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Friction Torque in Miniature Ball Bearings †

Denis Cojocaru, Gelu Ianuș, Vlad Cârlescu, Bogdan Chiriac and Dumitru Olaru *

Mechanical Engineering, Mechatronics and Robotics Department, Technical University "Gheorghe Asachi" Iași, 700050 Iași, Romania; denis.cojocaru@academic.tuiasi.ro (D.C.); gelu.ianus@academic.tuiasi.ro (G.I.); vlad.carlescu@academic.tuiasi.ro (V.C.); bogdan.chiriac@academic.tuiasi.ro (B.C.)

* Correspondence: dumitru.olaru@academic.tuiasi.ro

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Abstract: The problem of estimation the friction torque in operating miniature ball bearings lubricated with oil or grease is a complex one. Generally, in an angular contact ball bearing (ACBB), various types of losses can appear including losses caused by kinematics in ball-race contacts (rolling, sliding and pivoting), losses between the cage and the balls and between the cage and the guiding race and losses generated by lubricant, especially at high speeds. In the miniature ACBB, the applied loads have generally low values, and some losses can be ignored. In these circumstances, the most important contribution to the increase in the losses in miniature ACBB is the presence of the lubricant. In normal rolling bearings, the lubricant has an important contribution to decrease the losses and increase the reliability in miniature ball bearing; the lubricant (oil or grease) leads to the increase in the losses compared to the dry or limit lubrication conditions. The catalogues of various rolling bearing companies have not provided more details referring to the friction losses in miniature ball bearings. In order to evaluate the total friction torque in the rolling bearings, some empirical complex relations are presented via the SKF methodology, which can be applied only to moderate and high loads applied to the rolling bearings. Other empirical relations are presented by the Schaeffler catalogue. Based on previous experiments, the authors determined the friction torque in a 7000C ACBB with the spin-down method. The experimental results were correlated with the results obtained via the theoretical model developed by Houpert for IVR lubrication conditions. The theoretical results evidenced that the hydrodynamic rolling resistance generated by the lubricant is the most important component of the friction torque for 7000C ACBB. The experimental and theoretical results were compared to the results obtained according to the SKF and Schaeffler relations. The experimental results and the results obtained with the Houpert model generally had higher values compared to the results obtained with the SKF and Schaeffler relations.

Keywords: miniature ball bearings; friction torque; spin-down method

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1. Introduction

Miniature radial or angular contact ball bearings (ACBBs) are supports for rotating spindles in various micro-mechanical and mechatronics systems. The estimation of the friction torque in a miniature ball bearing lubricated with grease and oils is dependent on the grease composition, lubricant viscosity, temperature, the geometry of the ball bearings and rotational speed.

The literature [1–4] or bearing companies catalogues such as the SKF [5] and Schaeffler [6] indicate the methodologies and relations needed to determine the friction torque in usually oil- or grease-lubricated rolling bearings. Harris and Kotzalas [1] propose a methodology to determine friction torque in an ACBB as a sum of two components: a component including the product between viscosity and rotational speed, and the second component is an empirical relation depending on the applied external forces and the type of rolling bearing.

Houpert [2,3] carried out a complex analysis of all friction sources in an ACBB including the hydrodynamic resistance of the oil in rolling process, the effect of curvature of the two rolling races, the effect of elastic hysteresis in rolling contacts and the effect of the ball-cage contact. The component of the hydrodynamic rolling resistance has been upgraded by Biboulet and Houpert [7].

Wu et al. [4] conducted a complex review including several methodologies to determine the friction torque in rolling bearings, both for oil and grease lubrication conditions, ranging between 1945 and 2018. Our methodology based on a spin-down method has been included among the methodologies referring to reference [8].

The SKF [5] proposes a complex methodology to evaluate the friction torque in all types of rolling bearings. The methodology includes, as an important parameter, the product between the viscosity (or the base oil viscosity of grease) and the rotational speed. Also, Schaeffler [6] proposes a methodology to evaluate the friction torque for low loads, depending on the value of the product between the viscosity of oil (or the base oil viscosity for grease) and the rotational speed of the ball bearings.

Some researchers realized experimentally studies that focused on lubrication with greases. The film thickness of the grease-lubricated rolling bearings has been experimentally evaluated by Zhang and Glovnea [8], measuring the electrical capacity of the balls and the two races for greases and oils. De Laurentis et al. [9] determined the friction and film thickness for eight types of greases with base oil viscosities between 18 and 420 mm^2/s on a ball-on-disc tribometer. The authors compared the experimentally determined friction coefficients for all types of greases and noticed that higher values of friction coefficients were obtained by lithium soap greases.

Vengudusamy et al. [10] determined that at low speed, the dominant layer in the film thickness is the grease soap, and by increasing the speed, the film thickness is generated by the base oil using the ball–on-disc tests. For the friction coefficient, the authors established an important difference between greases with a low viscosity of the base oil and greases with a high viscosity of the base oil if the speed increases over a limit.

Cousseau et al. [11] carried out a complex experimental study with a 51107 thrust ball bearing loaded with 7000 N with the rotational speed set between 500 and 2000 rot/min. The authors evidenced the role of the different grease composition on the friction torque and proposed a new methodology to determine the friction coefficient in lubricated bearings according to the friction coefficient μ_{EHD} using the SKF friction torque methodology [5] and experimental results.

Kanazawa et al. [12] experimentally determined the friction torque in a 51107 thrust ball bearing and in two 81107 and 81105 cylindrical rollers thrust bearings using both the complex Lithium thickener and the urea thickener greases, operating at 200–1800 rot/min and loaded with 2000 N. Considering the experimental values of the friction torques, the authors use the SKF methodology [5] and determined the friction coefficient in the tested bearings. In addition, the authors experimentally determined the film thickness for all tested greases via optical interferometry in a ball-on-glass disc rig.

An important conclusion of the above-mentioned experiments is that for low lubricant parameter λ, the friction coefficient in the greased contacts is lower than the friction coefficient obtained with the base oil of the greases. By increasing the speed which increases the lubricant parameter, the differences can be ignored.

