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

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TABLE 1

	P	N	Rollers with fan			Athermos bearing			
			Air temp., deg. C.	Bearing temp., deg. C.	$f=\frac{1}{1000}$	Air temp., deg. C.	Brass temp., deg. C.	$f=\frac{1}{1000}$	
Group I. Clearance, 0·06 mm.	6,000	530	21	55·3	3·12	15	57	1·69	No fan used
	8,000	530	22·5	60·5	2·97	20	58·5	1·67	
	10,000	530	21	63·0	2·66	20·5	64·5	1·65	
	12,000	530	22	66·0	2·53	21·5	69·0	1·63	
	7,000	290	20·5	38·0	1·80	26·0	48·5	1·35	Observations after 4 hours' run
	7,000	580	20·0	60·5	2·37	—	—	—	
	7,000	700	22·0	83·5	3·82	24·0	66·0	1·23	
	12,000	700	20·0	95·0	4·55	—	—	—	Bearings not fitted with felts
	12,000	800	—	—	—	24·5	79·5	1·18	
			Grease, deg. C.	Bearing, deg. C.	$f=\frac{1}{1000}$	Oil, deg. C.	Brass, deg. C.	$f=\frac{1}{1000}$	
Group II. Clearance, 0·11 mm.	6,000	635	40·0	52·0	1·80	47·0	66·5	1·50*	Air temperature for the Group II tests was 22 deg. C.
	6,000	692	41·5	54·0	1·87	49·0	69·5	1·27	
	6,000	750	44·5	58·5	1·84	49·0	73·0	1·29	
	6,000	850	47·5	60·0	1·79	48·0	76·0	1·32	
	7,000	635	51·5	52·5	1·64	48·0	67·5	1·44	
	7,000	692	52·5	55·5	1·69	49·0	70·5	1·31	
	7,000	750	57·5	61·0	1·79	50·0	76·0	1·33	
	7,000	800	61·5	64·5	1·91	50·5	78·5	1·29	
	7,000	840	57·5	59·5	1·52	50·0	78·0	1·21	

* Rough-finished bearing with fairly hard whitemetal.

the roller bearing under the same load at the same speed and a clearance of 0·06 mm., the temperature rose to 92 deg. C. after 140 minutes and the test had to be stopped as the temperature continued to rise.

These apparently unexpected results are by no means in contradiction with results obtained in the writer's laboratory, using other methods.

Using the deceleration method (Fig. 1) the coefficient of friction of ordinary ball bearings was determined with a shaft 40 mm. in diameter. After starting up the combination of flywheel, shaft and bearing, the deceleration curve was obtained and the corresponding torque deduced.

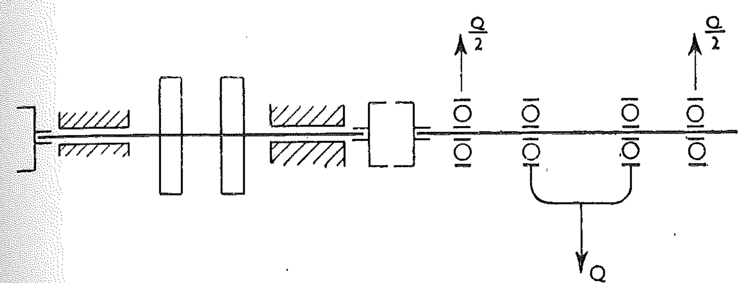

Fig. 1. Method of Testing

Then the torque M corresponding to the four ball-bearings could be obtained by difference, whence

$$f=\frac{M}{4\left(\frac{1}{2}Q\right)r}$$

The results, giving f 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 W absorbed by a ball bearing is made up of the following terms:—

(1) The loss by bearing friction can be taken as proportional to f_r, the coefficient of bearing friction, and to the load $\Sigma P'^{\frac{4}{3}}$, as was shown by Professor Dumas of Lausanne. Under these conditions we can write, for the whole

$$W_r=2f_r k_1 wr \Sigma P'^{\frac{4}{3}}=2f_r k_1 wr P^{\frac{4}{3}}\left[\left(\frac{P'}{P}\right)^{\frac{4}{3}}+\left(\frac{P''}{P}\right)^{\frac{4}{3}}+\ldots\right] \quad . \quad (1)$$

the coefficient k_1 representing the opposing force considered as applied to the periphery of the interior ball race, whereas wr represents the tangential speed of the shaft, all in relation to the radius r of the shaft.

(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

$$W_g = 2f_g k_2 wr \Sigma P' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

k_2 being inserted because the sliding speed is only a very small fraction of the peripheral speed wr.

