AN INVESTIGATION ON THERMOHYDRODYNAMIC LUBRICATION IN JOURNAL BEARINGS

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***Abstract**. Due to emissions law, as the Kyoto protocol, and customer's requirements, as lower fuel consumption, the automotive industry have been having a lot of difficulties to improve their engines. One way to reduce the emissions and the fuel consumption is increasing the engine's power. Turbochargers are a vital turbomachinery's class designed to improve the power for internal combustions engines, and, as a turbomachine, it has hydrodynamic bearings. In this case, the lubricant acts like a flexible liking element between the journal-bearing surfaces. The lubrication is essential for the turbocharger, because it reduces the wear between the internal parts and prevents the metal contact. Due to the shear stresses present in the lubricant, the temperature rises and, consequently, it changes the lubricant properties. The viscosity is strongly dependent on the temperature and it is the parameter that strongly influences the fluid flow and its dynamic behavior. Therefore, a thermohydrodynamics (THD) analysis allows a more accurate prediction of the bearings performance characteristics in rotating machines. So, due to the high importance of turbomachynery for engines' better performance, a rotor model to analyze the dynamic behavior of an automotive turbocharger is developed, focusing the study of bearings.*

***Keywords**: Thermohydrodynamic Lubrication; Journal Bearings; Stiffness Coefficients; Damping Coefficients.*

1. INTRODUCTION

The dynamic analysis of rotating machines is a complex task, because it involves the analysis of many parameters. Therefore, this investigation should not only take into account the dynamic behavior of the rotor, because it is necessary to analyze the interaction between other components of the same system, such as the supporting structure and the bearings.

In rotor-bearing-structure systems, the vibration transmitted from the rotor to the bearing generates motion in the supporting structure. The interaction between the supporting structure and the bearings retransmit the vibration to the rotor. Thus, the bearings play a very important role in this system by transmitting the forces from the rotor to the supporting structure and vice-versa. For that reason, in order to carry on a dynamic analysis in rotor-bearing-structure systems, the equivalent coefficient of stiffness and damping of the bearings that compose theses systems should be previously known.

The equivalent coefficients determination in hydrodynamic bearings has been a research theme for many years, because there is no global methodology applied in all types of bearings without the generation of uncertainties only a few methods are developed, in which can be noticed the perturbation analysis, the analytical analysis and the numerical approach.

In order to obtain these coefficients, various lubrication models have been developed to represent the behavior of hydrodynamic bearings. These models are classified according to the condition of lubrication, such as hydrodynamic (HD), thermohydrodynamic (THD), elastohydrodynamic (EHD) and thermoelastohydrodynamic (TEHD). However, regardless of the type of model, pressure determination is usually obtained through of the Reynolds' equation solution (Reynolds, 1886).

Thermohydrodynamic analysis, have been used in modeling of bearings considering the heating of the fluid due to the lubricant shearing during the operation. This heating affects the lubrication properties, since the viscosity decreases with the temperature increasing. Consequently, the viscosity reduction causes a decrease in the viscous friction and reduces the oil film sustaining capability.

These effects have been investigated by many authors and with different approach. One of the most significant papers on this subject was written by Dowson and March (1966). In this work, the authors suggest that the thermohydrodynamic behavior in journal bearing can be investigated from the conduction effects between the fluid film and the isothermal shaft. Within this context, Ferron *et al*. (1983) studied the thermohydrodynamic behavior in plain journal bearings and compared the experimental results with the results obtained through a theorical model. This model was developed by finite difference method and it takes into account the heat transfer between the film and both the shaft and the bearings. Because of the high computational costs, Lund and Hansen (1984) developed an approximated analysis of the temperature conditions in a journal bearing. Boncompain *et al.* (1986) developed a general THD theory, where the Reynolds' equation, the energy equation in the film, the heat transfer equation in the bearing and in the shaft are solved simultaneously. The results obtained in this method were in good agreement with experimental results. Similarly, Han and Paranjpe (1990) evaluated the thermohydrodynamic performance of journal bearing through a finite volume analysis, and concluded that the oil supply pressure and the oil input configuration significantly affect the bearing performance. Fitzgerald and Neal (1992) investigated the temperature distributions and the heat transfer in journal bearings through an experimental analysis. The results show that the neglecting of the bearing conduction does not lead to serious overestimation in the prediction of bearing operating temperature. This behavior has been observed by Cameron (1951) as well.

Thus, these researches show the importance in studying and investigating the thermohydrodynamic behavior of fluid film in journal bearing, because a robust analysis improves the reliability and design optimization, avoiding the occurrence of problems during the machine operation.

2. METHODOLOGY

The basis of the modern theory of hydrodynamic lubrication is the Reynolds' equation (Reynolds, 1886), which solution provides the pressure distribution in the oil film. This pressure distribution is the necessary information for solving most of the problems in hydrodynamic bearings analysis .The bearing hydrodynamic forces are obtained from the integration of the pressure field.

