Friction torque in thrust ball bearings grease lubricated

G Ianuș 1, A C Dumitrașcu1, V Cârlescu1 and D N Olaru1

1Mechanical Engineering, Mechatronics and Robotics Department,“Gheorghe Asachi”
Technical University of Iaşi, Romania

E-mail: gianus2002@yahoo.com

Abstract: The authors investigated experimentally and theoretically the friction torque in a modified thrust ball bearing having only 3 balls operating at low axial load and lubricated with NGLI-00 and NGLI-2 greases. The experiments were made by using spin-down methodology and the results were compared with the theoretical values based on Biboulet&Houpert’s rolling friction equations. Also, the results were compared with the theoretical values obtained with SKF friction model adapted for 3 balls. A very good correlation between experiments and Biboulet & Houpert’s predicted results was obtained for the two greases. Also was observed that the theoretical values for the friction torque calculated with SKF model adapted for a thrust ball bearing having only 3 balls are smaller that the experimental values.

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 between balls and cage, pivoting friction in the balls-race contacts, drag friction between balls and lubricant, micro-sliding friction in rolling contacts. For a thrust ball bearing without sealing system and operating with small quantity of oil the SKF methodology consider that the total friction torque $T_z$ can be determined as a sum of two components: the rolling component $M_{rr}$ and the sliding component $M_{sl}$:

$$ T_z = M_{rr} + M_{sl} \quad (1) $$

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

$$ M_{rr} = 1.06 \cdot 10^{-6} \cdot d_m^{1,83} \cdot Fa^{0,54} \cdot (\nu \cdot n)^{0,6} \quad [N\cdot mm] \quad (2) $$

where $d_m$ is expressed in mm, $Fa$ is expressed in N, $n$ is expressed in rpm and $\nu$ is expressed in mm$^2$/s.

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

$$ M_{sl} = 1.6 \cdot 10^{-2} \cdot d_{m}^{0,05} \cdot Fa^{4/3} \cdot \mu_{sl} \quad [N\cdot mm] \quad (3) $$

The friction coefficient $\mu_{sl}$ have values between 0.002 and 0.1 function 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 with a rotational speed ranging between 500 rpm and 2000 rpm lubricated with different greases. They compared their results with those calculated with SKF methodology, considering the viscosity of the base oil used for the grease. The differences between measured friction torques and calculated friction torques with SKF model were used by Cousseau et al. [2] to determine specific values for the friction coefficient $\mu_s$ as function of the grease type and rotational speed.

Ianuş et al. [3] investigated the friction torque in a 51205 modified thrust ball bearing having only 3 balls and without cage, by using small quantity of a synthetic grease as lubricant having about kinematics viscosity of base oil of about 1000 mm$^2$/s at the temperature of 24°C. Using spin-down methodology the authors obtained the experimental friction torque’s values for rotational speed between 100 and 300 rpm and a normal load on each ball of about 1.42 N. These experimental values were compared with theoretical values determined by using Biboulet & Houpert transition IVR – EHD (IsoViscousRigid – ElastoHydroDynamic) equations [4] applied for base oil of the grease and good agreement was obtained.

By using the SKF equations (1) – (3) adapted for the operating conditions in the experiments the authors obtained lower values of friction torque for 51205 modified thrust ball bearing than the experimental values. Also, for low axial loads Bălan et al [5] experimentally determined the total friction torque in a 51205 thrust ball bearing operating between 100 and 400 rpm, and lubricated with mineral oils with kinematic viscosities of 350 mm$^2$/s and 60 mm$^2$/s. The authors observed that the experimental values for total friction torques are higher than the values calculated with SKF methodology and are in good agreement with Biboulet & Houpert’s equations [4]. In reference [5] Balan et al. demonstrated that for low axial load $F_a$ applied to 51205 thrust ball bearing, the component $M_{sl}$ from the equation (1) can be neglected by comparing to the component $M_{rr}$ that means the dominant influence of the oil viscosity $\nu$ and rotational speed $n$ on the total friction torque.

In the present paper the authors carried out experimental investigations for total friction torque by using the spin – down methodology for a modified 51205 thrust ball bearing having only 3 balls, without cage, loaded with an axial load of 4.26 N and lubricated with small quantities of two types of grease having consistency corresponding to NGLI-00 and NGLI-2, respectively. The experiments were realized with small quantity of greases in the range of rotational speed between 60 and 300 rpm. The absence of the cage offers the possibility to evaluate only the friction between the three balls and the two races.

