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EXPERIMENTAL FRICTION TORQUE IN MINIATURE BALL BEARINGS GREASE LUBRICATED

BY

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Abstract. The friction torque in the miniature ball bearings grease - lubricated depends on many factors such as grease properties, rotational speed, applied loads and internal geometry. The catalogues of various rolling bearings manufacturers (SKF, Schaeffler, NSK) present empirical relations for determining friction torque in general purpose ball bearings. For miniature and low-load ball bearings, these relations lead to significant differences in the friction torque compared to the experimentally determined one. To demonstrate these differences, the authors experimentally determined the friction torques in the grease-lubricated 7000 C series angular contact ball bearing.

The experiments were conducted using three types of greases at rotational speeds between 200 and 700 rpm, and an axial load of 3.11 Newtons. The authors used the spin-down methodology and the resulted friction torques (T_{exp}) was determined by integrating the dynamic equation into the deceleration process using Python-based software. The input and output parameters were presented for each grease and rotational speed. The dependences between experimental friction

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torques and angular speeds were shown in logarithmic diagrams, and fitted relations were determined for all three greases.

Based on the SKF methodology, the friction torques was theoretical determined and compared with experimental results. It was observed that the calculated friction torques were (77-88)% lower values than the experiment values.

Keywords: ball bearing, grease, friction torque, micro friction, lubricated friction.

1. Introduction

In the last years, many micro motors and micro turbines have been developed using silicon wafer machining (Hanrahan *et al.*, 2015; McCarthy *et al.*, 2009a). Having in their structure micro ball bearings with micro balls of 0.285 mm diameter, the friction torque was determined by spin-down method. The spin-down method used by above-mentioned authors includes two phases. In the first phase the rotor supported by the micro rolling bearing is rotated to a given speed, monitored by a video camera. In the second stage the electric supply of the rotor is stopped and the deceleration of the rotor is monitored until it comes to a complete stop (McCarthy *et al.*, 2009a) proposed an exponential function for the variation of the rotor rotating angle φ as a function of time according to following equation:

$$\varphi(t) = A (1 - e^{-Bt}) \quad (1)$$

In the decelerating process the dynamic equilibrium of the rotor is expressed by the following equation:

$$J \cdot d^2(\varphi)/dt^2 + T_{\text{exp}} = 0 \quad (2)$$

where J is the rotor moment of inertia, $d^2(\varphi)/dt^2$ is the angular acceleration of the rotor in decelerating process and T_{exp} is the friction torque experimentally determined in the micro ball bearing.

According to the Eq. (1) the rotational speed during the decelerating process is obtained as, $d(\varphi)/dt = ABe^{-Bt}$ and angular acceleration can be expressed as $d^2(\varphi)/dt^2 = -B d(\varphi)/dt$, which means a linear relation between angular acceleration and angular speed. The constant B is determined by experiments. The friction torque obtained by (McCarthy *et al.*, 2009b) was linearly dependent on rotational speed, although the micro bearings used operated in dry conditions. The experiments conducted by (Olaru *et al.*, 2011) using the spin-down methodology on miniature axial ball bearings without cage and having only three balls showed that in dry conditions the friction torque does not depend on rotational speed. Similar conclusions were obtained by (Popescu *et al.*, 2020) for a 7205B

ball bearing, operating without lubricant and having only three balls. Also, the spin-down methodology has been developed in the experiments.

To extend the spin-down methodology for miniature radial or angular contact ball bearings, it is necessary to consider the effect of the lubricant in the friction torque analysis. Thus, the presence of oil results in the formation of a lubricating film between the balls and raceways, which reduces the friction coefficient during sliding, particularly in the pivotal motion of an angular contact ball bearing. The film thickness depends mostly on the oil viscosity and rotational speed (Hamrock and Dowson, 1981). At the same time, the presence of the oil in the balls-race contacts leads to the development of important hydrodynamic resistance forces (Houpert, 1999, 2002; Biboulet and Houpert, 2010). In the presence of the lubricant, the friction torque in miniature ball bearing depends on the rotational speed.