The theoretical solution of thermohydrodynamic problems requires the determination of the pressure distribution and, subsequently, the determination of the temperature variation in the oil film. Therefore, firstly, it is assumed the temperature field as being constant and the Reynolds' equation is solved.

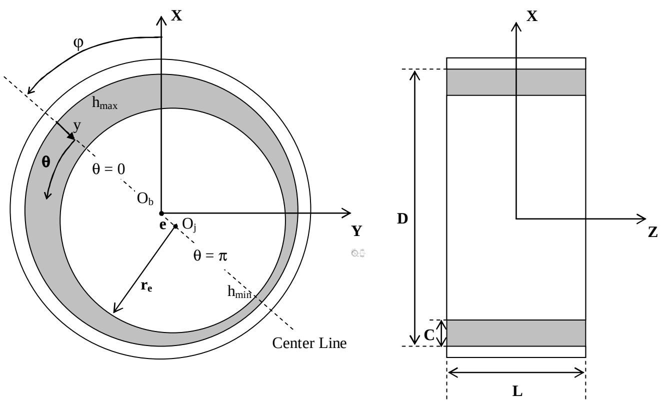


Figure 1. Schematic representation of the journal bearing.

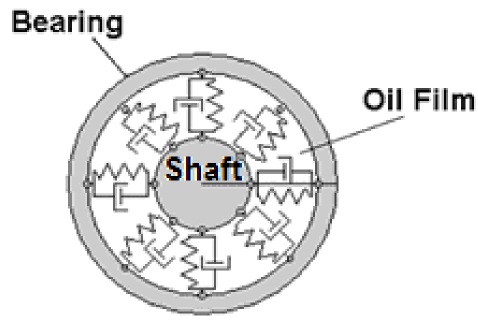


Figure 2: Spring-damper model for fluid film.

To deduce the Reynolds' equation is necessary to obtain the velocities in the circumferential (θ), radial direction (*y*), and axial (Z) direction (u, v e w) that are determined from the Navier-Stokes equations and the continuity equation as follow:

$$u = \left(\int_0^y \frac{y}{\mu} dy - \frac{F_1}{F_0} \cdot \int_0^y \frac{dy}{\mu} \right) \cdot \frac{\partial p}{\partial \theta} + \frac{r_e \cdot \omega}{F_0} \cdot \int_0^y \frac{dy}{\mu} \tag{1}$$

$$v = -\int_0^h \left(\frac{\partial u}{\partial \theta} + \frac{\partial w}{\partial Z} \right) dy + \frac{\partial h}{\partial t} \tag{2}$$

$$w = \left(\int_0^y \frac{y}{\mu} dy - \frac{F_1}{F_0} \cdot \int_0^y \frac{dy}{\mu} \right) \cdot \frac{\partial p}{\partial Z} \tag{3}$$

So, according to the scheme shown in Fig. 1, the Reynolds' equation can be written as (Dowson 1962):

$$\frac{\partial}{\partial\theta}\left[F_2'\cdot\frac{\partial p}{\partial\theta}\right]+\frac{\partial}{\partial Z}\left[F_2'\cdot\frac{\partial p}{\partial Z}\right]=r_e\cdot\omega\cdot\frac{\partial}{\partial y}\left[h-\frac{F_1}{F_0}\right]-\frac{\partial h}{\partial t} \tag{4}$$

Where $F_2'=\int_0^h\frac{y}{\mu}\cdot\left(y-\frac{F_1}{F_0}\right)dy$, $F_1=\int_0^h\frac{y}{\mu}dy=\bar{y}\cdot F_0$, $F_0=\int_0^h\frac{dy}{\mu}$, $p=p(\theta,Z)$ is the pressure distribution of the oil film, θ and Z are the circumferential and radial coordinates, respectively, μ is the absolute viscosity, r_e is the shaft radius, h is the thickness of the oil film and ω is the rotor rotating speed.

After obtaining the pressure distribution and the velocity field, it is necessary to determine the energy equation. The energy equation presented in this paper is simplified. Firstly, the fluid density is considered constant, because there is no heat generation source. The specific heat (Cp) and the thermal conductivity (k) are also considered constant.

Moreover, any heat conduction through the axial coordinate has not been considered, because such conduction is very small when compared with others present in the system as stated by Dowson (1966). Thus, the conduction through of the oil film is the most significant. For this reason, the term $\partial^2T/\partial Z^2$ can be vanished.

