A METHOD FOR DETERMINING THE ROLLING FRICTION TORQUE IN A THRUST BEARING. PART II

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Abstract: The work proposes a method and an experimental set-up for finding the rolling friction torque in a thrust ball-bearing. The technique is based on the remark that for the case of thrust ball bearing, the balls are axially equidistant due to the cage and the rolling components of the friction torques from the contact points balls-race give a system of vectors equivalent to zero. The experimental device is simple: one of the rings of the bearing is kept immobile and the other one has a fly-wheel attached and can perform a rotation motion with respect to its axis. The mobile ring and the fly-wheel are set to rotation motion and the law of variation of the position of the mobile ring is found. Afterwards, the angular velocity of the fly-wheel is found and the friction torque acting upon it.

Keywords: *thrust ball bearing, friction torque, experimental test-rig*

1. Introduction

The axial section through a thrust ball bearing is presented in Fig. 1. It is considered that the bearing has a number of n balls of r radius, placed angularly equidistant, which make contacts with the rings in points positioned on a circle of R radius.

The forces and moments acting upon the balls and the races are represented in Fig. 1: $M\boldsymbol{g}$ the weight of the upper ring together to the fly-wheel, $\boldsymbol{N}$ - the normal reaction, $\boldsymbol{T}$ - the tangential reaction, figured in the sketch as vectors normal to the plane of the drawing. The spinning torques, M_s are parallel to the axis of the bearing and the rolling friction torques are normal to the same axis. It is accepted the hypothesis that the mass of the balls is negligible with respect to the mass of the rings. The equation of projection of forces on vertical direction, upon the ring:

$$n\boldsymbol{N} - Mg = 0 \qquad (1)$$

permits finding the magnitude of the normal reaction N from contact:

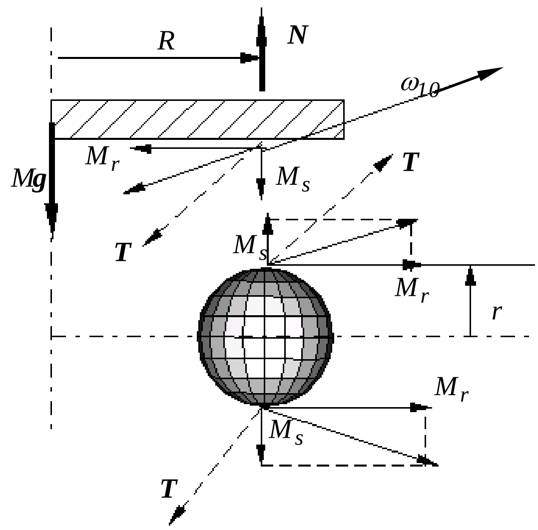

Fig.1. Forces and moments acting in a thrust ball bearing

The condition that the sum of the forces from a plane normal to the axis of rotation is zero is identically fulfilled since dues to the angularly equidistant distribution of the balls the friction forces form the n contacts are the sides of a regulate polygon with n sides and therefore:

$$\sum_{k=1}^{n} \boldsymbol{T_k} \equiv 0 \qquad (2)$$

The same remark is valid for the case of rolling friction moments:

$$\sum_{k=1}^{n} \boldsymbol{M}_{r\boldsymbol{k}} \equiv 0 \qquad (3)$$

The moment of momentum theorem [1] applied for a ball gives the following relation:

$$J_0 \varepsilon_0 \cong 0 = 2Tr - 2M_r \qquad (4)$$

The moment of momentum theorem applied for the ring when other external torques are zero, has the form:

$$J\varepsilon = nM_s + nTR \qquad (5)$$

where J_z is the moment of inertia of the ring with respect to the axis of rotation. Considering the relation 4, finally it is obtained:

$$J_z \varepsilon = n(M_s + \frac{R}{r} M_r) \qquad (6)$$

The relation 6 shows that when a rotation motion is applied to the external ring and afterwards this is set free, the ring will perform a steadily decreasing motion. If the angular braking acceleration is found experimentally, the total braking moment from a contact can be estimated.

$$M_s + \frac{R}{r} M_r = \frac{J_z \varepsilon}{n} \qquad (7)$$

In order to find the rolling friction torque, the spinning friction torque must be known. The methodology presented in [2] is applied in the estimation of the spinning friction moment. Briefly, the method consists in attaching a collar to the bearing ball, placing it into the race and then setting into motion. Using dynamical simulation software it is proved that towards the end of the motion, it is a motion around a vertical fixed axis, as presented in Fig. 2.

