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POWER LOSS COMPUTATION IN ANGULAR CONTACT BALL BEARINGS

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ABSTRACT

The paper presents a method for the computation of the power loss in angular contact ball bearings. Dynamic effects, generated at high-speed, are also included in the model, as centrifugal forces and gyroscopic moments acting on balls, balls and cage interactions, ball and cage drag and churning in the lubricating bath oil.

As concerning the estimated power loss in ball bearings, the model offers outputs in good agreement with the results found in literature.

Keywords: Ball bearings, Friction, Power loss, Oil lubrication.

1. INTRODUCTION

The power loss in rolling bearings is an important performance parameter, dictating the developed temperature and the lifetime of the bearing.

The rolling bearing manufacturers in the world possess adequate instruments for the prediction of the power loss, some sporadic information and ready to use formula being offered in their catalogues, especially to help their customers to choose the right bearing for a demanded application.

There are some dynamic analysis models for ball bearings, developed decades of years ago [1] [2]. Recently, Pouly et al. [3] estimated the power loss in rolling element bearing considering both ball and cage drag and hydrodynamic forces. The model is limited by the adoption of some simplifying kinematics assumptions.

Takabi and Khonsari [4] determines the friction heat in rolling bearings with the aid of the well-known Palmgren's formula for friction torque estimation, updated to consider the effect of induced thermal preload. To find friction torque in oil-lubricated ball bearings, Fernandes et al. [5] uses also a semi-empirical method based on Palmgren's equations, newly updated by SKF. These models are an easy to implement but rough approximation for the friction torque and power loss in bearings.

Paleu and Balan [6] presented results on the power loss in angular contact ball bearings from a complex dynamic model, treating only the case of axial preload and high-speed.

The aim of this paper is to presents a method for the computation of the power loss in angular contact ball bearings with the consideration of a complex load: axial and radial loads, and a shaft tilting torque. Dynamic effects, generated at high-speed, are also included in the model, as centrifugal forces and gyroscopic moments acting on balls, balls and cage interactions, ball and cage drag and churning in the lubricating bath oil.

2. BALL QUASI-DYNAMIC MODEL

Balls and races load forces result from the quasi-static model presented by Damian et al. [7].

To write the corresponding equilibrium equations for each ball, the forces and torques acting on ball and races and ball and cage contacts are presented in Figure 1, from which the corresponding equilibrium equations result.

The next equation represents the equilibrium of forces acting on ball:

$$FTR_o - FS_{yo} - FTA_{yo} - FTA_{yi} + FS_{yi} + FTA_{yi} - -FHD_{yo} + FHD_{yi} - FBAL_y - Q_c = 0 \tag{1}$$

For the torque equilibrium of each ball, it results:

$$\begin{aligned}
&\left(\sum M_w\right)_x : -x_{i0} \cdot \sin\alpha_i \cdot \left(FS_{yi} + FTA_{yi} - FTR_{yi}\right) + \\
&+ x_{o0} \cdot \sin\alpha_o \cdot \left(FS_{yo} + FTA_{yo} - FTR_{yo}\right) - TS_{yi} \cdot \cos\alpha_i = \\
&= J_w \cdot \frac{d\omega_{wx}}{dt} \\
&\left(\sum M_w\right)_y : -x_{i0} \cdot FS_{zi} = J_w \cdot \frac{d\omega_{wy}}{dt} \\
&\left(\sum M_w\right)_z : x_{i0} \cdot \cos\alpha_i \cdot \left(-FTR_{yi} + FS_{yi} + FTA_{yi}\right) + \\
&x_{o0} \cdot \left(-FTR_{yo} + FS_{yo} + FTA_{yo}\right) + \mu_c \cdot Q_c \cdot \frac{D_w}{2} + \\
&+ TBAL_z + TS_{xi} \cdot \sin\alpha_i = J_w \cdot \frac{d\omega_{wz}}{dt}
\end{aligned} \tag{2}$$

The ball-cage collision forces are obtained from equation (1), and the components of the angular velocity of the ball results from the differential equations (2). The fourth order Runge-Kutta method is used to solve the differential equations (2).

