Evaluation of spinning friction from a thrust ball bearing

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Abstract. The existence of ball-raceway relative motion imposes the simultaneous occurrence of spinning motion about an axis parallel to the normal in the contact point and a rolling motion about an axis contained in the common tangent plane intersecting the axis of the ball bearing. For the separate characterization of the two friction torques - spinning and rolling moments, the devices to be used must be specially designed. For finding the spin torque or the rolling torque, the test-rigs used must contain a housing washer and a ball between which only spinning or rolling motion, respectively, exists. A test rig for evaluation of the spinning moment assumes ensuring the rotation motion about the normal in the contact point and additionally allows for estimation of the friction torque. The main inconvenience of such a device is the necessity of bearings capable to ensure a fixed position for the axis of rotation of the ball. The work proposes a simple and precise method for estimation of the spin moment by analyzing the motion of an axi-symmetric body that is in contact with the housing washer of a thrust ball bearing. The experimental data are interpolated using a parabolic function which is a function characteristic to the rotation motion with constant friction torque. The (constant) angular acceleration is found from the law of motion and together with the moment of inertia of the body is used in estimation of the spinning torque. This moment is assumed equal to the one found experimentally and thus results the value of the coefficient of friction from the contact area. The concordance between the value of the coefficient of friction obtained by the present method and the values from literature validates the proposed technique.

1. Introduction
In machine building there are scarcely situations when an element operates separate from others and doesn't interact with any of the elements of the structure to which it belongs [1-2]. In most of the practical applications the element makes direct contact with the neighbouring elements of the assembly. There are two manners in which the two elements can contact, according to the conformity of the boundary surfaces [3]:
- when the contacting surfaces are identical, translation motion between these surfaces becomes possible, along axes contained in the tangent plane and for particular cases, the option of three rotations in the case of spherical surfaces (spherical pair), or of a single rotation for cylindrical surfaces (cylindrical pair) or plane surfaces (plane pair);
- when the surfaces are non-conformal and the theoretical contact is made in a point or on a curve, the possible number of relative motions increases; there is thus the option of rotation motions around axes contained in the common tangent plane. In the case of a point contact, the common normal is
completely defined. The relative angular velocity has two components, one parallel to the normal (spinning motion) and the other contained in the tangent plane (rolling motion), as shown in figure 1.

![Figure 1](image1.png)

**Figure 1.** Relative motions and the components of the friction torsor in the case of a point contact

To each of these motions a moment opposes, parallel and of opposite direction and having the related name. Another probable displacement between the two surfaces is the translation along a straight line from the tangent plane. A friction force parallel and of opposite sense will be opposed to it. From the above considerations it results the necessity of complete knowledge of relative motion between two contacting elements when the friction torsor is required [4-6]. This spinning torque problem occurs in the case of the shafts with appreciable axial loading, like axial wind turbine shafts [7-8]. Due to variable wind speed, the impact dynamics must be considered in the analysis of bearing elements. The efficiency of the assembly [9-11] depends directly on the values of the spinning torque.

2. The relative motions from a thrust bearing

Roller bearings are typical examples of machine elements where the rolling bodies make non-conformal contacts with the rolling raceway. In the case of radial ball bearings the balls have a plane-parallel motion while for the thrust bearings the motion is more complex. In figure 2 is presented the housing washer considered immobile and a ball in the running position. An adequate operation of the ball-bearing imposes that pure rolling exists in the contact point, \( \omega_C = 0 \) and the trajectory of the centre of the ball is circular. The velocity of the centre of the ball \( v \) is permanently tangent to the trajectory.

![Figure 2](image2.png)

**Figure 2.** The ball and the housing washer of a thrust bearing.

It is assumed that the velocity presents components on all the axes of the coordinate system from figure 2. The pure rolling condition [5-6] is written as:
\[
\begin{bmatrix}
  v_x \\
  0 \\
  0
\end{bmatrix} + \begin{bmatrix}
  \omega_x \\
  \omega_y \\
  \omega_z
\end{bmatrix} \times \begin{bmatrix}
  0 \\
  0 \\
  -r
\end{bmatrix} = \begin{bmatrix}
  v_x + r\omega_y \\
  -r\omega_z \\
  0
\end{bmatrix} = 0
\]

(1)

The equation (1) requires that the following relation is valid:

\[\omega_x = 0\]

(2)