### 2. Experimental procedure

The modified 51205-thrust ball bearing consists in replacing of the cage having 12 balls with 3 balls only, equally spaced on the circumference. The lower race is fixed on the rotating table. On the upper race is attached a disc and the weight of the disc and of the upper race consists in 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 to the lower race until both upper race and the attached disc reach a angular speed equal to that of lower race. In this moment the rotational table including the lower race is suddenly stopped and the upper race with disc starts to decelerate during a period time $t_{\text{max}}$ depend on the friction from all the six ball-race contacts.

By using white marks traced on the lower race and on the disc, the angular position of the two rotating elements can be visualized with a video camera. The images captured by the camera were recorded on the computer in real time and subsequently processed with an adequate program.

The experiments were conducted by using the Tribometer CETR UMT-2 from the Tribology Laboratory. In figure 1 is presented the modified 51205 thrust ball bearing and figure 2 presents the experimental set-up.
The geometrical parameters of the balls and races are the following: ball diameter $d = 7.938 \text{ mm}$ (5/16”), curvature radius of raceways $R_c = 4.2 \text{ mm}$, average radius of the races $r = 18 \text{ mm}$, average roughness for races $Ra_r = 0.065 \mu\text{m}$ and the average roughness of the balls was $Ra_b = 0.033 \mu\text{m}$.

In the experiments were used two types of greases. First grease was MOL Liton 00 having consistency grade NLGI-00 and the viscosity of the base oil at $40^\circ\text{C}$ of $40 \text{ mm}^2/\text{s}$. The second grease was MOL Alubia AK 2G having consistency grade NLGI-2 and the base oil viscosity at $40^\circ\text{C}$ of $150 \text{ mm}^2/\text{s}$. The laboratory temperature during the experiment was between $23^\circ\text{C}$ and $24^\circ\text{C}$.

The normal load on each ball $Q = G/3$, where $G = 4.26 \text{ N}$, the rotational speeds for upper race and disc, in rpm, were 60, 90, 120, 150, 180, 210, 240, 270 and 300 respectively.

Additionally, tests in dry conditions for the same load and speeds were made, to put in evidence the influence of the grease on the friction torque.

3. **Analytical models used for experimental determination of the friction torque**

In the deceleration process of the upper race can be use the dynamic balance of the moments acting on the rotating system:
where \( J \) is the moment of inertia of the ensemble formed by the upper race and disc and \( T(z(\omega_2)) \) is the total friction torque developed by friction generated in all 6 ball-race contacts as a function of angular speed of the upper disc \( \omega_2 \).

### 3.1. The model for lubricated conditions

In the lubricated conditions, for very low normal loads, Bălan et al. [5] demonstrated that the total friction torque \( T(z(\omega_2)) \) can be expressed only as a function of hydrodynamic forces \( FR \) developed in the ball race contacts, which, at their turn, depend on the angular speed of the upper race, \( \omega_2 \):

\[
T(z(\omega_2)) = 6 \cdot r \cdot FR(\omega_2) \tag{5}
\]

In a lubricated contact ball-race, depending on the geometry, lubricant viscosity, speed and normal load the lubricant regime can be a combination of IsoViscousRigid (IVR) regime and ElastoHydroDynamic (EHD) regime. Biboulet & Houpert [4] propose a hydrodynamic transition relation for \( FR \) by including both IVR and EHD regime according to following equation:

\[
FR_{Trans} = \frac{FR_{IVR} - FR_{EHD}}{1 + \frac{M}{6.6}} + FR_{EHD} \tag{6}
\]

The hydrodynamic forces \( FR_{IVR} \) and \( FR_{EHD} \) are described by the following equations [4]:

\[
FR_{IVR} = 2.9766 \cdot E^* \cdot R_x^2 \cdot k^{0.3316} \cdot W^{1/3} \cdot U^{2/3} \tag{7}
\]

\[
FR_{EHD} = 7.5826 \cdot E^* \cdot R_x^3 \cdot k^{0.4055} \cdot W^{1/3} \cdot U^{3/4} \tag{8}
\]