In the most of radial bearings, the shear flow is dominant and the temperature variation in the Z direction can be neglected as it is very small, i.e. $\partial T/\partial Z=0$. Therefore, the bi-dimensional energy equation (with temperature variation in circumferential and radial directions due to the boundaries conditions and the viscous shear) is given by the following expression:

$$\rho\cdot Cp\cdot\left(u\cdot\frac{\partial T}{\partial\theta}+v\cdot\frac{\partial T}{\partial y}\right)=k\cdot\left(\frac{\partial^2T}{\partial\theta^2}+\frac{\partial^2T}{\partial y^2}\right)+\mu\cdot\Phi \tag{5}$$

$$\Phi=\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2 \tag{6}$$

The viscosity function in the system is determined from the temperature distribution. The relation of the viscosity with respect to temperature and pressure is given by Larsson (2000):

$$\mu(p,T)=\mu_0(T)\cdot\exp\left\{\left[\ln[\mu_0(T)]+9.67\right\}\cdot\left[-1+\left(1+5.1\times10^{-9}\cdot p\right)^{Z(T)}\right]\right\} \tag{7}$$

$$\log[\log(\mu_0)+4.2]=-S_0\cdot\log\left(1+\frac{T}{135}\right)+\log(G_0) \tag{8}$$

$$Z(T)=D_Z+C_Z\cdot\log\left(1+\frac{T}{135}\right) \tag{9}$$

Where S_0, G_0, D_Z e C_Z are the lubricant parameters.

This sequence (steps of solution) is repeated until the equilibrium point is found. The equilibrium is reached when the forces, given by:

$$\begin{Bmatrix}F_X\\F_Y\end{Bmatrix}=-\int_{-L/2}^{L/2}\int_{\theta_1}^{\theta_2}p.\begin{Bmatrix}\cos\theta\\\sin\theta\end{Bmatrix}\cdot r_e\cdot d\theta\cdot dz \tag{10},$$

are $F_X=W$ and $F_Y=0$, where W is the static load.

Thus, the lubricant will be discretized as a spring-damping model, as can be seen in Figure 2, and characterized by equivalent stiffness and damping coefficients K and B, respectively.

The equivalence between oil film and equivalent spring and damper sets makes use of simple linearized equations, whose response matches with real systems studied.

In a coordinate system $x - y$ with origin in the center of the bearing, and the y axis in the static load direction (Figure 3), the reactions forces in the lubrication oil film are given by Equation (10).

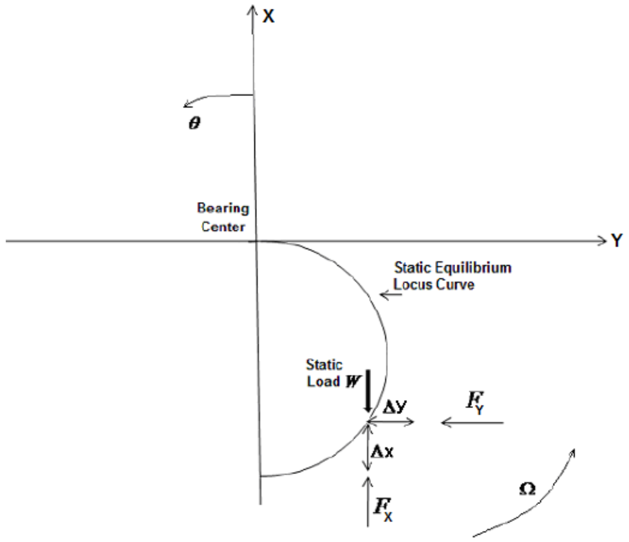


Figure 3. Coordinate system.

As said before, the differential equation, that describe the pressure in the lubricant film, is the Reynolds equation (Equation 1.).

The thickness of the oil film is given by:

$$h = Cr - x \cdot \cos\theta + y \cdot \sin\theta \qquad (11),$$

where Cr is the radial clearance, and x and y are the coordinate of the journal center.

It can be noticed that the reaction forces are function of coordinates x and y, and of the instantaneous velocity of the journal center $\dot{x}$ and $\dot{y}$ ("dot" indicates time derivate). So, for small deviations, Δx and Δy, measured form the static equilibrium (x_0 and y_0), a first order Taylor expansion gives:

$$F_X = F_{X0} + K_{XX} \cdot \Delta x + K_{XY} \cdot \Delta y + B_{XX} \cdot \Delta\dot{x} + B_{XY} \cdot \Delta\dot{y} \qquad (12a);$$

$$F_Y = F_{Y0} + K_{YX} \cdot \Delta x + K_{YY} \cdot \Delta y + B_{YX} \cdot \Delta\dot{x} + B_{YY} \cdot \Delta\dot{y} \qquad (12b),$$

where the coefficients are parcial derivates evaluated in the equilibrium point.

$$K_{XY} = \left(\frac{\partial F_X}{\partial Y}\right)_0 \quad B_{XY} = \left(\frac{\partial F_X}{\partial \dot{Y}}\right)_0 \qquad (13).$$

At the equilibrium point (x_0, y_0), as said before, $F_{X0} = W$ and $F_{Y0} = 0$.

