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EXAMINATION OF THE FRICTION MOMENT IN ROLLER BEARINGS OF THE CLOCK TYPE WORKING WITHOUT LUBRICATION

by

A. Wierciak

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By: A. Wierciak

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ABSTRACT

(U) Theoretical considerations have been made on the support of a journal by a bushing in a clock bearing. Described below are the results of studying an apparatus for measuring friction moment and the conclusions for designers of small mechanisms.

EXAMINATION OF THE FRICTION MOMENT IN ROLLER BEARINGS OF THE CLOCK TYPE WORKING WITHOUT LUBRICATION

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Theoretical considerations have been made on the support of a journal by a bushing in a clock bearing. Described below are the results of studying an apparatus for measuring friction moment and the conclusions for designers of small mechanisms.

In the design of small mechanisms, clock roller bearings are generally used. An installation of this type is shown in Fig. 1. The bearing assembly 1 (a gear shaft with gear wheel mounted on it) has roller journals 2, which revolve in bearing seats 3 machined in plates 4, which constitute simultaneously the skeleton of the mechanism. Because of the technology of the units and the assembly of these bearing mechanisms, the use of a large radial clearance is indispensible. Use of class 9 fittings, e.g., e9/H9 or d9/H9, is recommended for them.

In connection with the above, in a new bearing, during initial support, the journal comes in contact with the surface of the bushing almost linearly (the girdling angle is very small).

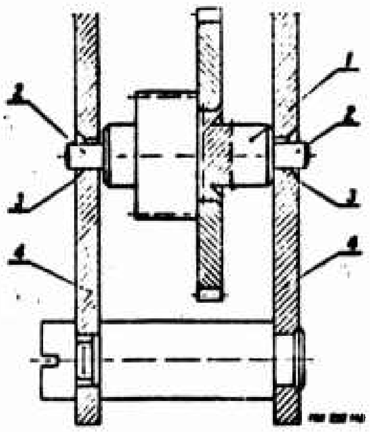

Fig. 1. Example of mounting a shaft in roller bearings.

For the purpose of investigating the effect of the radial clearance in a new bearing on the size of the contact surface of the journal with the bushing, we made a diagram of the width of the contact surface of the journal with the bushing $b_w = b_0/D$ as a function of the relative radial clearance value $L_w = L/D$. The diagram was made on the basis of the Hertz pattern

$$b_0 = 2\sqrt{\frac{p\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)}{\pi L\left(\frac{1}{r}-\frac{1}{R}\right)}}$$

where b_0 is the width of the contact band of the journal with the bushing; $D = 2R$ is the diameter of the bushing opening; L is the length of the bearing; $d = 2r$ is the journal diameter. It was further assumed that the journal was made of steel and the bushing of brass.

Bearing in mind that the length of the actual bearing has a finite value and the difference in the contact surface radii is small, the diagram should be treated as only illustrative of that dependence for actual bearings.

The diagram was made for the two values of the calculated unit pressure, $p = 5$ kg/cm^2 (≈ 0.5 MN/m^2) and 20 kg/mm^2 (≈ 2 MN/m^2), and presented in Fig. 2.

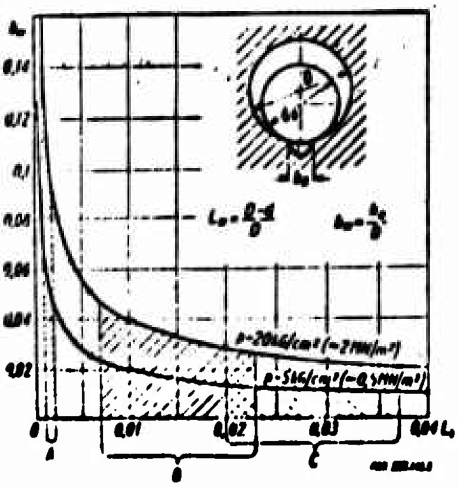

Fig. 2. Graph of relative width b_w of contact of the journal with the bushing, as a function of relative radial clearance L_w in the bearing with an individual calculation pressure $p = 5\ kg/cm^2$ ($\approx$0.5 MN/m^2 and 20 kg/cm^2 ($\approx$2 MN/m^2: A – range for ⌀ 30 f7/H7, B – range for ⌀ 3 d9/H9, C – range for ⌀ 3 c9/H9, d_0 – journal diameter, D – diameter of bushing opening, b_0 – width of contact surface calculated according to the Hertz pattern.