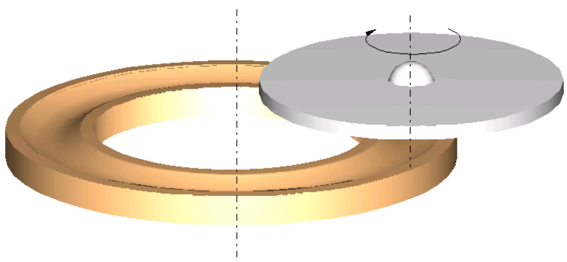

Fig. 2. Scheme for finding the spinning friction torque [2]

After finding by experimentally means the angular braking acceleration of the rotor, the coefficient of sliding friction from the contact area between the ball and the ring can be estimated.

$$\mu = \frac{J_{rot}\varepsilon}{\frac{\pi}{16} p_0 ab \int_0^{2\pi} \sqrt{a^2 \cos^2 \varphi + b^2 \sin^2 \varphi} d\varphi} = \frac{J_{rot}\varepsilon}{\frac{\pi}{4} p_0 a^2 b E\left(\sqrt{\frac{a^2 - b^2}{a^2}} \right)} \qquad (8)$$

In the above relation, p_0 is the maximum contact pressure and a, b are the half-axes of the contact ellipse, calculated according the relations from Hertzian contact theory [3].

2. Experimental test-rig and working methodology

The test rig used in the estimation of the rolling friction moment is presented schematically in Fig. 3.

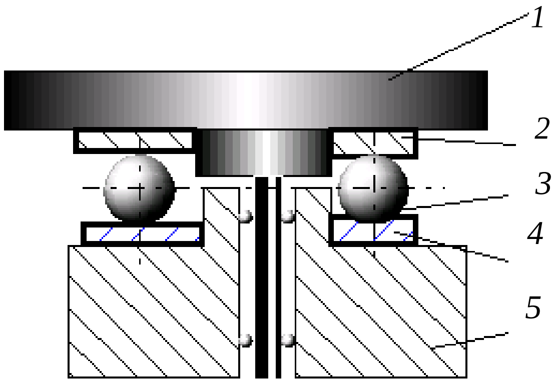

Fig. 3. The principle scheme of the test rig

The bearing ring 2 has a fly-wheel attached to it and the inferior ring 4 is mounted to the fixed part 5 of the device. The fly-wheel can perform rotation about the own axis due to the small radial ball bearings 6, fixed to shaft of the fly-wheel but also with a gap in the part 5, as to allow for axial motion without generating considerable axial friction forces. Between the two rings a number of $n=3$ balls of the bearing are positioned angularly equidistant. A greater number of balls may lead, due to dimensional errors of the balls, to different values of the normal reactions (the static indeterminate system case) [4]. The experimental set-up has a modular construction, as seen in Fig. 4.

Fig. 4. Laboratory test-rig

The following steps must be followed when an experiment is run:

-determining the mass and the moment of inertia of the fly-wheel;

-determining, according to [2] the coefficient of sliding friction on the contact area;

-mounting the test-rig with the balls and checking the fulfillment of the condition of angular equidistant positioning;

-getting the fly-wheel into motion;

-filming the running of the device, and splitting into frames;

-establishing the law of variation of the angle of position of the rotor by observing the position of an glued stamp;

-interpolating the experimental data with a parabolic function having the form:

$$\varphi = a_2 t^2 + a_1 t + a_0 \tag{9}$$

- finding the angular acceleration of the fly-wheel:

$$\varepsilon = \ddot{\varphi} = 2a_2 \tag{10}$$

- finding the total moment acting upon the rotor, using the relation 6;
- finding the spinning friction torque, using the relation given in [2]:

$$M_s = \mu \frac{\pi}{16} p_0 ab \int_0^{2\pi} \sqrt{a^2 \cos^2 \varphi + b^2 \sin^2 \varphi}\, d\varphi \tag{11}$$

where p_0, a, b are here the parameters of the contact between the ball and the race, considering that the normal force is:

$$Q = Mg/3 \tag{12}$$

3. Exemplification of methodology

The experimental results and the curve of interpolation for a launching of the fly-wheel are presented in Fig. 5. The coefficients of the polynomial of second degree used for interpolation are presented in the matrix next to the plot, starting with the constant term.

