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Power loss in grease lubricated ball bearings

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Abstract. The authors theoretically investigated and then validated the power loss generated both by sliding and rolling in a modified thrust ball bearing with 3 balls operating at low axial load and grease lubrication. Based on ball forces and moment equilibrium, it has been determined the tangential forces on contact ellipses. The hydrodynamic rolling friction forces have been determined by using Biboulet & Houpert's methodology considering base oil viscosity of the grease. Total power loss has been obtained as a sum of two components: power loss generated by sliding and power loss generated by rolling. Also, total friction torque has been theoretically determined and, as an alternative method, the total power loss has been calculated as a product between total friction torque and angular speed. Finally, based on previously experimental results, the power loss has been validated as the product between experimentally total friction torque and angular speed. A good correlation between the power losses determined by all three methods for 100, 200, 300 and 400 rpm have been obtained.

1. Introduction

The lubrication mechanism of the greases is a complex one and depends on grease penetration point, film formation, rheological behaviour, grease life [1]. The power loss dissipated in a greased thrust ball bearing is the result of some friction sources as: rolling and sliding in balls - race contacts, sliding between balls and cage, drag loss of balls and cage in grease. In practical applications the power loss dissipated in a ball bearing can be estimated as a product between friction torque and angular speed. To determine the friction torque the SKF methodology can be used if the applied load is equal to or larger than the recommended minimum load of the bearing [2]. This methodology generally includes ball bearing geometry, rotational speed, applied loads and base oil viscosity of the grease.

Cousseau et al. [3,4] determined by experiments the friction torque in a 51107 thrust ball bearing axially loaded with 7000 N and rotational speed ranging between 500 rpm and 2000 rpm. Various types of greases have been used by authors. Considering the SKF methodology, the authors obtained different values for the friction coefficient μ_{sl} as function of the grease type and rotational speed.

For very low axial loads, Bălan et al. [5] determined the friction torque in a modified thrust ball bearing with 3 balls and no cage. The authors evaluated the influence of the lubricant on the friction torque using the Biboulet & Houpert model for hydrodynamic rolling forces in ball and races contacts [6] and experimentally determined the friction torque by spin-down method to validate the theoretical model.

Ianuş et al. [7] determined both theoretical and experimental the friction torque in a modified thrust ball bearing lubricated with two types of greases. Considering the Biboulet & Houpert model for hydrodynamic rolling forces, the authors obtained good correspondence between theoretical and experimental friction torque values.

Popescu et al. [8] proposed a theoretical model to evaluate the ball motion in a modified 7205B angular contact ball bearing based on minimisation of the power loss in ball and race contacts. It considered a constant friction torque in ball-race contacts and obtained total power loss in a ball-race contact as a sum of sliding and rolling power losses.

In the present paper the authors propose an extension of the methodology used in [8] considering supplementary the influence of the hydrodynamic forces generated as result of grease lubrication and ball inertial effects. Also, to determine only the power loss in ball-race contacts, the authors applied the proposed methodology on a 51205 modified thrust ball bearing having only 3 balls and no cage. As an alternative method, the total power loss has been calculated as a product between total friction torque and angular speed. Finally, based on previously experimental results obtained in [7], the power loss has been validated as the product between experimentally total friction torque and angular speed.

2. Analytical model

2.1 Sliding speeds on contact ellipses

For a thrust ball bearing axially loaded with upper race rotating and fixed lower race, the angular and orbital speeds of the balls ω_b and ω_c respectively, can be determined with the following simplified relations [9]

$$\omega_b = \frac{\pi \cdot n_u}{60} \cdot \frac{d_m}{d_b} \tag{1}$$

$$\omega_c = \frac{\pi \cdot n_u}{60} \tag{2}$$

where n_u is rotational speed of the upper race, d_m is mean diameter of the ball bearing and d_b is ball diameter.