(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

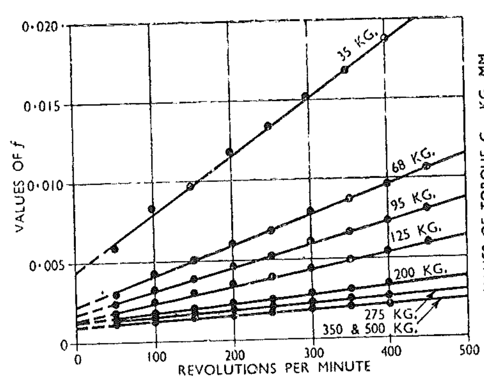

Fig. 2. The Value of f in Terms of N

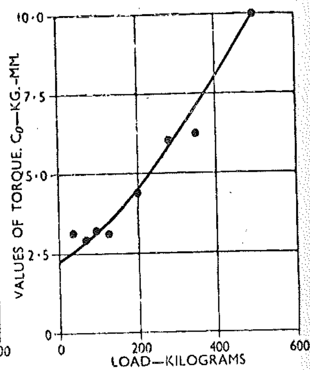

Fig. 3. Torque under Different Loads

race itself. Applying Newton's law, the force applied to the periphery of the balls can be written as

$$R = \mu k' \rho^2 \times \frac{k'' wr}{e} \times n$$

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, $k''wr$ the speed of slip, e the thickness of the oil film, and n the number of balls. Movement requirês 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

$$w_\mu = 2\mu(k' n \rho^2) \frac{k'' wr}{e} \times wr_i \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where $r_i = k''r$.

(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

$$W_i = C_i w \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Summing these four terms and seeing that the total power absorbed can be expressed in terms of the coefficient of friction as $Pfwr$, we have

$$f = 2k_1' f_r P^{\frac{1}{3}} + 2k_2 f_g + k' k''^2 n \frac{\rho}{e} \cdot \frac{\rho}{r} \cdot \frac{\mu V}{\frac{P}{2r}} + \frac{C_i}{Pr} \quad . \quad . \quad . \quad (5)$$

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 Pfr can be found for N=0 and the curves $Pfr = C_o$ (Fig. 3) can be drawn as a function of P. From this curve

$C_i = 2{\cdot}3$ kg.-mm. $\qquad 2k'_1 f_r r = 0{\cdot}0018 \qquad 2k_2 f_g r = 0{\cdot}00113.$

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 $\frac{2}{3}$ power, or,

$$\frac{(0{\cdot}01)^{\frac{2}{3}}}{2\rho}$$

where 0·01 is the diameter, in metres, of the balls of the bearing actually tested, and 2ρ is the diameter of the bearing under consideration.

As regards the expression C_i, 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 $2r/0{\cdot}04$.

The equation can now be written

$$f = 9 \times 10^{-5} P^{\frac{1}{3}} (0{\cdot}01/2\rho)^{\frac{2}{3}} + 5{\cdot}66 \times 10^{-5} + 2{\cdot}88 \frac{1}{\frac{P}{2r}} + 965 \frac{\mu V}{\frac{P}{2r}} \quad . \quad . \quad (6)$$

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 $2r = 120$ 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 Isothermos 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

$$f=10^{-5}(60{\cdot}7+5{\cdot}66+11{\cdot}5+95)=1{\cdot}73\times10^{-3}$$

which corresponds closely to the observed value ($1{\cdot}79\times10^{-3}$).

It would be of interest to work out the curve of μ in terms of the bearing temperature (which was close to that of the grease in contact

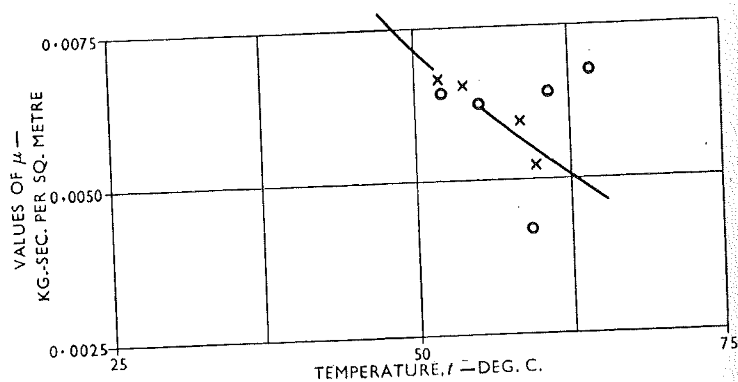

Fig. 4

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. Obviously, 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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