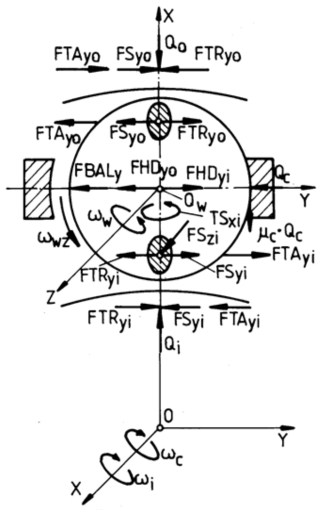

Fig. 1: Forces and moments acting on balls

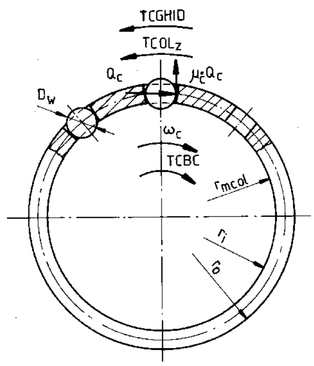

Fig. 2: Cage equilibrium

The forces and torques involved in ball's equilibrium are:

- The sliding traction force:

$$FS_{yi,o} = \iint_{A_{i,o}} \tau_{yi,o} dA \tag{3}$$

where $\tau_{yi,o}$ are the shear stresses in the lubrication film. The Ree-Eyring model, from Houpert [8, 9], was used to evaluate the lubricant shear stresses.

- The rolling traction force was evaluated with the equation proposed by Houpert [10, 11]:

$$FTR_{yi,o} = 2.86 \cdot E' \cdot R_{yi,o}^{2} \cdot K_{i,o}^{0.348} \cdot G_{i,o}^{0.022} \cdot W_{i,o}^{0.47} \tag{4}$$

- The asperity traction force was evaluated with the equation developed by Zhou [12]:

$$FTA_{yi,o} = 0.2 \cdot Q_{i,o} \cdot e^{-B \cdot \lambda_{i,o}^{C}} \tag{5}$$

- The hydrodynamic force was evaluated with the equation presented by Harris [13]:

$$FHD_{yi,o} = 18.4 \cdot (1-\gamma) \cdot E' \cdot R_{yi,o} \cdot a_{i,o} \cdot G_{i,o}^{-0.33} \cdot U_{i,o}^{0.7} \tag{6}$$

The spin friction torque at the inner contact, TS_{xi} and the resistant torque $TBAL_z$ due to ball's rotation in the air-lubricant mixture were evaluated with the equations proposed by Gupta [1].

3. CAGE EQUILIBRIUM

The condition of the dynamic equilibrium of the cage, which performs a rotation movement around Z axis, Figure 2, gives the equation:

$$\left(\sum M_c\right): \sum_{j=1}^{z} Q_c \cdot r_{mcol} - \mu_c \cdot \sum_{j=1}^{z} Q_c \frac{D_w}{2} - TCOL_z - TCGHID = J_c \frac{d\omega_c}{dt} \tag{7}$$

The numerical integration of the cage dynamic equation provides its angular velocity ω_c.

4. FRICTION TORQUE

The friction torque acting on the outer raceway is given by equation (8):

$$MF_o = \sum_{j=1}^{z} \left(\frac{dm}{2} + v_x + x_o \cdot \cos\alpha_o \right)_j \cdot \left(FS_{yo} + FTA_{yo} - FTR_{yo} \right)_j \tag{8}$$

At the inner race level, the friction torque is:

$$MF_i = \sum_{j=1}^{z} \left(\frac{dm}{2} + v_x + x_i \cdot \cos\alpha_i \right)_j \cdot \left(FS_{yi} + FTA_{yi} - FTR_{yi} \right)_j \tag{9}$$