Fischer et al. [13] determined the film thickness of two lithium greases and their corresponding base oils with a rolling speed between 1 mm/s to 4 m/s via optical interferometry using a ball-on-glass disc tribometer. The authors established that for low rolling speed, the grease film thickness is maintained at a higher level, compared to the film generated by the oil as result of thickener, which enters via the ball-race contact.

For an ACBB, some friction processes have differential contribution to the total friction torque and power losses. So, for normal and high loads, the friction generated via elastic rolling hysteresis in ball-race contacts and pivoting friction are dominant. For high-speed and moderate loads, the pivoting and lubricant friction are the parameters that limits operational speed by increasing the temperature and decreasing the film thickness in ball-race contacts which are the results of thermal effects and starvation.

For very low loads, the hysteresis and pivoting losses can be neglected, and the hydrodynamic rolling forces developed between the balls and races are the most important source for friction torque. Several experiments such as those achieved by Ianuş et al. [14] and Popescu et al. [15] evidenced the essential contribution of the hydrodynamic rolling forces on the friction torque, both for thrust and for ACBB operating with very low applied loads.

In the present paper, the authors compared the friction torque using the Houpert IVR methodology and the SKF and Schaeffler models for a 7000C ACBB loaded with 3.11 N for a rotational speed range of 100 to 700 rot/min. Important differences were obtained for the imposed viscosity.

To verify all three methods, the authors performed a substantial number of experiments and determined the friction torque for a 7000C ACBB using two types of greases and two types of oils. The tests were realized using the spin-down method, with an axial load of 3.11 N, and a rotational speed between 100 and 700 rot/min. The experimental results were compared across all three methods, applying the corresponding viscosity for each lubricant at the laboratory temperature.

The experimentally results, generally, correlated best with the Houpert IVR methodology [2,3].

2. Theoretical Model for Friction Torque

For an ACBB, Houpert [2,3] developed a complex methodology to determine the friction torque considering the following components: hydrodynamic rolling resistance, curvature effects, elastic rolling resistance, pivoting effects and additional braking effects caused by the cage. In the decelerating process, the inertial effect can also be included. So, the general Houpert IVR model applied for an ACBB considers the following equation for the total friction torque [2]:

$$T_Z = Z\cdot\left[2\cdot(FR_o + FR_i)\cdot\frac{R_o\cdot R_i}{d_m} + \frac{MC_i\cdot R_o + MC_o\cdot R_i}{d_b} + \frac{MER_i\cdot R_o + MER_o\cdot R_i}{d_b} + \left(\frac{MP_i\cdot + MP_o}{2}\right)\cdot\sin\alpha + \left(\frac{MB}{d_b} + \frac{FB}{2}\right)\cdot R_i\right] \quad (1)$$

In Equation (1), $FR_{o,i}$ are the hydrodynamic rolling forces in the ball-race contacts; $MC_{o,i}$ are the curvature friction moments developed in the ball-race contacts; $MER_{o,i}$ are the rolling resistance moments in the ball-race contacts as a result of elastic rolling hysteresis, and $MP_{o,i}$ are the pivoting moments between the balls and races on contact ellipses. MB and FB are the brake moments and contact forces developed in the ball-cage contacts, respectively.

A complex programme to determine the total friction torque in a 7000C ACBB was developed. The geometrical characteristics of the 7000C ACBB are evidenced in Figure 1, and their values are listed in Table 1. From Figure 1, it can be seen that the phenolic resin cage is guided along the inner ring.

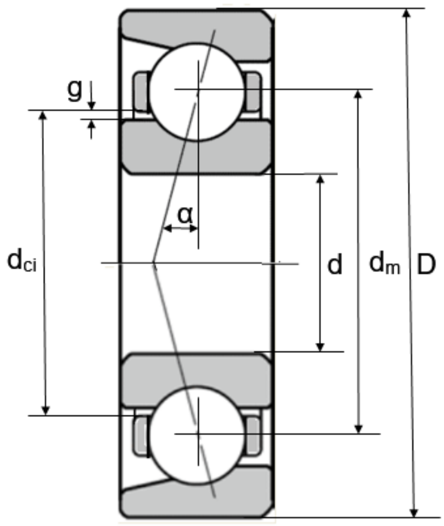


Figure 1. The main geometrical characteristics of the 7000C ACBB.

Table 1. Dimensional characteristics of the 7000C ACBB.

Dimensions of 7000C ACBB	Value	Unit Measurement
D—external diameter	26	mm
d—internal diameter	10	mm
d_m—pitch diameter $d_m = (D + d)/2$	18	mm
d_b—ball diameter	4.762	mm
α—contact angle	30	degrees
Z—number of balls	8	-
f_i, f_o—inner and outer races conformities	$f_i = f_o = 0.525$	-
g—the gap between phenolic resin cage and inner ring	$g = 0.15$	mm

All components of the total friction torque were determined for a rotational speed of the outer race between 100 and 700 rot/min; the axial load on ACBB was 3.11 N with a contact load between the balls and races of 0.763 N. The simulating programme was realized for both the two greases (SHELL Gadus S2 V100 2 and SKF LGHP 2/0.4) and for the two lubricant oils (engine oil SAE 5W-30 and transmission oil SAE T90 EP2). The most important characteristics of the lubricants are presented in Tables 2 and 3. The viscosities of the base oil for the two types of greases at the laboratory temperature were considered.

Table 2. Physical characteristics of the tested greases.

Properties	SKF LGHP 2/0.4	SHELL Gadus S2 V100 2
NLGI consistency	2–3	2
Soap type	Di-urea complex	Lithium Hydroxy stearate
Base oil type	Mineral	Mineral
Kinematic viscosity @40 °C (mm^2/s)	96	100
Kinematic viscosity @100 °C (mm^2/s)	10.5	11
Dropping point (°C)	240	180

Table 3. Physical characteristics of the tested oils.

