Influence of lubricant on the friction in an angular contact ball bearing under low load conditions

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Abstract. The presence of the lubricant in a ball-race contact has two principal effects on friction: reduces the sliding friction, as a result of developing a film thickness between the contact surfaces and increases the rolling friction, as result of developing a hydrodynamic rolling friction. The influence of the hydrodynamic rolling friction on the torque of a modified 7205B angular contact ball bearing has been theoretically and experimentally studied. The experiments demonstrated that, for very low contact loads between balls and races, the effect of the hydrodynamic rolling force is dominant in the total friction torque. The experiments have been validated by using Biboulet's transition model for hydrodynamic rolling forces developed in ball-race contacts.

Nomenclature

d_m	Mean diameter of the angular contact ball bearing [m]	$d_m = \frac{(D+d)}{2}$
d_b	Ball diameter [m]	
dT	Friction torque generated by a single ball [N·m]	
d	Inner diameter of the angular contact ball bearing [m]	
f	Ball-race conformity [-]	
k	Radius ratio [-]	$k_{i,o} = \frac{2 \cdot f_{i,o}}{(2 \cdot f_{i,o} - 1) \cdot (1 \mp \gamma)}$
n_o	Outer race rotational speed [rpm]	
v	Tangential speed in the rolling direction [m/s]	$v = \frac{\pi \cdot n_o \cdot (1-\gamma^2)}{30} \cdot \frac{d_m}{4}$
D	Outer diameter of the angular contact ball bearing [m]	
E^*	Equivalent Young's modulus [Pa]	
MC	Curvature friction moment [N·m]	
MER	Rolling resistance moment [N·m]	
MP	Pivoting moment [N·m]	
η_o	Oil dynamic viscosity [Pa·s]	
Q	Normal load on contact ellipse [N]	$Q = \frac{F_a}{3 \cdot \sin \alpha}$
$R_{i,o}$	Radius of the inner and outer races [m]	$R_{i,o} = \frac{d_m \cdot (1 \mp \gamma)}{2}$
R_X	Equivalent radii in the rolling direction [m]	$R_{Xo,i} = \frac{d_b}{2} \cdot \left(1 \pm \frac{d_b \cdot \cos \alpha}{d_m}\right)$
R_y	Equivalent radius of the curvature perpendicular to the rolling direction [m]	
T	Friction torque of the bearing [N·m]	

α	Contact angle of the angular contact ball bearing [rad]	
α_p	Piezo-viscosity coefficient of oil [Pa^{-1}]	
γ	Dimensionless parameter [-]	$\gamma = \frac{d_b \cdot \cos(\alpha)}{d_m}$
μ	Friction coefficient [-]	
ω_0	Angular speed of the outer race [rad/s]	
ω_b	Angular speed of the balls [rad/s]	

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o	Outer
i	Inner
s	Sliding
p	Pivoting
x, y	Contact ellipse coordinate

1. Introduction

In a ball-race contact, the power loss is due to many factors, including both losses from sliding friction on the contact ellipse and from rolling friction. Houpert [1] and [2] described all friction forces and resistant moments in a ball-race contact from an angular contact ball bearing and proposed a general equation for the total friction torque, including both losses generated by sliding and by rolling. One of the important resistant force for a low loaded ball-race contact is the hydrodynamic rolling force. Biboulet and Houpert [3] proposed a complex model for the hydrodynamic rolling force, by including both IVR (Isoviscous Rigid) and EHD (Elasto-Hydrodynamic) effects. The new Biboulet&Houpert's model has been validated by Balan et al. [4], [5] and [6] for a modified thrust ball bearing, having only three balls, without cage and lubricated with mineral oil. Ianuş et al. [7] also validated Biboulet&Houpert's model, applied to a grease lubricated low loaded thrust ball bearing.