Some experimental results (Bălan *et al.*, 2014; Ianuș *et al.*, 2022) considered the friction torque T_{exp} according to following relation:

$$T_{\text{exp}} = K\omega^n \quad (3)$$

where: K is a constant dependent on the lubricant and rolling ball bearings geometry, exponent $n < 1$ and angular speed $\omega = d(\theta)/dt$, where θ is the angular position expressed in radians.

In these conditions, Eq. (2) becomes:

$$J \cdot d\omega/dt + K \cdot [\omega]^n = 0 \quad (4)$$

The parameters K and n can be determined from the experimental results by integrating Eq. (4) and applying the boundary conditions.

Harris (Harris, 2001) presented a methodology for calculating the friction torque in a rolling bearing, M_F according to the relation:

$$M_F = M_0 + M_1 \quad (5)$$

M_0 is an empirical function of kinematic viscosity, bearing speed and geometry. M_1 is an empirical function including the external loads applied to the rolling bearing. The relationships for components M_0 and M_1 depend on the rolling bearing type.

A more complex methodology to determine the friction torque in a rolling bearing is presented by SKF company (SKF, 2025). Neglecting friction from the sealing systems and the friction caused by excess oil in the rolling bearing, the SKF relation is:

$$T_{\text{SKF}} = M_{\text{rr}} + M_{\text{sl}} \quad [\text{Nmm}] \quad (6)$$

The rolling component M_{rr} includes some coefficients depending on the thermal effects, starvation, applied loads and the product $(vN)^{0.6}$, where v is kinematic viscosity of the oil or of the base oil for grease, expressed in mm^2/s and N is rotational speed of the rolling bearing, expressed in rot/min .

The component M_{sl} is the sliding frictional component and depends on the sliding friction coefficient μ_{sl} developed in rolling contacts and is determined by the relation recommended by (SKF, 2025).

In the actual paper the authors developed Python-based software to solve the Eq. (4) and to determine parameters K , n and the experimental friction torque T_{exp} for a 7000C angular contact ball bearing, lubricated with three sorts of greases, in a rotational speed range of 200 and 700 rot/min . The experimental results were compared with those calculated using the SKF methodology under similar conditions.

2. Experimental procedure

2.1. Experimental equipment

Figure 1 presents the Tribometer CETR UMT-2 including the rotational table, testing rig mounted on the rotational speed and a video camera. The testing rig includes a standard 7000C angular contact ball bearing having inner ring mounted on rotational table and outer ring mounted in a housing, as can be seen in Fig. 2. The housing mounted on the outer ring generated an axial load $F_a = 3.11 \text{ N}$, corresponding to each normal ball-races contact load of $Q = 0.402 \text{ N}$. The maximum Hertzian ball-races contact pressures are values of 0.35 GPa and 0.45 GPa for outer and inner ball-races contacts, respectively.

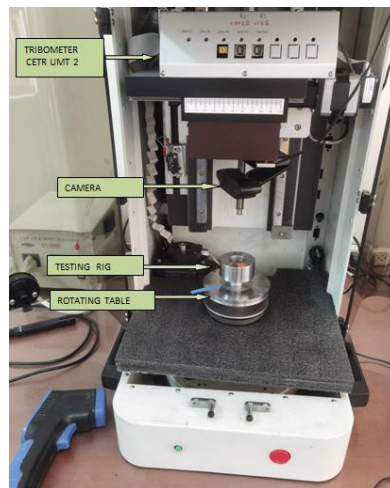


Fig. 1 – View of the CETR UMT 2 Tribometer and testing equipment.

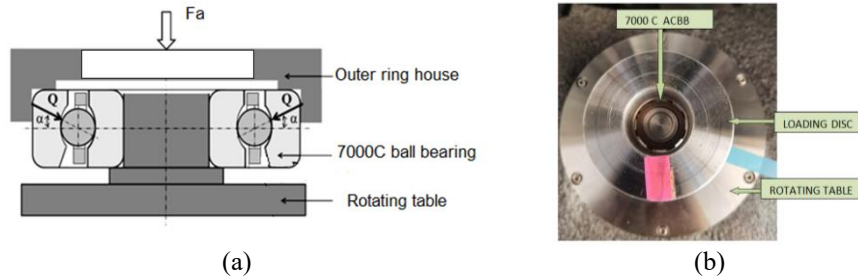


Fig. 2 – Testing ring with 7000C angular contact ball bearing: a) detail of the ball bearing mounting; b) upper view of the test rig.

