Influence of the cage on friction torque in low loaded thrust ball bearing operating in dry conditions

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Abstract. The authors investigated analytically and experimentally the friction torque in a modified thrust ball bearing operating at very low axial load in dry conditions by using only three balls and a cage. The experiments were conducted by using spin–down methodology. The results evidenced the influence of the sliding friction between the cage and the balls on the total friction torque. It was concluded that at very low loads the friction between cage and balls in a thrust ball bearing has an important contribution on total friction torque.

1. Introduction
The friction between the balls and the cage are usually ignored in evaluation of the friction torque for a rolling bearing. So, for a standard thrust ball bearing the SKF methodology [1] evaluate the total friction torque as function of oil viscosity, rotational speed, axial load, the bearing mean diameter and sealing system. No informations about the influence of the cage are included in the SKF methodology.

The research realized by Bălan et al. [2] on a 51205 thrust ball bearing evidenced that at very low loads the friction between cage and balls has an important contribution on total friction torque in the thrust ball bearing. The authors used spin–down methodology applied on 51205 thrust ball bearing, lubricated with oil having viscosity of 0.35 Pa·s at rotational speed between 100 to 400 rpm and loaded with an axial force of 4.26 N. The friction between balls and a steel cage estimated by Balan et al. [2] has a contribution between 10% and 40% from the measured friction torque, depending on the rotational speed.

The presence of the lubricant in a low loaded thrust ball bearing leads to important increases of the friction torque as result of hydrodynamic rolling forces developed in ball-race contacts [2], [3], [4].

For dry conditions Olaru et al. [5] determined the rolling friction torque in a modified thrust ball bearing having only three balls equidistance positioned at 120 degrees and without cage. So, for a modified 51100 thrust ball bearing having three balls of 1.588 mm with axial loads between 25 mN and 100 mN the authors obtained a total friction torque of (63 – 220) μN·mm.

In the present paper the authors developed an analytical model to evaluate the effect of friction between the balls and the steel cage in dry conditions for a 51100 thrust ball bearing. The numerical model was verified by experiments realized in dry conditions for a 51100 thrust ball bearing having only 3 balls with and without cage according to the spin–down methodology presented in [5].

2. Analytical model of the friction between balls and cage in dry conditions
The proposed analytical model is realized for a modified thrust ball bearing having only three balls and the cage, the three balls being positioned at 120 degrees. The lower race is mounted on the
rotating table of the CETR UMT–2 tribometer. On the upper cage is attached a disc of known weight, $G$ as is presented in figure 1.

![Figure 1. General presentation of the thrust ball bearing mounted on the rotating table of the tribometer.](image1)

The spin–down method has been used for analytical model. So, by imposing a constant angular speed both of the rotating table and on the lower race of thrust ball bearing, the upper race and disc reach an angular speed equal the one of lower race, over a time, as result of friction between balls and two races. Then the rotational table and the lower race are suddenly stopped, while upper race starts a deceleration motion until to the stop of rotating. In the deceleration motion of the upper race the forces and moments acting on a ball are presented in figure 2 and figure 3.

![Figure 2. The forces and moments acting on a ball in the ball rolling direction.](image2)
Figure 3. The moments acting on a ball in the ball pivoting motion.

In figure 2 are presented the forces and moments acting on a ball in the rolling direction of the ball. So, $Q_1$ and $Q_2$ are the normal forces in ball-races contact having following values: $Q_2 = G/3$ and $G = G_d + G_r$, where $G_d$ is the weight of the disc and $G_r$ is the weight of the upper race.

Because the cage is sustained only on the three balls, a supplementary normal force acts on a ball $Q_c$ given by relation $Q_c = G_c / 3$, where $G_c$ is the weight of the cage. By considering also the weight of the ball $Q_b$, it can be obtained the normal force between the ball and the lower race $Q_1$ from the following equation:

$$Q_1 = Q_2 + Q_c + Q_b$$

In the rolling direction $Ox$ acts the following forces: tangential forces $F_{S1}$ and $F_{S2}$ developed in the ball-race contacts and inertial force $F_{bc}$ generated by the inertia of the cage with three balls in the decelerating process.