On a slice from the lower and upper ball-race contact ellipses, denoted with l and u indexes, respectively the sliding speeds (in the rolling direction) v_{sl} and v_{su} are determined by the following relations [9]:

$$v_{sl,u}(x, n_u, \omega_c, \omega_b) = v_{l,u}(x, n_u, \omega_c) - v_{bl,u}(x, \omega_b) \tag{3}$$

where x is the distance from the centre of contact ellipse to a slice, as it is presented in Fig. 1 [8].

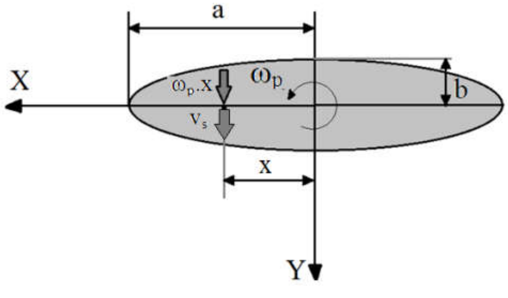

Figure 1. Sliding speeds on contact ellipse [8].

The tangential speeds v_l and v_u in the rolling direction can be expressed using Harris's equations adapted for a thrust ball bearing [9]:

$$v_l(x, n_u, \omega_c) = \left[\frac{d_m}{2} + \mathrm{x}\right] \cdot \omega_c \tag{4}$$

$$v_u(x, n_u, \omega_c) = \left[\frac{d_m}{2} + x\right] \cdot \left(\pi \cdot \frac{n_u}{30} - \omega_c\right) \tag{5}$$

The tangential speeds of a point from the ball $v_{bl,u}$ are determined by Harris's equations:

$$v_{bl,u}(x, \omega_b) = \omega_b \cdot B_{l,u}(x) \tag{6}$$

The distance from the centre of the ball to the point on the contact ellipses, $B_{l,u}(x)$ are determined by Harris's relations [9]:

$$B_{l,u}(x) = \left\{\left({Ra_{l,u}}^2 - x^2\right)^{0.5} - \left({Ra_{l,u}}^2 - {a_{l,u}}^2\right)^{0.5} + \left[(0.5 \cdot d_b)^2 - {a_{l,u}}^2\right]^{0.5}\right\} \tag{7}$$

where $Ra_{l,u}$ are determined with relations [9]:

$$Ra_{l,u} = d_b \cdot \frac{2 \cdot f_{l,u}}{(2 \cdot f_{l,u} + 1)} \tag{8}$$

The semi major and semi minor contact ellipse axis $a_{l,u}$ and $b_{l,u}$ are determined according to Houpert's relations [10].

$$a_{l,u} \approx 1.3085 \cdot R_{Yl,u} \cdot {k_{l,u}}^{0.4091} \cdot \left(\frac{Q}{E^* \cdot {R_{Yl,u}}^2}\right)^{\frac{1}{3}} \tag{9}$$

$$b_{l,u} \approx 1.1502 \cdot R_{Yl,u} \cdot {k_{l,u}}^{-0.1876} \cdot \left(\frac{Q}{E^* \cdot {R_{Yl,u}}^2}\right)^{\frac{1}{3}} \tag{10}$$

where E^* is the equivalent Young's modulus for the materials in contact, Q is the normal load, $R_{Yl,u}$ are the equivalent radii in the rolling direction determined by relation [10]:

$$R_{Yl} = R_{Yu} = \frac{d_b}{2} \tag{11}$$

The radius ratio $k_{l,u}$ are expressed by the relations [10]:

$$k_l = \frac{2 \cdot f_l}{(2 \cdot f_l - 1))} \qquad k_u = \frac{2 \cdot f_u}{(2 \cdot f_u - 1))} \tag{12}$$

where f_l and f_u are the conformities of the two races [9].