Properties	Shell HELIX HX8 (SAE 5W-30)	SAE 90 EP2
Density at 15.6 °C (kg/m^3)	859	889
Kinematic viscosity @40 °C (mm^2/s)	75	225
Kinematic viscosity @100 °C (mm^2/s)	11	18

Equation (1) includes the five sources of power losses in an ACBB and can be written as follows:

$$T_z = Z\cdot(T_{FR} + T_{MC} + T_{MP} + T_{MER} + T_{BC}) \tag{2}$$

In Equation (2), it is considered that T_{FR} includes the effects of hydrodynamic rolling friction; T_{MC} includes the effects of friction caused by the curvature of the races; T_{MP} includes the effects of friction caused by pivoting friction on contact ellipses; T_{MER} includes the effects of friction caused by elastic hysteresis in a rolling process, and T_{BC} includes the effect of friction between the cage and inner race and the friction generated by ball–cage interactions.

The major and minor semiaxis, a and b, of the inner and outer balls-race contacts were determined with relations presented in [1]. The following values are obtained: $a_i = a_o = 0.073$ mm, $b_i = 0.012$ mm and $b_o = 0.016$ mm. The lubricant regime is dominant IVR (iso-viscous-rigid), and the friction coefficient in ball-race contacts was obtained with values between 0.03 and 0.05. A demonstrative programme was carried out for a viscosity of 150 mm^2/s, corresponding to the base oil of Sheel Gadus grease at a temperature of (26–28) °C, which was determined during the experiments. The same viscosity was considered for the oil SAE 5W-30.

For the calculation of the components, T_{MC}, T_{MP} and T_{MER} were considered based on the relations developed by Houpert [2] and detailed by Popescu et al. in [15].

The friction torque generated by the cage is determined by considering the fluid friction in a gap between the cage and the inner guiding ring. The methodology to simulate this component is presented in Appendix A.

The components generated by the hydrodynamic rolling forces both on the outer and inner ball-race contacts, T_{FR}, were determined according to following equation [2]:

$$T_{FR} = Z\cdot 2\cdot(FR_0 + FR_i)\cdot\frac{R_o\cdot R_i}{d_m} \tag{3}$$

R_i and R_0 are the radii of the inner and outer races, respectively, and are determined via the following equations:

$$R_i = d_m\cdot(1-\gamma)/2;\;\; R_o = d_m\cdot(1+\gamma)/2 \tag{4}$$

where γ is given by the following relation [1]:

$$\gamma = d_b\cdot cos(\alpha)/d_m \tag{5}$$

the geometrical parameters of the ACBB (d_m, d_b, α, f_i, f_o) are defined in Table 1.

Hydrodynamic rolling resistances developed in the ball's outer and inner races contacts, FR_o and FR_i, respectively, are determined using the Houpert IVR equations [2]:

$$FR_{o,i} = 1.213\cdot E^*\cdot R_{x,o,i}^2\cdot k_{o,i}^{0.358}\cdot U_{o,i}^{0.636}\cdot W_{o,i}^{0.364} \tag{6}$$

where E^* is the reduced Young's modulus of the ball and races; $U_{o,i}$ and $W_{o,i}$ are the classical dimensionless speed and load parameters, respectively, determined for outer and inner ball-race contacts [2].

The ratios of the equivalent radii for inner and outer races are determined via the following relations [2]:

$$k_i = \frac{2\cdot f_i}{(2\cdot f_i - 1)\cdot(1-\gamma)};\; k_o = \frac{2\cdot f_o}{(2\cdot f_o - 1)\cdot(1+\gamma)} \tag{7}$$

where $R_{y,o,i}$ are the equivalent ball-race radius in a rolling direction and are determined via the following relations [2]:

$$R_{x,o} = \frac{d_b}{2}\cdot\left(1+\frac{d_b\cdot\cos\alpha}{d_m}\right); R_{x,i} = \frac{d_b}{2}\cdot\left(1-\frac{d_b\cdot\cos\alpha}{d_m}\right) \tag{8}$$

Imposing the geometrical characteristics of the 7000C ACBB, for an axial load of $F_a = 3.11$ N and for an oil viscosity of 150 mm^2/s, all the components included in Equation (1) were simulated and are presented in Figure 2.

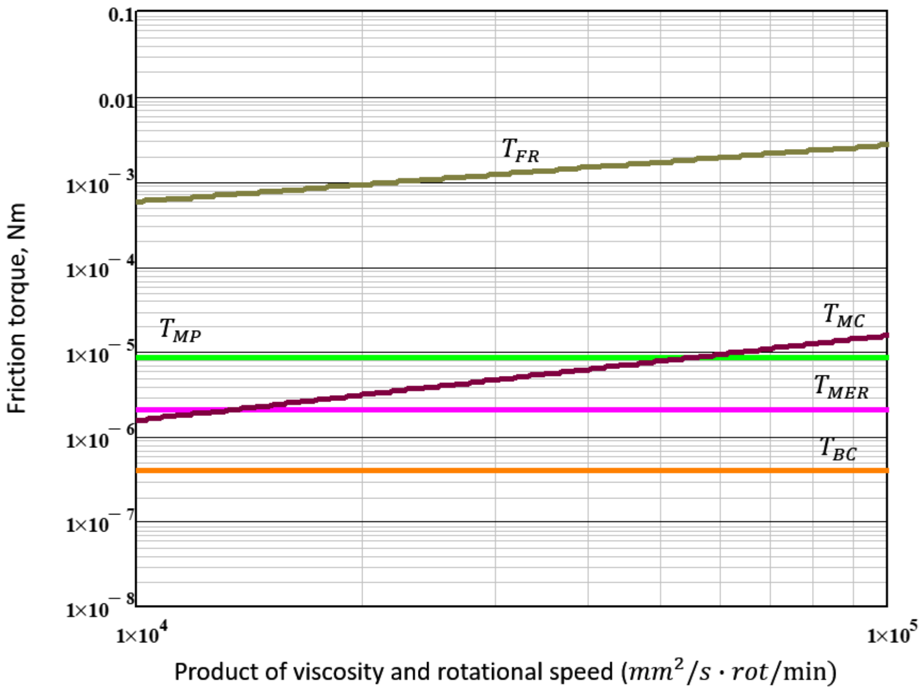


Figure 2. Variation in the five components for the total friction torque Tz simulated with the 7000C ACBB lubricated with an oil viscosity of 150 mm^2/s.