The total friction torque is:

$$MF_t = MF_i + MF_o \tag{10}$$

5. POWER LOSS COMPUTATION

The power loss is estimated at the inner and outer race, as follows:

-Power loss on the inner raceway:

$$PFCI = \sum_{j=1}^{z} \left(P_{if} + P_{iasp} \right)_j \tag{11}$$

- Power loss on the outer raceway:

$$PFCE = \sum_{j=1}^{z} \left(P_{of} + P_{oasp} \right)_j \tag{12}$$

- Drag and churning power loss, generated by the movement of balls and cage through the air-lubricant mixture:

$$PCBAL = \left(\sum_{j=1}^{z} FBAL_{yj} \cdot \omega_c \cdot r_{mcol} + \sum_{j=1}^{z} TBAL_{zj} \cdot \omega_{wj} \right) \tag{13}$$

$$PFCOL = TCOL_z \cdot \omega_c$$

- Power loss in the ball-cage contacts:

$$PFCBC = \sum_{j=1}^{z} \mu_c \cdot \left| Q_{cj} \right| \cdot \omega_{wj} \cdot \frac{D_w}{2} \tag{14}$$

The total power loss is obtained as the sum of the power loss in the entire bearing. This is given by equation (15):

$$PTOTAL = PFCI + PFCE + PCBAL + PFCOL + PFCBC \tag{15}$$

6. MODEL VALIDATION

The dynamic analysis model, developed for ball bearings loaded about 3 degrees of freedom, is validated by comparison of its numerical results with results found in literature.

Nelias [13] gives some results for a ball bearing with the geometrical parameters indicated in Table 1.

The running condition, simulated by Nelias [13], are: the rotational speed of the inner ring, 50,000 rev/min; the axial load, 1.500 N; the radial load, 500 N; the viscosity of the lubricant $\eta = 15.6 cP$ at 80^0C – the working temperature.

The results of the developed model agree well with those obtained by Nealis [13]. For comparison reasons, these are synthetically presented in Table 2.

Table. 1: Parameters of the bearing used by Nelias [13]

Geometrical parameter	Numerical value
Main catalogue dimensions d x D x B,[mm]	35 x 65 x 15
Ball diameter D_w [mm]	7.938
Number of balls, Z	16
Contact angle α_0 [^{0}]	31
Conformity of the inner raceway f_i	0.525
Conformity of the outer raceway f_o	0.51

Table. 2: Model validation

Particular power loss friction [W]	Authors	Nelias, [13]
Balls-inner ring contacts	172 W	105 W
Balls-outer ring contacts	24 W	1.6 W
Cage-balls contacts	77 W	45 W
Cage-inner ring (guiding land)	116 W	121 W
Cage-balls-lubricant (drag and churning)	31 W	67 W
Total power loss	420 W	339.6 W

7. MISALIGNMENT EFFECT ON FRICTIONAL POWER LOSS

The most important is that our model can take into account the misalignment of the shaft. The new formula developed by bearings manufacturers [14], does not cope with this problem, even they recognize that a misalignment increases the friction torque in bearings.

For a ball bearing from 7012 CTA – P4 series, the next running conditions are introduced in the simulation program: radial load F_x=2000 N, axial load F_z=350 N, inner ring speed N=6000 rev/min, oil dynamic viscosity at 45 ^{0}C, η_{45}=0.033 Pa.s

Figures 3 and 4 show that the increase of the tilting angle draws to an augmentation of the power loss within the bearings.

In balls and inner race contacts, the power loss PFCI is higher in relation to the outer race friction losses, PFCE, especially because of the higher sliding speed (rotating inner ring and fixed outer ring).

Increasing the tilting angle from zero to 15 degrees, the total power loss augments about 2.7 times. The same evolution is obtained for the power loss on the inner race.

