INFLUENCE OF THE LUBRICATION ON FRICTION IN MICROBALL BEARINGS

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Abstract: *A theoretical and experimental methodology to determine the rolling friction torque in microball bearings operating in full film lubrication regimes have been developed by the authors. The rolling friction torque determined by developed methodology was compared with the rolling friction torque determined according the SKF methodology. It was evidenced that for microball bearings the presence of the lubricant leads to important increasing of the rolling friction torque.*

Keywords: *micro-ball bearings, rolling friction torque, ball-race contacts, lubrication regime.*

1. Introduction

In the last period, the microball bearings were developed and tested to be use in micromotors and microturbines.

Ghalichechian et al. [1] experimentally determined the friction coefficient in an encapsulated rotary micro ball bearing realized by micro fabrication silicon and stainless steel micro balls of 0.285 mm diameter. By using spin – down method the authors determined the global friction and value of 5.62 μNm has been obtained for an angular speed of 20.5 rad/s and under an axial load of 48 mN.

McCarthy et al. [2] experimentally investigated the influence of the speed and of the normal load on rolling friction torque in an encapsulated microball bearing having 0.285 mm diameter stainless steel microballs and silicon races. Imposing a linear dependence between friction torque and rotational speed, the authors determined by spin down method the global friction torque for rotational speed between 250 rpm and 5000 rpm and for axial load between 10 mN and 50 mN. The global friction torque obtained was from 0.0625 μNm to 2.5 μNm for rotational speed between 250 rpm and 5000 rpm and for axial load between 10 mN and 50 mN, respectively. Based on the experimental results, the authors obtained following empirical equation for the global friction torque in the microball bearings: $M = 9 \cdot 10^{-5} \cdot F_N^{0.444} \cdot n$, where M is global friction torque in μNm, F_N is the axial load acting on the micro ball bearing in mN and n is rotational speed in rot/min.

Olaru et al. [3] evaluated the rolling friction torque in an original microtribometru realized from a micro thrust ball bearing by using only 3 micro balls. The experiments were realized in dry conditions with steel microballs having 1.588 mm diameter, with normal loads on the ball-race contact of (0.008 – 0.033) mN and rotational speed having values between 60 rpm to 210 rpm. Rolling friction torque in a modified thrust microball bearing having values between (0.06 – 0.23)10^{-3} Nmm were obtained for dry conditions. The obtained values of the friction torque are with 2–3 order of magnitude higher than the friction torque values calculated with classical equations in dry conditions. For lubricated conditions, from mixed to full film, Bălan [4] elaborated a theoretical model for rolling friction torque in a modified thrust ball bearing having 3 balls with diameter of 7.983mm. The experiments realized confirmed the theoretical model and evidenced the major influence of hydrodynamic rolling force in low loaded ball – race contacts.

In order to determine the rolling friction torque in the lubricated micro rolling systems low loaded, the authors used the new micro tribometer presented in [3,4]. Experimental investigations were realized on a modified thrust ball bearing 51100 having only 3 balls with diameter of 4.75 mm and operating between 60 to 210 rpm under the normal load of 33.2 mN and lubricated with a mineral oil having 0.05 Pas dynamic viscosity.

2. Equipment and procedure

Fig. 1 presents the modified thrust ball bearing. The driving disc 1 is rotated with a constant rotational speed and has a radial groove race. Three micro balls are in contact with the race of the disc 1 at the equidistance position (120 degrees). All the three micro balls sustain an inertial disc 2 with the same radial grove race and the micro balls are loaded with a force $Q = G/3$, where G is the weight of the disc 2.

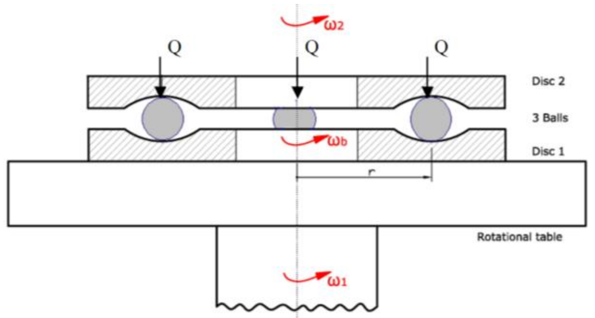

Figure 1: *Modified thrust ball bearing*

When the disc 1 starts to rotate with a constant angular speed ω_1, the balls start to roll on the raceway of the disc 1 and start to rotate the inertial disc 2, as a result of rolling friction forces between the balls and the disc 2. When the rotational speed of the inertial driven disc becomes constant, ω_0, the driven disc 1 is stopped and the inertial driven disc 2 starts a deceleration process until it completely stops due to rolling friction between the balls and the races. During this time the angular position of the disc 2, $\varphi_2(t)$, has a time variation from zero to a maximum value and the corresponding angular speed of the disc 2 has a time variation $\omega_2(t)$ from the initial value ω_0 to zero. By monitoring the angular position of the disc 2, information about the friction torque in the modified thrust ball bearing have been obtained. A high–speed camera Philips SPC900NC/00 VGA CCD with 90 frames/seconds was used to capture the angular position of the disc 2 during the tests. The images captured by the camera were processed frame by frame to determine the angular positions $\varphi_2(t)$, corresponding to every frame.