The simulation presented in Figure 2 shows that the dominant friction torque for the 7000C ACBB in the given conditions is generated by the hydrodynamic rolling resistances T_{FR}; all the other components can be neglected.

3. Comparison Between Houpert IVR Model and Catalogues Relations

The simulated Houpert IVR model adapted to the 7000C ACBB in the operation conditions presented in Paragraph 2 were compared to the SKF and Schaeffler relations in similar conditions.

3.1. The SKF Relations for the 7000C ACBB

For an ACBB ball bearing without a sealing system and operating with a small quantity of oil, the SKF methodology considers the total friction torque T_{SKF} which can be determined as a sum of two components: the rolling component M_{rr} and the sliding component M_{sl} [5].

$$\mathrm{T_{SKF}} = \mathrm{M_{rr}} + \mathrm{M_{sl}} \tag{9}$$

The rolling component M_{rr} can be determined via the following relation [5]:

$$\mathrm{M_{rr}} = \Phi_{\mathrm{ish}} \cdot \Phi_{\mathrm{rs}} \cdot \mathrm{G_{rr}} \cdot (\upsilon \cdot \mathrm{n})^{0.6}\ [\mathrm{N \cdot mm}] \tag{10}$$

where Φ_{ish} is the inlet shear heating reduction factor; Φ_{rs} is the kinematic starvation reduction factor; G_{rr} is a parameter dependent on the bearing type, dimensions as well as radial and axial loads; n is the rotational speed in rot/min, and ν is the oil or base oil viscosity in mm^2/s.

For a low rotational speed, the reduction factors Φ_{ish} and Φ_{rs} can be approximated by unity.

The parameter G_{rr}, adapted for an only axially loaded ACBB, has the following relation [5]:

$$\mathrm{G_{rr}} = 2.5 \cdot 10^{-7} \cdot d_m^{1.97} \cdot \left(3.55 \cdot 10^{-12} \cdot d_m^4 \cdot \mathrm{n}^2 + 3.64 \cdot \mathrm{F_a}\right)^{0.57}\ [\mathrm{N \cdot mm}] \tag{11}$$

where d_m is expressed in mm, and F_a is expressed in N.

The sliding frictional component M_{sl} is a function of a bearing's mean diameter d_m, axial load F_a and friction coefficient μ_{sl} according to the following equation [5]:

$$M_{sl} = 1.6 \cdot 10^{-2} \cdot d_m^{0.05} \cdot F_a^{\frac{4}{3}} \cdot \mu_{sl}\ [\mathrm{N \cdot mm}] \tag{12}$$

The friction coefficient μ_{sl} has an approximate value of 0.075.

3.2. The Schaeffler Relations for the 7000C ACBB

The Schaeffler catalogue proposes, for the product $(v \cdot n) > 2000$, the following relation to determine the total friction torque in an ACBB [6]:

$$T_{SC} = M_0 + M_1 \tag{13}$$

where M_o is dependent on the bearing type, geometry, lubricant and speed.

For an ACBB, the component M_o is determined via the following relation [6]:

$$M_0 = f_0 \cdot (v \cdot n)^{\frac{2}{3}} \cdot d_m^3 \cdot 10^{-7} \text{ [N·mm]} \tag{14}$$

For an ACBB, the coefficient f_0 = 1.3 (both for oil and grease). The component M_1 is given via the following relation [6]:

$$M_1 = f_1 \cdot F_a \cdot d_m \text{ [N·mm]} \tag{15}$$

where factor f_1 is determined via the following relation [6]:

$$f_1 = 0.001 \cdot \left(\frac{P_0}{C_0}\right)^{0.33} \tag{16}$$

For the 7000C ACBB, fatigue load limit P_0 = 0.071 kN, and basic static load rating C_o = 1.66 kN [6]. From Equation (16), the result is as follows: $f_1 = 3.534 \times 10^{-4}$.

A comparison between the simulated friction torques T_{FR}, T_{SKF} and T_{SC} is presented in Figure 3 for the 7000C ACBB operating with an axial load of 3.11 N with a kinematic viscosity of 150 mm^2/s. It can be seen that the friction torque determined using the Houpert IVR methodology has the highest values for imposed conditions.

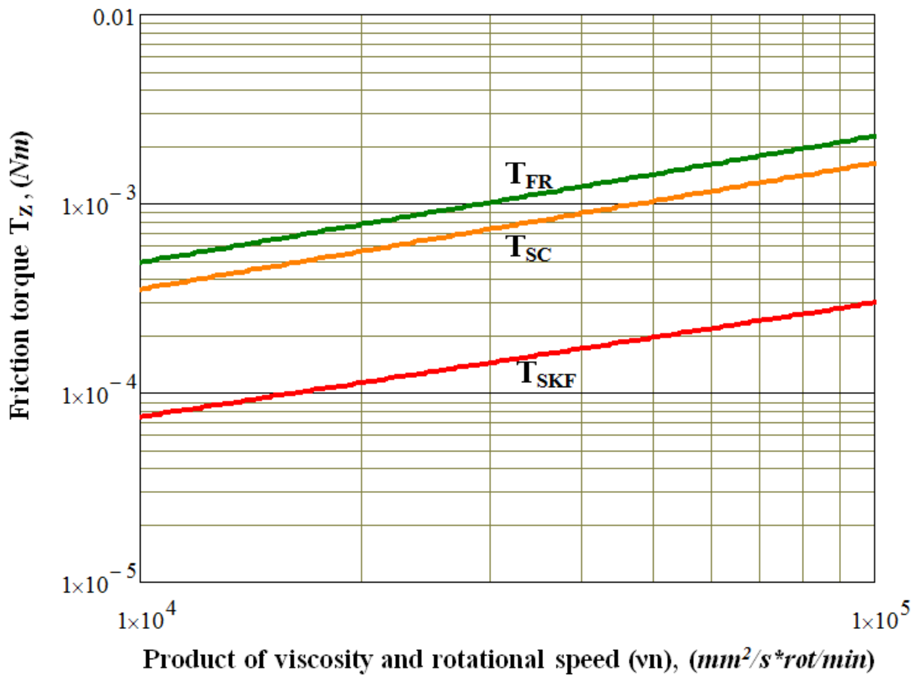


Figure 3. Comparation of the friction torques T_{FR}, T_{SKF} and T_{SC} simulated with the 7000C ACBB under an axial load F_a = 3.11N and with a kinematic viscosity of 150 mm^2/s.