In this paper the coefficients are directly evaluated by numerical differentiation by employing a perturbation solution. So, the fluid film thickness can be written as:

$$h = h_0 + \Delta h \qquad (14),$$

where:

$$h_0 = Cr - x_0 \cdot \cos\theta + y_0 \cdot \sin\theta \qquad (15);$$

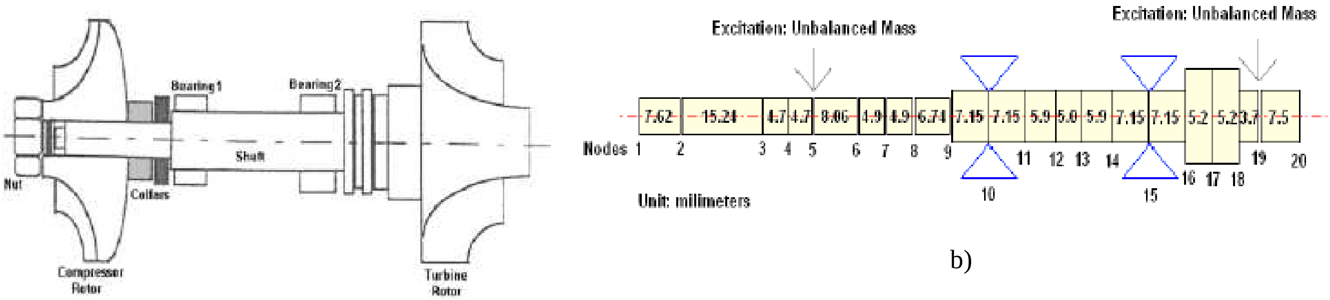
$$\Delta h = \Delta x \cdot \cos\theta + \Delta y \cdot \sin\theta \qquad (16);$$

$$\frac{\partial h}{\partial t} = \Delta \dot{x} \cdot \cos\theta + \Delta \dot{y} \cdot \sin\theta \qquad (17).$$

3. NUMERICAL MODEL

The modeling of the complete rotating system includes both the rotor journal and the bearings that connects the shaft to the supporting structure. The journal, modeled by finite elements, is represented mathematically by matrices of mass, stiffness and damping. The rotor is composed of 19 beam elements with circular section. The diameters of these elements are, 6 millimeters for the elements between nodes 1 and 9, 8.62 millimeters for the elements between nodes 9 and 16 and between nodes 18 and 20, and 15.37 millimeters for the elements between nodes 16 and 18. The numbers inside the elements, in Figure 4a, represent the length of the element in millimeters. The masses of the rotating parts, attached to the shaft, were introduced as lumped masses added in certain nodes. The unbalance forces were placed in nodes 5 and 19, corresponding to the location of lumped masses (0.232 milligrams) of the compressor and the turbine, lagged in 180° between each other and changing with the rotational speed. The hydrodynamic bearings are placed in nodes 10 and 15. In this rotor, a static analysis was performed giving the static load used in the bearings.

Once this work emphasizes the study of bearings, the details of the finite element model to the shaft can be seen in Genta e Gugliotta (1998).



b)

a)

Figure 4. a) Scheme of turbocharger. b) Discretized rotor used in numerical simulation.

However, a complete solution of the THD model is very difficult to obtain analytically, for this reason, the introduction of numerical methods is necessary. One of the most widely used methods is the finite difference method (Morton 2005). This numerical method evaluates the pressure distribution and, by integration, the sustaining forces, allowing the application of the perturbation method to calculate the bearings equivalent stiffness and damping coefficients.

In order to accomplish the numerical simulation, some approaches are applied for the boundary conditions of journal and bearing. In 1966, Dowson *et al.* (1966b) experimentally demonstrated that the journal has a small fluctuation of temperature, due to its rotational motion. This study allows the isothermal approach to the shaft without expressive losses of information. According Cameron (1951), the major part of the heat present in the work fluid is transferred through the journal, and Fitzgerald (1972) concluded that bearing conduction can be neglected with no serious overestimation in the prediction of the bearing operating temperature. This overestimation is, according to Khonsari (1996), about 3% higher. So, the adiabatic condition in the bearing shell was adopted. Therefore, the boundary conditions for the energy equation are: heat exchange with the isothermal shaft and adiabatic bearing shell. For the Reynolds' equation the well known Reynolds' boundary condition was adopted, as stated by Dowson (1962).

4. RESULTS

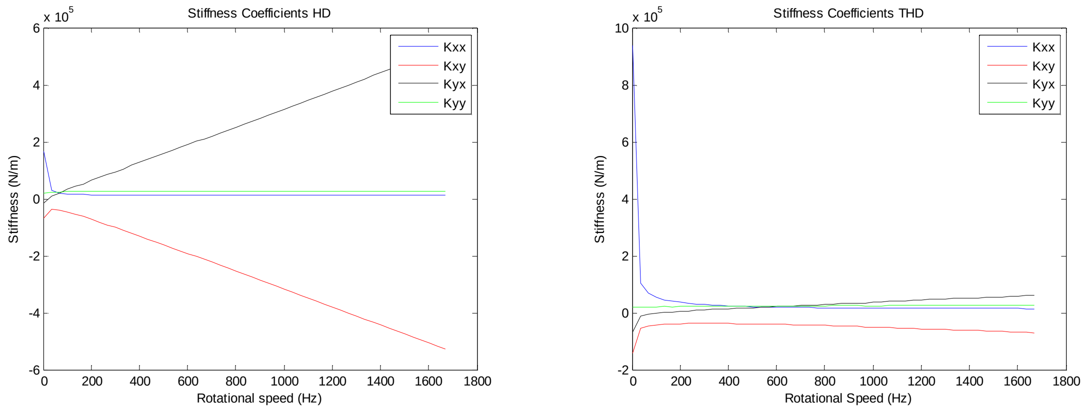
The input data for the numerical simulation are presented in Table 1.