By using a 7205B modified angular contact ball bearing, with 3 balls and without cage, Popescu et al. [8] experimentally determined the total friction torque in dry conditions. All friction sources (sliding and rolling in the ball-race contacts) were included in the measured friction torque, except the effect of the lubricant.

In the present paper the authors determined both theoretically and experimentally the influence of the oil on the total friction torque, in a modified 7205B ball bearing with three balls and no cage. Considering two oils with different viscosities and a low axial load, the authors evidenced the important increasing of the total friction torque as result of hydrodynamic rolling forces generated in ball-race contacts. Also were verified two equations for hydrodynamic rolling forces resulted from the Biboulet&Houpert's model.

2. Theoretical model for friction torque

In an angular contact ball bearing, Houpert [1] and [2] proposed following general equation for the friction torque, generated by a single ball:

$$dT = 2 \cdot (FRo + FRi) \cdot \frac{Ro \cdot Ri}{dm} + \frac{MC_i \cdot R_o + MC_o \cdot R_i}{d_b} + \frac{MER_i \cdot R_o + MER_o \cdot R_i}{d_b} + \frac{MP_i \cdot (1+\gamma) + MP_o \cdot (1-\gamma)}{2} \cdot \sin(\alpha) \quad (1)$$

where $FR_{o,i}$ is the hydrodynamic rolling friction in ball-race contacts, $MC_{o,i}$ is the curvature friction moment developed in ball-race contacts, $MER_{o,i}$ is the rolling resistance moments in ball-race contacts and $MP_{o,i}$ is the pivoting moments. According to the ball bearing geometry, the friction coefficient on ball-race contacts and normal load, as well as the moments *MC, MER* and *MP* can be determined according to Houpert's relations [1]. The geometrical parameters of the angular contact ball bearing (d_m, d_b, α, $R_{o,i}$ and γ) are defined in the nomenclature.

In the absence of lubricant, the moments *MC* and *MP* can be determined by imposing a constant friction coefficient on ball-race contacts and MER depends only on the normal load and ball-race geometry [1]. In the presence of lubricant, two important parameters must be included in equation (1): the hydrodynamic rolling forces FR_o and FR_i.

As it was demonstrated in previous research [4], [5] and [6], the transition model for hydrodynamic rolling friction force, proposed by Biboulet and Houpert [3], represents the best solution in low loaded ball-race contacts. Therefore, the transition model includes both IVR and EHD regimes, which usually exist in lubricated ball bearings but in different percentages [6].

The transition model for the hydrodynamic rolling force described by Biboulet and Houpert [3] is characterized by the following general equation:

$$FR_{Trans} = \frac{FR_{IVR} - FR_{EHL}}{1 + \frac{M}{6.6}} + FR_{EHL} \tag{2}$$

The two components of the hydrodynamic rolling forces FR_{IVR} and FR_{EHL} correspond to the IVR and EHD lubrication regimes, respectively. In [4] the two transition equations have been compared for rolling friction:

(i) Houpert's transition model which includes in equation (2) the Houpert's relations for FR_{IVR} and FR_{EDL} determined by the following [4]:

$$FR_{IVR\_H} = 1.213 \cdot E^* \cdot R_x^2 \cdot k^{0.358} \cdot U^{0.636} \cdot W^{0.364} \tag{3}$$

$$FR_{EHL\_H} = 2.765 \cdot E^* \cdot R_x^2 \cdot k^{0.35} \cdot U^{0.656} \cdot W^{0.466} \cdot G^{0.022} \tag{4}$$

(ii) Biboulet's transition model which includes in equation (2) the Biboulet&Houpert's relations for FR_{IVR} and FR_{EDL} determined by the following [3]:

$$FR_{IVR\_B} = 2.9766 \cdot E^* \cdot R_x^2 \cdot k^{0.3316} \cdot W^{1/3} \cdot U^{2/3} \tag{5}$$

$$FR_{EHL\_B} = 7.5826 \cdot E^* \cdot R_x^2 \cdot k^{0.4055} \cdot W^{1/3} \cdot U^{3/4} \tag{6}$$

