0·0015. The problem was to find the corresponding value for the same journal under the same load and speed of rotation when a smooth bearing was replaced by a roller bearing. The roller bearing was of the common railway type, and had two rings placed side by side; its dimensions were as follows:

- Journal diameter: 120 mm.
- External diameter: 260 mm.
- Diameter of the rollers in their plane of symmetry: 30 mm.
- Width of the rollers: 36 mm.
- Number of rollers: 64 in 4 rows.

Two sets of experiments were carried out: one with a diametral clearance on the bearing of 0·06 mm. (Group I) and the other with a diametral clearance of 0·11 mm. (Group II).

Table 1 (p. 75) gives the results obtained both with the roller bearing and an “Athermos” bearing. The latter is lubricated with oil throwers and the brass is of the type in which the oil is distributed by multiple jets, this type having been used in tests on bearings with partial brasses, the clearance being 3 mm. A helical domestic fan was used to obtain ventilation, the speed of circulation being some 16 kilometres per hour.

It was found that a clearance of 0·06 mm. was insufficient for proper working at the temperatures reached during the work, so a clearance of 0·11 mm. was employed. Lower values were then obtained and the temperature increased but slowly with increasing speed, while the coefficient of friction fell to below 0·002. For this reason, only the figures in Group II (Table 1, p. 75) should be considered, though the importance of the clearance in respect of the coefficient of friction is realized.

With a load of 12,000 kg. and a speed of 800 r.p.m., the temperature of the smooth Athermos brass remained at 79·5 deg. C., whereas with

* University of Liège.
### Table 1

<table>
<thead>
<tr>
<th>P</th>
<th>N</th>
<th>Rollers with fan</th>
<th>Athermos bearing</th>
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<tr>
<td></td>
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<td>Air temp., deg. C.</td>
<td>Bearing temp., deg. C.</td>
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<tr>
<td></td>
<td></td>
<td>6,000</td>
<td>530</td>
</tr>
<tr>
<td></td>
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<tr>
<td></td>
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<td>530</td>
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<tr>
<td></td>
<td></td>
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<td>530</td>
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<tr>
<td></td>
<td></td>
<td>7,000</td>
<td>290</td>
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<td>12,000</td>
<td>700</td>
</tr>
<tr>
<td></td>
<td></td>
<td>12,000</td>
<td>800</td>
</tr>
</tbody>
</table>

Group I. Clearance, 0.06 mm.

Group II. Clearance, 0.11 mm.

Grease, deg. C. | Bearing, deg. C. | f = f<sub>1</sub> &middot; 10<sup>6</sup> | Oil, deg. C. | Brass, deg. C. | f = f<sub>1</sub> &middot; 10<sup>6</sup> |
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<td>44-5</td>
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<td>56-0</td>
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* Rough-finished bearing with fairly hard whitemetal.

![Fig. 1. Method of Testing](image)

The values given in terms of N (revolutions per minute) are plotted in Fig. 2, and strongly suggest a linear expression for N. That this should be so can be proved, for it can be assumed that the power absorbed by a ball bearing is made up of the following terms:

\[
W = 4\pi fM = 2p + \frac{4\pi fM}{2p} + \frac{4\pi fM}{2p} + \cdots
\]

(1)

(2)

(3)

The coefficients obtained by Professor Dumas of Lassane. Under these conditions, we can write for the whole.

After starting up the combination of flywheel, shaft and bearing, the deceleration curve was obtained and the corresponding torque deduced.

The results obtained with results obtained in the writer's laboratory, using ordinary ball bearings were determined with a shaft 40 mm. in diameter.
(2) The loss due to the sliding friction of the balls in their race, the peripheral speed of the balls being strictly equal to the tangential speed of the race only in the plane of symmetry, whence

\[ W_e = 2k_2 k o \Sigma \omega r \]  

(2)

\( k_2 \) being inserted because the sliding speed is only a very small fraction of the peripheral speed \( \omega r \).

(3) The loss due to the slip of the layer of oil along the balls, the oil flowing back as the balls advance, and also to oil scraped off by the race itself. Applying Newton's law, the force applied to the periphery of the balls can be written as

\[ R = \mu k' \rho^2 \times \frac{k' \omega r}{e} 

where \( k' \rho^2 \) represents the surface of each ball affected by the removal of oil in the race and the reflux of oil on the race, \( k' \omega r \) the speed of slip, \( e \) the thickness of the oil film, and \( n \) the number of balls. Movement requires a force applied to the periphery of the interior ball race equal to \( 2R \) and therefore the equation for this loss can be written as

\[ \omega e = 2\mu (k' \rho^2) \frac{k' \omega r}{e} \times \omega r \]  

(3)

where \( \omega r = k' \rho r \).