Table 1. Operation Conditions

Diameter of the bearing	D = 8.62 mm
Length of the bearing	L = 5 mm
Radial clearance	C = 70 µm
Load	W_1 = 0.7472 N
	W_2 = 0.9714 N

Density of the lubricant	ρ= 860 Kg/m³
Thermal conductivity of the lubricant	k = 0.13 W/m.°C
Reference viscosity	ηi = 0.04541 Pa.s
Reference temperature	Ti = 60 °C
Shaft temperature	Te = 120 °C

Since it is difficult to precisely determine the average viscosity of the lubricant film, the supply temperature, which is the reference temperature (60 ^{o}C), is adopted to a lubricant viscosity for the isothermal analyses.



a) b)

Figure 6. Stiffness Coefficients for the bearing 1 (0.7472 N): a) HD model. b) THD Model.

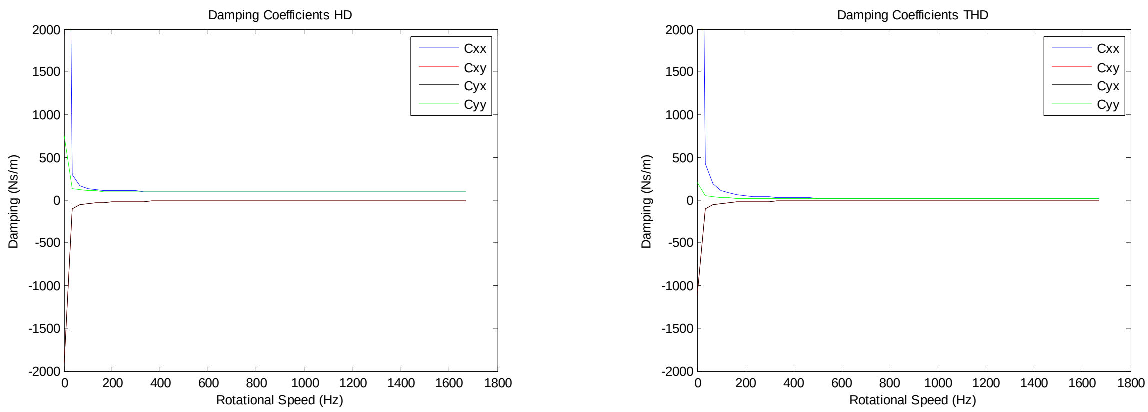


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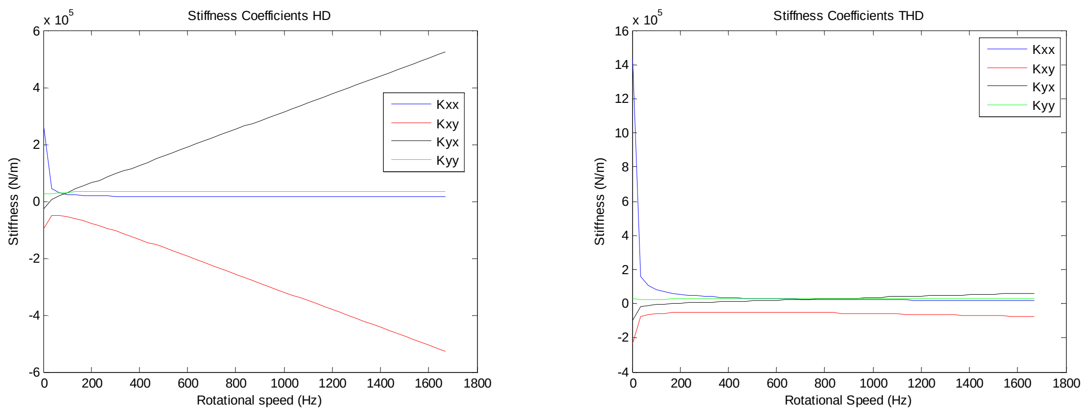
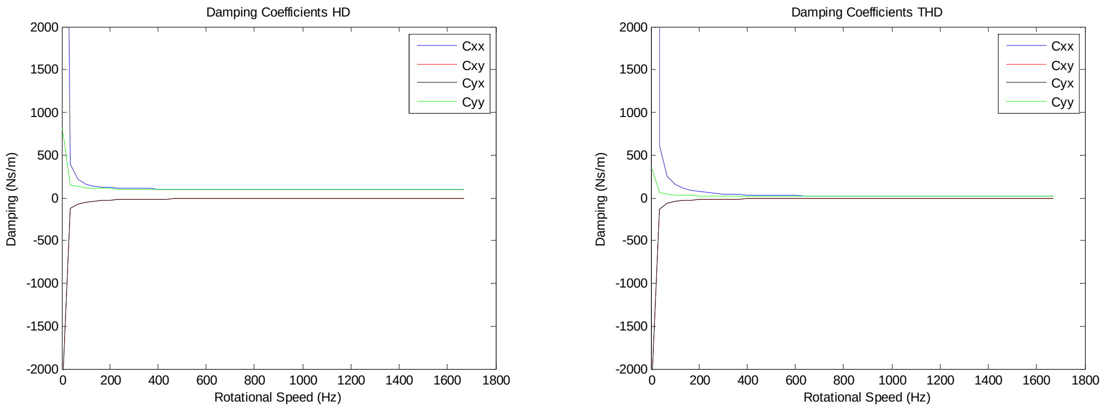
Figure 7. Damping Coefficients for the bearing 1 (0.7472 N): a) HD model. b) THD model.