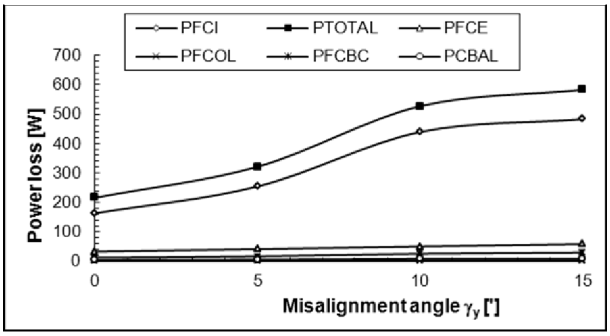

Fig. 3: Total power loss in 7012 bearing (Fx=2000 N, F=350 N, N=6000 rpm, η_{45}=0.033 Pa.s)

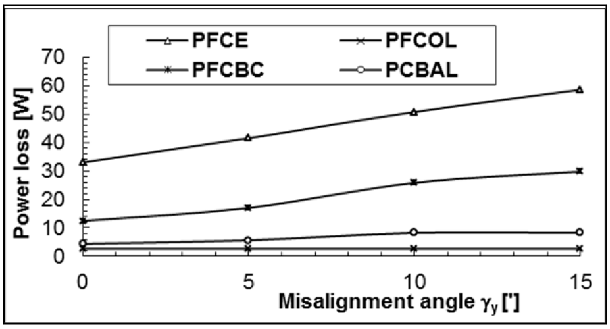

Fig. 4: Zoom in the lower zone of Fig. 2

7. CONCLUSIONS

The paper presents a method for the computation of the power loss in angular contact ball bearings. The dynamic effects, generated at high-speed, are also included in the model, as centrifugal forces and gyroscopic moments acting on balls, balls and cage interactions, ball and cage drag and churning in the lubricating bath oil.

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Increasing the tilting angle from zero to 15 degrees, the total power loss augments about 2.7 times.

8. NOMENCLATURE

B - bearing width
D_w - ball diameter
F_c - centrifugal force
F_x, F_z - external load
$\sum F$ - force equilibrium for entire bearing
FTR - rolling traction force
FS - sliding traction force
FTA - asperity level traction force
FHD - hydrodynamic force
FBAL - ball drag force
J_w - ball inertial torque
J_c - cage inertial torque
M_y - driven torque of the inner ring
MF - friction torque
N - rotation speed of the inner ring
TS - spin torque
TBAL - ball's friction torque due to the its rotation in air-lubricant mixture
$TCOL_z$ - z^{th} ball's drag torque
TCGHID - torque due to friction between cage and the outer guiding race
TCBC - resistant torque due to the contact load
O origin of inertial coordinate frame
O_i origin of the inner ring coordinate frame
O_o curvature center of the outer ring raceway
O_w ball's center
O_{wd} - ball's center final position
PFCI - power loss on inner raceway
PFCE - power loss on outer raceway
PFCBC - power loss in ball-cage contact
PCBAL - balls drag power loss (through the oil-air mixture)
$Q_{i,o}$ - contact load
$\sum Q$ - force equilibrium for ball
Q_c - load due to ball-cage collision
R_i curvature radius of the inner raceway
PFCOL - cage churning power loss (through the oil-air mixture) + cage guiding land power loss
PTOTAL total power loss
R_o curvature radius of the outer raceway
d_m bearing average diameter
$f_{i,o}$ conformity of the raceway
$\{n\}$ normal vector on the contact area
r_{mcol} - cage average radius
r_o - cage outer radius
r_i - cage inner radius
$x_{i,o}$ - distance from the action point of a force to the considered point of torque equilibrium;
z - number of balls
α_0 - free contact angle
$\alpha_{i,o}$ - operating contact angle
γ_y -inner ring misalignment
$\delta_{i,e}$ - contact deformation
ω_w -ball's angular rotation speed
ω_c - cage's angular rotation speed
μ_c -friction coefficient between ball and cage
η dynamic viscosity of the lubricant
- vector cross product

Lower scripts

i - inner ring
o - outer ring
x - X axis direction
y - Y axis direction
z - Z axis direction

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