3. A new methodology to determine friction torque

By neglecting the friction between disc 2 and air (rotational speeds were between 60 and 210 rpm) the differential equation of the upper disc 2 becomes:

$$\frac{J \cdot d\omega_2(t)}{dt} = -T_z \tag{1}$$

were J is the inertial moment of the disc 2 and Tz is friction torque acting on the disc 2 in deceleration process as result of rolling friction between balls and the two races.

For the torque Tz the following equation is proposed:

$$Tz = K \cdot \omega_2^n \tag{2}$$

where K is a constant and n is an exponent having values lower than 1.

For $n < 1$ by integrating Eq. (1) results the following equation for $\omega_2(t)$ and $\varphi_2(t)$, respectively:

$$\omega_2(t) = \left[\omega_0^{1-n} - \frac{K \cdot (1-n)}{J} \cdot t\right]^{\frac{1}{1-n}} \tag{3}$$

$$\varphi_2(t) = \frac{J}{K \cdot (2-n)} \cdot \omega_0^{2-n} - \left[\omega_0^{1-n} - \frac{K \cdot (1-n)}{J} \cdot t\right]^{\frac{2-n}{1-n}} \tag{4}$$

The initial conditions in the integrating processes were: for $t = 0$, $\omega_2(t) = \omega_0$ and $\varphi(t) = 0$, where ω_0 is the upper race angular speed at the start of deceleration process.

The K parameter and exponent n were established on the experimental basis by imposing the following conditions: the disc is stopped at a time $t_{\max}$ determined by testing and the measured cumulative position angle $\varphi \max$ has to correspond with the values given by the equations:

$$\varphi_2(t_{\max}) = \varphi \max \tag{5}$$

$$\omega_2(t_{\max}) = 0 \tag{6}$$

Conditions (5) and (6) lead to following nonlinear equations:

$$\left[\omega o^{1-n} - \frac{K}{J}\cdot(1-n)\cdot tmax\right]^{\frac{1}{1-n}} = 0 \tag{7}$$

$$\frac{J}{K\cdot(2-n)}\cdot\omega o^{2-n} - \left[\omega o^{1-n} - \frac{K}{J}\cdot(1-n)\cdot to\right]^{\frac{2-n}{1-n}} - \phi max = 0 \tag{8}$$

Based on the experiment, for a given rotational speed ω_0 we obtained the values for the parameter K and exponent n and finally, by Eq. (2) was determined the rolling friction torque *Tz*.

4. SKF methodology

The SKF methodology [6] for determining the friction torque in a ball bearing takes into account four components considered as being important in friction torque estimation. Thus, the rolling friction torque in a bearing is given by following equation:

$$M = M_{rr} + M_{sl} + M_{seal} + M_{drag} \tag{9}$$

where M_{rr} is the rolling component, M_{sl} is the slip component, M_{seal} is the seal systems component, M_{drag} is the component due to friction of the rolling elements in oil bath. In the absence of seals and for lubrication conditions using reduced oil quantities, only the first two components are of importance. The SKF rolling friction components can be calculated as:

$$M_{rr} = G_{rr}\cdot(\upsilon\cdot n)^{0.6} \tag{10}$$

where n is the bearing speed, in rpm, υ is the kinematics viscosity of the lubricant, in mm^2/s, G_{rr} is a variable depending on type of bearing, mean diameter and load. Supplementary, the rolling friction resistance is affected by two factors:

(i) Inlet shear heating factor ϕ_{ish} determined by the equation:

$$\phi_{ish} = \frac{1}{1+1.84\cdot10^{-9}\cdot(n\cdot d_m)^{1.28}\cdot\upsilon^{0.64}} \tag{11}$$

(ii) Kinematical starvation factor ϕ_{rs} given by:

$$\phi_{rs} = \frac{1}{\exp\left\{3\cdot10^{-8}\cdot(\upsilon\cdot n)\cdot(d_a+D)\cdot\left[\frac{3.8}{2\cdot(D-d_a)}\right]^{0.5}\right\}} \tag{12}$$

where d_a and D are the diameters of the inner and outer thrust ball bearing's races respectively, in mm.

