The Mechanism of Microslip between the Balls and the Raceways of Bearings

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Abstract. In the article it was shown that one of the factors that significantly affect the mechanism of rolling friction is the mechanism of micro slippage between the balls and the raceway bearing. In this connection was defined the character of friction torque distribution along the contact area of contacting elements.

Introduction

In the recent years numerous scientific publications have been devoted to modelling friction in rolling element bearings [1-6, etc.]. Most of these papers are devoted to the study of rolling friction occurring between two cylinders, or friction of one cylinder along the rail under the traction load on the rolling element. However, little attention was given to the problem of free rolling friction of elements at great curvature of rolling raceways and rolling elements that has a great importance for rolling contact bearings. This paper gives an insight into the given problem.

Problem Statement

Let us consider the instantaneous position of two elastic bodies—the spherical element with the diameter \( d_s \) and toroidal raceway with the radius \( r_g \) and diameter \( D_g \) during the rolling procedure. The contacting elements when in the rolling process are under constant external load \( P \), which is directed along the line connecting the centre of the sphere and the initial point of the contact elements. The contact strip between the two contacting elements occurs under the influence of \( P \) and is equal to \( \delta_o \), which results in a curved contact area having elliptical contour and semiaxes \( a \) and \( b \). The semiaxis \( a \) is placed in the direction of rolling, whereas the semiaxis \( b \) is in the main cross-section of the contact area.

The cross-section of contacting bodies is shown in Fig.1. The instantaneous axis of relative pivotal point in contact surfaces occurs in the course of rolling. It is placed in the cross-section of the contact zone and crosses it in the points A and B, where no microslip in the contact surfaces is found.

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Let us design the Cartesian reference system centred around symmetry centre of the contact area. The axis \( Oz \) is directed along the normal toward the contact area, the axis \( Ox \) in the rolling direction, and the axis \( Oy \) —in the transverse direction. Let us set the following restrictions: 1. The coefficient of sliding friction \( f \) between contacting surfaces is constant on the whole contact area. 2. The friction forces, acting along the contact area due to significant curvature, considerably exceed the adhesive forces, including the forces caused by elastic hysteresis of body materials. Therefore we do
not take into account the impact of these forces. 3. The size of the contact area \( a \) is very small compared to the size of the rolling bodies; therefore the curvature of the contact area in the rolling direction can be neglected. 4. The shape of the initial clearance between the bodies in the main sections is described as parabolic:

\[
\begin{align*}
\Delta z(y) &= \left(\frac{1}{d_s} - \frac{1}{2r_g}\right)y^2 \\
\Delta z_x &= \left(\frac{1}{d_s} \pm \frac{1}{D_g}\right)x^2
\end{align*}
\]

where “plus” refers to the contact between the rolling elements and the rolling raceway of the inner ring, while “minus” refers to the contact between the rolling element and the rolling raceway of the inner ring.

The error caused by the latter restriction does not exceed 4% at \( b \leq 0.2d_s \). Let us solve the equation for the cross-section of the contact area. Additionally, let assume that elastic strain at an arbitrary point of the contact area for each contacting body is in inverse proportion to material modulus of elasticity:

\[
\frac{\delta_s(y)}{\delta_g(y)} = \frac{E_g \cdot (1-\mu_s^2)}{E_s \cdot (1-\mu_g^2)} = k_E
\]

where \( \delta_s(y) \) and \( \delta_g(y) \) refer to elastic strain of the rolling element and the raceway respectively at the section point with the coordinate \( y \); \( E_s \) and \( E_g \) refer to the material modulus of elasticity for these bodies, \( \mu_s \) and \( \mu_g \) are the Poisson's ratios for contacting bodies.

Using the simple geometrical constructions based on the scheme shown in Fig. 1 and equations (1) and (2), we can define the equation to the cross-section of the contact area:

\[
z_p(y) = R_M y^2 \quad \text{at} \quad -b \leq y \leq b
\]

\[
R_M = \frac{1}{1 + k_E \left(\frac{k_E}{2r_s} + \frac{1}{2r_g}\right)}
\]

where
Defining Friction Torque in the Bearing

Let us consider rolling resistance force at the arbitrary point M of the contact area. Let us select an area sized $dx$ and $dy$ at the given point in the contact area. The rolling resistance force caused by microslip in contact surfaces over the given surface element is equal to:

$$dF_t(x, y) = f \cdot q(x, y) \cdot dx \cdot dy$$

where $f$ is the coefficient of rolling resistance depending on the material of the contact elements, lubricants and other conditions of friction; $q(x, y)$ is the value of contact stresses at the point in the contact zone with the coordinate $x, y$.

The force $dF_t(x, y)$ acts in direction of the velocity vector relating the relative slip of contact surfaces along the axis $OX$. This force results in the rolling resistance to the instantaneous rotation axis of the surfaces. With regard to the small size of the contact area in the rolling direction compared to the size of the elements, the arm, which is under the impact of the element forces acting along the velocity vector of slippage between the surfaces, is constant and equals the ordinate $(z_o - z(y))$.

