INFLUENCE OF THE GREASE ON FRICTION IN BALL-RACE CONTACTS

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Abstract: *The authors investigated experimentally the friction torque in a modified thrust ball bearing having only 3 balls operating at very low axial load and lubricated with greases. The experiments were realized by using spin-down methodology and the results were compared with the theoretical values based on Biboulet- Houpert's rolling friction methodology. A very good correlation between experiments and theoretical methodology has been obtained for synthetic grease SSX having base oil viscosity of 1000 mm²/s.*

Keywords: *Rolling friction, Thrust ball bearing, Grease, Spin-down methodology.*

1. Introduction

The friction torque in a thrust ball bearing is a result of some friction sources: rolling friction in balls-race contacts, sliding friction in balls-cage contacts, pivoting friction in the balls-race contacts, drag friction between balls and oil. For a thrust ball bearing without sealing system and operating with small quantity of oil the SKF methodology consider that total friction torque Tz can be determined as a sum of two components: the rolling component M_{rr} and the sliding component M_{sl}, Ref.[1]:

$$Tz = M_{rr} + M_{sl} \tag{1}$$

The rolling component M_{rr} is a function of the rotational speed n , oil kinematics viscosity υ, axial load *Fa,* bearing mean diameter *dm* according to the equation:

$$M_{rr} = 1.06 \cdot 10^{-6} \cdot d_m^{1.83} \cdot Fa^{0.54} \cdot (\upsilon \cdot n)^{0.6} \text{ [Nmm]} \tag{2}$$

where d_m in mm, *Fa* is in N, n is in rpm and υ is in mm²/s.

The sliding frictional component M_{sl} is a function of bearing mean diameter dm, axial load Fa and friction coefficient μ_{sl} according to the equation:

$$M_{sl} = 1.6 \cdot 10^{-2} \cdot d_m^{0.05} \cdot Fa^{4/3} \cdot \mu_{sl} \text{ [Nmm]} \tag{3}$$

The friction coefficient μ_{sl} have values in the range 0.002 – 0.1 in terms of bearing type and lubrication conditions.

Cousseau et al. [2] experimentally determined the friction torque in a 51107 thrust ball bearing operating with an axial load of 7000 N in a rotational speed range between 500 and 2000 rpm by using several different greases and compared the results with the SKF methodology. The authors determined the sliding friction component caused by grease as a different between measured total friction and SKF rolling component M_{rr} calculated with the viscosity of the base oil used in greases. Also, they determined the real values of friction coefficient μ_{sl} for greases between 0.02 to 0.06, depending on the grease type and rotational speed.

By using very low axial loads, Bălan et al [3] experimentally determined the total friction torque in a 51205 thrust ball bearing operating between 100 and 400 rpm, loaded with Fa = 4.26 N and lubricated with mineral oils having 350 mm²/s and 60 mm²/s. The authors observed that the experimental values for total friction torques are higher than the values calculated with SKF methodology with about an order of magnitude. Also, the authors established that for very low axial load and for the viscosity of 350 mm^2/s the sliding frictional component M_{sl} is about two orders of magnitude smaller that the component M_{rr} that means the major influence of the lubricant viscosity in the total friction torque.

Biboulet and Houpert [4] established a set of equations to evaluate the friction resistance moment in a ball-race contact operating in IVR and EHL lubrication conditions. Based on the Biboulet and Houpert's equations and using a spin - down methodology, Bălan et al. [5] experimentally determined the friction torque in a three balls thrust ball bearing very low axial loaded and lubricated with mineral oils having various viscosities from 350 mm^2/s to 50 mm^2/s. The experimental results confirmed that the hydrodynamic rolling forces *FR* generate the dominance effect in total friction torque.

In the present paper the authors realized experimental investigations for total friction torque by using the spin – down methodology developed by Bălan et al [5] for a modified thrust ball bearing having only 3 balls, loaded with Fa = 4.26 N and lubricated with small quantities of synthetic grease SSX. The experiments were realized with small quantity of grease in the range of rotational speed between 100 and 300 rpm.

2. Experimental procedure

In Fig. 1 is presented the modified thrust ball bearing having only 3 balls and without cage. The three balls are mounted between the races of a 51205 thrust ball bearing at equidistant angular position of 120 degrees. The lower race 1 is fixed on the rotating table. On the upper race is fixed a disc and the weight of the disc and of the upper race have the role of the axial load acting on the three balls G, each ball will take a load $Q = G/3$. The spin-down method consists of imposing a constant angular speed on race 1 until race 2 and the attached upper disc reach a synchronous angular speed equal to that of race 1. In this moment the rotational table and race 1 are suddenly stopped and race 2 starts to decelerate during a time t_{max} depending of the frictions from all the six ball-race contacts.

