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Investigation of nonlinear parameters of force on lubricated elliptical contact with load and rotation variations

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Abstract: The development of optimized products through time-consuming projects is one of the current researches in the global machinery and equipment industry. The computational simulation allowed to reduce both the project time and costs with prototypes. In this context, the present study aims to evaluate the behavior of contact forces parameters in angular contact ball bearings with the variations of load and rotation applied to this contact, applying the resulting forces simulated in complex systems that contain this type of bearing. From the characterization of the bearing and the parameters for a specific set of rotations and loads, it is possible to study its behavior and estimate parameter values for different load conditions without having to perform a new characterization for each set of parameters that are tested. Initially, the balance of moments and forces in the bearing is calculated using the Hertz contact force model and the force distribution is determined. Sequentially, the system of elastohydrodynamic (EHD) equations is solved by applying the previously calculated parameters. Then, an optimization of the parameters of non-linear contact forces is made for each sphere (contact with the internal and external raceway) and the balance of forces and moments are recalculated using these estimated parameters. The process is repeated until the contact forces adjusted and calculated by the balance of forces and moments reach an established tolerance. The final calculated parameters of these iterations allow to evaluate the behavior of the bearing contact with the variations of rotation and load and to test the resultant forces on simulations of a complex rotor system, for example.

Keywords: contact model, ball bearing, nonlinear parameters, lubrication

INTRODUCTION

The improvements of rotating machinery, widely applied in several industrial sectors, consists in search of performance and reduction on project phase time and tested prototypes. The ball bearings have a key role in this type of equipment and their numerical simulation and characterization contribute to advances in the complete rotating system model.

In order to analyze this type of bearing, the model of nonlinear contact force for elliptical contact (ball/raceway), proposed by Nonato and Cavalca (2014), was applied along the bearing forces and moments equilibrium equations and elastohydrodynamic (EHD) lubrication system of equations, resulting in optimized values of nonlinear contact forces parameters as presented by Bizarre, Nonato and Cavalca (2018). The main focusing of the present work is the study of these nonlinear parameters variations with different conditions of loads and rotational speeds, allowing to evaluate its behavior and reducing the time of characterization of each contact, and consequently, the total time of simulation of the rotational system that applies this type of bearing.

METHODOLOGY

The nonlinear force parameters study demands to solve the full bearing model and to achieve the convergence, resulting in the parameters of interest. Thus, the methodology applied during this work was divided into three main parts: the ball bearing forces and moments equilibrium, the elastohydrodynamic (EHD) lubrication model and the nonlinear contact forces model.

Ball bearing forces and momentum equilibrium

The model used to describe the bearing dynamics considers five degrees of freedom (x (axial), y and z (radial) and θ_y and θ_z (angular displacement)), the centripetal force, gyroscopic moment and the angular contact between the inner and the outer raceway as presented in Bizarre, Nonato and Cavalca (2018). The Fig. 1 contains the directions (x,y and z) and the representation of the forces and moments acting on the sphere and on the bearing.

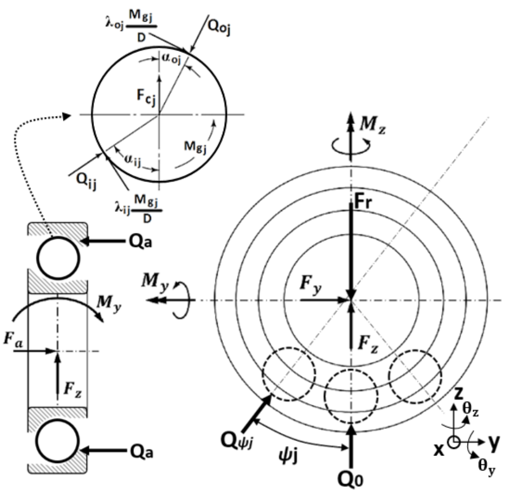

Figure 1 – Angular contact ball bearing load, forces and directions scheme (adapted from Harris (1991)).