2.2. Lubricant characteristics

Three commercial greases- **Shell Gadus S2 V100 2 (Grease A)**, **SKF LGHP 2/0.4 (Grease B)** and **Cogelsa Ultraplex LT2 (Grease C)** - were applied in small amounts to lubricate the standard 7000C angular contact ball bearing. The physical properties of the **Grease A** are presented in (Cojocaru *et al.*, 2025). The **Grease B** includes the Di-urea complex soap and kinematic viscosities of the base oil at 40 and 100°C are 96 and 10.5 mm²/s, respectively. The **Grease C** includes the Lithium complex soap and kinematic viscosities of the base oil at 40 and 100°C are 220 and 21 mm²/s, respectively. The three greases have NLGI consistency between 2 and 3. The geometrical characteristics of the 7000C angular contact ball bearing are presented in (Cojocaru *et al.*, 2025).

2.3. Experimental procedure

The following steps were performed for each rotational speed:

- (i) The inner ring was rotated at a given rotational speed N (rpm) until, as a result of friction in the ball bearing, the housing and outer ring reached the same speed of the inner ring. The synchronization of the two speeds was checked using the video camera and the two indicators shown in Fig. 2b.
- (ii) In the moment of speeds synchronization, the rotating Tribometer table was stopped and the both housing and outer race start a deceleration phase until it comes to a complete stop. The total rotating angle φ_0 and the decelerating time t_0 were monitored by video camera.
- (iii) The differential Eq. (4) was solved in two successively steps:
 - In the first step was integrated Eq. (4) to obtain the relation for angular speed $\omega(t)$ with initial condition: at time $t=0$, $\omega(0) = \omega_0$, that means the start of decelerating process;
 - In the second step was integrated the expression of the $\omega(t)=d\varphi/dt$ and was obtained the relation for the rotating angle $\varphi(t)$ with initial condition: at time $t=0$, $\varphi(0) = 0$.

Knowing the values of the total cumulative angle until stopping φ_0 and the time in which the braking process took place t_0 , two nonlinear equations with unknowns $\mathbf{K}$ and $\mathbf{n}$ have been obtained.

$$\left[\omega_0^{(1-n)} - \frac{K \cdot (1-n)}{J} \cdot t_0 \right]^{\left(\frac{1}{1-n}\right)} = 0 \quad (7)$$

$$\frac{J}{K \cdot (2-n)} \cdot \omega_0^{(2-n)} + \frac{J}{K \cdot (2-n)} \cdot \left[\omega_0^{(1-n)} - \frac{K \cdot (1-n)}{J} \cdot t_0 \right]^{\left(\frac{2-n}{1-n}\right)} - \varphi_0 = 0 \quad (8)$$

where: ω_0 is the initial angular speed in rad/s, t_0 is the maximum rotating time in decelerating process (seconds) and φ_0 is the total angle accumulated during the deceleration process, expressed in radians.

3. Experimental results

The experiments were carried out for each grease type at the following rotational speeds N : 200; 300; 400; 500; 600 and 700 rpm. During the experiments, the laboratory temperature was approximately 26 to 28°C. The inertia moment of the housing including the outer ring was $J = 1.197 \times 10^{-4} \text{ kg.m}^2$.

To solve the nonlinear Eqs. (7), (8) the authors developed Python-based software and determined parameters K and n for each rotational speed and type of greases. The authors also determined the friction torques, T_{exp} .

The developed Python application includes a dedicated numerical module for solving the nonlinear system of equations (7) and (8) for each experimental dataset. The software automatically determines the parameters K and n by iterative fitting and computes the corresponding experimental torque T_{exp} . A graphical user interface was implemented to facilitate data import from CSV files and visualization of the computed results. The program also generates PDF tables and logarithmic plots of $T_{\text{exp}} = K \cdot N^n$ for each tested grease, ensuring a fully automated and reproducible workflow.