On the diagram the terminal values of the radial clearance for fittings used in clock bearings are designated, namely ⌀ 3e9/H9 and ⌀ 3d9/H9, as the example of the fitting used in machine bearings, ⌀ 30f7/H7.

From the diagram it is evident that the initial fitting angle ϕ in clock bearings, whose value is simply proportional to b_w, is several times smaller than the fitting angle in machine bearings. During the running of the bearing the bushing yields to wear and the fitting angle increases. In order that the contact surfaces of journal and bushing can be covered over a fitting angle of 180°, the bushing must wear out at the value Z (Fig. 3).

Figure 4 represents a relationship diagram of the relative wear $Z_w = Z/D$, corresponding to an increase in the fitting angle up to 180° as a function of the radial clearance in the bearing. The terminal values Z_w corresponding to the above given fittings are designated in the diagram.

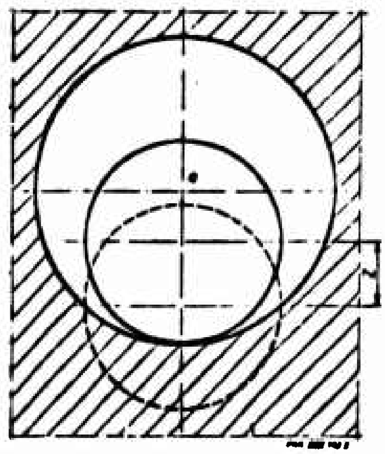

Fig. 3. Diagram explaining the wear value Z corresponding to an increase in the fitting angle up to 180°.

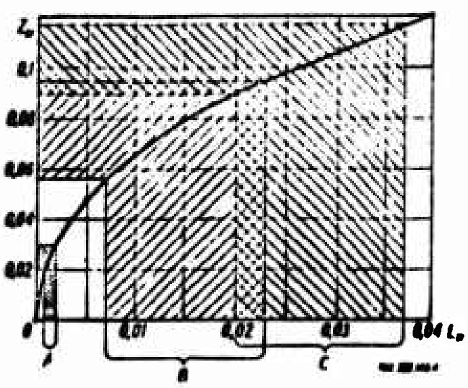

Fig. 4. Dependence of relative wear of bushing Z_w, corresponding to fitting angle of 180°, as a function of the relative radial clearance L_w in the bearing: A – range for ø 30 f7/H7, B – range for ø 3 d9/H9, C – range for ø c9/H9.

As is evident from the diagram, in order to obtain a fitting angle of exactly 180° in a clock bearing, the value Z_w may amount to 12% of the diameter of a nominal bearing in a borderline case.

Such great wear of the bushing is not allowable in most instances from a functional viewpoint. Therefore, in clock bearings the fitting angle does not reach a value of 180°.

The friction moment in a sheeve bearing is calculated from the formula

$$M_t = Q\mu_{ob}\frac{d}{2}$$

where Q is the bearing load; μ_{ob} is the calculated friction coefficient; d is the diameter of the journal. At the start of bearing operation, when contact with the bushing is approximately linear, μ_{ob} is equal to the friction coefficient μ.

In technical literature we find the relationships between the value of the friction coefficient and the value of the calculated friction coefficient for bearings in which the fitting angle amounts to 180°. The value of the calculated friction coefficient depends in this case upon the assumptions made by individual authors (for example, the constant unit pressure on the contact surface [1], proportional wear for unit pressure [2]).

