INFLUENCE OF THE CONTACT PRESSURE ON THE ROLLING RESISTANCE MOMENTS IN DRY BALL- RACE CONTACTS

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Abstract. *Based on a theoretical model and an experimental methodology for defining the rolling resistance moments in a modified thrust ball bearing having only 3 balls without cage, the authors experimentally investigated the influence of the Hertzian contact pressure on rolling resistance moments between a ball and a race. The experiments were realized with balls having diameters between 1.588 mm and 4.762 mm with maximum Hertzian pressure between 0.2GPa and 1GPa, operating for rotational speed between 60rpm to 210 rpm. The experiments evidenced that the measured values of the rolling resistance moments have higher values that the theoretical hysteresis and curvature rolling resistance moments for low contact pressure. By increasing of the contact pressure to 1GPa the experimental values for rolling resistance moments are in good agreement with the theoretical models.*

Keywords: *thrust ball bearing, 3 balls, rolling resistance, contact pressure.*

Introduction

The rolling resistant moment of a ball rolling on a raceway in dry conditions is due to elastic hysteresis losses and micro slip in the contact, curvature effects, roughness and form deviations effects [1,2]. For ball bearing applications Houpert [1,2] developed equations to evaluate the friction torque by considering the effects of hysteresis losses, micro slip in the contact, curvature effects and lubricant effect.

Olaru et al. [3] developed a methodology to determine the rolling friction resistance in a modified thrust ball bearing having only 3 balls without cage and operating in dry conditions. The experiments investigated rolling friction resistances between balls having 1.588 mm and races from 5100 thrust ball bearing operating at very low loads (between 8.2 to 33 mN) and with rotational speed from 30 rpm to 210 rpm. The results presented in [3] demonstrated that at low loads, the theoretical elastic hysteresis and curvature effect resistances do not exceed 12 percent from the experimentally rolling resistance moment. Dumitraşcu [4] determined experimentally the rolling friction resistance in dry conditions by using steel balls having between 1.588 mm to 4.762mm operating with normal loads of (8.2 - 33) mN. Was evidenced that the rolling resistance increase with the ball diameter and normal load.

Bălan et al. [5] developed theoretical model and an experimental methodology for defining the friction torque in a modified thrust ball bearing, operating in mixed and full film lubrication conditions. The experiments realized with low normal contact pressure (0.246GPa) evidenced that the hydrodynamic effect in a ball race contact is dominant.

Based on a theoretical model and an experimental methodology for defining the rolling resistance moments in a modified thrust ball bearing having only 3 balls and presented in [3,4], the authors experimentally investigated the influence of the Hertzian contact pressure on rolling resistance moments between balls and races.

Theoretical background

Figure 1 presents the modified thrust ball bearing having only 3 balls. The driving disc 1 is rotated with a constant rotational speed and as a result of rolling friction the inertial disc 2 start to rotate until to the synchronism with the disc 1. The three balls are loaded with normal loads Q =G/3, where G is the weight of the disc 2. When the rotational speed of the inertial driven disc becomes constant, $\omega_{2,0}$, the driven disc 1 is stopped and the inertial driven disc 2 starts a deceleration process until it completely stops due to friction. During this time the angular position of the disc 2, $\varphi_2(t)$, has a time variation from zero to a maximum value and the corresponding angular speed of the disc 2 has a time variation $\omega_2(t)$ from the initial value $\omega_{2,0}$ to zero.The variation of the angular position of the disc 2 were monitored by a video camera obtaining the real variation of the $\varphi_2(t)$ as function of the time.

