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ROLLING FRICTION TORQUE IN LUBRICATED CONTACTS FROM MICRO TO MACRO SCALE

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ABSTRACT

Using a modified axial ball bearing with three balls and a new analytical methodology, this paper investigates, both theoretically and experimentally, the variation of the rolling friction torque in a ball-race contact in dry and lubricated conditions. It was determined that for the microballs having 1.588 mm in diameter, the rolling friction torque determined with the help of the proposed methodology in lubricated conditions is higher with one order of magnitude than the rolling friction torque determined with the help of Biboulet–Houpert equation. Increasing the diameter of the balls to 7.938 mm, the rolling friction torque determined by the proposed methodology corresponds to the values obtained by Biboulet–Houpert equation. Also, when the maximum Hertzian contact pressure between a ball and a race in pure rolling motion has low values (0.1–0.4 GPa), it was shown that in a ball-race contact operating from dry conditions to fully flooded lubricated conditions, both in micro scale and macro scale, the rolling friction torque has a continuum increases, dry conditions realizing the minimum friction torque.

Keywords: Rolling friction torque, dry and lubricated contacts, axial ball bearing, friction test methods, dynamic modelling

1. INTRODUCTION

For a steel ball that rolls over a steel rolling track having certain conformity, as for the ball bearings, Houpert [5] introduces the following equation for the rolling friction torque, Mr:

$$Mr = 7.48 \cdot 10^{-7} \left(\frac{d}{2}\right)^{0.33} Q^{1.33} \left[1 - 3.519 \cdot 10^{-3} (k-1)^{0.806}\right] \quad (1)$$

in which $k = R_y / R_x$, R_y and R_x are the equivalent radii of curvature of the ball on rolling track in transverse and rolling directions, respectively, and d is the ball diameter. It can be noted that this rolling friction torque depends only on the material, the load and the contact geometry.

At the contact between a ball and the rolling track in a bearing, in the absence of an additional spin or gyroscopic motions, only two lines of pure rolling will exist on the contact ellipse [5]. In the rest of the contact area, the rolling will be accompanied by slips in the same or opposite direction of the motion, as a result of the geometrical effect.

As a result of the raceway's curvature an additional friction torque, Mc, is obtained, which has the following expression [6, 7]:

$$Mc = 0.08 \cdot \mu_s \cdot \frac{Q \cdot a_c^{\,2}}{Rd} \qquad (2)$$

where μ_s is the friction coefficient on the contact ellipse, which depends on the lubrication regime, a_c is the major semi-axis of the contact ellipse, Rd is the radius of the deformed contact surface, $Rd = \frac{2 \cdot d \cdot R_c}{2 \cdot R_c + d}$, in which R_c is the transversal curvature radius of the rolling track.

Thus, for a ball race contact the totally developed rolling friction torque developed can be obtained by summing Eqs. (1) and (2):

$$M_{rol} = Mr + Mc \qquad (3)$$

According to Eq. (2), the presence of the lubricant in the rolling contact must lead to a decrease in the rolling friction torque by reducing the friction coefficient μ_s on the contact ellipse.

In fact, the presence of The lubricant in the the rolling contact is responsible for an additional increase in rolling friction torque due to the hydrodynamic effect. In this context, Biboulet and Houpert [3] have established equations to compute the rolling friction torques in the point contacts, with fully flooded lubrication and pure rolling conditions, for the elastohydrodynamic (EHL) and IsoViscousRigid (IVR) regimes. In [1, 2], Biboulet Houpert equations are presented in details.

If we consider the ball-race contact from a thrust ball bearing operating at very low axial load, both in dry conditions and in fully flooded lubrication conditions, the high difference between the rolling friction torque in dry conditions and in fully flooded conditions can be observed.

In our opinion, it is normal to consider that, at low normal contact pressures, from dry to fully flooded lubricant conditions, the rolling friction torque between a ball and a race will increase i.e in the mixed lubrication regimes, the rolling friction will have a continuous increase according to the increase of the lubricant parameter Λ.

In this paper, Λ is defined as the ratio between the minimum film thickness and the equivalent roughness of the ball-race surfaces.

Using a developed analytical model to determine the rolling friction torques between a ball and the races for a thrust ball bearing from the lubricant parameter $\Lambda=0$ (dry conditions) to $\Lambda>3$ (fully flooded lubricant conditions), as described in [1, 2], this paper investigates, both theoretically and experimentally, the variation of the rolling friction torque in a ball–race contact from dry to lubricated conditions.