The transition parameter M is described by Biboulet and Houpert [3] with the following equation:

$$M = 0.5549 \cdot k^{-0.6029} \cdot W \cdot U^{-0.75} \tag{7}$$

Therefore, in the proposed simulation program two models for the transition of the hydrodynamic rolling force, FR_{Trans} were considered:

Houpert's transition model:

$$FR_{Trans\_H} = \frac{FR_{IVR\_H} - FR_{EHL\_H}}{1 + M/6.6} + FR_{EHL\_H} \tag{8}$$

Biboulet's transition model:

$$FR_{Trans\_B} = \frac{FR_{IVR\_B} - FR_{EHL\_B}}{1 + M/6.6} + FR_{EHL\_B} \tag{9}$$

The parameters included in equations (3) - (7) are defined as follows:

U is the dimensionless speed parameter:

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

W is the dimensionless load parameter:

$$W = \frac{Q}{E^* \cdot R_x^2} \tag{11}$$

G is the material parameter: $$G = \alpha_p \cdot E^* \tag{12}$$

in which η_o is the oil dynamic viscosity in Pa·s, at the operating temperature of the contact; α_p is the piezo-viscosity coefficient of oil in Pa^{-1}, $v = (v_1 + v_2)/2$ is the average entrainment speed in m/s; E^* is the equivalent Young modulus of the two elements in contact (for steel ball – race contact, $E^* \approx 2.3 \cdot 10^{11}$ Pa), k is the radii ratio R_y/R_x, with R_y defined as the equivalent radius of curvature in the *y* direction (perpendicular to the rolling direction) and R_x the equivalent radius in the rolling direction.

3. Simulation of the friction torque for a modified angular ball bearing

The theoretical model has been applied to a modified 7205B angular contact ball bearing, having the following geometrical parameters: d=0.025 m; D=0.052 m; d_b=0.0079375 m; d_m=0.0385 m, f_i=0.515; f_e=0.522 and a nominal contact angle α=15 degrees. Also, the roughness parameters on races and balls are: R_{qr}=0.06 µm and R_{qb}=0.03 µm. Three balls have been distributed at 120 degrees between the inner and outer race and axially loaded with a 5 N weight cylinder.

Figure 1 presents a view of a section through the modified 7205 angular contact ball bearing.

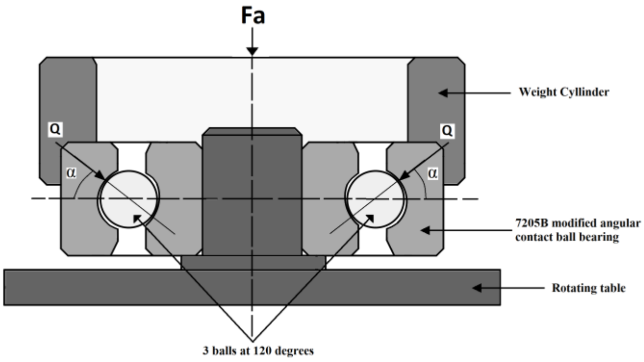

Figure 1. View of a section through the modified 7205B ball bearing assembly

The simulation of the total friction torque has been performed for the kinematic conditions generated by the spin-down method, in which the inner race is rotating with a given rotational speed, between 100 and 400 rpm. After some time, as result of the friction in all six ball races contacts, the outer race and attached cylinder rotates with a rotational speed similar to the inner race rotational speed. In that moment, the inner race is stopped and all kinetic energy of the outer race and cylinder are dissipated by friction, in all six ball-races contacts, in a deceleration process during a short or long time, depending on the friction resistance in the ball-races contacts.

The simulation of the total friction torque has been realized for two mineral oils, having the dynamic viscosity of η of 0.085 Pa·s and 0.35 Pa·s at laboratory temperature (22-24^o C) and with the piezo-viscosity coefficient $\alpha_p = 2.2\cdot10^{-8}$ Pa^{-1}.