White marks were traced on race 1 and on the disc to visualize the angular position for the two rotating elements. A video camera with 90 frames/second was mounted above the disc. The images captured by the camera were recorded on the computer in real time and subsequently processed with an adequate program.

The experiments were realized by using the Tribometer CETR UMT-2 from the Tribology Laboratory .

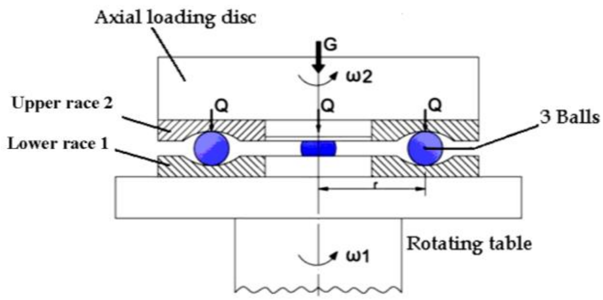

Figure 1: *Modified thrust ball bearing*

Geometrical parameters: ball diameter of 7.938 mm (5/16"), curvature radius of raceways Rc = 4.2 mm, radius r = 18 mm, roughness for races Ra_r = 0.065 μm and the average roughness of the balls was Ra_b = 0.033 μm.

Grease properties: density at 20^0C = (950 – 990) kg/m^3, penetration at 25^0C =(200-280)/10 mm, drop temperature = 250^0C, viscosity of base oil at 25^0C = (950 – 1050) mm^2/s.

Load and rotational speed: the normal load on every ball Q =G/3, the rotational speeds for upper race and disc were 100, 150, 200, 250 and 300 rpm.

3. Testing methodology

In the deceleration process of the upper race and attached disc can be use the dynamic balance of the moments acting on the rotating system:

$$J \cdot \frac{d\omega_2}{dt} + Tz(\omega_2) = 0 . \tag{4}$$

where J is the moment of inertia of the ensemble formed by the upper race and disc and Tz(ω_2) is the total friction force developed by friction generated in all 6 ball-race contacts.

For very low normal loads Bălan et al. [5] demonstrated that the total friction torque *Tz* can be expressed only as a function of hydrodynamic force *FR*:

$$Tz = 6 \cdot r \cdot FR \tag{5}$$

As in Ref. [5] we propose to use the Houpert's transition hydrodynamic force FR_{Trans_H} described by following equation, Ref. [4,5,6]:

$$\begin{aligned} FR_{Trans\_H} &= [1/(1+M/6.6)] \cdot FR_{IVR\_H} + \\ &+ [(M/6.6)/(1+M/6.6)] \cdot FR_{EHL\_H} \end{aligned} \tag{6}$$

The Eq. 6 includes both IVR and EHL hydrodynamic rolling forces expressed by following equations, Ref. [4]:

$$\begin{aligned} FR_{IVR\_H} &= 2.9766 \cdot E^* \cdot R_x^2 \cdot k^{0.3316} \cdot \\ &\cdot W^{1/3} \cdot U^{2/3} \end{aligned} \tag{7}$$

$$\begin{aligned} FR_{EHL\_H} &= 7.5826 \cdot E^* \cdot R_x^2 \cdot k^{0.4055} \cdot \\ &\cdot W^{1/3} \cdot U^{3/4} \end{aligned} \tag{8}$$

The parameter M is included by Biboulet and Houpert to evidence the transition from IVR lubrication regime to EHL lubrication regime and is approximated by equation, Ref. [4]:

$$M = 0.5549 \cdot k^{-0.6029} \cdot W \cdot U^{-0.75} \tag{9}$$

U is the dimensionless speed parameter and *W* is the dimensionless load parameter described by following equations:

$$U = \frac{\eta_0 \cdot v}{E^* \cdot R_x} \qquad W = \frac{Q}{E^* \cdot R_x^2} \tag{10}$$

where η_0 is the oil dynamic viscosity in Pa·s at the operating temperature of the contact, $v = (v_1 + v_2)/2$ is the average entrainment speed in m/s; *E** is the equivalent Young modulus of the balls and races and $k = R_y/R_x$. R_y is the equivalent radius of curvature in the *y* direction (perpendicular to the rolling direction) and R_x is the equivalent radius in the rolling direction.