2. ANALYTICAL MODEL. SHORT PRESENTATION

2.1. Modified Ball Bearing Set-Up

Figure 1 depicts the modified thrust ball bearing used to study the rolling friction in ball races contacts [1, 2]. Between the rings of a thrust ball bearing three balls are mounted at an angular position of 120^0. The lower ring *1* is fastened to the table of a tribometer UMT 2 which can be rotated with the angular speed ω_1. On the upper ring *2* a disk with known weight G is attached, in order to obtain the axial loading upon the three balls. Because the balls are symmetrically placed, each ball will take over a load $Q = G/3$.

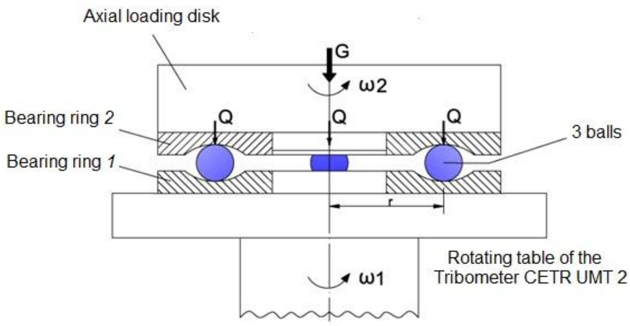

Fig. 1. Modified thrust ball bearing

The rolling friction torques between the balls and the two raceways is measured using the spin-down method. This method consists in the rotation of the ring *1* with a constant angular speed until, as a result of frictions between balls and raceways, the ring *2* together with the attached upper disk will reached the synchronism angular speed (its speed will be equal with that of the ring *1*). At this moment, the rotational table of the tribometer is suddenly stopped together with the ring *1*, while the ring *2* together with the disk, starts a deceleration process until it reaches a complete stop, as all the kinetic energy of the disk and the ring *2* is consumed by the friction in the six rolling contacts and with the air.

2.2. Forces and Moments Acting on a Ball and Computing the Tangential Force

The forces developed in the rolling process of a ball on the raceways in a modified thrust bearing and which act upon a ball, in the presence of the lubricant, are represented in Fig. 2.

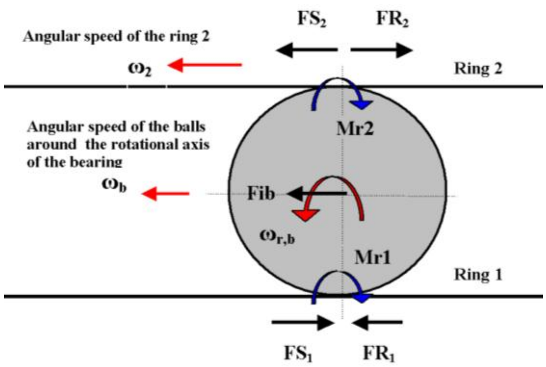

Fig. 2. Forces and moments acting on a ball

The forces *FR1* and *FR2* are the hydrodynamic forces depending on the viscosity, the speed, the normal load, the materials and the contact geometry and they can be computed with the relation presented in [1, 4, 5].

Fib, the inertial force of the ball which acts in the ball center, is given by:

$$Fib = \frac{m_b \cdot r}{2} \cdot \frac{d\omega_2}{dt} \quad (4)$$

where m_b is the ball mass and ω_b is the angular speed of the balls in the rotational motion around the bearing axis.

The rolling friction torques Mr1 and Mr2 developed between the ball and the two races are unknown in this analysis and will be determined later. FS1 and FS2 are the traction forces developed in the ball-races contacts [5].