In its first iteration, the bearing dynamic applied the model proposed by Hertz to describe the contact force between spheres and raceways, gives by Eq. 1. The load (Q) was related to the displacement of the raceway ($\delta_{i,o}$) and the contact stiffness ($K_{i,o}$). Considering the load applied radially and axially along with the bearing rotation, it is possible to define the load distribution on each sphere that is used as input on the lubrication section. Equations [2-6] contains the main equilibrium forces and moments of the bearing (solved by the Newton-Raphson method) as described in Harris (1991). From the second iteration, the contact force model proposed by Nonato and Cavalca (2014) replaces the Hertz contact model.

$$Q_{i,o} = K_{i,o}\delta_{i,o}^{1.5} \tag{1}$$

$$F_a - \sum_{j=1}^{j=z}\left(Q_{i_j}sin(\alpha_{i_j}) - \frac{\left(\lambda_{ij}M_{g_j}\right)}{D}cos(\alpha_{i_j})\right) = 0 \tag{2}$$

$$F_z - \sum_{j=1}^{j=z}\left(Q_{i_j}cos(\alpha_{i_j}) - \frac{\left(\lambda_{ij}M_{g_j}\right)}{D}sin(\alpha_{i_j})\right)cos\psi_j = 0 \tag{3}$$

$$F_y - \sum_{j=1}^{j=z}\left(Q_{i_j}cos(\alpha_{i_j}) - \frac{\left(\lambda_{ij}M_{g_j}\right)}{D}sin(\alpha_{i_j})\right)sen\psi_j = 0 \tag{4}$$

$$M_z - \sum_{j=1}^{j=z}\left(\left(Q_{i_j}\,sin\left(\alpha_{i_j}\right) - \frac{\left(\lambda_{ij}M_{g_j}\right)}{D}cos\left(\alpha_{i_j}\right)\right)\Re_i + \lambda_{ij}f_iM_{gj}\right)sen\psi_j = 0 \tag{5}$$

$$M_y - \sum_{j=1}^{j=z}\left(\left(Q_{i_j}\,sin\left(\alpha_{i_j}\right) - \frac{\left(\lambda_{ij}M_{g_j}\right)}{D}cos\left(\alpha_{i_j}\right)\right)\Re_i + \lambda_{ij}f_iM_{gj}\right)cos\psi_j = 0 \tag{5}$$

The variables presented in Eqs. 1-5 were the number of spheres (Z), the inner and outer contact angle (α_i and α_o), the sphere diameter (D), the variable that determines how much the gyroscopic moment was resisted by the inner raceway (λ_{ij}), the gyroscopic moment (M_{g_j}), the distance between the geometric center of the bearing and the center of curvature of the inner raceway ($\Re_i$), the osculation ratio (f_i), the azimuth angle (ψ_j), the resultant moments in axis z and y (M_z and M_y) and forces in each direction (F_a, F_y and F_z). From these equilibrium, the displacements and equilibrium position was calculated, allowing proceeding with the solution of the lubrication model. The sub-index i and o represents the inner and outer raceways, respectively.

Elastohydrodynamic lubrication model

The region of contact was modeled with the elastohydrodynamic (EHD) lubrication theory, which is applied to non-conforming bodies, generating a small contact region. Due to this reduced contact area, high pressure was developed, generating an increase in the fluid viscosity and surface deformation proportional to the oil film thickness on the contact. The main difference between EHD lubrication and Hertz dry contact is the consideration of the bodies' deformation in contact, and the variation of fluid properties (viscosity (η) and density (ρ)) with the pressure distribution.

Initially, results of load and velocity were used to calculate the load (M) and lubricant (L) Moes parameters (Eqs. 7 and 8) (related to the contact properties: load, velocity, curvature, and fluid viscosity) along with the ellipticity, were used to characterize each contact. The term w is the load on contact, E' is the reduced modulus of elasticity, R_x is the curvature sum on x direction, η_0 is the viscosity on ambient pressure, u_s is the sum of the surface velocities and α is the pressure-viscosity index.

$$M = \frac{w}{E' R_x^2}\left(\frac{\eta_0 u_s}{E' R_x}\right)^{-\frac{3}{4}} \tag{7}$$

$$L = \alpha E' \left(\frac{\eta_0 u_s}{E' R_x}\right)^{\frac{1}{4}} \tag{8}$$

A system of equations was applied to describe the lubrication model of the contact. The elastic deformation associated with the geometry of the space where the lubricant is contained allows calculating the oil film thickness. Sequentially, Reynold's equation was applied to model the contact pressure distribution. It is also necessary to use relations between viscosity-pressure and density-pressure. In this work, was considered the relations proposed by Roelands (1966) and Dowson and Higginson (1977). The equation of forces balance provides the integral of pressure over the film area (contact force) to be balanced by the external load applied normal to the contact area and, the last equation of the system, contains the equation of motion for the EHD problem, with a harmonic excitation on the contact. The full description of each equation can be found on Bizarre, Nonato and Cavalca (2018) and a multi-level algorithm as described in Venner (2001) solved this system of equations.