The measured parameters N , ω_0 , t_0 and φ_0 were included as input data for each sort of greases. The parameters K , n and the friction torque T_{exp} were presented as output data. Tables 1 and 2 present the input and output data for the Grease A, respectively, and Fig. 3 shows the variation of the T_{exp} as function of N for Grease A.

Table 1

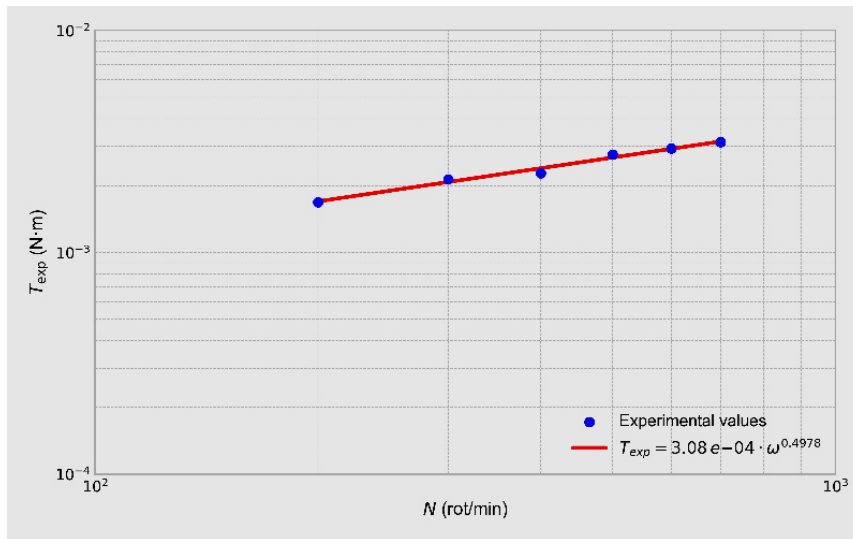
Input data for Grease Shell Gadus S2 V100 2

N (rot/min)	ω_0 (rad/s)	t_0 (s)	φ_0 (rad)
200.0	20.943	2.766	20.769
300.0	31.415	3.7	37.698
400.0	41.887	4.4	61.696
500.0	52.358	5.133	85.868
600.0	62.83	5.833	113.967
700.0	73.302	6.433	144.509

Table 2

Output data for Grease Shell Gadus S2 V100 2

N (rot/min)	ω_0 (rad/s)	t_0 (s)	ϕ_0 (rad)	K	n	T_{exp} (N·m)
200.0	20.943	2.766	20.7688	0.0003658	0.50206	0.001684
300.0	31.415	3.7	37.698	0.0003395	0.53367	0.002137
400.0	41.887	4.4	61.6956	0.0003497	0.5009	0.002271
500.0	52.358	5.133	85.8677	0.0002372	0.62034	0.002764
600.0	62.83	5.833	113.9666	0.0002565	0.58891	0.002938
700.0	73.302	6.433	144.509	0.0002579	0.5819	0.003138

Fig. 3 – Variation of the T_{exp} as function of N for the Grease A.

Tables 3 and 4 present the input and output data for the Grease B, respectively, and Fig.4 shows the variation of the T_{exp} as function of N for Grease B.