a) b)

Figure 8. Stiffness Coefficients for the bearing 2 (0.9714 N): a) HD model. b) THD model

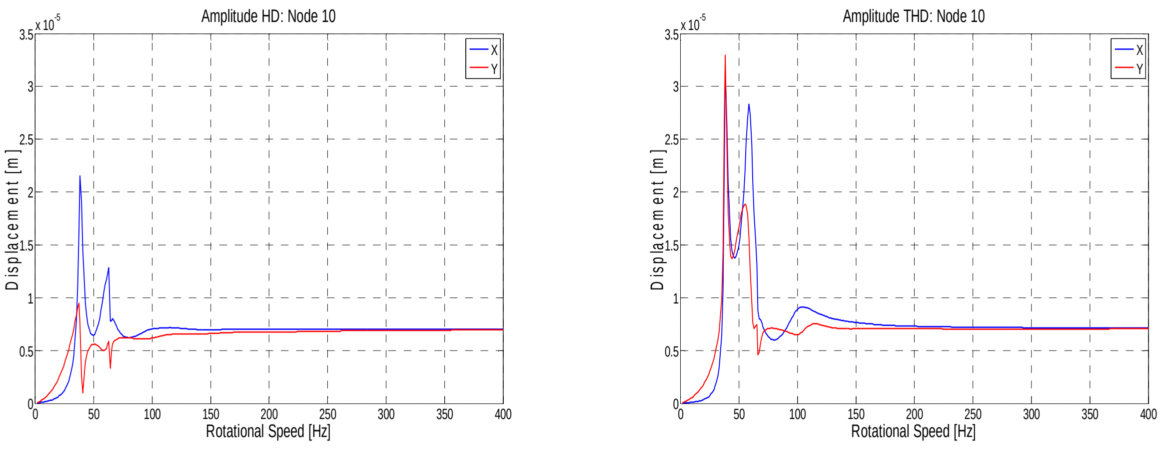


a) b)

Figure 9. Damping Coefficients for the bearing 2 (0.9714 N): a) HD model. b) THD model.

Figures 6 to 9 shows that both stiffness and damping coefficients have the same standard behavior. However, it is clear that the HD and THD models present different slopes, because for each rotational speed, the shaft has a different equilibrium position. This difference can be better seen in the cross-couple stiffness coefficients, K_{xy} and K_{yx}, where, these coefficients reach lower values for the THD model then for the HD model, as the rotational speeds increases. This happens due to the viscosity decreasing as a consequence of the temperature increasing. The direct stiffness coefficients do not present differences in the shape, but in the magnitude, due to the shaft lower location inside the bearing in the THD model, at the same rotation speed.

Besides, both models presents the same tendency to the damping coefficients, except the direct coefficients in the THD model are lower than in the HD model. About the cross-couple coefficients, because they are given by an self-adjoint operator, they have the same values, and consequently, the damping matrix is symmetric and it presents orthogonal basis formed by eigenvectors. However, this effect does not occur in the stiffness matrice, so the cross-couple stiffness coefficients are different.



a) b)

Figure 10. Frequency Response for the bearing 1 (0.7172 N): Amplitude. a) HD model. b) THD Model.