The condition (2) stipulates that the radial direction \(O_y\) is the rolling direction while the spinning motion is \(\omega_z\); the two motions may coexist with the remark that the angular rolling velocity \(\omega_y\) is always nonzero. It results that when the spinning friction is sought after for the running conditions of the bearing, the rolling friction should be known as well as the spinning friction. Hence the conclusion that it is easier to find the spinning torque considering a ball-raceway contact where only spinning motion exists. The impediment occurring in this situation is that the free-ball in contact with the raceway may execute all three rotations and practically it is impossible to separate only the spinning component. A solution consists in imposing a controlled spinning motion to the ball using bearings [12] with the mention that the friction in these bearings must be considerably smaller than the spinning friction (use of magnetic or gas bearings). In the present work, based on the conclusions from [13] the contact between a single ball and a housing washer was the choice made with the mention that in order to ensure steady rotation motion of the ball around the axis, a rim, made from a disc, capable to ensure to the assembly an constancy gyroscopic effect [14-18] was attached to the ball. A first stage studied the effect of the rim using a dynamical simulation software, as shown in figure 3. With the assumption that the center of the ball has the initial velocity \(v\) and an angular velocity directed along the normal in the contact point, the motion of the ball was simulated for three cases: first, a contact characterized by the absence of friction \(\mu = 0\), then a contact with low friction \(\mu = 0.05\) and finally a contact with important friction, \(\mu = 0.3\). The trajectory described by the center of the collar was represented in order to evaluate the effect of steadiness of the motion. As it can be observed from figure 4 no correlation can be made between the value of the coefficient of friction and the stability of the motion. Thus, in the case of no friction, the center of the rim performs an oscillatory motion along the theoretical trajectory.

**Figure 3.** Simulation made using a dynamical analysis software.

**Figure 4.** The results of dynamical simulation for the horizontal projection of the trajectory of the center of the ball.
When the friction is present, an oscillatory motion on the direction normal to the theoretical trajectory occurs and the velocity of the center of the ball becomes zero before the end of the motion.

3. Experimental set-up. Construction and operating
The dry contact between a bearing ball and a concave spherical lens was studied in [13] and from experiments it was noticed that, regardless of the manner in which the ball is launched, after a time, the motion of the ball becomes a steady rotation around a fixed axis. A bearing 51120 series with balls of diameter \( \frac{1}{2} \text{inch} = 12.54 \text{mm} \) was used for the experimental tests. The rim attached to the ball is \( 4.85 \text{mm} \) thick and has the external diameter of \( 50 \text{mm} \). The ball with the attached rim is shown in figure 5 and the experimental set-up is presented in figure 6.

![Figure 5. Bearing ball with attached rim.](image)

![Figure 6. Experimental set-up.](image)

The test-rig is rather simple: the housing washer, the ball with the rim and a high-speed camera mounted on a tripod to captures the motion of the collar. A test consists in the launching of the ball and starting the camera when it is considered that the steady motion of the rim is attained. Afterwards, the film of the motion of the rim is split into frames and the moments of complete rotations of the rim are identified. For the actual case, there were considered the instants when complete sets of ten-rotations were made. Under the hypothesis of constant spinning torque, the equation of motion of the ball-rim assembly for steady motion is [6]:

\[
J_z \ddot{\phi} = -M_s
\]  

(3)

With the initial conditions:

\[
\phi(0) = \phi_0, \quad \dot{\phi}(0) = \omega_0
\]  

(4)

the solution of the differential equation (4) has the form:

\[
\phi(t) = -\frac{1}{2} \frac{M_s}{J_z} t^2 + \omega_0 t + \phi_0
\]  

(5)

The numerical dependence of the rotation angle versus time is interpolated by the parabola:

\[
\phi(t) = a_2 t^2 + a_1 t + a_0
\]  

(6)

By identifying the coefficients of the polynomials from equations (4) and (5), the value of the spinning torque can be found:

\[
M_s = -2J_z a_2
\]  

(7)
The experimental data and the interpolation curves for two launches of the ball-rim assembly are presented in figure 7 together with the values of the coefficient of interpolation polynomials.

![Figure 7. Experimental data and interpolation curves](image)

![Figure 8. Coefficients of the interpolation polynomials: a) 4-th degree polynomial; b) parabola](image)

The moment of inertia of the ball-rim assembly is:

\[
J_z = \frac{2}{5} m_p r^2 + \frac{1}{2} \pi (r_{\text{max}}^4 - r^4) \rho = 2.32 \cdot 10^{-5} \text{kg} \cdot \text{m}^2
\]

where \( r_{\text{max}} = 0.025 \) is the external radius of the rim and \( \rho \) is the density of the steel. Thus, the spinning moment has the value:

\[
M_{\text{spin}} = 9.9 \cdot 10^{-6} \text{N} \cdot \text{m}
\]

An interesting feature is evidenced when an estimation of the coefficient of friction \( \mu \) on the contact area is sought. To this end, the dimensions of the contact area must be established, namely the \( a \) and \( b \) semi-axis, and therefore the transcendental equation that offers the eccentricity of the contact [4], [19] must be solved:

\[
(1 - ex^2) \frac{D(ex)}{K(ex) - D(ex)} = \frac{A}{B}
\]

where:

\[
K(ex) = \int_0^{\pi/2} \frac{d\varphi}{\sqrt{1 - ex^2 \sin^2 \varphi}}
\]

and

\[
E(ex) = \int_0^{\pi/2} \sqrt{1 - ex^2 \sin^2 \varphi} d\varphi
\]

are the complete elliptical integrals of first and second kind, respectively and \( D(ex) \) has the expression:
The contact stiffness $\eta$ has the expression:

$$\eta = \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2}$$

where $E_{1,2}$ and $\nu_{1,2}$ are the Young moduli and the Poisson coefficients, respectively, for the two materials.