The parameter \( M \) is included by Biboulet & Houpert to take into consideration the transition from IVR to EHD lubrication regimes and it is calculated with the following equation [4]:

\[
M = 0.5549 \cdot k^{-0.6020} \cdot W \cdot U^{-0.75} \tag{9}
\]

The following parameters were included in equations (7) – (9): \( E^* \) is the equivalent Young modulus of the balls and races (for steel balls and races \( E^* = 2.3 \cdot 10^5 \text{ Pa} \)), \( R_x \) is the equivalent radius in the rolling direction (for thrust ball bearing \( R_x = d/2 \)), \( k = R_y/R_x \) and \( R_y \) is the equivalent radius of curvature in the transverse direction expressed by:

\[
R_y = \frac{R_x \cdot d}{2 \cdot R_x - d} \tag{10}
\]

where \( U \) is the dimensionless speed parameter and \( W \) is the dimensionless load parameter. These are described by the following equations:

\[
U = \frac{\eta_0 \cdot v}{E^* \cdot R_x} \tag{11}
\]

\[
W = \frac{Q}{E^* \cdot R_x^2} \tag{12}
\]

where \( \eta_0 \) is the dynamic viscosity of base oil in Pa·s at the operating temperature and \( v = (v_1 + v_2)/2 \) is the average entrainment speed in ball-race contact expressed in m/s.

Assuming pure rolling of the balls over the races, the average entrainment speed \( v \) in the ball - race contacts is given by:
\[
v = \frac{\omega_2 \cdot r}{2}
\]  
\(13\)

For a given geometry, lubricant viscosity and normal load, the hydrodynamic transition force \(F_{R_{\text{Trans}}}\) can be expressed only as a function of angular speed \(\omega_2\) and according to equation (5) it can be use following relation for the total friction torque \(T_z(\omega_2)\) [5]:

\[
T_z(\omega_2) = K^* \cdot \omega_2^n
\]  
\(14\)

where \(K^*\) does not depend on angular speed and the exponent \(\alpha < 1\).

The dynamic equation (4) becomes:

\[
J \cdot \frac{d\omega_2}{dt} + K^* \cdot \omega_2^n = 0
\]  
\(15\)

Bălan et al. [5, 6] solved analytically equation (15) and the following relations for variation of the angular speed of the upper race \(\omega_2(t)\) and angular position of the upper race \(\varphi_2(t)\) were obtained:

\[
\omega_2(t) = \left[ \omega_{2,0}^{1-n} - \frac{K^* \cdot (1-n) \cdot J}{t} \right]^{\frac{1}{1-n}}
\]  
\(16\)

\[
\varphi_2(t) = \frac{J}{K^* \cdot (2-n) \cdot \omega_{2,0}^{2-n}} \left[ \omega_{2,0}^{1-n} - \frac{K^* \cdot (1-n) \cdot J}{t} \right]^{\frac{2-n}{1-n}}
\]  
\(17\)

The values for \(K^*\) and \(\alpha\) were determined by solving the nonlinear equations (16) and (17) with following two data obtained in the experiments:

(i) At the initial moment \(t = 0\), the angular speed of the upper race is that measured when the lower race was stopped, \(\omega_2(0) = \omega_{2,0}\);

(ii) The total angle of rotation during the deceleration until the complete stop of the upper race \(\varphi_{2,\text{max}}\) is obtained during the time \(t_{\text{max}}\) and following equation was used: \(\varphi_{2}(t_{\text{max}}) = \varphi_{2,\text{max}}\). The values of \(t_{\text{max}}\) and \(\varphi_{2,\text{max}}\) are obtained by analyzing of the video recording;

3.2. The model for dry friction conditions

In the absence of the lubricants between the three balls and the races the total friction torque \(T_z\) does not depend on the rotational speed [6, 7] and equation (4) becomes:

\[
J \cdot \frac{d\omega_2}{dt} + T_z = 0
\]  
\(18\)

Equation (18) can be easily integrated twice, resulting the following relations for variation of the angular speed \(\omega_2(t)\) and the angular position \(\varphi_2(t)\):

\[
\omega_2(t) = \omega_{2,0} - \frac{T_z}{J} \cdot t
\]  
\(19\)

\[
\varphi_2(t) = \omega_{2,0} \cdot t - \frac{T_z}{2 \cdot J} \cdot t^2
\]  
\(20\)
During the deceleration process of the upper race, the angular speed \( \omega_2 \) decreases from the initial value \( \omega_{2,0} \) to zero during a period of time \( t_{\text{max}} \) and a total angular position \( \varphi_{2,\text{max}} \) has been cumulated by the upper race in this time.