From Figure 3, it can be observed that the friction torque values calculated using SKF relations have the smallest values. These results are because the axial load for tests has a smaller value that minimum axial load recommended by SKF methodology [5]. However, we also considered this method for comparison, since the M_{rr} component contains the product $(v \cdot n)^{0.6}$, as an important parameter for our comparation.

4. Experimental Equipment and Procedure

4.1. Experimental Equipment

Figure 4 shows the general view of the Tribometer CETR UMT-2 with the testing ring mounted on the rotating table. Figure 5 presents the details of the testing ring components. The standard 7000C ACBB has 8 balls separated by a phenolic resin cage guided on the inner race. As presented in Figure 5b, a loading disc mounted on the outer ring generated an axial load of F_a = 3.11 N, corresponding to the contact load Q = 0.763 N for each normal ball-races. The maximum Hertzian ball-race contact pressures are 0.3 GPa and 0.40 GPa for outer and inner ball-race contacts, respectively.

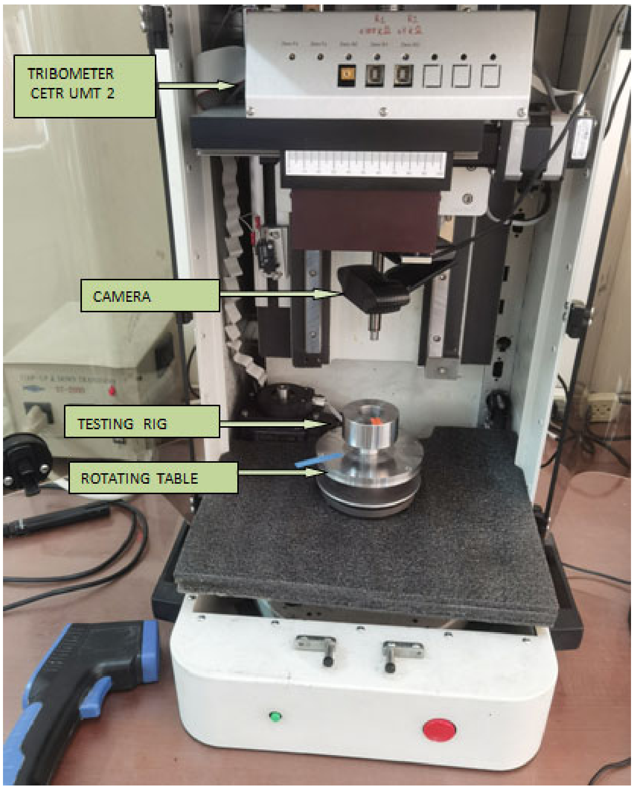
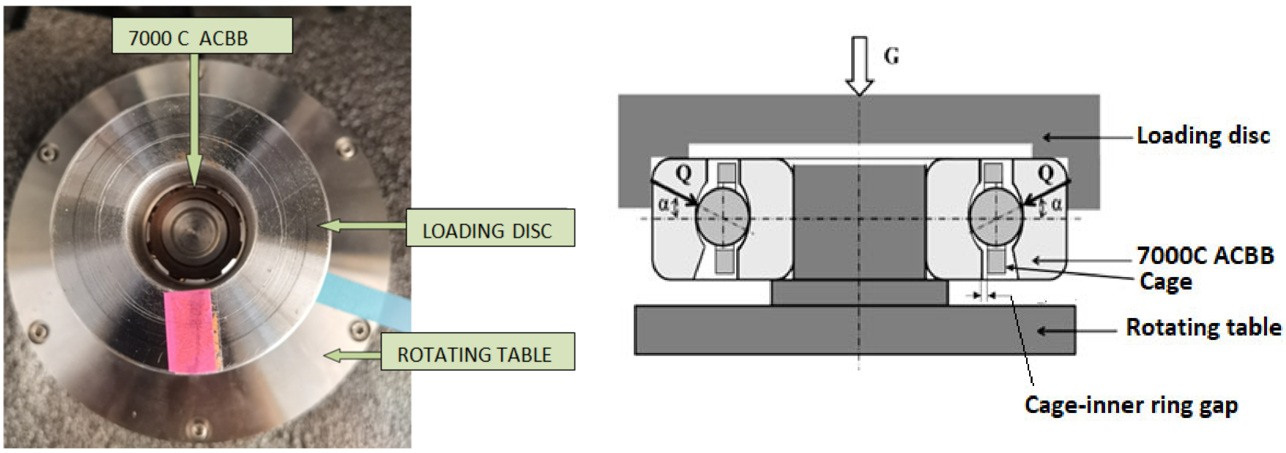


Figure 4. View of the Tribometer CETR UMT 2 and testing equipment for an ACBB.



(**a**) (**b**)

Figure 5. Testing ring with the 7000C ACBB: (**a**) upper view; (**b**) schematic lateral view and details.

4.2. Lubricant Characteristics

Two commercial greases in small quantities were used to lubricate the standard 7000C ACBB: Shell Gadus S2 V100 2 and SKF LGHP 2/0.4. The physical characteristics are presented in Table 2.

The tests were also carried out with a viscosity grade SAE 5W-30 oil, and SAE T90 EP2 had the most important characteristics which are presented in Table 3.

During the experiments, the temperature in the laboratory was about 26–28 °C.