For a given geometry, lubricant and normal load, the hydrodynamic force FR_{Trans_H} is a function only of the rotational speed. In this circumstance the friction torque $Tz(\omega_2)$ can be expressed by following equation:

$$Tz(\omega_2) = k^* \cdot \omega_2^{\alpha} \tag{11}$$

where *k** is not depending on angular speed and the exponent $\alpha < 1$.

The dynamic equation (4) becomes:

$$J \cdot \frac{d\omega_2}{dt} + k^* \cdot \omega_2^{\alpha} = 0 \tag{12}$$

Eq. (12) is analytically solved and following relations for variation of the angular speed of the upper race $\omega_2(t)$ and angular position of the upper race $\varphi_2(t)$ have been obtained:

$$\omega_2(t) = \left[\omega_{2,0}^{1-\alpha} - \frac{k^* \cdot (1-\alpha)}{J} \cdot t \right]^{\frac{1}{1-\alpha}} \tag{13}$$

$$\phi_2(t) = \frac{J}{k^* \cdot (2-\alpha)} \cdot \omega_{2,0}^{2-\alpha} - \left[\omega_{2,0}^{1-\alpha} - \frac{k^* \cdot (1-\alpha)}{J} \cdot t \right]^{\frac{2-\alpha}{1-\alpha}} \tag{14}$$

The values for *k** and *α* were determined by solving the nonlinearly Eqs. (13) and (14) by using following conditions obtained in the experiments:

a) At the initial time $t = 0$, $\omega_2(0) = \omega_{2,0}$, where $\omega_{2,0}$ is the angular speed of the upper race and disc when the lower race is stopped;

b) The upper race and disc stops at a time t_{max} determined by testing and the measured cumulative position angle $\varphi_{2,max}$ corresponds with the value given by the equation: $\varphi_2(t_{max}) = \varphi_{2,max}$.

For every experiments the values for t_{max} and $\varphi_{2,max}$ were obtained by analyzing the records realized with the camera.

4. Experimental results

In Fig. 2 are presented the experimental values for total friction torque determined by using Eq. 11 for five rotational speeds from 100 to 300 rpm. The quantity of the grease was 0.018 grams on each race.

On the same diagram is presented with red line the theoretical total friction torque Tz calculated by Eq. 5 where we used for hydrodynamic force FR the Houpert's transition force FR_{Trans_H} described by Eq. 6. It can be mentioned that to calculate the force FR_{Trans_H} was used the dynamic viscosity of the base oil of the SSX grease η_0 = 1Pas (equivalent of approx. 1000 mm^2/s). Also, to evidence the effect of the lubricant on the total friction torque at very low axial load, in Fig. 2 are presented the total friction torque for modified thrust ball bearing with three balls operating in dry conditions.

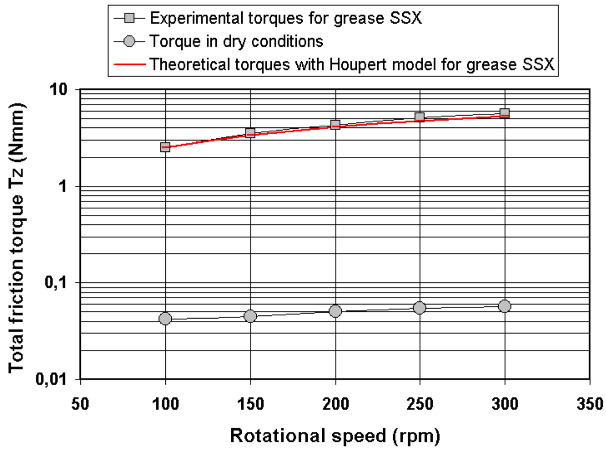

Fig. 2: *The variation of total friction torque with rotational speed, experimentally and theoretically determined*

5. Conclusions

1. The spin – down methodology was used to evaluate experimentally the friction torque in a modified thrust ball bearing having only 3 balls, loaded with very low axial force and lubricated with synthetic SSX grease.

2. The experimental values obtained for friction torque are in good correlation with the theoretical model developed in Ref. [5] based on the Houpert's hydrodynamic transition force FR_{Trans_H}. The Houpert's hydrodynamic transition force FR_{Trans_H} was calculated bu using base oil viscosity of the grease.

6. References

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