If it is considered pure rolling of the three balls on the races, in a thrust ball bearing the angular speed of the cage with balls $\omega_{bc}$ is given by the equation:

$$\omega_{bc} = \frac{1}{2} \cdot \omega_2$$

The inertia force $F_{bc}$ can be developed from the inertia moment of the cage and of the three balls by following equation:

$$F_{bc} = (m_c + 3 \cdot m_b) \cdot r \cdot \frac{d\omega_c}{dt}$$

where $m_c$ is the mass of the cage, $m_b$ is the mass of the ball and $r$ is the rotating radius of the balls (see figure 1). The moment of inertia of the cage was approximated by equation for thin cylindrical shell with open ends, having mass $m_c$ and the average radius $r$. 


The tangential force $F_{s2}$ is obtained by the equilibrium of the forces and moments acting on the ball according to the figure 2.

The rolling resistant moment $MER$ is generated by the elastic hysteresis losses in the rolling process. For steel ball and race, the equation for $MER$ is as follows [5]:

$$MER = 7.481 \cdot 10^{-7} \cdot \left(\frac{d}{2}\right)^{0.33} \cdot Q^{1.33} \cdot \left[1 - 3.519 \cdot 10^{-3} \cdot (k - 1)^{0.806}\right]$$

$$[N\cdot m]$$ (4)

where $d$ is the ball diameter, in m, $Q$ is the normal load, in N, $k = R_y / R_x$, with $R_y$ and $R_x$ defined as the equivalent radius of curvature in the $y$ direction and $x$ direction, respectively ($x$ direction being the rolling direction and $y$ direction being perpendicular to the rolling direction).

For the ball-race contact in thrust ball bearing, the ratio $k$ has the following equation:

$$k = \frac{2 \cdot R}{2 \cdot R_y - d}$$

(5)

where $R_y$ is the curvature radius of the race.

The friction moment $M_{bc}$ acting on a ball as a result of the ball-cage contact when the ball realizes the rolling motion has been determined in the Appendix 1 and has the following equation:

$$M_{bc} = \frac{2 \cdot \mu \cdot G \cdot r_{bc} \cdot d}{3 \pi \cdot \sqrt{d^2 - 4 \cdot r_{bc}^2}}$$

(6)

where $\mu$ is the friction coefficient between ball and cage in dry conditions.

From the equilibrium of the forces and moments acting on the ball in rolling direction results the equation of the tangential force $F_{s2}$:

$$F_{s2} = \frac{(MER1 + MER2 + M_{bc})}{d} \cdot F_{s2} - \frac{F_{bc}}{2}$$

(7)

The tangential force $F_{s2}$ for all three balls realize a resistant torque $T_{z2}$ acting on the rotating upper race and disc in decelerating process given by relation:

$$T_{z2} = 3 \cdot F_{s2} \cdot r$$

(8)

By considering the expression of the tangential force $F_{s2}$, see equation (7), the friction torque $T_{z2}$ has the following three components:

$$T_{z2} = T_{ZMER} + T_{Zbc} - T_{Zibc}$$

(9)

where $T_{ZMER} = 3 \cdot \frac{d}{d} (MER1 + MER2) \cdot r$; $T_{Zbc} = 3 \cdot \frac{d}{d} M_{bc} \cdot r$; $T_{Zibc} = \frac{3}{2} F_{s2} \cdot r$.

Supplementary, as shown in figure 3, on the upper race act three pivoting torques $M_{p2}$.
The pivoting torque $M_{P2}$ is a sum of the two components $M_{P1}$ and $M_{Pc}$. The pivoting torque between the ball and lower race $M_{P1}$ can be calculated by equation [5]:

$$M_{P1} = \frac{3}{8} \cdot \mu_p \cdot Q_1 \cdot a_{c1}$$  \hspace{1cm} (10)

where $\mu_p$ is the friction coefficient between ball and race in pivoting motion and $a_{c1}$ is semi major axis of the contact ellipse between ball and race.

The pivoting torque in ball-race contact for dry conditions was developed in Appendix 2 and has following equation:

$$M_{PC} = \frac{1}{3} \mu_c \cdot G_c \cdot r_{bc} \cdot d \sqrt{d^2 - 4 \cdot r_{bc}^2}$$  \hspace{1cm} (11)

The total friction torque acting on the upper race and disc becomes:

$$T_{tot} = T_{ZMER} + T_{Zbc} + T_{Zibc} + T_{ZP1} + T_{ZPC}$$  \hspace{1cm} (12)

where $T_{ZP1} = 3 \cdot M_{P1}$ and $T_{ZPC} = 3 \cdot M_{PC}$.

