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Load distribution in cylindrical roller bearing with EHD lubricated contact force

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Abstract: In roller bearings, elastohydrodynamic (EHD) lubrication is presented in contacts between rolling elements and raceways. The objective of this work is to study the effect of the EHD contact forces in load distribution of cylindrical roller bearings in contrast to classic Hertzian contact. The EHD line contact force model is based on elastohydrodynamic stiffness and EHD constant surface separation force. These force parameters are obtained by solving numerically the full EHD system of equations. Subsequently, the force equilibrium in cylindrical roller bearing is evaluated. The roller bearing is simulated at 5000 rpm and under a radial load of 50 kN with different amount of clearance. The application of EHD contact force model impacts the load distribution in cylindrical roller bearings and the major influence is related to the displacement of the roller elements. The variation of displacement amplitude indicates that the inclusion of EHD lubricated contact in roller bearings can affect the response in rotor dynamic.

Keywords: cylindrical roller bearing, elastohydrodynamic lubrication, contact force, EHD stiffness, load distribution

INTRODUCTION

The dynamic of roller bearings has neglected the presence of oil film in contacts during decades. However, due to the increasing demand to lower levels of vibration and machinery components with a longer lifetime, the study of lubrication in roller bearings has increased in the past few years.

Wiegert, Hetzler and Seemann (2013) proposed an elastohydrodynamic contact force for line contacts combining hydrodynamic and Hertzian models. Qin, Chao and Duan (2015) suggested a load-dependent oil film stiffness and applied to one-degree-of-freedom cam-follower mechanism. Further, Zhang et al. (2016) studied EHD stiffness and damping considering the vibration of line contact. The previous works were focused on elastohydrodynamic lubricated contact force models rather than dynamic modeling. Nonato and Cavalca (2014) investigated the influence of EHD lubrication in deep groove ball bearings and proposed a model for elliptical contact based on nonlinear EHD stiffness. Later, Bizarre, Nonato and Cavalca (2018) applied the previous work in five degrees of freedom ball bearings. Tsuha, Nonato and Cavalca (2016, 2017) proposed an explicit load-displacement relation for infinite line contacts.

The main objective of this work is to analyze the influence of EHD contact force model in load distribution of cylindrical roller bearings in comparison with classic Hertzian theory. The EHD contact reduced order force model is evaluated in function of an elastohydrodynamic contact stiffness and an EHD constant surface separation force parameter proposed in Tsuha, Nonato and Cavalca (2017).

METHODOLOGY

The content of the present work is divided into two parts. First, the parameters of the elastohydrodynamic lubricated contact force model are obtained by solving numerically the full EHD system of equations. Afterward, the force equilibrium in cylindrical roller bearing is evaluated accounting the EHD contact force previously characterized.

EHD contact force model

In order to obtain the EHD contact force model parameters, full EHD problem must be solved considering the dynamic of fluid film, the local elastic deformation and the contact force balance. In infinite line contact, the dynamic of fluid film can be obtained by Reynolds Equation:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{6\eta}\frac{\partial p}{\partial x}\right) - \frac{\partial (u_s \rho h)}{\partial x} = 0 \tag{1}$$

where p is the pressure, h is the lubricant film thickness, u_s is the sum of velocities between both contact surfaces, η is the oil viscosity and ρ is the fluid film density.

Since δ is the approximation between the contact bodies, E' is the reduced modulus of elasticity, and R is the reduced radius of curvature $R^{-1} = R_1^{-1} + R_2^{-1}$, the film thickness equation in line contact is:

$$h(x) = -\delta + \frac{x^2}{2R} - \frac{4}{\pi E'}\int_{-\infty}^{\infty} p(x') ln\left|\frac{(x-x')}{x_0}\right| dx' \tag{2}$$

Due to extreme conditions of operation between the contact bodies, the viscosity and density vary with pressure. The density-pressure and viscosity-pressure relations are given respectively by Dowson and Higginson (1977) and Roelands (1966).