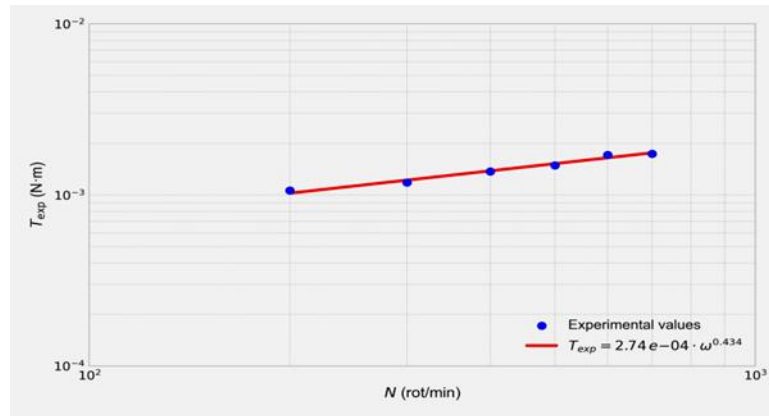
Table 3

Input data for Grease SKF LGHP 2|0.4

N (rot/min)	ω_0 (rad/s)	t_0 (s)	ϕ_0 (rad)
200.0	20.943	4.567	32.986
300.0	31.415	6.167	66.582
400.0	41.887	7.3	101.924
500.0	52.358	8.433	140.844
600.0	62.83	9.567	188.926
700.0	73.302	10.666	244.513

Table 4**Output data for Grease SKF LGHP 2|0.4**

N (rot/min)	ω_0 (rad/s)	t_0 (s)	ϕ_0 (rad)	K	n	T_{exp} (N·m)
200.0	20.943	4.567	32.9858	0.0002318	0.5	0.001061
300.0	31.415	6.167	66.5823	0.000211	0.5	0.001183
400.0	41.887	7.3	101.9242	0.0002122	0.5	0.001374
500.0	52.358	8.433	140.8439	0.0002053	0.50019	0.001487
600.0	62.83	9.567	188.9263	0.0001817	0.54227	0.001716
700.0	73.302	10.666	244.5134	0.0001782	0.53107	0.001743

**Fig. 4 – Variation of the T_{exp} as function of N for the Grease B.**

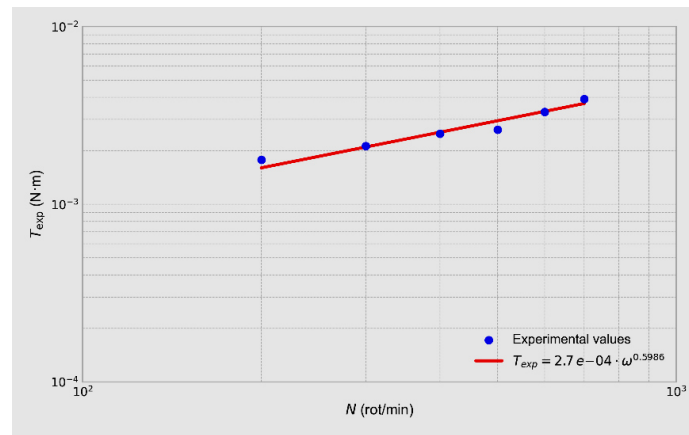
Tables 5 and 6 present the input and output data for the Grease C, respectively, and Fig. 5 shows the variation of the T_{exp} as function of N for Grease C.

Table 5**Input data for Grease Cogelsa Ultrplex LT2**

N (rot/min)	ω_0 (rad/s)	t_0 (s)	ϕ_0 (rad)
200.0	20.943	2.667	15.708
300.0	31.415	3.4	31.415
400.0	41.887	3.867	46.424
500.0	52.358	4.6	70.771
600.0	62.83	5.6	102.971
700.0	73.302	5.8	121.646

Table 6

Output data for Grease Cogelsa Ultrplex LT2						
N (rot/min)	ω_0 (rad/s)	t_0 (s)	ϕ_0 (rad)	K	n	T _{exp} (N·m)
200.0	20.943	2.667	15.7076	0.000385	0.5029	0.001778
300.0	31.415	3.4	31.415	0.0003778	0.50185	0.002131
400.0	41.887	3.867	46.4244	0.0003831	0.50185	0.002497
500.0	52.358	4.6	70.771	0.0003605	0.50187	0.002628
600.0	62.83	5.6	102.9714	0.0002613	0.6131	0.003309
700.0	73.302	5.8	121.6459	0.0002403	0.64974	0.003914

Fig. 5 – Variation of the T_{exp} as function of N for the Grease C.