3. Numerical results

The proposed analytical model was applied to the 51100 thrust ball bearing having only three balls and a steel cage. The geometrical parameters for balls-race and ball-cage contacts are: balls diameter $d = 4.762$ mm (3/16’’), race curvature radius $R_c = 2.63$ mm, rolling radius of the balls $r = 8.4$ mm, the radius of the cage hole $r_{bc} = 2.05$ mm. The mass of the disc and upper race $m_d = 128.35$ grams, the mass of the cage $m_c = 2.29$ grams and the mass of a ball $m_b = 0.44$ grams.

Equation (12) was numerical solved by imposing a friction coefficient $\mu_p = 0.15$ [6] and for the friction coefficient between ball and cage $\mu_c$ we’re considered a variation between 0.1 to 0.3. By the experiments presented in the next section, the angular deceleration $\frac{d\omega}{dt}$ does not exceeds 1 rad/s$^2$ for maximum rotational speed of 210 rpm and this value was considered in equation (3).

The variation of the total friction torque $T_{tot}$ obtained from equation (12) as function of the friction coefficient between balls and cage $\mu_c$ is presented in figure 4.

It can be observed that the value of friction coefficient between balls and cage in dry contact has an important contribution to the value of the total friction torque $T_{tot}$. Also, the two components of the total friction torque $T_{ZMER}$ and $T_{Zibc}$ can be neglected having values between $10^{-7}$ and $10^{-8}$ N·m.
4. Experimental procedure

4.1. Testing equipments

The experiments were carried out by using the Tribometer CETR UMT 2, from the Tribology Laboratory of Mechanical Engineering Faculty from Iasi. A thrust ball bearing 51100 was mounted on the rotational table of the tribometer. On the upper race was attached a disc of known weight \( G \) which determine the axial load acting on the thrust ball bearing, as in figure 1.

To measure the rolling friction torque in a thrust ball bearing the spin–down method developed by the authors was used in [2], [3], [4], [5]. Thus the lower race is rotate with given angular speed \( \omega_1 \). The upper race and disc start to rotate by friction generated in the thrust ball bearing with an angular speed \( \omega_2 \). After a period the angular speeds of the two races becomes approximate equals and in this moment the rotating table with lower race is stopped and all the kinetic energy of the disc and upper race are dissipated by friction into thrust ball bearing.

The upper race and disc start a deceleration process from the angular speed \( \omega_2 \) to zero. The angular position for the disc was visualized by using white marks and a video camera with 90 frames/second. The images captured by the camera were recorded on the computer in real time and subsequently processed with an adequate program. A general view of the testing equipments is presented in figure 5.

Two types of experiments were realized. First, the experiments were realized by using the two races of the 51100 thrust ball bearing, the disc and only three balls having diameter \( d = 4.762 \text{ mm (3/16")} \) without the cage. The experiments were realized in dry conditions for rotation speed of the upper race between 60 and 210 rpm. Secondly, the experiments were realized with the same races and disc but using three balls mounted in a steel cage at 120 degrees. The experiments were realized in dry conditions and at same rotational speed of upper race between 60 and 210 rpm.

From both types of experiments were monitored the total angular position of the upper race and the time in deceleration process specific to the spin-down method.
4.2. Experimental friction torque in dry conditions

During the deceleration process of the upper race 2 and attached disc, the angular speed $\omega_2$ decreases from an initial value $\omega_{2,0}$ to zero during a time $t_{max}$. Using a dynamic balance of the moments acting on the upper race 2 and attached disc and neglecting the friction between disc and air, one obtains:

$$J \cdot \frac{d\omega_2}{dt} + T_{ztot} = 0$$

(13)

where $J$ is the moment of inertia of the ensemble formed by the upper race and disc which can be calculated by equation:

$$J = 0.5 \cdot m_d \cdot R^2,$$  

[Kg·m²]  

(14)

For the tests having only three balls (without cage), in the equation (12) of the total friction torque $T_{ztot}$ must be excluded the two components generated by cage balls contacts $T_{Zic}$ and $T_{ZPC}$. Also, the mass of the cage must be excluded from the components $T_{ZMER}$, $T_{Zhec}$ and $T_{zP1}$.