In a thrust ball bearing the ball is supplementary rotated with an angular pivoting speeds ω_p expressed by the following equation [9]:

$$\omega_p = \omega_c \tag{13}$$

The total sliding speeds in the rolling directions $v_{tl,u}$ are determined with the following equations:

$$v_{tl,u}(x, n_u, \omega_b, \omega_c) = v_{sl,u}(x, n_u, \omega_b, \omega_c) + \omega_p \cdot x \tag{14}$$

2.2. Friction coefficient on contact ellipses

The presence of the grease in the ball-race contacts develops an EHD film with the minimum thickness determined by Hamrock & Dowson's equations [10,11]:

$$h_{min_{l,u}} = 3.63 \cdot R_{yl,u} \cdot {U_{l,u}}^{0.68} \cdot G^{0.49} \cdot {W_{l,u}}^{-0.073} \cdot \left(1 - 0.61 \cdot e^{-0.68 \cdot k_{l,u}}\right) \tag{15}$$

where ***U, G*** and ***W*** are the dimensionless speed parameter, material parameter and load parameter respectively, determined by the classical relations:

$$U_{l,u} = \frac{\eta_0 \cdot v_{l,u}}{E^* \cdot R_{yl,u}}; \quad G = \alpha_p \cdot E^*; \quad W_{l,u} = \frac{Q}{E^* \cdot R_{l,u}^2} \tag{16}$$

In Eqs. (16) η_0 is the base oil dynamic viscosity at the operating temperature, α_p is the piesoviscosity exponent, E^* is equivalent elastic modulus, Q is normal load and $v_{l,u}$ are the average entrainment speed in ball-race contacts defined by equation:

$$v_{l,u} \approx \frac{\pi \cdot n_u \cdot d_m}{120} \tag{17}$$

In a mixed lubrication regime, the friction coefficient is dependent of the lubricant parameters $\lambda_{l,u}$ and it can be estimated by the following equations [12]:

$$\mu_{sl,u} = \mu_e \cdot 0.82 \cdot \lambda_{l,u}{}^{0.28} + \mu_a \cdot \left(1 - 0.82 \cdot \lambda_{l,u}{}^{0.28}\right) \tag{18}$$

where μ_e and μ_a are the friction coefficient in full lubrication and in limit lubrication conditions, respectively.

The lubrication parameters $\lambda_{l,u}$ are determined by the following relations:

$$\lambda_{l,u} = \frac{h_{min_{l,u}}}{\sqrt{R_{ql,u}{}^2 + R_{qb}{}^2}} \tag{19}$$

where $R_{ql,u}$ are the RMS roughness on the two race surfaces and R_{qb} is the RMS roughness on the ball surfaces.

2.3. *Forces and moments on contact ellipses*

When upper race is rotating and lower race is stopped, the forces and moments acting on a ball in a thrust ball bearing without cage are presented in figure 2 [12].

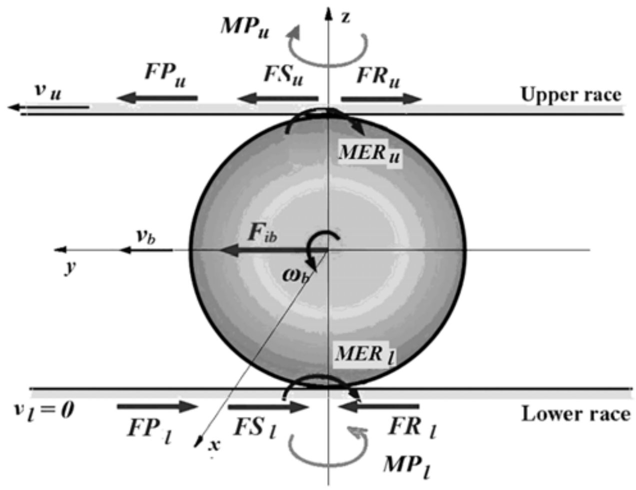

Figure 2. Forces and moments acting on a ball [12].

The presence of the lubricant in ball-race contacts leads to both hydrodynamic rolling forces $FR_{l,u}$ and pressure forces $FP_{l,u}$.