From the equilibrium of the forces and the moments acting on a ball, the following analytical expressions for the traction forces FS1 and FS2 result:

$$FS1 = \frac{Mr1 + Mr2}{d} + \frac{Fib}{2} + FR2 \quad (5)$$

$$FS2 = \frac{Mr1 + Mr2}{d} - \frac{Fib}{2} + FR1 \quad (6)$$

At the ball-raceway *2* contact, the hydrodynamic force *FR2*, which acts on the ball, has the same direction as the hydrodynamic force acting on the raceway, while the traction force FS2 has an opposite direction on the ball and the raceway [5]. Accordingly, the ball will act on the raceway of ring 2 with a tangential force of contact Ft2, obtained by summing the two components FS2 and FR2:

$$Ft2 = \frac{Mr1 + Mr2}{d} + FR1 + FR2 - \frac{Fib}{2} \quad (7)$$

2.3. Differential Equation of the Upper Disk

During the deceleration process of the ring *2* together with the disk, the angular speed ω_2 decreases from an initial value $\omega_{2,0}$ to zero in a period of time t. From the dynamic balance of the moments that act on the ring *2* and on the disc, it can be written:

$$J \cdot \frac{d\omega_2}{dt} - 3 \cdot Ft_2 \cdot r - M_f = 0 \quad (8)$$

where J is the moment of inertia of the ensemble formed by the ring *2* and the disk and M_f is the friction torque between the disk and the air.

In [1, 2], details are presented regarding the development of Eq. (8) and the following differential equation results:

$$\frac{d\omega_2}{dt} = a \cdot [(M_{r1} + M_{r2}) + d \cdot (FR1 + FR2)] + b1 \cdot \omega_2^2 \quad (9)$$

where a and $b1$ are constants defined by the relations:

$$a = \frac{3 \cdot r}{d \cdot (J + \frac{3}{4} \cdot r^2 \cdot m_b)}; \qquad b1 = \frac{c_f}{(J + \frac{3}{4} \cdot r^2 \cdot m_b)} \quad (10)$$

In [8], a similar differential equation was developed, but neglecting the hydrodynamic forces FR and with the imposed hypothesis that the sum of friction torques ($Mr1 + Mr2$) does not depend on the angular speed, since the experiments were carried out in dry conditions.

In the presence of the lubricant, both the hydrodynamic forces and the friction torques with the lubricant, depend on the speed with exponent values between 0.66 and 0.75.

In order to integrate the differential Eq. (9) having only the angular speed as a variable, in identical geometrical and loading conditions, the following approximation is introduced:

$$[(M_{r1} + M_{r2}) + d \cdot (FR1 + FR2)] = K \cdot \omega_2 \qquad (11)$$

in which the constant K is speed independent.

The friction loss due to the pivoting effect of the balls on the raceways, which appears in a thrust ball bearing, is neglected. Using the hypothesis expressed by the Eq. (8), the differential Eq. (9) becomes:

$$\frac{d\omega_2(t)}{dt} = a \cdot K \cdot \omega_2(t) + b1 \cdot \omega_2^2(t) \qquad (12)$$

If it is noted the product $a \cdot K = a^*$ Eq. (12) is analytically solved, the expression for the angular speed $\omega_2(t)$ as function of time *t* results:

$$\omega_2(t) = \frac{a^* \cdot \exp(-a^* \cdot t + K1)}{1 - b1 \cdot \exp(-a^* \cdot t + K1)} \qquad (13)$$

where $K1$ is a constant determined from the initial condition: at $t = 0$, the angular speed of the disc is $\omega_2(t) = \omega_{2,0}$. The value of the $K1$ constant is given by:

$$K1 = \ln\left(\frac{\omega_{2,0}}{a^* + b1 \cdot \omega_{2,0}}\right) \qquad (14)$$

Considering that $\omega_2(t) = \frac{d\varphi_2(t)}{dt}$, Eq. (13) is integrated regarding the time and it results the analytical expression of the variation in time for the angular position of the disc $\varphi_2(t)$ during the deceleration period:

$$\varphi_2(t) = \frac{1}{b}\ln\left[1 - b1 \cdot \exp(-a^* t + K1)\right] - \frac{1}{b1}\ln\left[1 - b1 \cdot \exp(K1)\right] \qquad (15)$$

with the initial condition: at $t = 0$, $\varphi_2(t) = 0$.

The K parameter is established on the experimental basis imposing the following conditions: the disk has to stop at a time t_0 determined by testing and the measured cumulative position angle $\varphi_{2,total}$ to correspond to the value given by the equation:

$$\varphi_2(t_0) = \varphi_{2,total} \qquad (16)$$

In the operating conditions with bearing rings having the same conformity and roughness on raceways, if the weight of the balls is negligible, it can be considered that the torques *Mr1* and *Mr2* are identical and also the forces FR1 and FR2 are equal, the tangential speeds at the ball contacts on the two raceways being identical.