For the tests having three balls and the cage the total friction torque $T_{ztot}$ is given by equation (12).

Because the friction torques generated by inertial effect in deceleration process have values with two order of magnitude than the other components from total friction torque, can be neglected and it can be observed that all the other components from total friction torque is not depending of the angular speed $\omega_2$.

So, the equation (13) can be integrated resulting the following relations for variation of the angular speed $\omega_2(t)$ and angular position $\varphi_2(t)$:

$$\omega_2(t) = \omega_{2,0} - \frac{T_{ztot}}{J} \cdot t$$

(15)

$$\varphi_2(t) = \omega_{2,0} \cdot t - \frac{T_{ztot}}{2J} \cdot t^2$$

(16)
where $\omega_{2,0}$ is the angular speed of the upper race and disc when the lower race is stopped and start the decelerating process.

The time $t_{\text{max}}$ and the total angular position $\phi_{2,\text{max}}$ cumulated by the upper race and disc are experimentally determined for every type of tests and for every rotational speed. From equation (16), by imposing the experimental values for $t_{\text{max}}$ and $\phi_{2,\text{max}}$ one can define for every rotational speed the experimental values of $T_z$ in dry conditions:

$$T_{z,\text{tot}} = 2 \cdot J \cdot (\omega_{2,0} \cdot t_{\text{max}} - \phi_{2,\text{max}})$$

(17)

4.3. Experimental results

In figure 6 are presented the variations of the measured total friction torque $T_{z,\text{tot}}$ both for three balls without cage and for three balls with cage.

The experimental values of total friction torque presented in figure 6 confirm the existence of supplementary friction sources in balls-cage contacts, the differences between the values of total friction torques in the two types of tests having values between $6 \cdot 10^{-5}$ N·m and $7 \cdot 10^{-5}$ N·m.

As was presented in [4] the theoretical rolling friction moment $MER$ given by equation (4) applied for small ball-races contacts and very low loaded leads to values smaller that the measured values and supplementary friction resistance moments generated by roughness, by small dimension deviations or small sliding processes must be considered. For example, the total friction torque calculated by using equation (12) adapted only for three balls without cage, for dry conditions leads to a total friction torque $T_{z,\text{tot}} \approx 4.4 \cdot 10^{-6}$ N·m that means about 40% from average measured total friction torque ($T_{z,\text{tot, balls}} \approx 1.75 \cdot 10^{-5}$ N·m) as it can see in figure 6.

In these circumstances we consider that to evidence only the influence of the cage on total friction torque we must use instead of the equation (12) following equation:
\[ T_{\text{tot\_cage}} = T_{\text{tot\_balls}} + T_{zbc} + T_{ZPC} \]  \hspace{1cm} (18)

where \(T_{\text{tot\_balls}}\) is the total friction torque measured in conditions with only three balls without cage.

The variation of total friction torque given by equation (18) as function of the friction coefficient between balls and cage \(\mu_c\) is presented in figure 7.

**Figure 7.** Variation of total friction torque for three balls and cage according to equation (18) as function of friction coefficient between balls and cage.

From figure 7 it can observe that if the friction coefficient between balls and cage in dry condition is 0.3 the total friction torque for three balls and cage determined by equation (18) have values of \(7.5 \cdot 10^{-5}\ \text{N\cdot m}\) and the average total friction torque experimental determined is about \(8.5 \cdot 10^{-5}\ \text{N\cdot m}\).

This result confirm that the proposed two components for friction moments between balls and cage in dry conditions realize a good approximation of the friction contribution between balls and cage in small thrust bearings, operating in dry conditions.

More investigations are in working for lubricated conditions.