The friction torque generated by a single ball and given by equation (1) can be divided in two components: a component generated by hydrodynamic rolling forces dT_H and a component generated by sliding and rolling friction dT_{S+R}:

$$dT=dT_H+dT_{S+R} \tag{13}$$

where the two components are given by the following equations:

$$dT_H = 2\cdot(FRo+FRi)\cdot\frac{Ro\cdot Ri}{dm} \tag{14}$$

$$dT_{S+R} = \frac{MC_i\cdot R_o+MC_o\cdot R_i}{d_b}+\frac{MER_i\cdot R_o+MER_o\cdot R_i}{d_b}+\frac{MP_i\cdot(1+\gamma)+MP_o\cdot(1-\gamma)}{2}\cdot sin(\alpha) \tag{15}$$

To determine the component dT_{S+R}, the friction coefficient on the contact ellipses has been calculated with equation (16) as function of the lubricant parameter λ. The component dT_{S+R} is presented in detail in reference [9], which was determined for dry condition by imposing a constant friction coefficient. The friction coefficient is determined for the inner and outer ball-races contacts by the following relation [4]:

$$\mu_s = \mu_e\cdot0.82\cdot{\lambda_{i,o}}^{0.28}+\mu_a\cdot\left(1-0.82\cdot{\lambda_{i,o}}^{0.28}\right) \tag{16}$$

μ_a is the friction coefficient for limits conditions, usually μ_a=0,11, μ_e is the friction coefficient in EHD lubricant film, usually considered as μ_e=0.05 and the lubricant parameter $\lambda_{i,o}$ is calculated with relation:

$$\lambda_{i,o} = \frac{h_{min_{i,o}}}{\sqrt{{R_{qr}}^2+{R_{qb}}^2}} \tag{17}$$

The minimum film thickness on inner and outer races has been evaluated with Hamrock&Dowson's equations [1] and [2]:

$$h_{min_i} = 3.63\cdot R_{xi}\cdot {U_i}^{0.68}\cdot G^{0.49}\cdot {W_i}^{-0.073}\cdot\left(1-0.61\cdot e^{-0.68\cdot k_i}\right) \tag{18}$$

$$h_{min_o} = 3.63\cdot R_{xo}\cdot {U_o}^{0.68}\cdot G^{0.49}\cdot {W_o}^{-0.073}\cdot\left(1-0.61\cdot e^{-0.68\cdot k_o}\right) \tag{19}$$

The variation of the minimum film thickness and lubricant parameter for inner and outer ball-races contact have been calculated for both oil viscosities between 100 and 400 rpm. The lubricant parameters obtained have values between 1 and 2.8, for the oil with the viscosity of 0.085 Pa·s. Also, the lubricant parameters obtained have values between 3 and 7, for the oil with the viscosity of 0.35 Pa·s. The correspondent friction coefficients varied between 0.06 and 0.05.

Important differences between Houpert's transition model and Biboulet's transition model can be observed for both oil viscosities. These variations can be explained by the differences between equation (3) – (4) and (5) – (6). The hydrodynamic rolling friction variation as function of rotational speed for the two oil viscosities are presented in figures 2 and 3. Equations (8) and (9) have been used for Houpert's transition model and Biboulet's transition model, respectively.

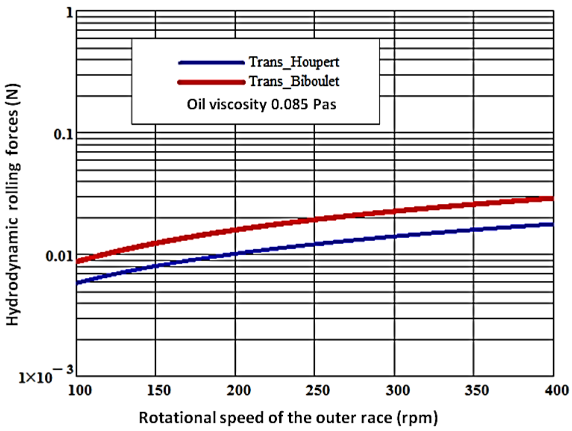

Figure 2. Variation of the hydrodynamic force FR_{Trans} as function of the rotational speed for an oil viscosity of 0.085 Pa·s.