4.3. Experimental Procedure

The friction torque of a 7000C ACBB has been experimentally determined by the authors based on the spin-down method described in detail in [14,15]. In fact, the friction torque is indirectly determined according to the time and total angular position obtained during the deceleration process of the outer race and of the disc from an imposed rotational speed to the stopping process. So, all the kinetic energy of the rotating disc and outer race is dissipated in the friction processes between the balls, races and cage. Figure 4 presents the Tribometer CETR UMT-2 with the 7000C ACBB mounted on the rotational table, and a camera was used to capture the rotation of the loading disc and of the outer race during the decelerating process. Figure 5a presents a frontal view of the rotational table and an ACBB with coloured marks used to count the number of rotations and braking time, φ_{max} and t_{max}, respectively. Figure 5b presents a section in the 7000C ACBB with the loading disc mounted on the outer race and the inner race mounted on the spindle of the rotational table of the tribometer.

The procedure follows two phases:

(1) In the first phase, both the rotating table and the inner race are rotated at an imposed rotational speed. As a result of the friction on the rolling bearing, the outer race and the loading disc start to rotate until it reaches the speed of the inner race;

(2) In the second phase, the rotating table is suddenly stopped, and the outer race and the disc continue to rotate until the kinetic energy is depleted. All the deceleration process is monitored by a video camera, and finally, the video recording has been processed to establish the total rotating angle and corresponding time.

4.4. Evaluation of the Friction Torque Based on the Tests

Based on Equation (6), it can be noticed that for a given geometry, lubricant viscosity and normal load Equation (6) can be expressed as a product between a constant parameter and the speed parameters $U_{o,i}$ with a power of 0.636. Both the speed parameters $U_{o,i}$ can also be expressed as a product between a constant and the angular speed of the ball bearing outer race, ω. Finally, we proposed that the total friction torque for experiments follows the following general equation [14,15]:

$$\mathrm{T_z(\omega) = K \cdot \omega^n} \tag{17}$$

where $T_z(\omega)$ is the total friction torque in the ball bearing; K is a constant depending on the lubricant viscosity, load and geometry; ω is the angular speed of the outer race, and n is an exponent smaller than one ($n < 1$).

Applying the dynamic equilibrium of the rotating disc and outer race in the deceleration process with the total friction torque given by Equation (17), the following differential equation for the dynamic equilibrium for the outer race and loading disc was obtained as follows [14,15]:

$$\mathrm{J \cdot \frac{d\omega}{dt} + K \cdot \omega^n = 0} \tag{18}$$

where J is the inertia moment of the rotating mass and the outer race of an ACBB.

Equation (18) was analytically solved, and the variation in the angular speed $\omega(t)$ and the variation in angular position $\varphi(t)$ as function of time are determined by the following relations [14,15]:

$$\mathrm{\omega(t) = \left[\omega_0^{1-n} - \frac{K \cdot (1-n)}{J} \cdot t\right]^{\frac{1}{1-n}}} \tag{19}$$

$$\mathrm{\varphi(t) = \frac{J}{K \cdot (2-n)} \cdot \omega_0^{2-n} - \left[\omega_0^{1-n} - \frac{K \cdot (1-n)}{J} \cdot t\right]^{\frac{2-n}{1-n}}} \tag{20}$$

where ω_0 is the angular speed at the starting decelerating process when time t = 0. Imposing the experimentally values for the total decelerating angle and time φ_{max} and t_{max}, respectively, the nonlinear Equations (19) and (20) were solved, and the parameters K and n were determined for each angular speed ω_o.

4.5. Comparison of the Experimental Friction Torque with Houpert IVR Model, SKF and Schaeffler Simulated Friction Torque

Figures 6 and 7 present the variation in the experimental friction torque for the standard 7000C ACBB, greased with Shell Gadus S2 V100 2 and SKF LGHP 2/0.4, with a quantity of approx. 0.17 cm^3 which means that half of the recommended quantity was used by SKF for the precision of an ACBB [16]. This quantity was adopted to avoid the supplementary friction caused by the churning of the grease.

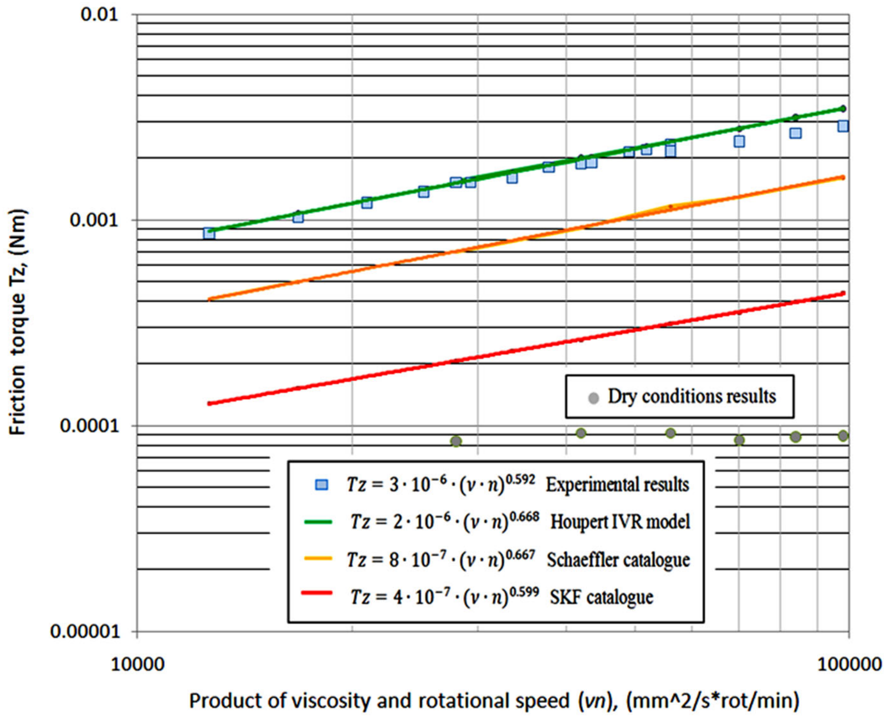


Figure 6. The variation in the friction torque Tz for the 7000C ACBB, greased with Shell Gadus S2 V100 2, experimentally obtained and calculated with the SKF, Schaeffler relations, the Houpert IVR methodology and in dry conditions.