The sliding component of rolling friction torque is:

$$M_{sl} = \mu_{sl}\cdot G_{sl} \tag{13}$$

in which G_{sl} is a variable depending on type of bearing, mean diameter and load, μ_{sl} is the friction coefficient with values in the range 0.002 – 0.1 in terms of bearing type and lubrication conditions.

The SKF Catalogue [5] specifies that the use of the relationship above is valid for full film lubrication conditions when the κ ratio between oil viscosity υ and SKF reference viscosity υ_1 (depending on speed and diameter) is larger then 2. For mixed lubrication ($\kappa < 2$), SKF recommends corrections of the sliding friction coefficient μ_{sl} so that it can be estimated by equation:

$$\mu_{sl} = \varphi_{bl}\cdot\mu_{bl} + (1-\varphi_{bl})\cdot\mu_{EHL} \tag{14}$$

where μ_{bl} is the friction coefficient between the interacting asperities, μ_{EHL} is the friction coefficient due to lubricant shearing, φ_{bl} is a weighting factor including the influence of the asperities on lubricant shearing effect:

$$\varphi_{bl} = \frac{1}{e^{2.6\cdot10^{-8}\cdot(\upsilon\cdot n)^{1.4}\cdot d_m}} \tag{15}$$

Due to the lubrication conditions (i.e. a reduced quantity of oil) and the absence of seals the friction torque in the bearing using the SKF model was computed as:

$$M_{SKF}(n) = M_{rr}(n) + M_{sl}(n) \qquad (16)$$

where the two components $M_{rr}(n)$ and $M_{sl}(n)$ are given according to the Eqs. (10-15) respectively. The variables in the above mentioned equations were calculated, as indicates the SKF Catalogue [5], with the equations:

$$Grr = 1{,}06 \cdot 10^{-6} \cdot dm^{1{,}83} \cdot Fa^{0{,}54} \qquad (17)$$

$$Gsl = 1{,}6 \cdot 10^{-2} \cdot dm^{0{,}05} \cdot Fa^{\frac{4}{3}} \qquad (18)$$

in which *dm* is the bearing mean diameter, in mm, *Fa* is the axial load force, in N, the total friction torque $M_{SKF}(n)$ resulting in N·mm.

The axial load *Fa* was considered for a normal thrust ball bearing 51000 having 9 balls with diameter of 4.75mm. So, if in modified thrust ball bearing was used only 3 balls with normal load Q for every ball, to be the same normal load in the standard thrust ball bearing, the axial load was imposed *Fa = 9Q*. The dimensions of the standard 51100 ball bearing are: d=10mm, D=24mm and d_m=17mm.

4. Lubrication regimes

To determine the film thickness between the balls and races, it will be necessary to verify if the lubrication regime is IVR or EHL. Using the Hamrock's methodology [6] Balan et al. [7] developed a computer program to realize the lubrication maps for general point contacts. Bălan [4] adapted the computer program for a modified thrust ball bearing.

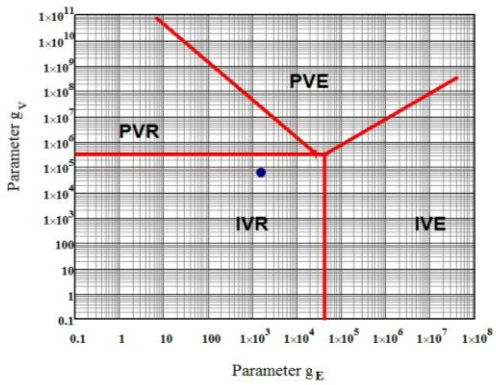

Figure 2: *Map of lubrication regimes for oil with viscosity of 0.05 Pas and rotational speed of 210 rpm*

So, by using the program developed in [4] was realized the maps of lubrication regimes for lubricant having the dynamic viscosity of 0.050 Pa·s at the temperature of 27^{0}C and for the geometry of the standard 51100 ball bearing, for a normal load of 0.033N on every ball race contacts. Fig. 2 presents the map of lubrication regimes for the contact between a ball and the raceway for a rotational speed of the disc 2 of 210 rpm.

It can be observed that the lubrication regime is dominant IVR. The dimensionless viscosity parameter g_V and dimensionless elasticity parameter g_E are determined according to the equations developed in [4].

The minimum film thickness in the contact between the ball and the races for IVR lubrication regime was determined according to the methodology presented in[4].