With these conditions, having determined the constant parameter K, it is possible to determine the rolling friction torque between the ball and a raceway $M_R(\omega_2)$, as a function of the disk *2* angular speed, by the following equation:

$$M_R(\omega_2) = \frac{K \cdot \omega_2}{2} - d \cdot FR(\omega_2) \qquad (17)$$

in which $FR(\omega_2)$ is computed as a function of the angular speed of the disk *2*, finally resulting the rolling friction torque in a contact, $M_R(\omega_2)$, as a function of the angular speed.

3. TESTING METHODOLOGY

The experiments were carried out on Tribometer CETR UMT 2 in the Tribology Laboratory from Mechanical Engineering Faculty of Iasi. The ring *1* was fixed to the rotational table of the tribometer so that it rotates together with the table. On the ring *1* raceway, the 3 balls at equidistant position of 120^0 each other were placed and the ring *2* was mounted over them.

For micro scale, the rings *1* and *2* were taken from an axial ball bearing 51100 and three microballs have 1.588 mm in diameter. Also, for the same rings were used three balls having 4.75 mm diameter. The normal load on each microballs was 8.68 mN and 33 mN, respectively.

For macro scale, the rings *1* and *2* of the thrust ball bearing 51205 were used and 3 balls having 7.938 mm in diameter, the normal load on each ball being 1.45 N.

White marks were traced on the ring *1* and ring *2* in order to visualize the angular position for the two elements in rotating motion.

The normal load is generated by the weight of the ring *2* (at the diameter of 1.588 mm) and by additional disks for the balls of 4.75 mm and 7.938 mm in diameter, respectively. Above the disk, a video high speed camera Philips was mounted. The images captured by the camera were recorded on the computer in real time and subsequently processed with the program VIRTUAL DUB.

The testing equipments for macro scale and micro scale are presented in Figures 3 and 4, respectively.

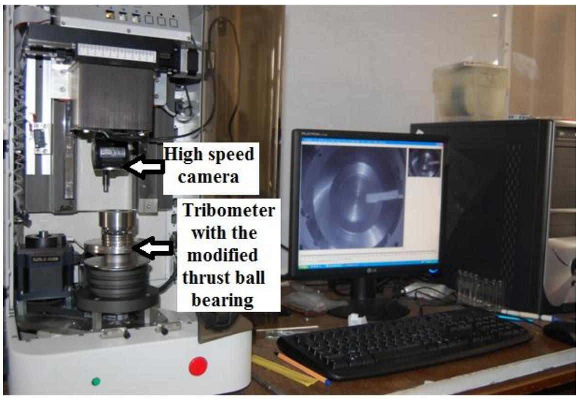

Fig. 3. The experimental equipments for macro scale

For testing conditions, two types of mineral oil were used: transmission oil with the viscosity of 0.35 Pa·s at 27^0C and mechanisms oil with 0.05 Pa·s viscosity at 27^0C. The oil quantities were a few drops in order to avoid the drag losses. The roughness parameter was measured with Taylor Hobson Profilometer, the values of the parameter *Ra* were: $Ra_r = 0.04\,\mu m$ for the two rolling races and $Ra_b = 0.03\,\mu m$ for the balls.

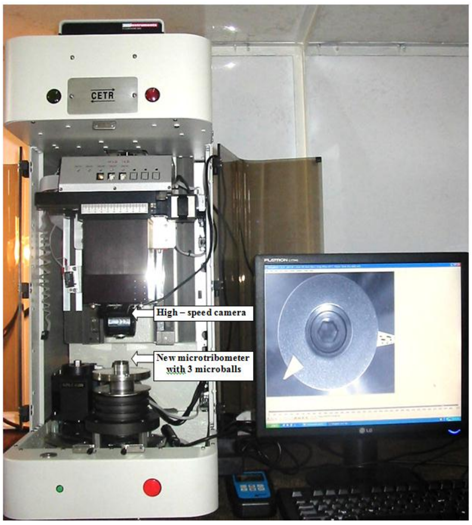

Fig. 4. The experimental equipments for micro scale

The experiments were carried out at speeds between 60 rpm and 250 rpm. For this speed range, at the existing roughness and with the used oils, it was possible to cover the entire range of the lubrication regimes.

4. EXPERIMENTAL RESULTS

A lot of tests concerning the time evaluation of the angular position φ_2 in dry and lubricated conditions were performed with the two devices. Based on the experimental results, with Eq. (17) the rolling friction torques Mr were determined as a function of the rotational speed of the ring *2*.