5. Conclusions

The authors investigated analytically and experimentally the friction torque in a modified thrust ball bearing operating at very low axial load in dry conditions by using only three balls and a cage with intention to evaluate the influence of the friction between the balls and cage on total friction torque of the bearing. Two analytical models were realized in order to determine the friction moments between a ball and a steel cage in a thrust ball bearing by considering both the rolling and pivoting motion of the ball. The two friction moments were included in a general formula to evaluate the total friction torque for a thrust ball bearing with three balls and a cage. The numerical results of the total friction torque were determined for a 51100 thrust ball bearing, having three balls with diameter of \(d = 4.762\ \text{mm}\) mounted in a steel cage and axial loaded with 1.29N. The important influence of the friction coefficient between balls and cage has been evidenced in dry conditions. To validate the proposed models for the two friction moments developed between balls and cage, two types of experiments were realized: (i) a set of experiments in which was used only three balls without cage and (ii) a set of experiments in which was used three balls mounted in a steel standard cage. All experiments were realized in dry conditions with the same axial load and for rotational speed of the modified thrust ball bearing between 60 and 210 rpm. The tests were realized by using spin-down method and to calculate
the total friction torque in the two cases the simple integration of the differential equation of the upper race and disc was operated. Both the analytical model and experimental values evidenced that the friction between the balls and the standard steel cage has an important contribution to increases the friction torque of the thrust ball bearing. By correlation the theoretical model with experimental results was evidenced that if the friction coefficient between balls and cage in dry contact has a value of about 0.3, the theoretical model can approximate with good accuracy the experimental values. New models for lubricated conditions have been developed and will be presented in a future paper.

6. Appendices

Appendix A
In figure A1 is presented the contact between a ball and cage.

![Figure A1. Ball–cage contact in rolling direction.](image)

The weight of the cage on every ball \( G_c/3 \) is distributed on the circular contact having radius \( r_c \) on a radial direction as in figure A1.

The distributed force \( q \) has the following equation:

\[
q = \frac{G_c}{3} \cdot \frac{1}{2 \cdot \pi \cdot r_c} \cdot \frac{1}{\cos \alpha}
\]  

(A1)

The rotation of the ball with the angular speed \( \omega_c \) in the hole of the cage, as in figure A2, generated a friction moment \( M_{bc} \).
The elementary friction moment generated at distance $x$ from the rotation on axis $Ox$ can be expressed by equation:

$$ dM(\varphi) = \mu_c \cdot q \cdot r_c^2 \cdot \sin \varphi \cdot d\varphi $$

(A2)

The total friction moment generated between ball and cage from the rotating axis $oy$ is obtained by integration the elementary moment $dM(\alpha)$ between $\alpha = 0$ and $\pi / 2$ and multiply by four:

$$ M_{bc} = 4 \cdot \mu_c \cdot q \cdot r_c \quad \text{(A3)} $$

From equation (A1) and (A3) and expressed $\cos \alpha$ as function of $r_c$ and $d$, the following equation was obtained:

$$ M_{bc} = \frac{2 \cdot \mu_c \cdot G_c \cdot r_c \cdot d}{3 \cdot \pi \cdot \sqrt{d^2 - 4 \cdot r_c^2}} $$

(A4)

**Appendix B**

In figure B1 is presented the elementary friction force $dF_c$ developed between ball and cage in pivoting motion.

In dry conditions it can be written:

$$ dF_c = \mu_c \cdot q \cdot dl \quad \text{(B1)} $$
By including in equation (B1) the relationships for $q$ and $dl$ has been obtained the elementary friction force $dF_c$:

$$dF_c = \mu_c \cdot \frac{G_c}{6 \cdot \pi \cdot r_c \cdot \cos \alpha} \cdot r_c \cdot d\varphi$$ (B2)

The elementary friction moment $dM_{pc}$ becomes:

$$dM_{pc} = \mu_c \cdot \frac{G_c}{6 \cdot \pi \cdot \cos \alpha} \cdot r_c \cdot d\varphi$$ (B3)

From figure A1, it can be expressed $\cos \alpha$ by equation:

$$\cos \alpha = \sqrt{d^2 - 4 \cdot r_c^2} \cdot \frac{1}{d}$$ (B4)

By integrating equation (B3) from $\varphi = 0$ to $\varphi = 2\pi$ results:

$$M_{pc} = \frac{1}{3} \cdot \frac{\mu_c \cdot G_c \cdot r_c \cdot d}{\sqrt{d^2 - 4 \cdot r_c^2}}$$ (B5)

7. References
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