(4) The loss absorbed initially under no load owing to the positional tension of the balls and imperfections in manufacture, owing to which the torque is not strictly zero under no load

\[ W_e = C_0 \varepsilon \]  

(4)

Summing these four terms and seeing that the total power absorbed can be expressed in terms of the coefficient of friction as \( P_{fr} \), we have

\[ f = 2k_1 f_{fr} p^3 + 2k_2 f_{fr} + k' k' n \frac{\rho^2}{e} \frac{\mu V}{r} \frac{P}{P} + C_1 \]  

(5)

where \( V \) represents the peripheral speed of the shaft, \( k_1 \) being distinguished from \( k_1 \) in order to take into account the bracketed term in equation (1).

From equation (5) it is seen that \( f \) should appear as a linear function of \( N \) in the experiments carried out under constant conditions of load and temperature. If the curves of Fig. 2 are extended to the ordinate axis, the value of \( P_{fr} \) can be found for \( N = 0 \) and the curves \( P_{fr} = C_0 \) (Fig. 3) can be drawn as a function of \( P \). From this curve

\[ C_0 = 2.3 \text{ kg-mm.} \quad 2k_1 f_{fr} = 0.0018 \quad 2k_2 f_{fr} = 0.0013. \]

To render equation (5) applicable to all geometrically similar bearings the specific pressure must be introduced in the first term instead of \( P \); in other words the first term must be multiplied by the ratio of the squares of the radii of the balls to the \( \frac{3}{2} \) power, or,

\[ \frac{(0.01)^{\frac{3}{2}}}{2p} \]

where 0.01 is the diameter, in metres, of the balls of the bearing actually tested, and 2p is the diameter of the bearing under consideration.

As regards the expression \( C_1 \), which depends essentially on the mechanical finish, it can be said that, provided the quality is the same, it certainly increases with \( 2r \), so that for the equation to be generally applicable, the expression must be multiplied by \( 2r / 0.04 \).

The equation can now be written

\[ f = 9 \times 10^{-5} P_2 (0.01 / 2p)^{\frac{3}{2}} + 4 \times 10^{-5} + 2.88 \left( \frac{1}{P} + 965 \frac{\mu V}{P} \right) \frac{1}{2} \frac{2r}{2r} \]

(6)

Let this equation be used to calculate the coefficient of friction of a ball bearing geometrically similar to that used in these experiments, but suitable for a shaft diameter \( 2r = 120 \text{ mm.} \), i.e. three times the size of the experimental ball bearing (40 mm.). This bearing had practically the same dimensions as the roller bearings tested by the Isotermos Laboratory and had 30 mm. rollers arranged in two rows of 16 rollers each, as in the 40 mm. bearing. The load \( P / 2r \) corresponding to the
total stress of 6,000 kg. must be taken as equal to 3,000/0.120 = 25,000 kg.
since the bearing is double.

To calculate the value of the last term, a value must be assumed for
the viscosity of an oil such as was used, the viscosity at 60 deg. C. being
probably 0.005. With this value and N = 800 r.p.m., then
\[ f = 10^{-3}(60.7 + 5.66 + 11.5 + 95) = 1.73 \times 10^{-3} \]
which corresponds closely to the observed value (1.79 \times 10^{-3}).

It would be of interest to work out the curve of \( \mu \) in terms of the
bearing temperature (which was close to that of the grease in contact
with the ball race), as the experimental values could then be made to
agree with the calculated values. Thus the Group II experiments
under a load of 6,000 kg. yield the curve of Fig. 4, which certainly
resembles the viscosity-temperature curve. (The circles refer to
experiments with a load of 7,000 kg.) It will be seen that the last
experiment carried out during running-in under that load gave a very
low figure comparable to that obtained with a load of 6,000 kg.

There is thus sufficient ground for affirming that the proposed
equation can be applied fairly widely as well as fairly strictly. Ob-
viously, it must be remembered that a roller bearing is mechanically
very complex and that the different coefficients in the equation may
be modified when one type is replaced by another, according to the
mechanical finish and even to the temperature when the clearances
are not correctly adjusted for a particular working temperature.
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