Figure 5 presents the comparison among the rolling friction torques determined by our methodology, in dry and lubricated conditions and the rolling friction torque determined by Biboulet–Houpert equation in lubricated conditions, the microballs having 1.588 mm diameter, under a normal load of 8.68 mN, the oil viscosity being 0.05 Pas.

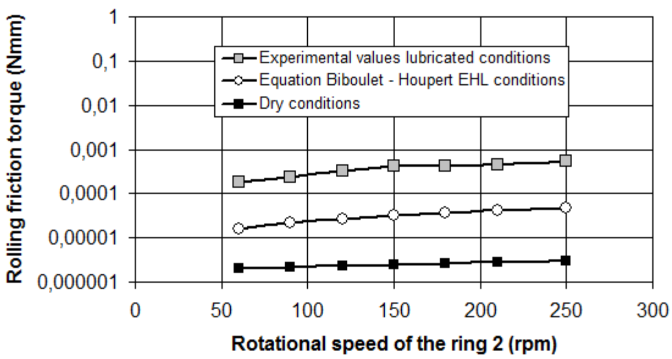

Fig. 5. The rolling friction torque vs. the rotational speed for the microballs having 1.588 mm in diameter

Figure 6 shows a comparison between the rolling friction torques determined by our methodology in lubricated conditions and the rolling friction torque determined with

Biboulet–Houpert equation in lubricated conditions (the microballs having 4.75 mm in diameter, under a normal load of 33 mN, the oil viscosity being 0.05 Pas).

Figure 7 presents the comparison between rolling friction torques determined by our methodology in lubricated conditions and rolling friction torque determined with Biboulet–Houpert equation in lubricated conditions, the balls having 7.938 mm indiameter, under a normal load of 1.45 N, the lubricant viscosity being 0.35 Pas.

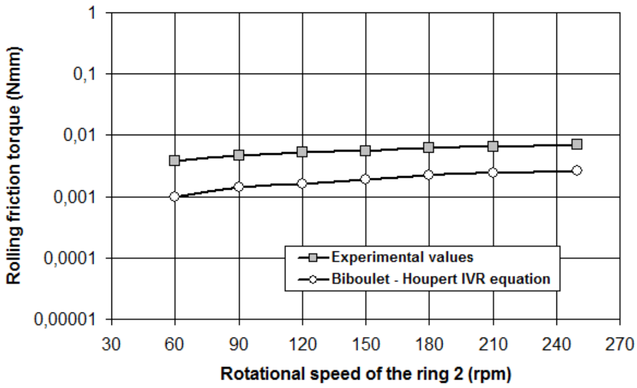

Fig. 6. The rolling friction torque vs. the rotational speed for the microballs having 4.75 mm in diameter

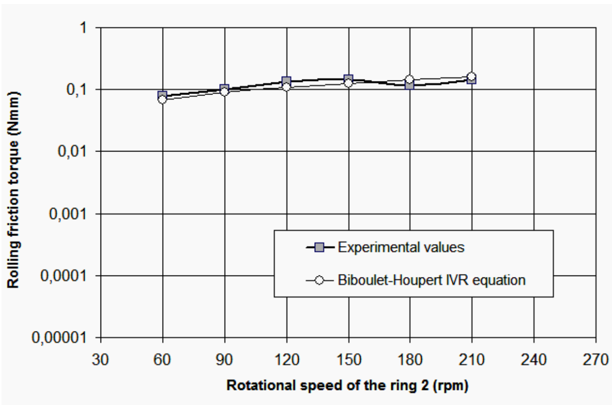

Fig. 7. The rolling friction torque vs. the rotational speed for the balls of 7.938 mm in diameter

Considering all the results obtained for dry, mixed and fully flooded lubrication conditions at macro scale, a cumulative diagram was realized in terms of the lubricant parameter Λ.

Figure 8 presents the variation of the rolling friction torque between a ball and a raceway for Λ values in the range of 0 to 17, with a contact pressure of $\sigma_0 = 0.264\,\text{GPa}$ for the balls having 7.938 mm in diameter. On the same diagram, the values of the rolling friction torques when $\Lambda > 3$, using the relations given by Biboulet-Houpert [3] are also indicated.