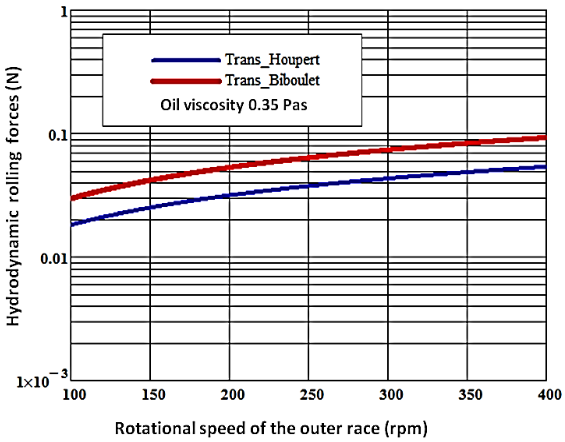

Figure 3. Variation of the hydrodynamic force FR_{Trans} as function of the rotational speed for an oil viscosity of 0.35 Pa·s.

In the absence of the cage, the total friction torque in the modified 7205B ball bearing is obtained by following equation:

$$T = 3 \cdot dT \tag{20}$$

The total friction torques have been calculated for the two oils using both simulation models for transition hydrodynamic rolling forces. These variations of total friction torque are presented in figures 4 and 5.

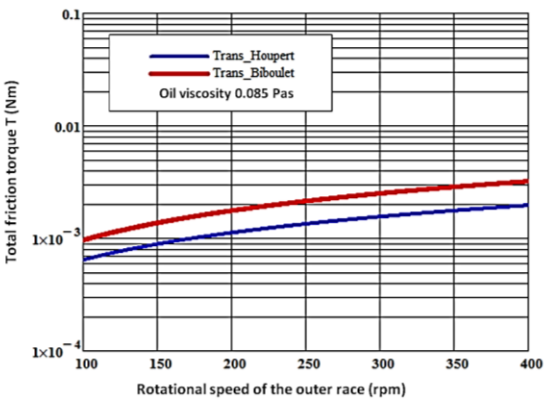

Figure 4. Variation of the total friction torque as function of the rotational speed for oil viscosity of 0.085 Pa·s.

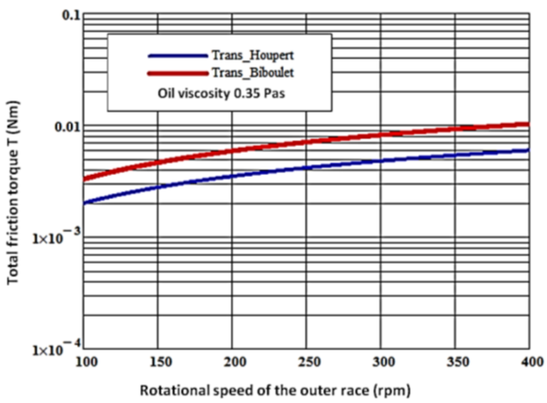

Figure 5. Variation of the total friction torque as function of the rotational speed for oil viscosity of 0.35 Pa·s.

The next step in our approach consist in validating the model by use of the experiments to adequate the transition hydrodynamic rolling force model for the modified 7205 ball bearing.