The hydrodynamic rolling forces are determined for the two contacts by Biboulet & Houpert's transition equations considering both IVR and EHL lubrication regimes in ball race contacts [5],[6]:

$$FR_{l,u} = \left[1/(1 + M_{l,u}/6.6)\right] \cdot FR_{IVR,l,u} + \left[(M_{l,u}/6.6)/(1 + M_{l,u}/6.6)\right] \cdot FR_{EHL,l,u} \tag{20}$$

where the IVR and EHL components of FR are determined by Biboulet's relations [5]:

$$FR_{IVR,l,u} = 2.9766 \cdot E^* \cdot R_{yl,u}^2 \cdot k_{l,u}{}^{0.3316} \cdot W_{l,u}{}^{1/3} \cdot U_{l,u}{}^{2/3} \tag{21}$$

$$FR_{EHL,l,u} = 7.5826 \cdot E^* \cdot R_{yl,u}^2 \cdot k_{l,u}{}^{0.4055} \cdot W_{l,u}{}^{1/3} \cdot U_{l,u}{}^{3/4} \tag{22}$$

The transition parameter $M_{l,u}$ from the IVR to EHL lubrication conditions can be determined with equations [6]:

$$M_{l,u} = 0.5549 \cdot k_{l,u}^{-0.6029} \cdot W_{l,u} \cdot U_{l,u}^{-0.75} \tag{23}$$

When the race radius is infinite in the rolling direction (as in a thrust ball bearing), the pressure forces are the followings $FP_{l,u}= 2\ FR_{l,u}$.

The inertial force acting on the ball centre F_{ib} is given by equation:

$$F_{ib} = -\frac{m_b \cdot d_m}{2} \cdot \frac{d\omega_c}{dt} \tag{24}$$

where m_b is the ball mass.

$MER_{l,u}$ are the rolling resistant moments due to elastic hysteresis losses in compression during the rolling process, and can be determined by equations [10]:

$$MER_{l,d} = 7.4810^{-7} \cdot \left(\frac{d_b}{2}\right)^{0.33} .Q^{1.33} .\left\{1 - 3.519\ 10^{-3} (k_{l,u} - 1)^{0.8063}\right\} \tag{25}$$

The traction forces $FS_{l,u}$ are determined as a result of the equilibrium of the forces and moments acting on a ball and results:

$$FS_{l,u} = \frac{MER_l + MER_u}{d_b} \pm \frac{Fib}{2} + FR_{u,l} \tag{26}$$

where the negative sign is for upper ball- race contact.

The total tangential force developed between a ball and the upper race Ft_u is the sum of sliding, rolling and pressure forces, according to following relation:

$$Ft_u = \frac{MER_l + MER_u}{d_b} + (FR_u + FR_l) - \frac{Fib}{2} \tag{27}$$

The pivoting friction moments $MP_{l,u}$,normal to the center of the contact ellipse, are determined with the following relations [10]:

$$MP_{l,u} = \frac{3}{8} \cdot \mu_{sl,u} \cdot Q \cdot a_{l,u} \tag{28}$$

The friction torque generated at upper race for a ball is obtained by relation:

$$dTz_u = Ft_u \cdot \frac{d_m}{2} + MP_u \tag{29}$$

For a modified thrust ball bearing with only three balls, the total friction torque acting on upper race can be written:

$$Tz_u = \frac{3 \cdot (MER_l + MER_u) \cdot d_m}{2 \cdot d_b} + \frac{3 \cdot (FR_u + FR_l) \cdot d_m}{2} + 3 \cdot MP_u - \frac{3 \cdot Fib \cdot d_m}{4} \tag{30}$$

Four components of the total friction torque have been included in the Eq. (30) as following: the components of the rolling friction torque generated by hysteresis, by hydrodynamic effect, by pivoting effects, and by inertial effect, respectively.

2.4. Evaluation of the total power loss

A complex method to estimate the power loss consisting in summing the power loss generated by sliding effect and power loss generated by rolling effect has been proposed by the authors.