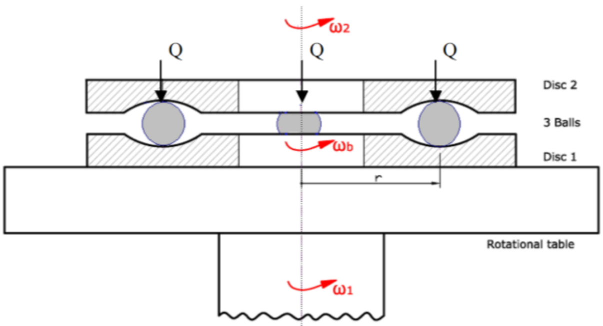

Figure 1. *Modified thrust ball bearing with 3 balls*

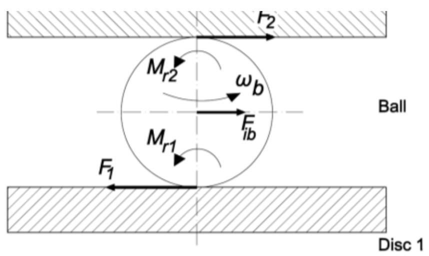

Figure 2. *The forces and the moments acting on a microball in deceleration process.*

In the deceleration process of the disc 2, when the angular speed $\omega_2(t)$ decreases from a constant value $\omega_{2,0}$ to zero, by considering both the friction forces between the disc 2 with the three microballs in contact and the friction of the disc 2 with the air, the following differential equation can be used for the inertial disc 2:

$$J \cdot \frac{d\omega_2}{dt} + 3 \cdot F_2 \cdot r = 0 \quad , \tag{1}$$

were J is the inertial moment of the disc 2, F_2 is the tangential force developed in the contact between a microball and the disc 2, r is the radius of the discs path. The friction between rotating disc 2 and air is neglected. According to figure 2 the tangential forces F_2 are determined by using the forces and moments equilibrium equations for a microball:

$$F_2 = (M_{r1} + M_{r2}) / d_b - F_{ib} / 2 \, , \tag{2}$$

where d_b is the microball diameter, Mr1 and Mr2 are the rolling resistance in dry rolling contacts between balls and races and F_{ib} is the inertial force acting in the center of the microballs .

The inertial force acting in the center of the microball is determined by the equation:

$$F_{ib} = m_b \cdot \frac{d\omega_c}{dt} \cdot r \tag{3}$$

where m_b is the mass of the microball and ω_c is the angular speed of the microball in the revolution motion around the center of the two discs and r is the radius of the rolling path. Considering the pure rolling motion of the microballs, the angular speed ω_c can be expressed as $\omega_c = 0.5 \cdot \omega_2$ and the Eq. 3 can be written:

$$F_{ib} = \frac{m_b \cdot r}{2} \cdot \frac{d\omega_2}{dt} \tag{4}$$

From Eqs. (1), (2) and (4) results following differential equation:

$$\frac{d\omega_2}{dt} = a \cdot (M_{r1} + M_{r2}) \, , \tag{5}$$

where a is a constant defined by the relation:

$$a = 12 \cdot r / \left[d_b \cdot (3 \cdot r^2 \cdot m_b - 4 \cdot J) \right] \qquad (6)$$

where m_b is the mass of the micro balls.

For dry conditions it was considered that the rolling resistance moments M_{r1} and M_{r2} are not depending on the speed and equation (5) was integrated and following analytical solutions for variation of angular speed of the disc 2 in decelerating process $\omega_2\ (t)$ and variation of the angular position of the disc 2 in deceleration process $\varphi_2(t)$ are:

$$\omega_2(t) = \omega_{2,o} - a \cdot (M_{r1} + M_{r2}) \cdot t\,, \qquad (7)$$

$$\varphi_2(t) = \omega_{2,o} \cdot t - a \cdot (M_{r1} + M_{r2}) \cdot t^2 / 2\,, \qquad (8)$$

Following initial conditions were imposed: for $t = 0$, $\omega_2(t) = \omega_{2,0}$ and $\varphi_2(t) = 0$.