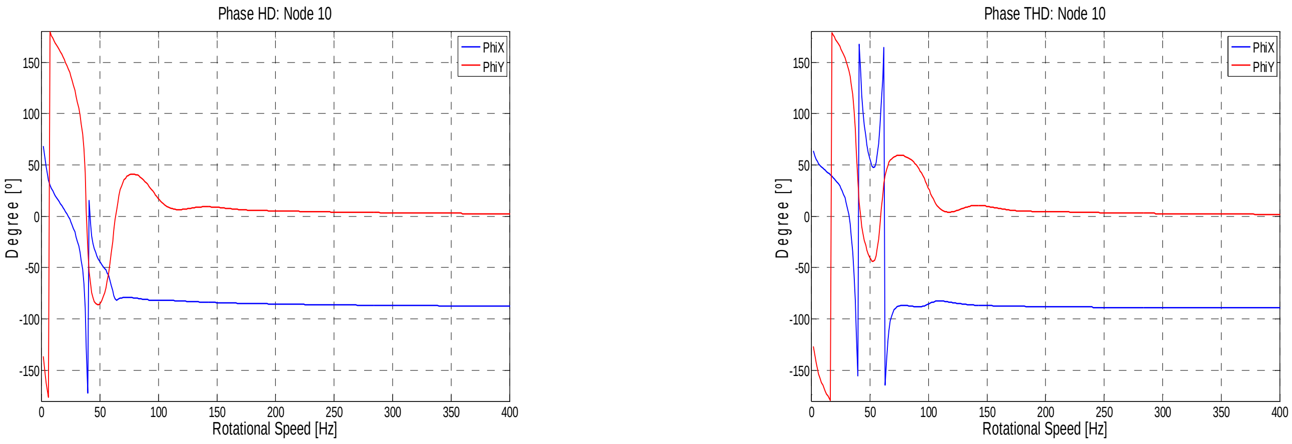


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Figure 11. Frequency Response for the bearing 1 (0.7472 N): Phase. a) HD model. b) THD Model.

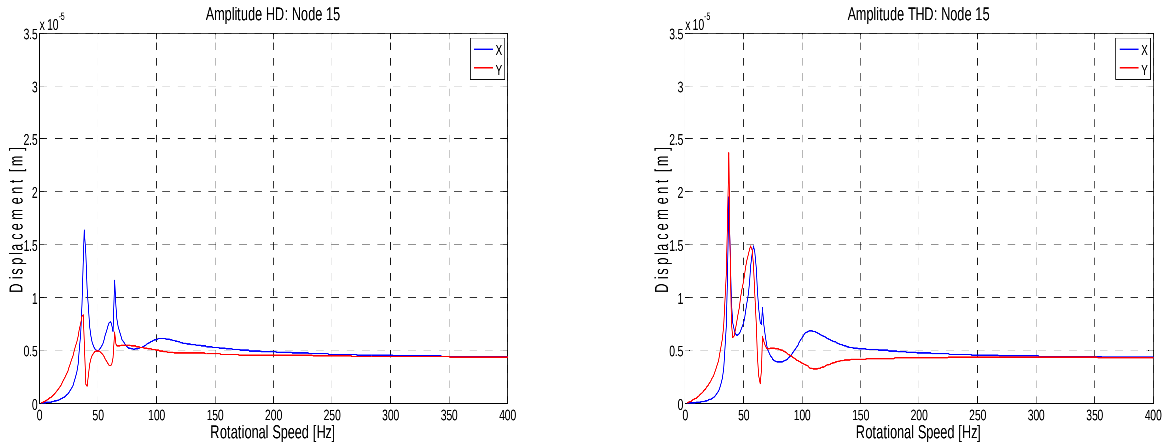

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Figure 12. Frequency Response for the bearing 2 (0.9714 N): Amplitude. a) HD model. b) THD Model.