The power losses generated by sliding speeds $v_{sl,u}$ are determined using the following relations [8]:

$$P_{S,l,u} = \frac{1}{a_{l,u}} \int_{-a_{l,u}}^{a_{l,u}} \left| v_{tl,u}(x, n_o, \omega_b, \omega_c) \right| \cdot dF_{s,l,u}(x) \cdot dx \tag{31}$$

For the elementary forces $dF_{sl,u}(x)$ acting in the rolling direction on contact ellipses, have been adapted the Houpert's equation [10] as follows:

$$dF_{s,l,u}(x) = \frac{3}{4} \cdot \mu_{sl,u} \cdot \frac{Q}{a_{l,u}} \cdot \sqrt{1 - \left(\frac{x}{a_{l,u}}\right)^2} \tag{32}$$

The total power loss due to sliding in the rolling direction for a single ball is considered the sum of the power loss on lower and upper race contact:

$$P_S = P_{S,l} + P_{S,u} \tag{33}$$

The total power loss generated by rolling for a single ball includes the power loss due to the moments $MER_{i,u}$ and the power loss due to the inertial effect. It can be expressed with the following relation:

$$P_R = (MER_l + MER_u) \cdot \omega_b + (FR_l + FR_u) \cdot d_b \cdot \omega_b - Fbi \cdot \frac{d_m}{2} \cdot \omega_c \tag{34}$$

The total power loss for the modified thrust ball bearing with three balls is determined by the following equation:

$$P_t = 3 \cdot (P_S + P_R) \tag{35}$$

In order to validate the results obtained with Eq. (35) two methods have been proposed:

A first method consist in estimating the total power loss in the modified thrust ball bearing according to the total friction torque determined with Eq. (30), using following relation:

$$P_{t,I} = Tz_u \cdot \frac{\pi \cdot n_u}{30} \tag{36}$$

The second method consist in estimating the total power loss according to experimental friction torque using the following relation:

$$P_{t,II} = Tz_{u,exp} \cdot \frac{\pi \cdot n_u}{30} \tag{37}$$

where $Tz_{u,exp}$ is the measured total friction torque [7].

3. Results and discussion

The simulation of the total power loss has been realized for a modified 51205 thrust ball bearing having only 3 balls, without cage and operating in deceleration process according to spin-down method [5, 7]. The axial load *Fa,* was 4.26 N and normal load for each ball *Q*, was 1.42N. The grease type used was MOL Liton 00 and the viscosity of the base oil at 40^{0}C of 40 mm^2/s has been considered for simulation. The piesoviscosity exponent α_p was considered of $2.5 \cdot 10^{-8} Pa^{-1}$. The following values for geometrical parameters of ball bearing have been considered: d_b = 7.938 mm (5/16*"), $f_l = f_u$* = 0.53, d_m = 36 mm, the average roughness for races $R_{ql,u}$ = 0.06 µm and the average roughness of the balls was R_{qb} = 0.03 µm. Rotational speeds n_u between 100 and 400 rpm have been considered for simulation. Also, the laboratory temperature of 23^{o}C -24^{o}C has been considered for base oil viscosity. In Eq. 18 the following values for the coefficients μ_a and μ_e have been adopted: μ_a=0.11 and μ_e=0.05 [10].

For the identical geometry, load and tangential speed on lower and upper raceways, the ball-race contact ellipses have the same dimensions ($a_l = a_u$). Also the moments *MER, MP* and the hydrodynamic forces *FR* have the same values for both ball-race contacts.

The experimental methodology and the values for the total friction torque determined with the grease by using spin down procedure were presented in detail by Ianuş et al. [7]. Based on these experiments, it were determined the values for upper race decelerations $d\omega_c/dt$ and the experimental total friction torques $T_{zu,exp}$ corresponding to 100, 200, 300 and 400 rpm rotational speed of upper race.

The most important parameters obtained by simulation are presented in the table 1.

A very good correlation between the power loss has been obtained for four different rotational speeds: n_u = 100, 200, 300 and 400 rpm using any of the following three approaches:

(a) The sum of sliding, rolling and inertial power loss - P_t;

(b) The product between theoretical total friction torque and angular speed- $P_{t,I}$;

(c) The product between experimentally measured total friction torque and angular speed- $P_{t,II}$.

Table 1. The simulation results obtained for upper race rotational speed of 100, 200, 300 and 400 rpm.