For an imposed geometry of the microballs and of the discs, and for the imposed initial angular speed of the disc 2, $\omega_{2,0}$, the parameter a, from Eq. (5) is constant, and must be determined only the sum of the resistance moments $(M_{r1} + M_{r2})$ according to the experiments. So, for every experiment was imposed that at the time t_{limit} measured from the start of deceleration process to the stop of the disc 2, the analytical value of the angular position given by Eq. (8) to be equal with $\varphi_{2,\lim it}$ measured by video registration.

Experimental methodology

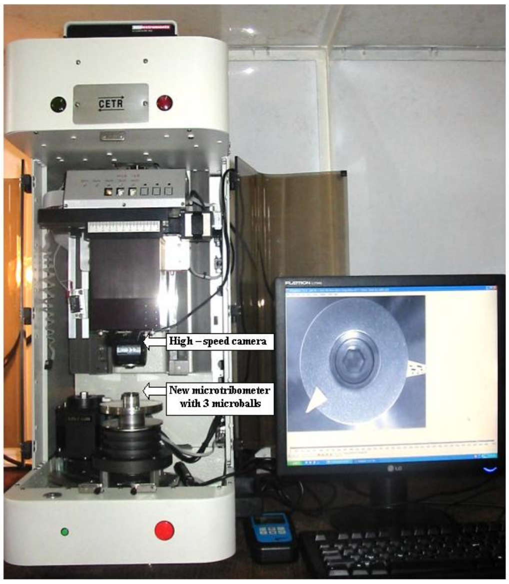

Figure 3. *General view of the experimental equipments*

In figure 3 is presented the modified thrust ball bearing mounted on the rotational table of the CETR-UMT Tribometer. The discs 1 and 2 are the steel rings of an axial ball bearing (series 51100) having a rolling path with a radius r = 8.4 mm and a transversal curvature radius R_c = 2.63 mm. The weight of the disc 2 imposed a minimum normal load on every microball Q = 8.68 mN. To increase the normal load on the microballs, a lot of new discs similar to the disc 2 were attached on the disc 2, thus obtaining the following values for the normal load: 8.8 mN, 15 mN, 33 mN. The microballs having 1.588 mm, 1.97 mm, 2.47 mm and 4,76 mm were used in the experiments. The roughness of the races were Ra = 0.030 μm and the roughness of the balls were Ra = (0.02-0,025) μm. The tests were realized for the following rotational speed of the disc 2 : 60 rpm, 90 rpm, 120 rpm, 150 rpm, 180 rpm, 210 rpm. For every set of three balls were realized experiments for normal load of 8.8 mN, 15 mN and 33 mN and were obtained values for the sum of the rolling friction moments Mr1+Mr2. Because the roughness of the two races was equally and neglecting the weight of the balls compared to Q was considered that rolling resistance moment Mr on a ball-race contact is Mr = (Mr1+Mr2)/2. The maximum contact pressure in balls-races contacts varied between 0.08GPa (for balls having 4.76mm) and 0.47 GPa (for balls having 1.588mm).

Experimental results

In figure 4 and figure 5 are presented the variation of the rolling friction resistance Mr as function of rotational speed and normal loads for balls having 1.588mm and 4.76mm respectively. Excepting the minimum normal load (8.8 mN) no important variations of the rolling resistance moment were obtained between 60 rpm and 210 rpm. Also was evidenced important increasing of the rolling resistance moment by increasing of the balls diameter.