The experiments were also ran with the lubricant Grade SAE oil T90, and the results are presented in Figure 8. To compare the experimental results to the results obtained using the Houpert IVR model, the SKF and Schaeffler relations, the variation in the experimental friction torque was presented as a function of the product between the kinematic viscosity and the rotational speed ($\nu \cdot n$) where ν is given in mm^2/s, and n is given in rot/min.

As a first conclusion from the results presented in Figure 6, the experimental results of the friction torque obtained with grease Shell Gadus S2 V100 2 had values higher than the values determined using the SKF and Schaeffler relations. Similar results were obtained in previous experiments where the experiments were compared to the SKF results [14].

At some time, we noticed that there is a good correlation between our experimental results and the theoretical Houpert IVR model presented in Paragraph 2.

Between the experimental results obtained with the Shell Gadus S2 V100 2 grease in the SKF and Schaeffler relations, one can notice that the parameter ($\upsilon \cdot n$) has similar exponents, 0.592 from the experiments, and 0.599 given by the SKF methodology. It is clear that the dimensions and the axial load included in the parameter G_{rr} lead to important decreases in the friction torque compared with our experiment.

The exponent of the product ($\upsilon \cdot n$) obtained via Schaeffler relations and the Houpert IVR model have values between 0.667 and 0.668.

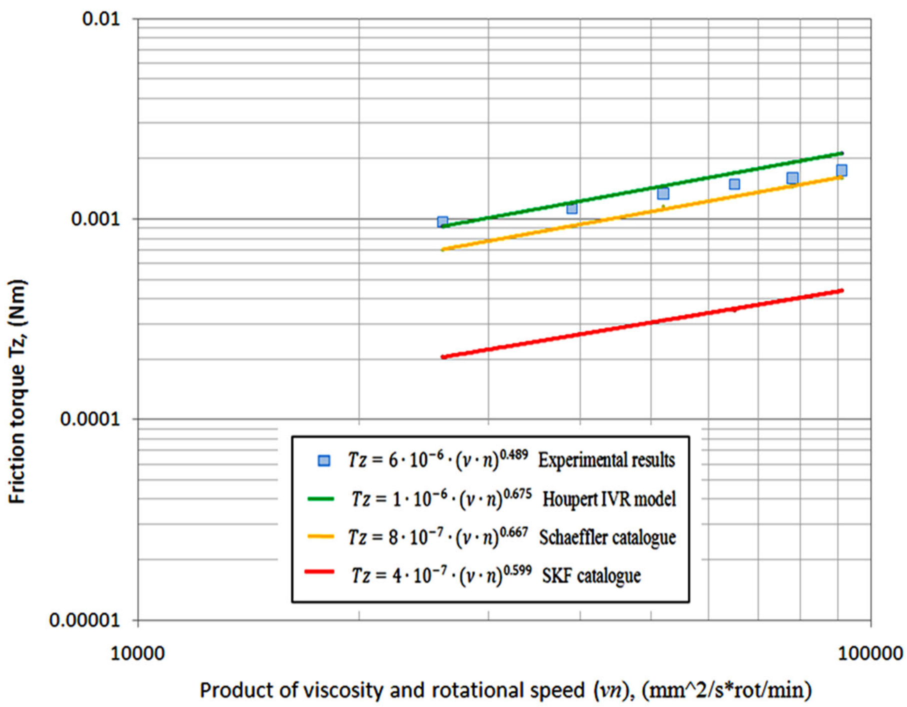


Figure 7. The variation in the friction torque *Tz* for the 7000C ACBB, greased with SKF LGHP 2/0.4 and experimentally determined and calculated using the SKF, Schaeffler relations and the Houpert IVR methodology.

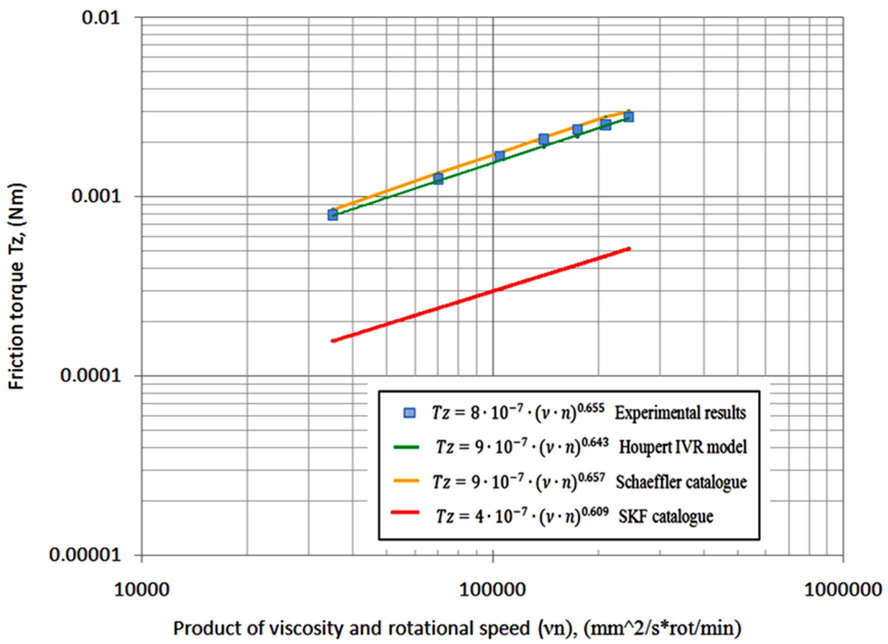


Figure 8. The variation in friction torque *Tz* for the 7000C ACBB, lubricated with viscosity grade SAE oil T90; experimentally determined and calculated with the SKF methodology and with the Houpert IVR methodology.