Nevertheless, considering l the effective length of contact surface, the force balance between contact bodies is:

$$l \int_{-\infty}^{+\infty} p(x) dx = f_{i,o} \tag{3}$$

The numerical solution of EHD system of equations, given by Eq. (1), (2) and (3) was obtained by multilevel techniques (Venner and Lubrecht, 2000). Multigrid is applied to Reynolds Equation and Multilevel Multi-Integration (MLMI) is used in film thickness equation to integrate the deformation term.

Since the EHD numerical solution was obtained, the contact force model can be evaluated. On the point of view of contact dynamics, the EHD reduced model can be well represented by an equivalent single degree of freedom (Wijnant, Wensing and Nijen, 1999). Equation (4) shows an explicit load-displacement relation for EHD line contact suggested by Tsuha, Nonato and Cavalca (2016, 2017) over the classic Hertzian dry contact approximation given by Eq. (5) (Harris, 1991).

$$f_{i,o} = K_{EHD_{i,o}} \delta_{i,o} + \Delta F_{i,o} \tag{4}$$

$$f_{i,o} = K_{Hertz} {\delta_{i,o}}^{10/9} \tag{5}$$

where K_{Hertz} is the dry contact stiffness, K_{EHD} is the EHD stiffness, ΔF is an EHD constant surface separation force and δ is the approximation between the contact bodies for external load f. Subscripts relate i for inner raceway and o for outer raceway.

Stiffness K_{EHD} and ΔF are independent of load but are related to geometry and velocity. Thus, it is necessary to obtain different force parameters for inner and outer raceways. K_{EHD} and ΔF can be evaluated by curve fitting the displacement δ to several contact loads f. Afterward, these parameters can be used in dynamic study of bearings.

Load distribution in radial cylindrical roller bearing

Figure 1 indicates the forces distribution on a cylindrical roller bearing subjected to a radial force F_r. Considering equilibrium of forces in each rolling element j, where F_c is the roller inertia force:

$$f_{o_j} - f_{i_j} - F_c = 0 \tag{6}$$

f_{o_j} and f_{i_j} can be rewritten for each cylindrical roller j in function of K_{EHD} and ΔF, according to Eq. (4) for EHD lubrication, and contact stiffness K_{Hertz} giving by Eq. (5) for dry contact. Equation (7) indicates equilibrium of forces in the direction of external radial load. ψ_j is the azimuth angle of element j. Furthermore, inner and outer raceways can be related geometrically in Eq. (8) with bearing radial displacement δ_r and diametral clearance P_d. Solution of the system of equations (6), (7) and (8) using Newton-Raphson method leads to the load distribution in radial roller bearing.