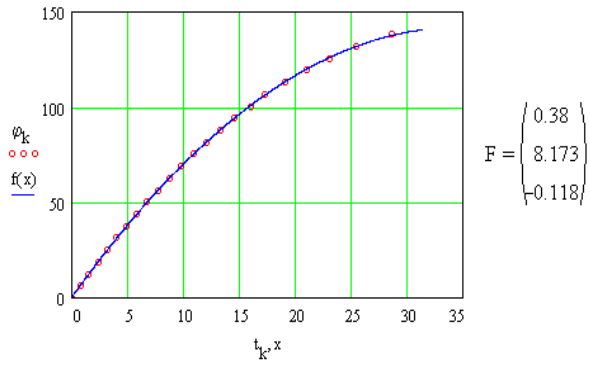

Fig. 5. Experimental data and the polynomial of interpolation

According to relation 10, it results the angular acceleration is:

$$\varepsilon = 0.236 rad / sec^2 \tag{13}$$

The fly-wheel from Fig. 4 is made by two identical cylinders of radius $r_1 = 0.08m$ and height $h_1 = 0.041m$ and a disc of radius $r_2 = 0.07m$ and height $h_2 = 0.015m$. All parts are from steel and therefore the fly-wheel assembly has the mass $M = 14.72kg$, the moment of inertia $J_z = 0.046 kg \cdot m^2$ which according to relation 7 gives:

$$M_s + \frac{R}{r} M_r = 3.59 \cdot 10^{-3} Nm \tag{14}$$

In the above relation, R is the radius of the trajectory of the centre of ball and r is the radius of the ball. The tests were performed using parts of a 51108 series thrust bearing, that has $r = 3.5mm$ and $R = 50mm$.

The parameters of the ball-race contact can be established after stipulating the principal curvature radii of the toroidal groove which is in fact the rolling race, namely $\rho_1 = \infty$ and $\rho_2 = -7mm$.

According to relation (12), the normal force in the Hertzian contact is $Q = 49N$. For this value, the contact parameters were calculated in Mathcad and the results are presented in Fig. 6.

$$\begin{bmatrix} ex \\ p_0 \\ a \\ b \end{bmatrix} = \begin{bmatrix} 0.256 \\ 1.796 \cdot 10^{9} \\ 1.267 \cdot 10^{-4} \\ 1.028 \cdot 10^{-4} \end{bmatrix}$$

Fig. 6. The values of the ball-race contact parameters

Using for the dynamic sliding coefficient of friction between ball and race the value $\mu = 0.5$, experimentally found [2], the rolling friction torque is obtained applying the relation 7, and it results $M_r = 2.777 \cdot 10^{-4} Nm$.

Under the assumption of linear dependence between the torque and normal force, to this value a spinning friction coefficient $s_r = 5.66 \mu m$ corresponds, that is in full agreement to the values from technical literature and to the values obtained in a series of recent works [5-8].

4. Conclusions

The second part of the paper presents the methodology and the test-rig used for the evaluation of the rolling friction torque in a thrust ball-bearing, accepting that during the entire motion of the bearing the balls are angularly equidistant.

When one of the rings is kept immobile, the braking of the other ring is produced by the spinning torque and the friction forces from contacts. The equation of motion of the balls proves a direct relationship between the friction forces and rolling friction torque and therefore it results that the braking of the mobile ring is produced by both components of the friction torque.

The paper proposes a method and device for finding a linear combination between the spinning and rolling friction torques.

The methodology supposes finding the law of variation of the angle of position for a fly-wheel attached to the mobile ring. In order to estimate the rolling moment, the value of the spinning moment must be first experimentally found, applying the technique presented by authors in a recent paper. An actual example validates the accuracy of the proposed method.

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