Consequently, considering the equation (3), the element moment emerging due to the force $dF_t(x, y)$ is equal to

$$dM_t(x, y) = f \cdot (z_o - z(y)) \cdot q(x, y)dx \cdot dy = R_M f \cdot \left( y_o^2 - y^2 \right) \cdot q(x, y)dx \cdot dy$$

where $z_o$ is the ordinate for the instantaneous axis of rotating contact surfaces when the elements perform a rolling motion; $z(y)$ is the ordinate of the cross-section of the contact area; $y_o$ is the abscissa value of the section point in the contact zone with the instantaneous axis of rotating contact surfaces passing through its vertex.

It is worth noting that tangential rolling resistance forces located lower than the instantaneous axis of rotating contact surfaces act in one direction, whereas the forces located above the instantaneous axis act in the opposite direction. Therefore, the rolling resistance moments caused by tangential forces located below or above the instantaneous axis, can be determined by integrating the equation (5) along the elliptical contact area:

$$M_{m}(x, y) = 4R_M f \int_{y_o}^{y} \int_{0}^{x} \left( y_o^2 - y^2 \right) q(x, y)dx \cdot dy$$

$$M_{nv}(x, y) = -4R_M f \int_{y_o}^{b} \int_{0}^{x} \left( y_o^2 - y^2 \right) q(x, y)dx \cdot dy$$

where $M_{m}$ and $M_{nv}$ are friction torque moments found below and above the instantaneous axis respectively $x = a \sqrt{1 - \frac{y^2}{b^2}}$.

If we sum up the stress components $M_{m}$ and $M_{nv}$ in accordance with the absolute value and set them to zero, we can find the value $y_o$ from the obtained equation. If we sum up the moduli of the given values, we can get the rolling friction moment. For example, if we assume that distribution of contact stresses complies with the Hertz theory:

$$q(x, y) = \frac{3P}{2\pi ab} \sqrt{1 - \frac{x^2}{a^2} - \frac{y^2}{b^2}}$$

where $P$ is the external load acting on the rolling element, and on rearranging the equation we obtain:
\[ y_o = \sqrt{\frac{1}{5}} \cdot b \]

\[ M_s = 0.172 R_M f \cdot P \cdot b^2 \]  

(8)

where \( M_s \) is the friction torque of the raceway with one rolling element.

If the number of rolling elements in the bearing and distribution of external load between rolling elements are established, then the moment of rolling friction in the bearing can be defined. For example, if the bearing is under the axial load \( A \), the number of balls in the bearing is equal to \( z_s \), and the section radii of the rolling raceways for external and inner rings are equal, then based on the equation (8), the value of the friction torque in the bearing is given by

\[ M_s = 0.172 R_M f \cdot \frac{A}{\sin \beta} \left( b_v^2 + b_n^2 \right) \]  

(9)

where \( b_v \) and \( b_n \) are the semiaxes of the contact area formed in the external and inner rings respectively, and depending on the ball load; \( \beta \) is the contact angle in the bearing.

Values \( b_v \) and \( b_n \) depend on a number of factors, but can be easily defined using the standard methodology as in the angular contact bearing 36206 \( \beta = 12^\circ \), \( d_s = 9.53 \text{[mm]} \), \( r_g = 4.94 \text{[mm]} \), \( k_E = 1 \), \( z_s = 16 \). When the bearing is under the axial load \( A = 2500 \text{[H]} \), the size of areas are equal to: \( b_v = 1.306 \text{[mm]} \), \( b_n = 1.069 \text{[mm]} \). Using the given values for the equation (9), and taking into account \( f = 0.12 \), we get \( M_s = 72.9 \text{[H}\cdot\text{mm]} \). In all the catalogues for bearings, statistical dependence is used to estimate the friction torque in the ball bearings:

\[ M_s = 0.002 \cdot A \cdot \frac{d}{2}, \]  

(10)

where \( d \) is the diameter of the bearing bore, mm.

In the considered example \( d = 30 \text{[mm]} \), we get \( M_k = 75 \text{[H}\cdot\text{mm]} \). However, as distinguished from the existing methodology, the proposed method for estimating the friction torque reflects physics of the friction process and impact of the main factors. Therefore, our method allows us to improve the accuracy of the estimates, and to effectively control the friction process.

Among the significant advantages of the given methodology is that it helps differentiate the friction torque along the section of the contact surface. Using the equation (5) for the minor contact area, we get:

\[ m_t(y) = 2R_M f \left( y^2 - y_o^2 \right) \int_0^{y_o} q(x, y) dx = 2R_M f \left( y^2 - y_o^2 \right) \]  

(11)

where \( m_t(y) \) is the specific friction torque per length unit of the section in the contact area.

**Conclusion**

Figure 2 shows distribution of the friction torque along the cross-section of the contact area for conditions described in the above mentioned example. As is seen, distribution of friction torque along the contact section is uneven, which proves the S.V. Pinegin's experiment [8] that showed that wear of the rolling raceways occurs along the centre line and the areas located closer to the sides of the contact area.

Thus, the proposed mathematical model describes the mechanism of rolling friction and provides a possibility to control the friction torque through perfecting the design of the rolling bearing.
Figure 2. Distribution of the antitorque moment along the big axis caused by microslip of contacting bodies.

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