When the rotational speed of the upper ring 2 varies between 60 rpm and 210 rpm, and the normal load on ball-race contact is Q = 0.033 N, the minimum film thickness $h_{min(IVR)}$ has a variation between 0.24μm to 1.32 μm respectively. For the roughness of the races $Ra_r = 0.06$ μm and for the roughness of the balls $Ra_b = 0.03$ μm the lubricant parameter Λ defined by the equation:

$$\Lambda = \frac{h_{min}}{1.15 \cdot \sqrt{Ra_r^{\,2} + Ra_b^{\,2}}} \qquad (19)$$

The values for lubricant parameter Λ varies between 2.4 and 15, when the rotational speed varies between 60 rpm to 210 rpm.

5. Results

A lot of experimental investigations were realized on the modified thrust ball bearing 51100 having only 3 balls with diameter of 4.75 mm and operating between 60rpm to 210rpm under the normal load of 33.2 mN. For this normal load the maximum Hertz normal contact between a ball and races was 0.129 GPa with about an order of magnitude lower than in normal ball bearing contacts. Was used small quantity of mineral oil having 0.05 Pas dynamic viscosity at the temperature of 27^{o}C. For every experiment was determined the maximum time t_{max} and maximum cumulative position angle φmax. By solving the nonlinear equations (7), (8) the parameter K and exponent n were determined and friction torque Tz was obtained. In Figure 3 are presented the variation of the friction torque Tz with rotational speed of the disc 2. Also, by using the SKF methodology the friction torques for a modified thrust ball bearing having 3 balls was calculated. The numerical values were also presented on the Figure 3.

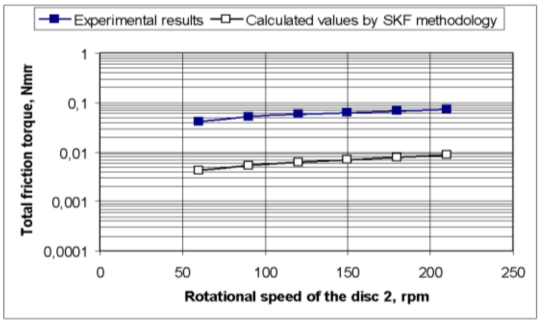

Figure 3: *The variation of the friction torque in modified thrust ball bearing: oil viscosity 0.05Pas, normal load on ball race contact 0.033N*

It can be observed that for very low normal loads the friction torque experimentally determined is higher with about an order of magnitude than the friction torque calculated by SKF methodology. Similar results were obtained for dry conditions and very low normal loads.

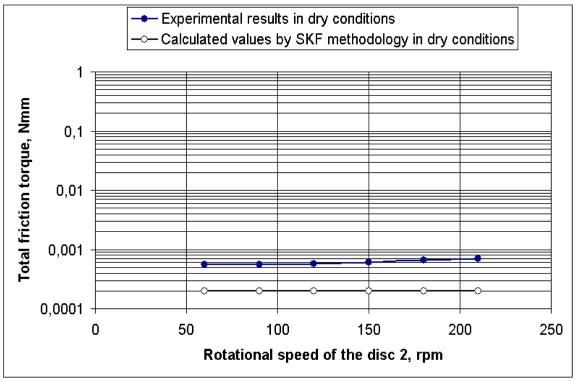

Figure 4: *The variation of the friction torque in modified thrust ball bearing: dry conditions, normal load on ball race contact 0.033N*

In figure 4 are presented the variations of the total friction torque for modified thrust ball bearing operating in dry conditions both by experiments and by SKF methodology. For dry conditions the experimental results were obtained by using the methodology presented in [3]. Also, for dry conditions the M_{rr} component from the SKF equation (10) is zero as a result of absence of the viscosity.

By comparing the values for the total friction torque obtained by proposed methodology it can be observed that for very low load conditions the presence of the lubricant in a microball bearing leads to an important increasing of the friction torque as a result of hydrodynamic effect. In our experiments from dry to full film conditions we obtained an increasing of the total friction torque with about 2 orders of magnitude.

6. Conclusions

A theoretical and experimental methodology to determine the rolling friction torque in microball bearings operating in full film lubrication regimes have been developed by the authors. The rolling friction torque determined by developed methodology was compared with the rolling friction torque determined according the SKF methodology.

It was evidenced that for very low normal loads the presence of the lubricant leads to important increasing of the rolling friction torque.

The friction torque obtained by the proposed methodology is higher with about an order of magnitude than the friction torque calculated by SKF methodology. It can be concluded that the SKF methodology can not be applied for very low normal loads.

The friction torque obtained for dry conditions and very low normal loads are with 2 orders of magnitude lower than the friction torque obtained for full lubrication film.

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