Parameters	Rotational speed, n_u (rpm)			
	100	200	300	400
ω_b [rad/s]	23.74	47.49	71.24	94.98
ω_c [rad/s]	5.23	10.47	15.70	20.94
$h_{minl,u}$ [μm]	0.11	0.17	0.23	0.27
$\lambda_{l,u}$	1.42	2.26	2.96	3.6
$\mu_{l,u}$	0.06	0.05	0.05	0.05
$FR_{l,u}$ [N]	0.59×10^{-2}	1.1×10^{-2}	1.5×10^{-2}	1.9×10^{-2}
$d\omega_c/dt$ [rad/s^2]	3.68	7.3	10.4	13.5
$Fib_{l,u}$ [N]	1.36×10^{-4}	2.68×10^{-4}	3.86×10^{-4}	4.9×10^{-4}
Ft_u [N]	1.17×10^{-2}	2.19×10^{-2}	2.98×10^{-2}	3.78×10^{-2}
$MP_{i,u}$ [Nm]	3.26×10^{-6}	3.25×10^{-6}	3.25×10^{-6}	3.25×10^{-6}
$MER_{i,u}$ [Nm]	1.75×10^{-7}	1.75×10^{-7}	1.75×10^{-7}	1.75×10^{-7}
Tz_u [Nm]	0.64×10^{-3}	1.19×10^{-3}	1.62×10^{-3}	2.06×10^{-3}
P_S [W]	5.03×10^{-4}	4.17×10^{-4}	4.19×10^{-4}	4.22×10^{-4}
P_R [W]	2.67×10^{-2}	4.98×10^{-2}	6.79×10^{-2}	8.6×10^{-2}
P_t [W]	2.72×10^{-2}	5.03×10^{-2}	6.83×10^{-2}	8.64×10^{-2}
$P_{t,\,I}$ [W]	2.69×10^{-2}	4.99×10^{-2}	6.79×10^{-2}	8.58×10^{-2}
$P_{t,\,II}$ [W]	2.68×10^{-2}	4.94×10^{-2}	6.7×10^{-2}	8.5×10^{-2}

As seen from table 1, the hydrodynamic rolling friction forces *FR* have major contribution on total power loss for all four rotational speeds. Also, was demonstrated that the use of base oil viscosity of the grease in the theoretical Biboulet&Houpert model for hydrodynamic rolling friction forces given by Eqs: (20) – (23), can conduct to correct values for total power loss. For given operation conditions (very low load on ball- race contacts and low values of friction coefficient as result of the film thickness) the total power loss generated by sliding friction (sliding in rolling direction and pivoting sliding on contact ellipses) does not exceed 2% from the total power loss generated by rolling friction. Based on these results, to reduce the total power loss in a very low loaded ball bearing, is recommended to use grease with a very low base oil viscosity.

4. Conclusions

To evaluate the power loss of grease lubricated thrust ball bearings with no cage, the authors developed a simulation program consisting in the establishment of sliding, rolling and inertial power loss. Using an analytical method, the kinematics, sliding and rolling forces, rolling and pivoting moments, total friction torque and total power loss in a modified 51205 thrust ball bearing were determined. The modified 51205 rolling bearing had only 3 balls and no cage, being lubricated by MOL Liton 00 grease.

The simulations were realized for an axial load of 4.26N and a variation of the rotational speed from 100 to 400 rpm. The most important conclusions are:

- The total power loss obtained by the proposed analytical model is in good correlation with the total power loss obtained using two other methods:
    - the product between analytical friction torque and angular speed;
    - the product between the experimentally measured friction torque and the angular speed.
- It was demonstrated that using the base oil viscosity of the grease in the theoretical model for hydrodynamic rolling friction forces computation, can draw to correct values of computed total power loss.
- For imposed conditions, the hydrodynamic rolling friction forces *FR* have major contribution on total power loss. The total power loss generated by sliding friction does not exceeds 2% from the total power loss generated by rolling friction.
- To reduce the total power loss in a ball bearing very low loaded, is recommended to use grease with a very low base oil viscosity.

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