To determine the actual value of the calculated friction coefficient within the range of bearing operation, a laboratory device was developed, enabling us to measure the friction moment of a bearing as a function of the number of its revolutions. In Fig. 5 a diagram of this is shown. The assembly being tested (1) was placed in the bushings (2) fastened to the walls of the housing (3). Loading of the bearings was accomplished with a spring (4) by means of a roller bearing (5). The load value was regulated by a screw (6). Drive of assembly 1 is provided by an electric motor (7) through a multi-speed gearbox (8) and the device measuring the driving moment (9). A counter (10), driven by an output shaft of the transmission, records the number of revolutions made by the tested assembly.

In the clock setup the journal is able to roll over the surface of the bushing and occupy a position which depends upon the friction coefficient value (Fig. 6).

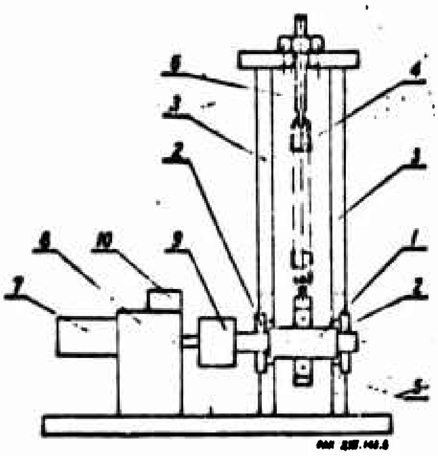

Fig. 5. Diagram of the device allowing the measurement of the friction moment of the bearing as a function of the number of revolutions: 1 – investigated assembly, 2 – bushings, 3 – walls of the housing, 4 – load spring, 5 – roller bearing, 6 – regulation screw, 7 – driving motor, 8 – multi-speed transmission box, 9 – torque meter, 10 – counter.

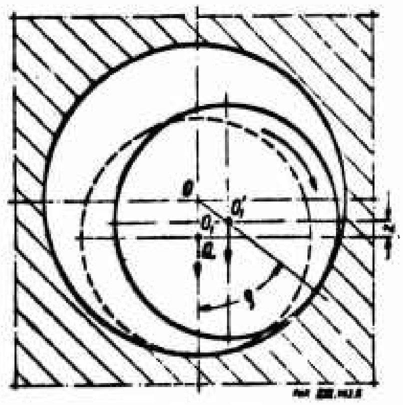

Fig. 6. Position of the journal with respect to the bushing during operation $\rho = \mathrm{tg}\ \mu$, where μ is the friction coefficient.

The device for the tests assures the freedom of the journal in occupying such a position in the bushing. This was achieved by using a load spring of flat characteristics and also by solving the problem of driving the investigated system so that it would not impose additional load on it.

Data Obtained in Tests

The tests were carried out on factory-made journals and bushings. The openings were cut out and then machined. The thickness of the layer cut out amounted to 0.02 mm. Surface brittleness of the working bushing was $\nabla 9$. The journals were ground on an automatic machine and rolled. The brittleness of the working surface of the journals amounted to $\nabla 10$. As a result of the technology of bushing openings, the traces of cracks along the treated surface of the bushing were perpendicular to the sliding direction.

For the journals and bushings we used materials most often employed for clock bearings in the production of small mechanisms. The journals were made from automatic steel A10, the bushings from spring clock brass M58.

The nominal diameter of the bearing adopted for the tests was 3 mm, the length of the bearing 1 mm.

Prior to testing, the bushings and journals were washed twice in gasoline and then twice in trichloroethylene with the aid of a small brush. The tests were conducted at sliding rates for the bearing contact surfaces of: 0.1; 1.0; 10.0 cm/min, and with an adopted bearing diameter of 3 mm this corresponds to about 0.1; 1.0; 10.0 rpm. The calculated unit pressure adopted was within the limits of 0 to 80 kg/cm^2 (≈ 8 MN/m^2).

Every test for the described rate and pressure was repeated five times, which means that ten bearings were examined in every series.

Investigation Results

The investigations showed that the friction moment in the initial working period of the bearing rises very rapidly, after which a more gradual rise in the amount sets in. Figure 7 represents a graph of the change in the average value of the friction moment as a function of the number of revolutions at a calculated pressure of $p = 40$ kg/cm^2 (≈ 4 MN/m^2) and sliding rate of $V = 0.1$ cm/min.