4. Experimental validation of the theoretical model

The experiments were performed in the Tribology Laboratory of the Mechanical Engineering Faculty of Iasi. The modified 7205B angular contact ball bearing presented in figure 1 has been tested. The inner ring was fixed to the rotational table of the Tribometer CETR UMT2, so that it rotates with the table (figure 7). On the outer ring a steel cylindrical support was fixed, imposing an axial load of 5.0 N, with an inertial moment $J=4.354\cdot10^{-4}$ kg·m^2. No cage was used in the experiments. A video camera recorded with 60 frames/second the rotation of the outer race in the deceleration process. The video captured by the camera was recorded on a computer in real time and subsequently processed with a specialized program. For every rotational speed, two experimental parameters have been determined: the total angular position cumulated by the outer race in the decelerating process, from an initial rotational speed to stop, expressed in radians and the time during the deceleration process, expressed in seconds.

In the deceleration process of the outer race, the dynamic balance of the moments acting on the rotating system has been used:

$$J \cdot \frac{d\omega_o}{dt} + T(\omega_o) = 0 \tag{21}$$

where J is the inertia moment of the outer race and cylinder and $T(\omega_o)$ is the total friction torque developed in the modified ball bearing, as a function of the angular speed of the outer race ω_o. For a low axial load, the hydrodynamic component of the torque is dominant and the total friction torque $T(\omega_o)$ can be approximated by a formal relation: $T(\omega_o) = K^*\cdot\omega_o^n$, where K^* and n was determined by integration of equation (21) and according to the measured parameters (total angular position of the outer race and deceleration time) for each rotational speed. Details of the integration of equation (21) are presented in [4]. A general view of the experimental equipment is presented in figure 6 and some details are presented in figure 7.

Figure 6. General view of the CETR UMT 2 Tribometer.

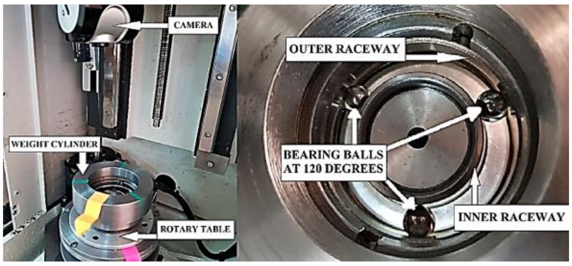

Figure 7. Details of the Tribometer and 7205B modified angular contact ball bearing [8].

Some repeated tested have been performed for rotational speeds of the outer race between 100 and 400 rpm for both viscosity of 0.085 Pa·s and 0.35 Pa·s.

The experimental friction torques obtained by using the spin-down method have been indicated in figures 8 and 9. Also, the theoretical variation of the friction torque determined by the Biboulet's transition model was superposed on the experimental results in figure 8 and 9. A very good correlation has been obtained for both oil viscosities.

Comparing these results with the results obtained by Balan et al. in [4] (figures 10, 12 and 13) on the modified 51205 thrust ball bearing with similar oil viscosities, low axial load and comparable rotational speed, the following comment can be made: the total friction torques experimentally determined in [4] have been validated with Houpert's transition model for hydrodynamic rolling forces and not with Biboulet's transition model as in our experiments.

It can be explained by following important differences for the transition parameter M. Thus, for a modified 7205B ball bearing lubricated with oil having 0.085 Pa·s, the parameter M has values between 25 and 9 when the rotational speed varied between 100 and 400 rpm, while for the modified 51205 ball bearing, for similar oil and speeds, the parameter M varied between 7 and 2. Similarly, for oil viscosity of 0.35 Pa·s the parameter M varied between 7 and 2 for the modified 7205B ball bearing and between 2 and 0.9 for the modified 51205 ball bearing.