4. Calculation of the friction torque based on SKF methodology

The viscosities at operating temperature (27°C) ν (mm²/s) for the base oils of the three greases was determined according to the Khonsary and Booser methodology (Khonsary and Booser, 2017) and presented in detail for base oil of the Grease A by Cojocaru et al. in (Cojocaru *et al.*, 2025). Follows values for kinematic viscosities were obtained: $\nu = 213$ mm²/s for **Grease A**, $\nu = 206$ mm²/s for **Grease B** and $\nu = 482$ mm²/s for **Grease C**.

Equation (6) was applied to the angular contact ball bearing 7000C considering the base oil viscosities of the three greases mentioned above.

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

$$M_{rr} = \Phi_{ish} \cdot \Phi_{rs} \cdot G_{rr} \cdot (\nu \cdot N)^{0.6} \quad [\text{N} \cdot \text{mm}] \quad (9)$$

Φ_{ish} and Φ_{rs} are reduction factors caused of the inlet oil shear heating and of the kinematic starvation, respectively;

G_{rr} is a complex empirical function given by Eq. (10), adapted for the 7000C ball bearing axially loaded [SKF]: At low rotational speeds, the reduction factors Φ_{ish} and Φ_{rs} can be approximated as unity.

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

where d_m is the pitch diameter expressed in mm, F_a is axial load expressed in Newtons and N is rotational speed expressed in rpm.

The sliding friction component M_{sl} is given by Equation (11) adapted for the 7000C ball bearing axially loaded [SKF].

$$M_{sl} = 1.6 \cdot 10^{-2} \cdot d_m^{0.05} \cdot F_a^3 \cdot \mu_{sl} \quad [N \cdot mm] \quad (11)$$

where d_m is the pitch diameter expressed in mm, F_a is axial load expressed in Newtons and μ_{sl} is a friction coefficient depending on the lubrication regime in the ball bearing [SKF]. As it was experimentally observed by Cojocaru et al. (Cojocaru *et al.*, 2025) the lubrication regime in the 7000C ball bearing grease lubricated and operating between 200 and 700 rpm is dominant a mixed regime. In these circumstances the value of the friction coefficient μ_{sl} was approximated by 0.075.

The variations of the friction torque T_{SKF} determined according to the methodology presented in [SKF] and adapted using Eqs. (6), (9), (10) and (11), are shown in Figs. 6, 7 and 8 for Grease A, Grease B and Grease C respectively. The experimental values for the three greases were compared in Figs. 6, 7 and 8 with the values calculated based on SKF methodology.