Nonlinear contact forces model

After performing the balance of forces and moments on the bearing and calculate the EHD lubrication on the contact, the forces and displacements data resulting from the harmonic simulation of the EHD simulation can be applied to compute the nonlinear contact force parameters: $K_{i,o}, d_{i,o}, \Delta F_{i,o}$. The algorithm of Levenberg-Marquardt (Marquardt (1963)) was used to optimize the parameters and, inside the convergence process of the whole bearing, the Hertz model of contact force was replaced by the equation (8), where $K_{i,o}$ is the contact stiffness with inner (i) and outer (o) raceways, $d_{i,o}$ is de nonlinear index and $\Delta F_{i,o}$ is an offset of force that represents the hydrodynamic lift force when the approach between the bodies is null.

$$Q_{EHD\,i,o} = K_{i,o}\left(\delta_{EHD\,i,o}\right)^{d_{i,o}} + \Delta F_{i,o} \tag{8}$$

The nonlinear parameters for each sphere turn possible to calculate the force acting on the bearing and this force can be applied as an input for rotating systems that contain this type of bearing.

RESULTS AND DISCUSSION

The angular contact ball bearing tested during this work was the 7006 and its geometrical characteristics and load conditions are presented on Tab. 1.

Table 1 – 7006 ball bearing geometric and operational characteristics

7006 ball bearing geometric characteristics	
Sphere diameter (mm)	42.5
Pitch diameter (mm)	7.0
Osculation ratio of the inner raceway (%)	102
Osculation ratio of the outer raceway (%)	108
Number of spheres (-)	11
Nominal contact angle (°)	15
Radial clearance (mm)	0
Operation conditions	
Force in radial/axial directions (N)	1000
Rotational speed of inner raceway (rpm)	5000, 4000, 3000
Rotational speed of outer raceway (rpm)	0
Angular displacement (°)	0.001

During the simulations, three levels of rotation were tested: case A, with 5000 rpm; case B, with 4000 rpm and case C, with 3000 rpm. A load of 1000N was applied in both directions (radial (z and y) and axial (x)) along with an angular displacement of 0.001°. Following the methodology proposed, values of nonlinear contact force parameters were converged, and the results were shown in Fig. 2.

An examination of the results presented in Fig.2 revealed a trend in the variation of the stiffness (K) and the nonlinear index (d), demonstrating that, these two parameters are not influenced by the rotation variation, having variations only with the different loads present in the contacts. Still, the parameter of force offset (ΔF) showed different tendencies with the rotation speed variations, for each level of rotation, the values followed a trend curve. It is also important to point out the behavior of the points of ΔF with the increase of rotation, as the rotation increases, the force offset also increases, agreeing with the hydrodynamic lift force.

Another behavior noted on the results is the tendency to the Hertzian contact with the increase of load on the contact with the same rotation, the nonlinear index d, for example, tends to the value of 1.5.

From the results obtained, it can be possible to model the nonlinear force contact parameters, adjusting a curve that provides the resultant parameters for a combination of load and rotation speed (for stiffness and nonlinear index, rotation do not generate a relevant variation). Finally, this adjust can be inserted on the complete models of rotating machinery, reducing the time of numeric simulation to obtain the forces behavior.