$$F_r = \sum_{j=1}^{j=Z} f_{i_j} \cos \psi_j \tag{7}$$

$$\delta_r \cos \psi_j - \frac{P_d}{2} = \delta_{o_j} + \delta_{i_j} \tag{8}$$

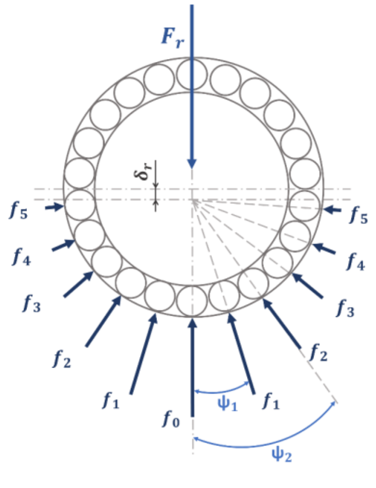
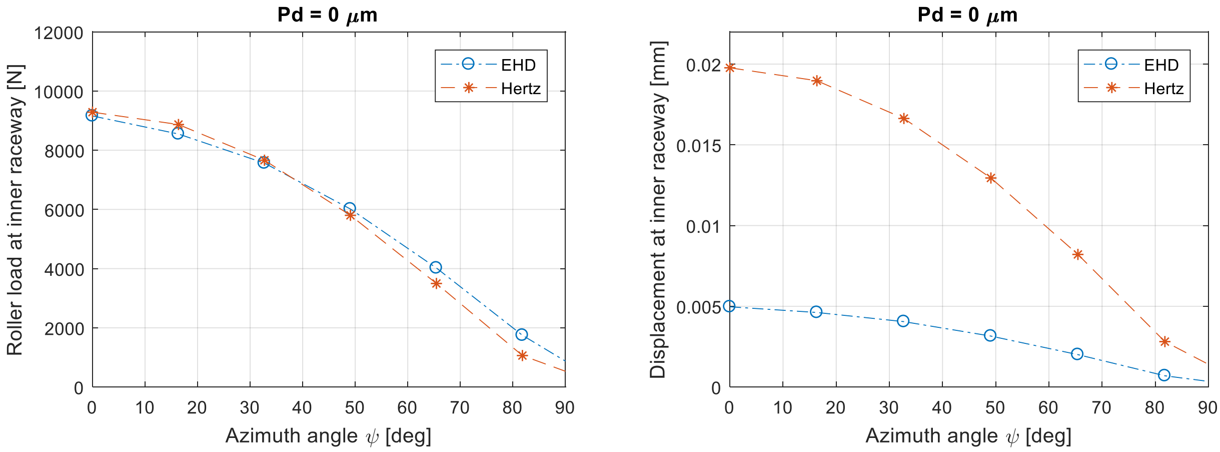


Figure 1 – Load distribution in radial roller bearing.

RESULTS

To evaluate the influence of EHD lubricated contact in roller bearing dynamic, NA4908 needle roller bearing was simulated at 5000 rpm and under radial load of 50 kN. NA4908 roller bearing has $Z = 22$ elements, effective contact length is 0.122 m, pitch diameter is 0.053 m and roller diameter is 0.005 m. Both Hertzian contact force and EHD force model were applied in order to compare the influence of contact stiffness in load distribution.

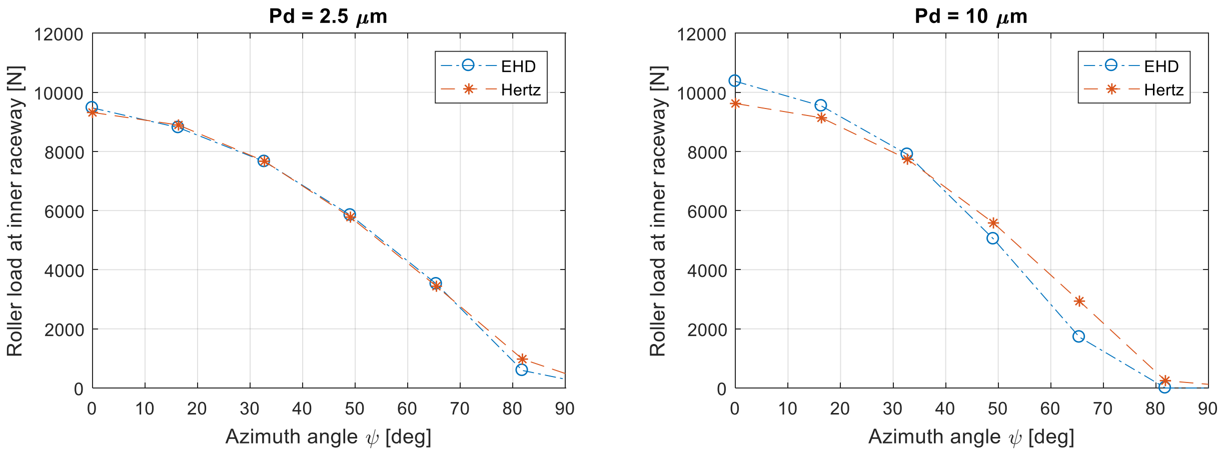
Figure 2 shows the difference between lubricated and dry contact forces in roller bearing with no clearance. The load distribution is similar. However, in the most loaded region, near azimuth angle of 0°, the EHD model tends to reduce the roller load at inner raceway. In contrast, as azimuth angle increases, the EHD model tends to decrease the force at inner raceway in comparison with Hertzian contact model. The displacements at inner raceway in are considerably smaller in EHD lubricated contacts. This difference is caused by the oil film presented between contact bodies in elastohydrodynamic lubrication that can decrease the load displacement.