For a normal force given by the weight of the assembly $Q = (m_b + m_c)g$ the semi-axis of the contact area $a, b$ are found using the relations:

$$a, b = \frac{3}{\pi} \eta \left( \frac{Q}{k} \right)^{1/3}$$

and for the maximum contact pressure the following expression is applied:

$$p_0 = \frac{n_p}{\pi} \left[ \frac{3}{2} \eta \left( \frac{Q}{k} \right)^{1/3} \right]^{1/3}$$

In the above relations, the coefficients $n_{a,b}$ and $n_p$ are established with the relations:

$$n_a = \frac{2}{\pi} \sqrt{\frac{1 + B}{A} D(ex)} \quad n_b = \frac{2}{\pi} \sqrt{\frac{1 + B}{A} K(ex) - D(ex) \sqrt{1 - ex^2}} \quad n_p = n_an_b$$

Finally, it results:

$$a = 5.606 \cdot 10^{-5} \text{ m}, b = 3.006 \cdot 10^{-5} \text{ m}$$

$$p_0 = 2.164 \cdot 10^8 \text{ Pa}$$

Considering a current point of coordinate $(x, y)$ on the contact area, the pressure in this point is:

$$p(x, y) = p_0 \sqrt{1 - \frac{x^2}{a^2} - \frac{y^2}{b^2}}$$
The total moment of friction is obtained by integrating the elementary torque of friction on the entire contact area:

\[ M_f = \int_{\mu} dM_f = \int_{\mu} \sqrt{x^2 + y^2} \mu p_0 \sqrt{1 - \frac{x^2}{a^2} - \frac{y^2}{b^2}} dxdy \]  

(24)

The next change of variable is made in relation (24):

\[
\begin{align*}
    x &= a\rho \cos \varphi \\
    y &= b\rho \sin \varphi
\end{align*}
\]

(25)

and after some simplifications it results:

\[
M_f = \mu \frac{\pi}{16} p_0 ab \int_0^{2\pi} \sqrt{a^2 \cos^2 \varphi + b^2 \sin^2 \varphi} \varphi d\varphi
\]

(26)

The value of the coefficient of friction is found by equalling the right members of equations (7) and (26):

\[
\mu = \frac{\pi}{16} \frac{p_0 ab}{J \varepsilon} \int_0^{2\pi} \sqrt{a^2 \cos^2 \varphi + b^2 \sin^2 \varphi} \varphi d\varphi = \frac{\pi}{4} \frac{J \varepsilon}{b E} \left( \frac{a^2 - b^2}{a^2} \right) = 0.49
\]

(27)

which is in complete agreement to the values from technical literature [20].

4. Conclusions

The paper analyses the relative motion between the balls and the raceway of a thrust bearing and the conclusion that the angular velocity of a spherical body presents components both on the normal and in the tangent plane in the contact point is reached. The friction spinning torque and the rolling friction torque correspond respectively to the two components of angular velocity and permanently coexist. It is concluded that for finding the spinning moments or rolling moments, distinct devices must be used, employing a housing washer and a ball between which only spinning or rolling motion, respectively, should exist.

As a principle, for measuring the spinning moment, the device must ensure a rotation motion about the normal in the contact point and allow for estimation of the friction torque. The major drawback of such a test-rig consists in the necessity of bearings that should ensure a fixed position of the axis of rotation of the ball. The presence of these bearings supposes that the internal friction moment must be either much smaller than the moment to be measured or very well known as not to alter the values of the moments to be measured.

The present work proposes a very straightforward and precise methodology for estimating the spinning moment by analyzing the motion of an axi-symmetric body that is contacting the housing washer of a thrust bearing. More precisely, the revolution body is a ball of the analyzed bearing pressed into a metallic rim with the thickness smaller than the diameter of the ball. The ball is brought into contact with the raceway and a rotation motion around an axis approximately vertical is initiated; it is noticed that after a while, the body presents a steady rotating motion around a vertical axis. A stamp was glued to the frontal face of the rim and its motion is filmed. The final part of the film, corresponding to the stable motion is split into frames using image analysis software and the experimental dependence between the rotation angle and time is found.

The experimental data are interpolated using a parabolic function which is characteristic to the rotation motion with constant friction torque. The value of angular acceleration (constant), determined from the law of motion together with the axial moment of inertia of the body permit finding the spinning torque.
The precision of the evaluation of the spinning torque was estimated after finding the characteristics of the Hertzian ball-raceway contact and calculating the spinning moment of the contact area under dry friction conditions. The value of the coefficient of sliding friction from the contact area is found from the equality between this spinning moment and the one found experimentally. The complete concordance between the coefficient of sliding friction obtained by the present manner and the values from literature validate the proposed technique.

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