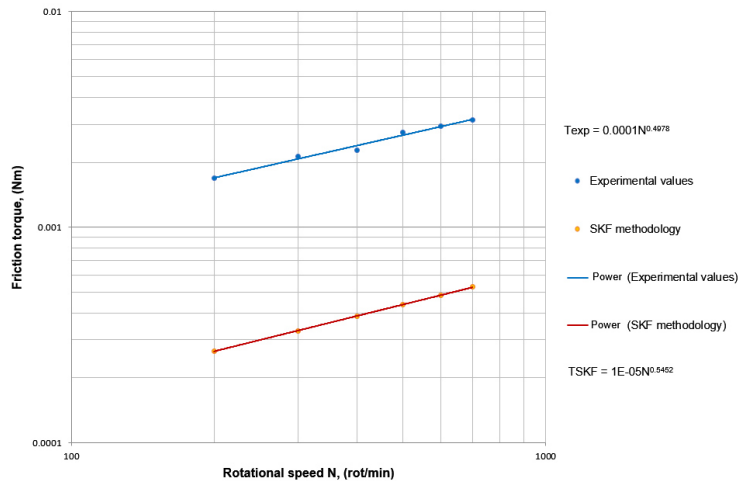
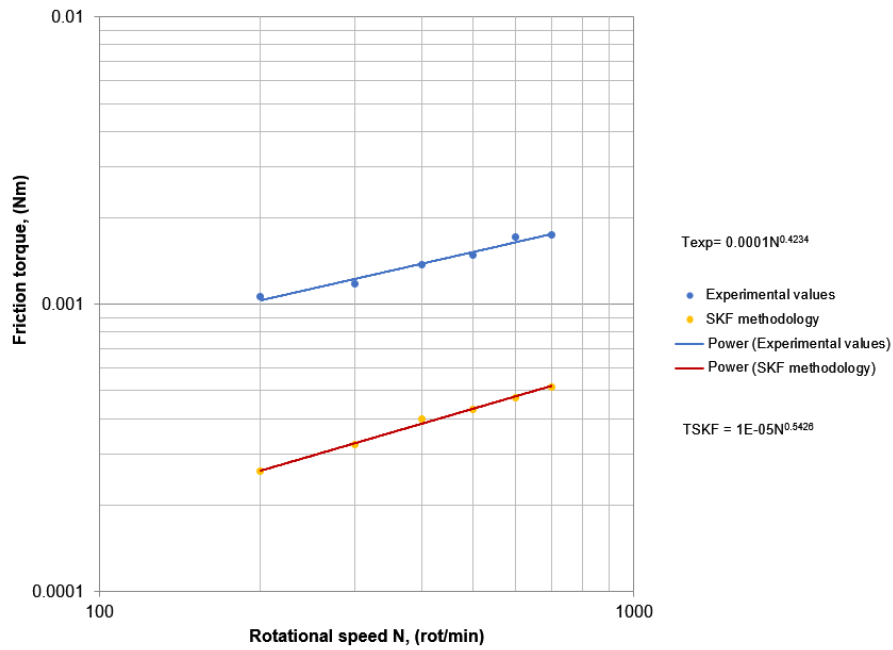
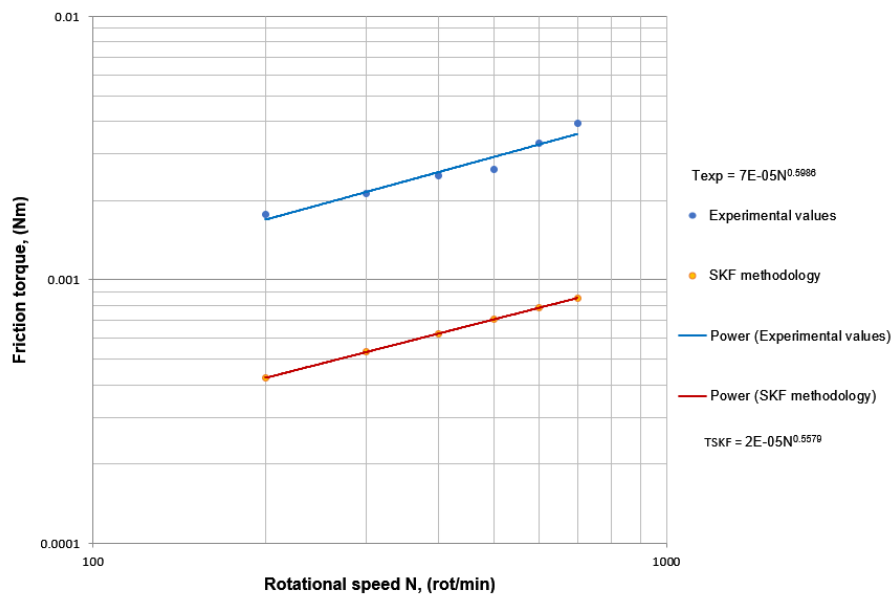


Fig. 6 – Variation of the T_{exp} and T_{SKF} with speed for Grease A.

Fig.7 – Variation of the T_{exp} and T_{SKF} with speed for Grease B.Fig. 8 – Variation of the T_{exp} and T_{SKF} with speed for Grease C.

For all three greases tested, the calculated values of the friction torque using the SKF methodology are (77-88)% lower than the measured values. The largest difference was obtained for **SHELL Gadus S2 V100 2** grease (87-88)% and the smallest difference was obtained for **Cogelsa Ultraplex LT2** grease (77-79)%. The smallest difference can be explained by approximately doubled base oil viscosity at the testing temperature.