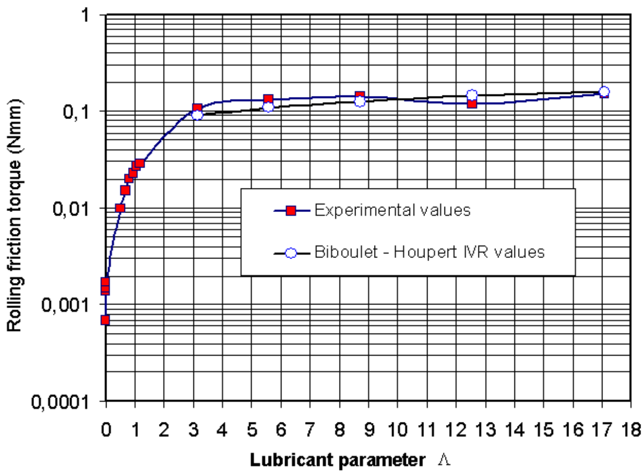

Fig. 8. The variation of the rolling friction torque between a 7.938 mm diameter ball and its raceway vs. the lubricant parameter Λ

A continuous increase of the rolling friction torque with increasing of the Λ parameter can be observed from the values having the order of magnitude of 10^{-3} N·mm in dry conditions, to 10^{-1} N·mm in the presence of a fully lubricating film.

5. CONCLUSIONS

Using a modified axial ball bearing with three balls and a new analytical methodology, our paper investigates, both theoretically and experimentally, the variation of the rolling friction torque in a ball–race contact in dry and lubricated conditions.

The rolling friction torque for three steel balls with a diameter of 1.588 mm, 4.75 mm and 7.938 mm, respectively, operating with the rotational speed between 60 rpm and 250 rpm, axially loaded with normal forces between 8.68 mN and 1.45 N and having as lubricant a mineral oil with the dynamic viscosity of 0.35 Pa·s at 27^0C - the operating temperature, was investigated.

One can notice that, for the balls having 1.588 mm in diameter, the rolling friction torque determined by the proposed methodology is higher with one order of magnitude as compared to the rolling friction torque determined by Biboulet–Houpert equation.

Increasing the diameter of the balls to 7.938 mm, the rolling friction torque determined by proposed methodology, corresponds to the values obtained by Biboulet–Houpert equation.

Also, when the maximum Hertzian contact pressure between a ball and the race in pure rolling motion has low values (0.1–0.4 GPa), it was evidenced that in a ball-race contact operating from dry conditions to fully flooded lubricated conditions, both in micro scale and macro scale, the rolling friction torque has a continuum increase, the dry conditions realizing the minimum friction torque.

Finally, a cumulative diagram including the variation of the rolling friction torque in a ball-race contact as a function of the lubrication parameter Λ between $\Lambda=0$ to $\Lambda=17$, was realized.

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A continuous increase of the rolling friction torque, from dry conditions to fully flooded lubrication conditions, was obtained.

REFERENCES

1. **Bălan M.R.**, 2011, Theoretical and experimental contributions concerning contact behaviour of rough surfaces, PhD Thesis (in Romanian).
2. **Bălan M.R., Olaru D.**, 2012, New Methodology to Evaluate Rolling Friction Torque in Dry and Mixed Lubrication Regimes, *ModTech International Conference*, Sinaia, Romania.
3. **Biboulet N., Houpert L.**, 2010, Hydrodynamic force and moment in pure rolling lubricated contacts: part II, point contacts, *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, 224, pp. 777-787.
4. **Houpert L., Leenders P.**, 1985, A Theoretical and Experimental Investigation into Rolling Bearing Friction, *Proceeding of the Eurotrib Conference*, Lyon, France.
5. **Houpert L.**, 1999, Numerical and Analytical Calculations in Ball Bearings, *Proc. of 8th European Space Mechanism and Tribology Symp.*, Toulouse, France.
6. **Houpert L.**, 2002, Ball Bearing and Tapered Roller Bearing Torque: Analytical and Experimental Results, *The 57-th STLE Annual Meeting*, Houston, USA.
7. **Johnson K.L.**, 1987, Contact Mechanics, Cambridge University Press, Cambridge.
8. **Olaru D., Stamate C., Dumitraşcu A., Prisăcaru G.**, 2011, A New Microtribometer for Rolling Friction, *Wear*, 271, pp. 842-852.