From equations (19) and (20) results following relation for total friction torque \( T_z \):

\[
T_z = \frac{2 \cdot J \cdot (\omega_{2,0} \cdot t_{\text{max}} - \varphi_{2,\text{max}})}{t_{\text{max}}^2}
\]

(21)

### 3.3. Application of the SKF friction torque model

To compare the proposed methodology with the SKF friction torque model, equations (1) - (3) have been calculated for the two types of greases assuming that:

(i) The kinematic viscosity was considered that of the base oil of each grease and at laboratory temperature (23³C- 24³C);

(ii) The axial load was calculated by relation \( F_z = z \cdot Q \), where \( z \) is the number of the balls in a standard 51205 thrust ball bearing (for standard 51205 ball bearing \( z = 12 \)). By imposing in equations (2) and (3) the axial force \( F_z = z \cdot Q \), for each ball in standard ball bearing will be a normal force \( Q \) similar to our experiments.

(iii) To obtain the only the influence of the balls on total friction torque, the calculated friction torque with the previous assumptions was divided by \( z/4 \), neglecting the influence of the cage.

### 4. Experimental results

In figure 3 are presented the experimental data for total friction torque determined by solving Equations (16) and (17) for grease MOL Liton 00. A small quantity of grease was used on each race.

On the same graph is presented with red line the theoretical total friction torque \( T_z (\omega_2) \) calculated by equations (5) – (13) where the dynamic viscosity of the base oil was of 0.085 Pa·s at the laboratory temperature. In the same figure are presented the variation of the friction torque determined by SKF model adapted for 3 balls and the friction torque experimentally determined for dry conditions.

**Figure 3.** The variation of total friction torque with rotational speed, experimentally and theoretically determined for MOL Liton 00 grease.
In figure 4 are presented the experimental values for total friction torque determined by solving Equations (16) and (17) for grease MOL Alubia AK 2G. Small quantity of grease was used on each race.

On the same diagram is presented with red line the theoretical total friction torque $T_z (\omega_2)$ calculated by equations (5) – (13) where the dynamic viscosity of base oil was of 0.370 Pa·s at the laboratory temperature. Also, on the same figure are presented the variation of the friction torque determined by SKF model adapted for 3 balls and the friction torque experimentally determined for dry conditions.

By comparing the experimental results with theoretically predicted results in the conditions of very low axial load the following remarks can be made:

(i) For both lubricant greases the total friction torque experimentally determined are in good correlation with the theoretical model for friction torque based on the Biboulet & Houpert’s transition IVR-EHD equation for hydrodynamic force $FR$ by considering the base oils viscosity used in the greases.

(ii) The presence of a small quantity of grease in the balls – race contacts leads to important increases of the total friction torque compared with the dry conditions (increases with one to two orders of magnitude depending of the speed and base oil viscosity).

(iii) The SKF friction torque methodology applied to the modified thrust ball bearing leads to smaller values for the total friction torque compared with the experiments.

Figure 4. The variation of total friction torque with rotational speed, experimentally and theoretically determined for MOL Alubia AK 2G grease.

5. Conclusions
The authors investigated experimentally and theoretically the friction torque in a modified thrust ball bearing having only 3 balls operating at low axial load and lubricated with two types of greases having the consistency grade NGLI-00 and NGLI-2. Also, the authors investigated experimentally the friction torque in the modified thrust ball bearing operating in dry conditions. The experiments were realized on the CETR UMT-2 Tribometer by using spin-down method and the results obtained for lubricated conditions were compared with the theoretical values calculated using Biboulet & Houpert’s rolling
friction equation applied for base oil of the two greases. Also, the results were compared with the theoretical values obtained by the SKF friction torque model adapted for a thrust ball bearing having only 3 balls. A very good correlation between experimental values and theoretical values obtained by using the Biboulet & Houpert’s methodology has been obtained for the two greases. It was also observed that the theoretically predicted values for the friction torque calculated with SKF model adapted for a thrust ball bearing having only 3 balls are smaller than the experimental values.

6. References
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