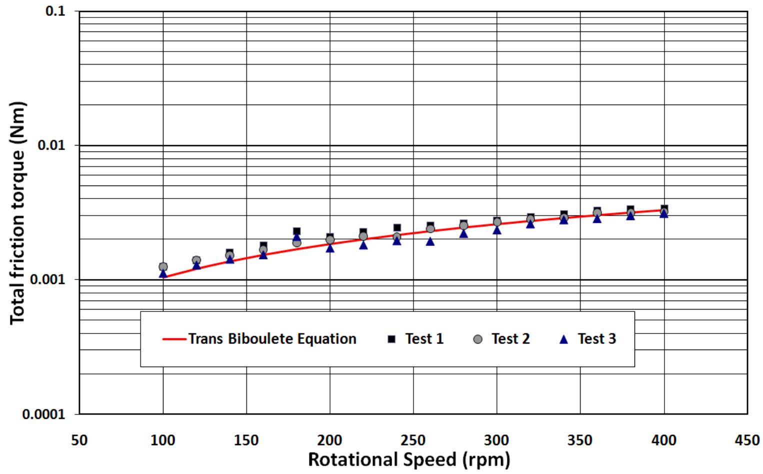

Figure 8. Variation of the theoretical and experimental friction torque for a modified 7205B ball bearing and for an oil viscosity of 0.085 Pa·s.

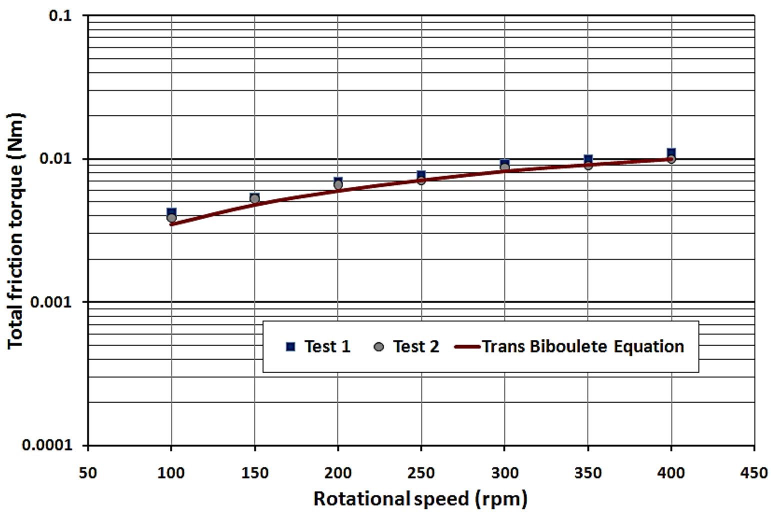

Figure 9. Variation of the theoretical and experimental friction torque for a modified 7205B ball bearing and for an oil viscosity of 0.35 Pa·s.

Supplementary the geometrical parameters which are included in the IVR and EHD rolling friction equations as Rx and k have higher values for the 7205B ball bearing than for 51205 ball bearing. Also, the normal load for each ball-race contact is 6.44N for the 7205B ball bearing and 1.42 N for the 51205 ball bearing.

All these differences lead to disparities between the total friction torque for the two modified ball bearings considered in this paper and in [4], even if the ball diameter, oil viscosity and rotational speed are similar and the applied loads are very close.

5. Conclusions

A complex model for determining the influence of the lubricant on the friction torque in a modified angular contact ball bearing has been theoretical simulated and experimentally validated.

The simulation was realised for an angular contact ball bearing, having only three balls, without cage and axially loaded with a very low load. Only the friction losses in ball-race contacts were evidenced.

Two transition models for hydrodynamic rolling force in ball-race contacts, Houpert's transition model and Biboulet's transition model obtained by combining different equations for IVR and EHD lubrication regimes have been simulated. Two different oil viscosities (0.085 Pa·s and 0.35 Pa·s) and a variation of the rotational speed between 100 and 400 rot/min have been used in the simulation.

The experiments were realised with spin-down methodology for various outer race rotational speed between 100 and 400 rpm for both oil viscosities and with an axial load of 5N.

The experimental values validated Biboulet's transition model for hydrodynamic rolling forces for both viscosities.

6. References

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