As it can be observed in the SKF methodology, the axial force F_a and the pitch diameter d_m are included in both components G_r and M_{sl} . For a very low axial force applied to a miniature angular contact ball bearing, rolling resistance becomes negligible and the hydrodynamic resistance force generated by the base oil becomes the dominant contributor for the friction torque. This explains the large differences between the experimentally determined friction torques and those predicted with the SKF model. The SKF model is a general model, primarily applicable to conventional-sized bearings used in mechanical systems.

Most friction torque calculation methodologies use the product $(v.N)^\alpha$, where the oil viscosity v is expressed in mm^2/s , the rotational speed N is expressed in rpm, and exponent α is a sub-unitary coefficient.

To be in accordance with general relations, both the experimental and calculated friction torque values were fitted using power-law relations involving viscosity and rotational speed with exponent between 0.42 and 0.59. The fitted relations are presented for each greases in Figs. 6, 7 and 8.

5. Conclusions

Using the spin-down methodology, the friction torque of a greased-lubricated miniature 7000C angular contact ball bearing was determined.

Three sorts of commercial greases **SKF LGHP 2/0.4**, **SHELL Gadus S2 V100 2** and **Cogelsa Ultraplex LT2** were tested at a temperature of (27-28)°C, within a rotational speed range between 200 and 700 rpm.

The axial load was generated by the weight of the outer ring housing and had a small value of 3.11N.

The experimental values for friction torque were compared with the friction torque values calculated using the SKF methodology.

The calculated values of the friction torque obtained from the SKF methodology are (77-88)% lower than the measured values.

After the grease is laterally displaced from the raceways, the bleed oil with a viscosity similar to the base oil realizes practically ball-race contacts. The hydrodynamic resistance forces generated in the ball-race contact lead to an increase in the friction torque.

As a general conclusion, for miniature ball bearings grease-lubricated and very low loaded the measured friction torque is more than 75% higher than the friction torque calculated using the SKF methodology based on base oil viscosity.

We consider this finding to be significant for miniature ball bearings used in systems related to precision mechanics and mechatronics.

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DETERMINAREA EXPERIMENTALĂ A MOMENTULUI DE FRECARĂ ÎN RULMENȚII MINIATURALI LUBRIFICAȚI CU UNSOARE

(Rezumat)

Momentul de frecare în rulmenții cu bile miniaturale, lubrifiați cu unsoare, depinde de numeroși factori, cum ar fi: proprietățile unsoarelor, viteza de rotație, sarcinile aplicate și geometria internă a rulmentului.

Pentru a evalua momentul de frecare în rulmenții cu bile, sunt prezentate relații empirice complexe în cataloagele celor mai importante companii producătoare de rulmenți, precum SKF, Schaeffler, NSK, relații care nu se potrivesc, de regulă, la rulmenții miniaturali.

Pe baza experimentelor anterioare, autorii au determinat momentul de frecare într-un rulment radial axial cu bile seria 7000C lubrifiat cu unsoare, pe baza metodei spin-down, considerând că momentul de frecare depinde de viteza de rotație. Experimentele au fost efectuate pe trei tipuri de unsoare la turații cuprinse între 200 și 700 rot/min, iar rezultatele momentului de frecare au fost obținute cu un software realizat în limbajul de programare Python. Parametrii de intrare și de ieșire din program au fost prezentați pentru fiecare unsoare și viteză de rotație. Dependințele dintre momentele de frecare experimentale și vitezele unghiulare au fost prezentate în diagrame logaritmice și, prin aproximare, s-au obținut relații pentru momentele de frecare corespunzătoare celor trei unsoare.

În condiții similare au fost calculate momentele de frecare utilizând metodologia SKF. Prin compararea momentelor de frecare măsurate cu momentele de frecare calculate s-a observat că momentele de frecare calculate au valori cu (77-88)% mai mici decât valorile experimentale. Autorii consideră că aceste diferențe dintre valorile momentelor de frecare determinate experimental și cele calculate sunt suficient de mari și trebuie luate în considerare în sistemele specifice mecanicii fine și mecatronice unde se găsesc rulmenți miniaturali, cu rol predominant cinematic, fără sarcini semnificative.