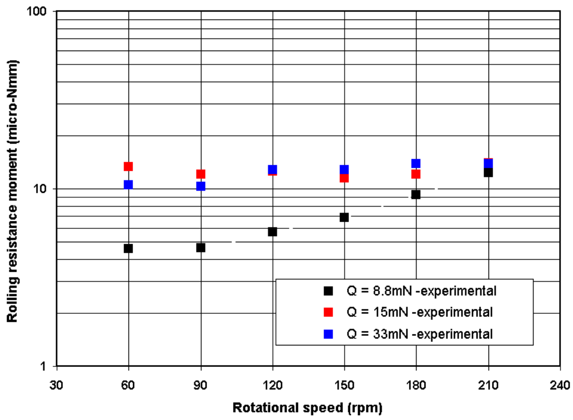

Figure 4. *Rolling resistance moment Mr for balls having 1.588 mm*

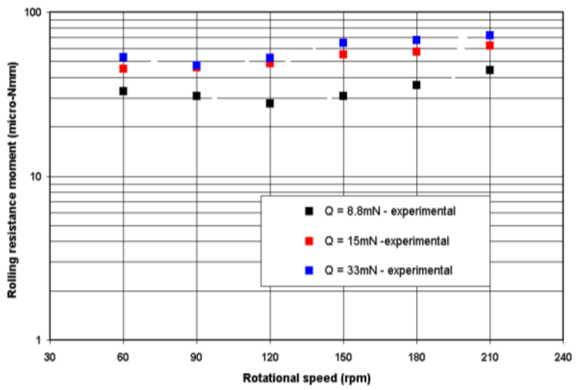

Figure 5. *Rolling resistance moment Mr for balls having 4.76 mm*

In figure 6 are presented the variations of the rolling resistance moment Mr for all four types of balls and for the three normal loads. The results correspond to 120 rpm.

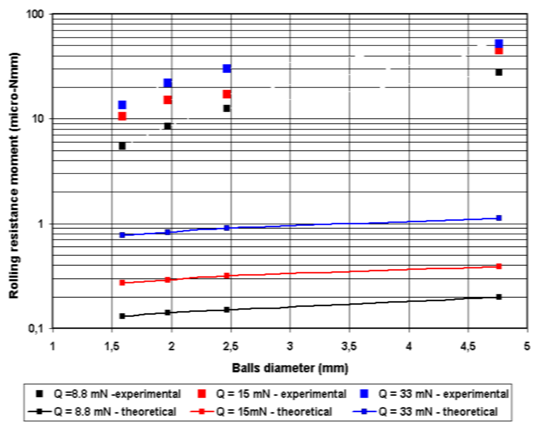

Figure 6. *Experimental and theoretical rolling resistance moment Mr for balls having following diameters:1.588 mm, 1.97 mm, 2.47 mm and 4.76 mm*

Supplementary, on the some diagrams are presented the theoretical rolling resistance moment for a dry ball-race contact determined as a sum of following two resistance moments MER and MC, where MER is rolling resistance caused by the elastic hysteresis and MC is caused by the curvature effect. Following equations were used [1,2]:

$$MER = 7.48\,10^{-7}\,(d/2)^{0.33}.Q^{1.33}.\left\{1 - 3.519\ \ 10^{-3}\,(k-1)^{0.8063}\right\}, \quad [\text{N}\cdot\text{m}] \qquad (9)$$

where R_x and R_y are the reduced radii of curvature in the rolling direction and the transverse direction, respectively, and k represents the ratio R_y/R_x.

$$MC = 0.16\cdot\mu_s\cdot Q\cdot a^2/d \quad ,[\text{Nm}] \qquad (10)$$

where μ_s is the average friction coefficient on the contact ellipse and a is the major semi-axis of the contact ellipse. The sliding friction coefficient μ_s has a maximum value of the order of 0.11 in dry contact conditions [3,4].

It can be observed from figure 6 that theoretical rolling resistance moments are less than 10% from experimental values. In Figs. 7 and 8 are presented the variation of the experimental rolling resistance moments with maximum Hertzian contact pressure between balls and races and with ball-race contact area *Ac*, respectively.