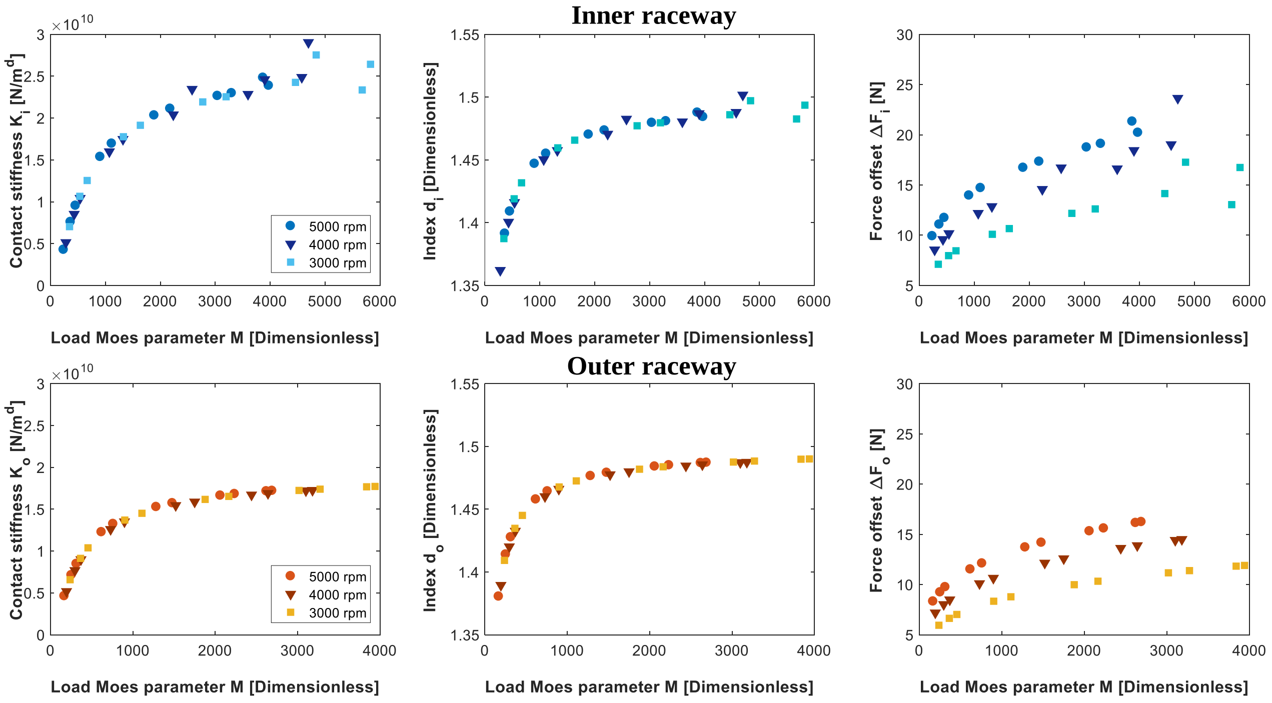

Figure 2 – Results of stiffness ($K_{i,o}$), nonlinear index (d) and force offset (ΔF) with different speed rotations (Cases A, B and C) and load level on contact, considering inner and outer raceways.

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REFERENCES

Nonato, F., Cavalca, K. L., 2014, "An approach for including the stiffness and damping of elastohydrodynamic point contacts in deep groove ball bearing equilibrium models", Journal of Sound and Vibration, Vol. 333, pp. 6960-6978.

Bizarre, L., Nonato, F., Cavalca, K. L., 2018, "Formulation of five degrees of freedom ball bearing model accounting for the nonlinear stiffness and damping of elastohydrodynamic point contacts", Mechanism and Machine Theory, Vol. 124, pp. 179-196.

Harris, T.A., 1991, "Rolling Bearing Analysis", John Wiley & Sons, New York.

Roelands, C. J. A., 1966, "Correlational Aspects of the Viscosity-Temperature-Pressure Relationship of Lubricating Oils", Ph.D. Thesis, Technical University Delft, Delft, The Netherlands.

Dowson, D., Higginson, G. R., , 1977, "Elasto-hydrodynamic Lubrication", SI Edition, Pergamon Press, Oxford.

Venner, C. H., Lubrecht, A. A., 2001, "Multilevel Methods in Lubrication. Netherlands", Elsevier, Vol. 37, 400p.

Marquardt, D. W., 1963, "An Algorithm for Least-Squares Estimation of Nonlinear Parameters", Journal of the Society for Industrial and Applied Mathematics, Vol. 11, pp. 431-441.

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