So, considering all exponents obtained for the product ($\upsilon \cdot n$), it can be accepted that the friction torque in a miniature ACBB, with a low load and operated in low rotational speed, can be approximated by the following relation:

$$\mathrm{T_{EXP}} = \mathrm{G_{EXP}} \cdot (\upsilon \cdot \mathrm{n})^{(0.600\,-\,0.668)}\ [\mathrm{N \cdot m}] \quad (21)$$

where G_{EXP} is a parameter depending on the ball bearing geometry and axial load, with values with a magnitude order of 10^{-6}.

Figure 6 shows that if the ball bearing is operating under dry conditions, the friction torque values range between 8.4 and 9.2 $\times$ 10^{-5} N·m, with the lowest values obtained compared to the values obtained using the greased tests and SKF methodology.

Figure 7 presents the variation in the friction torque for the grease SKF LGHP 2/0.4 as a function of the product ($v \cdot n$). The experimental results presented in Figure 7 confirm similar conclusions as the results presented in Figure 6 with small difference for the exponent of the product ($v \cdot n$). So, an important difference in the magnitude between the experiment values and the calculated values with the SKF method was observed. For this grease, the resulting exponent of the product ($v \cdot n$) for experimental values is 0.489, which is smaller than the exponent of the SKF, Schaefler relations and the Houpert IVR methodology. This means that when the rotational speed increases, a lower increase of the friction torque for the grease SKF LGHP 2/0.4 is seen. Although the base oil viscosities of the two greases are very close, it turns out that the differences are given by the soap type.

Figure 8 presents the variation in the friction torque for grade SAE oil T90 as a function of the product ($v \cdot n$) which was compared with the results obtained using the SKF, Schaeffler relations and the Houpert IVR methodology.

In the experiments with SAE oil T90, we obtained values for total friction torque higher than the values obtained with the SKF methodology but very close to the Schaeffler and Houpert IVR methodologies. The exponent of the product ($v \cdot n$) can be approximated with an average value of 0.65.

We consider that by increasing the viscosity, the Schaeffler relations can realize a good approximation of the friction torque for the 7000C ACBB, similar to the Houpert IVR methodology.

5. Conclusions

To evaluate the friction torque in the miniature ball bearings, lubricated with greases and oils, the authors developed an experimental programme on the 7000C ACBB based on the spin-down methodology. The experiments were realized for two types of greases and oils under a very low axially load, in a range of rotational speed between 100 and 700 rot/min. The experimental results were compared to the Houpert IVR methodology and to the SKF and Schaeffler catalogue relations. A simulating programme to evaluate the components of the friction torque to the complex Houpert IVR methodology was developed.

The most important conclusions drawn from our work are as follows:

(1) The dominant component of the Houpert IVR friction torque for the 7000C ACBB in the given conditions included in the experiments is generated by the hydrodynamic rolling resistances, and all the other friction components can be neglected;

(2) The experimental results of the friction torque both for greases and oils tests are in good correlation with the friction torque generated by the hydrodynamic rolling resistance developed between the balls and races, described using the Houpert IVR methodology;

(3) Both the experimental friction torque results and the Houpert IVR friction torque values are higher than the friction torque values obtained with the SKF relations, both for greases and for low or high oil viscosity;

(4) For high lubricant viscosity, the experimental friction torque results are very close to the friction torque values obtained with the Schaeffler relations. The results obtained so far show us that in the case of the low-loaded miniature ball bearings, the Schaeffler relations for the friction torque can be used instead of the Houpert IVR model with good approximation, both for greases and for oils with a high viscosity;

(5) In the absence of the lubricant in the miniature ball bearing, the lowest values were obtained for the total friction torque;

(6) As a general recommendation, to evaluate the friction torque in the miniature ball bearings, operating at moderate rotational speed and under low loads, we propose the calculation of the friction torque with the Houpert IVR model, although it involves a complex calculation;

(7) Finally, more research must be made with various types of greases and oils viscosities to obtain a more simplified relationship.

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Appendix A

Between the cage and the inner guide, the lubricant was considered to have a Newtonian behavior. So, the shear stress in the interstitial gap $\tau(n_0)$, as function of rotational speed of outer race, is determined using the following Newton's relation:

$$\tau(n_0) = \eta \cdot \upsilon_c(n_0)/g \tag{A1}$$

where $v_c(n_0)$ is the tangential speed of the cage; g is considered the lubricant film thickness between the cage and the inner guiding ring, and η is the dynamic viscosity.

If the outer race is rotating, and the inner race is stationary, the angular speed of the cage in an angular contact ball bearing is obtained using the following relation [1]:

$$\omega_c(n_0) = \left(\pi \cdot \frac{n_0}{60}\right) \cdot (1+\gamma) \tag{A2}$$

where $\gamma = d_b \cdot \frac{\cos(\alpha)}{d_m}$.

The tangential speed of the cage is obtained by the following relation:

$$v_c(n_0) = \omega_c(n_0) \cdot d_{ci}/2 \tag{A3}$$

where d_{ci} is the internal diameter of the cage.

The tangential moment acting on the cage as result of the viscous friction is given by the following relation:

$$M_c(n_0) = \tau(n_0) \cdot \pi \cdot d_{ci} \cdot 2c \cdot d_{ci}/2 \tag{A4}$$

where c is the width of the cage-guided inner ring contact.

This moment is compensated by the contact forces between the eight balls and cage. So, the contact forces between the balls and the cage can be obtained according to following relation:

$$FB(n_0) = M_c\,(n_0) \cdot 2/(8 \cdot d_m) \tag{A5}$$

The moment MB acting on the balls as a result of the tangential forces developed between the balls and the cage is determined using the following relation:

$$MB(n_0) = \mu_c \cdot F_b(n_0) \cdot d_b/2 \tag{A6}$$

According to Equation (1), the component T_{BC} is obtained using the following relation:

$$T_{BC}(n_0) = R_i(\mu_c + 1) \cdot F_b(n_0)/2 \tag{A7}$$

where μ_c is the friction coefficient between the balls and the cage.

In the simulated programme, the following parameters were adopted: d_{ci} = 14.49 mm; g = 0.15 mm; c = 1.5 mm and μ_c = 0.05.

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