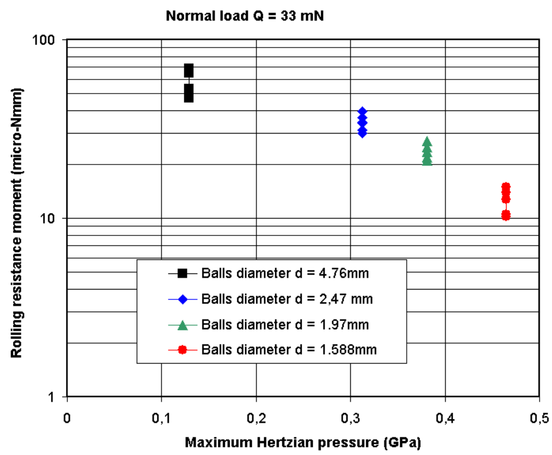

Figure 7. *Variation of the experimental resistance moments as function of contact pressure*

and maintaining the normal load, increases the maximum contact pressure between balls and races but decreases the rolling resistance moment. By considering the ball-race contact area Ac ( ) where a and b are the major and minor semi-axis of contact ellipses, it can be observed in figure 8 an increasing of the resistance moment according to increasing of the ball-race contact area. These results suggest that the ball - race contact area influences direct the rolling resistance moments in low normal loads while the contact pressure is not an essential parameter for rolling resistance moment.

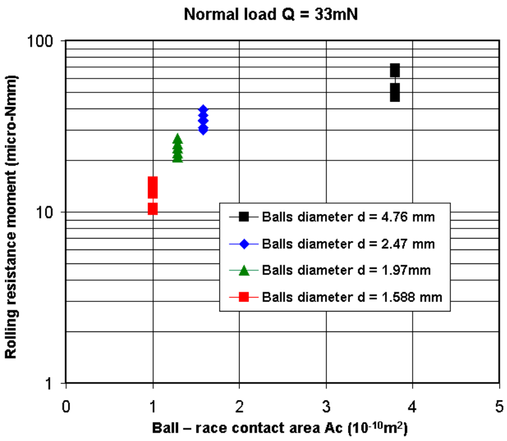

Figure 8. *Variation of the experimental resistance moments as function of contact area*

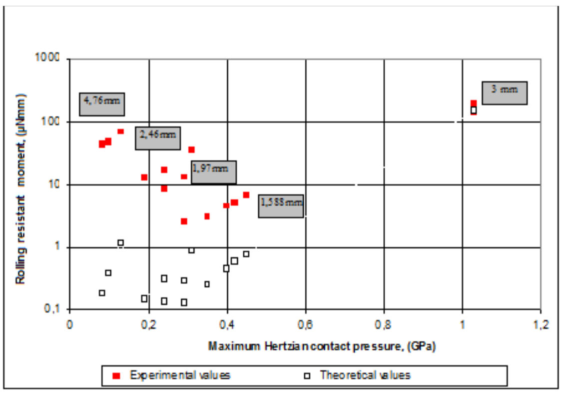

Figure 9. *Variation of the experimental and theoretical rolling resistance moments as function of contact pressure*

Based on these observations it can be explained that for low load conditions the roughness and form deviation are essentially for rolling resistance moment (depending of the contact surfaces) while the hysteresis effect generated by the contact pressure are very small influence.

If the contact pressure increases to about 1GPa or more it was observed a good correspondence between theoretical rolling resistance moment and experimental results. In figure 9 are present global diagrams including the experimental and theoretical rolling resistance moments for all tested balls with normal loads between 8.8mN and 33mN. Supplementary, in the figure 9 is included the tests realized with a modified 51205 thrust ball bearing having 3 balls with diameter of 3mm loaded with 1.45 N, that means a maximum ball-race contact pressure of 1.03 GPa.

Conclusions

The experiments realized with small balls and low normal loads evidenced a high difference between experimental and theoretical rolling resistance moments (of about two orders of magnitude for a contact pressure of 0.08 GPa).

By increasing of the normal load increases the ball-race contact pressure and the difference between experimental and theoretical values of rolling resistance moments decrease to one order of magnitude (at 0.45GPa). Increasing the ball-race contact pressure over 1 GPa leads to a concordance between experimental and theoretical rolling resistance moments.

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