Along the right side of the diagram a scale of the calculated friction coefficient value is plotted. The high value of the friction coefficient here deserves attention. It is much higher than the value given in textbooks for materials subjected to testing.

The limits of maximum and minimum friction moment values appearing at the time of operation of the examined bearing are indicated in the diagram. As became evident from the tests, the value of the friction moment is variable within the limits of a single revolution, during which time the fluctuations reach 40% of the average value of this moment.

To determine the effect of the break-in process of the contact surfaces on the position of the journal with respect to the bushing during the testing, measurements of the variation of the journal position were made with respect to the axis of the bushing opening. Figure 8 represents a diagram of the position of the journal axis with respect to the bushing axis during bearing operation at a sliding rate of V = 0.1 cm/min and a calculated pressure of $p = 40$ kg/cm^2 (≈ 4 MN/m^2). Shifting of journal z is measured in the direction of load action, as indicated in Fig. 6. From the diagram in Fig. 7 it is evident that during the first 300 revolutions there is a considerable rise in the friction coefficient. This is due to the break-in of the bearing contact surfaces. The increased friction coefficient at the initial running produces a continually higher rolling-in of the journal over the surface of the bushing, until the average value of its coefficient has been established. This is confirmed by the diagram in Fig. 8, from which it is clear that during the first 300 revolutions the journal shifts in the direction opposite the load, and only then does the bearing become effective.

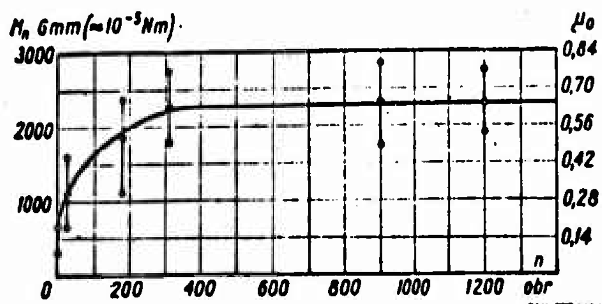

Fig. 7. Diagram of the average driving moment as a function of the number of revolutions at a calculated unit pressure of $p = 40$ kg/cm^2 (≈ 4 MN/m^2) and a sliding rate of V = 0.1 cm/min; M_n – driving moment, n – number of revolutions, μ_{ob} – calculated friction coefficient; vertical lines describe the maximum and minimum values of the moment obtained during the tests.

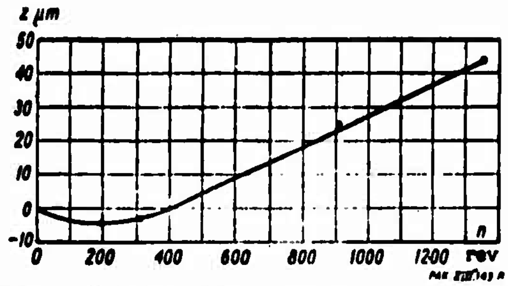

Fig. 8. Diagram of the journal position in the clock bearing bushing, as a function of the number of revolutions with calculated unit pressure of p = 40 kg/cm^2 ($\approx$4 MN/m^2) and a sliding rate of V = 0.1 cm/min; z — shifting of journal with respect to bushing in the direction of load action, n — number of revolutions.

To illustrate the effect of unit pressure on the magnitude of friction moment, the results obtained were plotted in a graph (Fig. 9).

In practice, it can be agreed that the dependence of friction moment on unit pressure is linear. The effect of sliding rate on the value of the calculated friction coefficient is significant, as is evident from Fig. 9, where, with a rise in the rate, the value of the friction coefficient rises slightly.

Figure 10 illustrates the average values of the friction coefficient obtained from examinations before and after seating the bearing contact surfaces. From the illustration it is clear that the average value of the friction coefficient rises by a factor of 3.5 after seating.