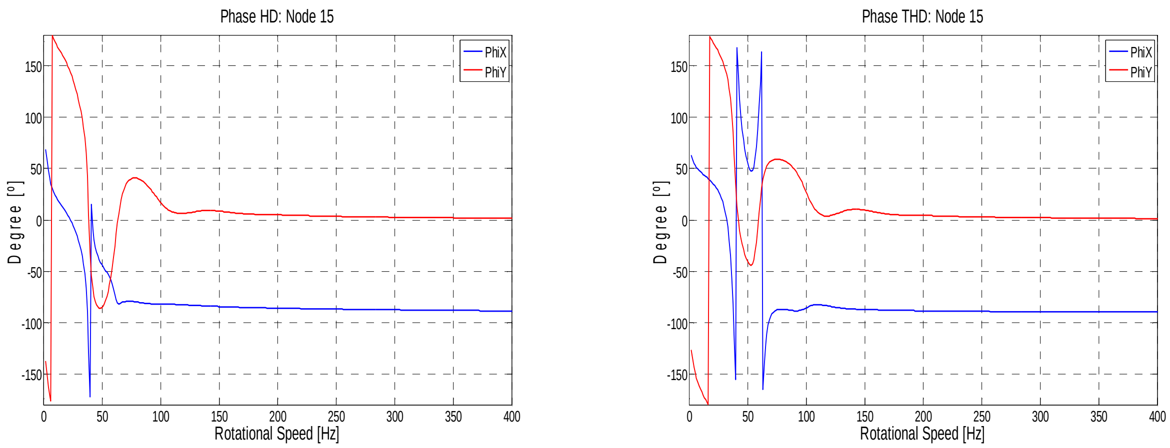

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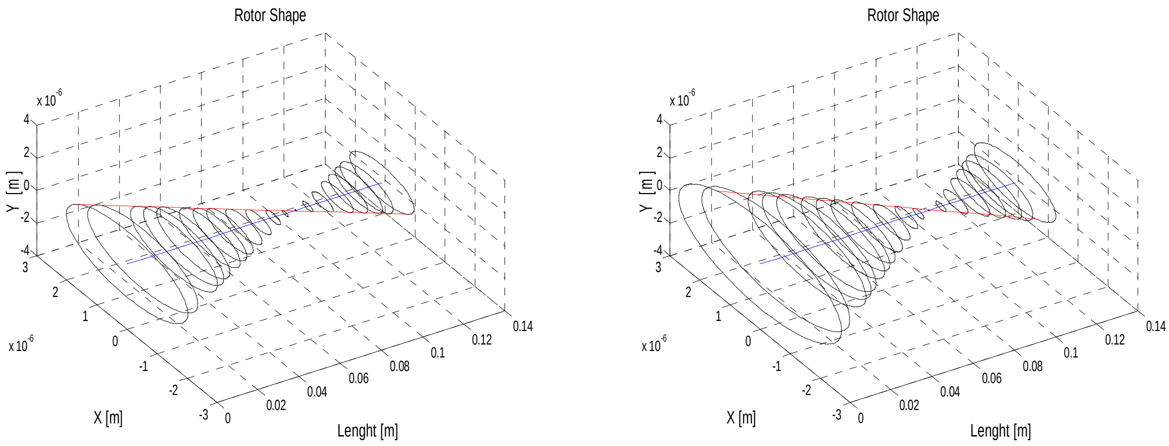
Figure 13. Frequency Response for the bearing 2 (0.9714 N): Phase. a) HD model. b) THD Model.

The Figures (10) and (11), are about the node 10, that correspond to the bearing 1 described in table (1), which load is 0.7472N, while the Figures (12) and (13) correspond to the node 15, where is located the bearing 2 described in table (1), with load of 0.9714N. So, analyzing this figures, can be noticed some critical velocities, where the vibration is higher, that are located in the initial range of velocities. Also, in the THD model, the vibration peak happens on a difference frequency than in the HD model, i.e., in the THD model, we have a displacement of the critical velocity for the right. Moreover, is easy to notice that, due to the decrease of viscosity, the vibration amplitude tends to be higher, cause the stiffness of the bearing decrease too. The Figures (14) and (15) shows the rotor shape for both models, and for two different rotational speeds. Can be seen that the amplitudes in the THD model are greater than in HD model, due to the lower oil viscosity. However, for both models, the operational mode is of rigid body, which is plausible, because the natural frequencies of the shaft, corresponding to its bending modes, are much higher than the analyzed range, so they are not being excited.



a) b)

Figure 14. Rotor shape at 39 Hz: a) HD model. b) THD model.

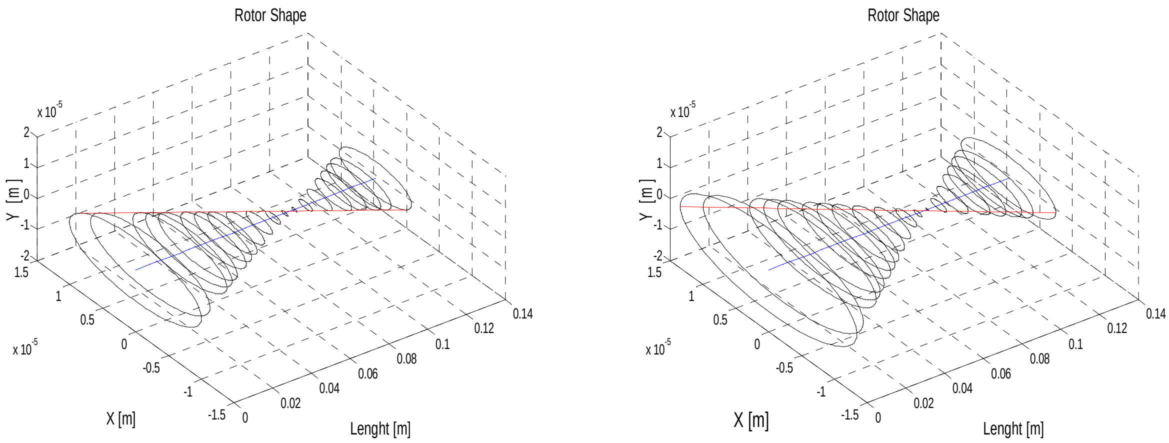


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Figure 15. Rotor shape at 65 Hz: a) HD model. b) THD model.

5. CONCLUSIONS

According to the outcome results, some conclusions about the thermohydrodynamic model applied to bearings can be stated:

- Due to the lower viscosity, the equivalent coefficients are different in the THD and HD models. This difference is more expressive in the case of cross-couple stiffness coefficients and direct damping coefficients.
- The thermal influence causes expressive changes in the location of the maximum peak of vibration.
- The vibration in the THD model present higher magnitude than in the HD model due to the bearings stiffness coefficients decreasing.

The finite difference solution proposed for this problem seems to be very promising and the results present an acceptable consistence.

6. ACKNOWLEDGEMENTS

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