(a) (b)

Figure 2 – Cylindrical roller bearing at 5000 rpm for Pd = 0 μm. (a) Load distribution. (b) Displacement δ_i at inner raceway for EHD and Hertzian contact force.

The same roller bearing is simulated at 5000 rpm and 50 kN, but different clearances are considered. Figure 3 shows the load distribution for diametral clearance of 2.5 μm and 10 μm. For the minor clearance, load distribution of EHD force model approached the Hertzian theory. On the other hand, the larger clearance leads to an opposite behavior from Fig. 2-(a). In the region near azimuth angle 0°, the EHD model tends to increase the roller load in comparison to dry contact and decreases the force as azimuth angle increases.



(a) (b)

Figure 3 – Load distribution in cylindrical roller bearing at 5000 rpm. (a) Pd = 2.5 μm. (b) Pd = 10 μm.

Figure 4 indicates the differences in displacement between amount values of clearances for Hertzian and EHD contacts. Independent of the clearance applied, there is a larger displacement in dry contact than in EHD lubricated one. As the diametral clearance increases, displacement of rollers near azimuth 0° tends to decrease and elements near azimuth angle 90° tend to increase. The same behavior is verified in both contact models, although the differences of amplitude of displacement.

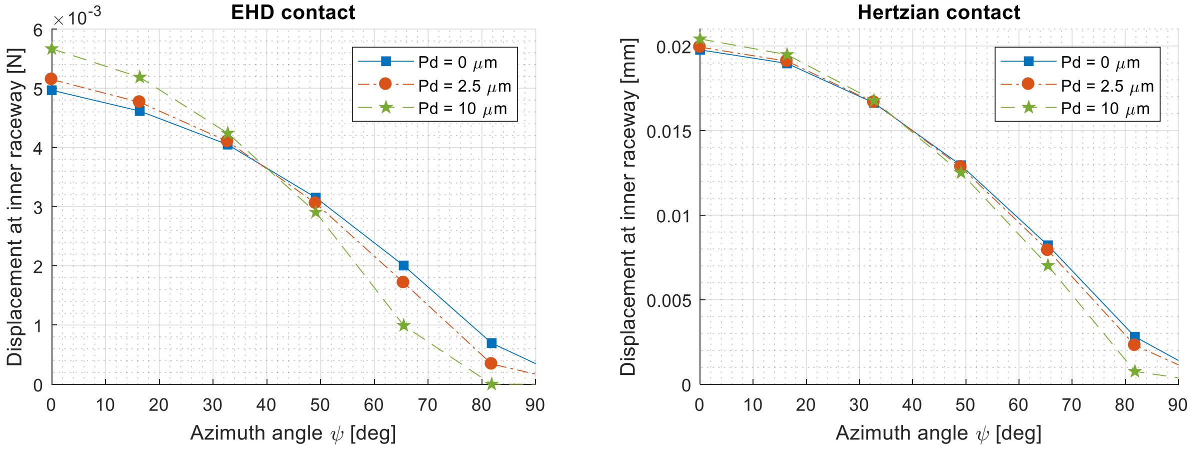


Figure 4 – Displacement δ_i at inner raceway for different clearances. (a) Hertzian contact. (b) EHD contact.

CONCLUSIONS

The application of EHD contact force model instead of classic Hertzian contact theory impacts the equilibrium of forces in radial cylindrical roller bearings. The major influence is related to displacement of the roller elements, although the EHD model also changes the load distribution. The variation of displacement amplitude indicates that the inclusion of EHD lubricated contact in roller bearings has potential to affect the responses in rotors and other mechanical systems.

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