During bearing operation the journal rolls over the surface of the bushing at an angle corresponding to the angle of friction. Since the tests have shown that the value of the friction coefficient is variable within a single revolution, the journal changes its position during the running period and comes in contact with the bushing in various places. The roll-in angle changes from q_{min} to q_{max} and this means that the form of the bushing, wearing surface is not used as the roller surface of the journal radius. This has been confirmed by observations of the form of bushing wear. As to the journal's running on a variable rolling angle, the angle of journal fit cannot reach a value of 180°.

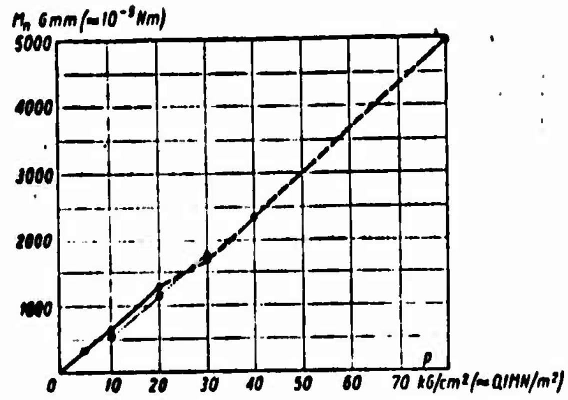

Fig. 9. Diagram of the average driving moment as a function of unit pressure at various sliding rates: ----- diagram for a sliding rate of 0.1 cm/min, ... diagram for a sliding rate of 1.0 cm/min, ——— diagram for a sliding rate of 10.0 cm/min, p — calculated unit pressure in kg/cm^2 ($\approx$0.1 MN/m^2), M_n — driving moment in G·mm ($\approx 10^5$ Nm).

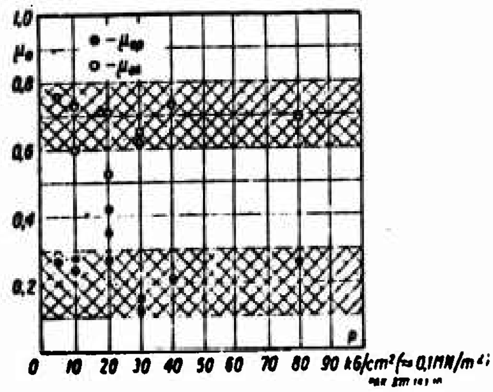

Fig. 10. Calculated value of the friction coefficient μ_0 with various values of p in kg/m^2 ($\approx$0.1 MN/cm^2), μ_{op} — average calculated friction coefficient appearing at first running of the bearing, μ_{ok} — average calculated friction coefficient appearing in the bearing after the contact surfaces have been broken in.

Conclusions

1. The running period of a clock bearing can be divided into two stages from the viewpoint of the moment value.

State I is characterized by a rapid rise in the calculated friction coefficient which sets in during the first several hundred revolutions.

During this running stage there is practically no bearing utilization, because the contact points of the journal and bushing keep changing constantly, during which time the fitting angle increases slightly and the bearing contact surfaces are broken in.

State II is characterized by a small increase in the calculated friction coefficient.

The slight rise in the calculated friction coefficient is most likely caused by a rise in the fitting angle because in this stage the process of wear and tear of the bearing has already begun.

2. The friction coefficient does not depend upon the calculated pressure value.

3. The rise in the rate within the limits adopted for the tests (0.1 to 10 cm/min) has an insignificant effect on the value of the calculated coefficient of friction and can practically be disregarded.

4. The value of the friction coefficient for materials used in the tests (steel journal, brass bushing) is considerably greater than the value given in technical textbooks.

In the design of clock bearings which work in dry state, without lubrication, a calculated value of the friction coefficient of about 0.7 must be used. Failure to use approximately this friction coefficient value in the design may cause the mechanisms, after factory installation and temporary satisfactory performance, to stop functioning during use after the break-in period for the bearing surfaces. This is especially important in mechanisms in which the driving motor has a limited moment, e.g., in mechanisms with a spring drive.

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