AN INVESTIGATION ON THERMOHYDRODYNAMIC LUBRICATION IN JOURNAL BEARINGS

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Abstract. *Hydrodynamic bearings are used in a great number of applications, especially in the automotive industry. In this case, the lubricant acts like a flexible liking element between the journal-bearing surfaces. The lubrication is essential for the engine, 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 characterizes the fluid flow and its dynamic behavior. Any temperature change induces a consequent modification in the lubricant behavior. Excessive maximum temperatures are one of the basic failures caused in hydrodynamic bearings and uncertain predictions can generate an unreliable design. Therefore, a thermohydrodynamics (THD) analysis allows a more accurate prediction of the bearings performance characteristics and it also generates a temperature distribution in the bearing. Thus, a general THD analysis of a finite journal bearings has been developed. The solution of this problem is obtained from the simultaneous solution of the Reynolds' Equation and the energy equation. The thermohydrodynamic analysis is initially made only to the lubricant, neglecting the heat transfer in the solid parts, so the boundary conditions for the heat transfer were simplified.*

Keywords: *Thermohydrodynamic Lubrication; Journal Bearings; Finite-difference Method.*

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 foundation and the bearings.

In rotor-bearing-foundation 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 foundation and vice-versa. For that reason, in order to carry on a dynamic analysis in rotor-bearing-foundation systems, one should know previously the equivalent coefficient of stiffness and damping of the bearings that compose theses systems.

The equivalents 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, whereby there are many methods.

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 to model 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 generates a decrease in the viscous friction and reduces the oil film pressure. In this way, the hydrodynamic forces are reduced.

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. This work represented a great contribution and it has been the basis for numerous researches. 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. In this method, the differential equations that involve the problem are resolved applying the Fourier series expansion, which obtained results shows that this method presents sufficient accuracy for the most of practical purposes. 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 are compared with experimental results with a good agreement. 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 of the thermohydrodynamic behavior of fluid film in journal bearing, because the analysis helps the reliability and design optimization, avoiding the occurrence of problems during the machine operation.

2. METHODOLOGY

The basis of the modern theory of the 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.

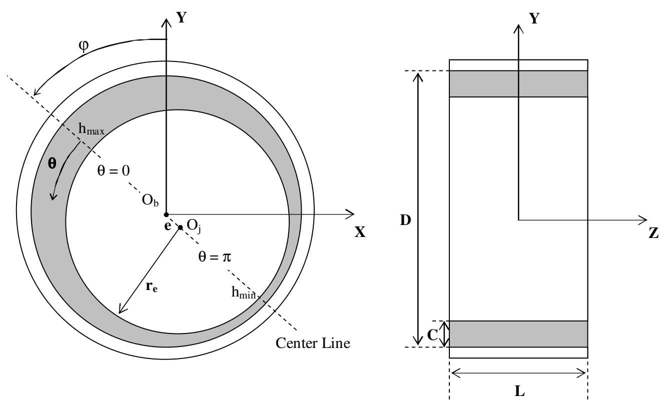


Figure 1. Schematic representation of the journal bearing.

According to the scheme shown in Fig. 1, the Reynolds' equation can be written as:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\cdot\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\cdot\frac{\partial p}{\partial z}\right)=6\cdot r_e\cdot\omega\cdot\frac{\partial h}{\partial x} \tag{1}$$

Where $p = p(x, z)$ is the pressure distribution of the oil film, x and z are the rectangular coordinates, η is the absolute viscosity, r_e is the bearing radius, h is the thickness of the oil film and ω is the rotor rotating speed.

The Reynolds' formulation is obtained from the Navier-Stokes equations, considering the following hypotheses:

• The effects of viscous shear is predominant, so the only important parameter of the fluid is the absolute viscosity;
• The inertia forces of the fluid are neglected;
• The fluid is incompressible;
• The film thickness is small, which allows the consideration of a constant pressure through the film;
• The curvature of the film is negligible;
• There is no sliding at the interface between the fluid and the surfaces of the shaft and the bearing.

After determining the pressure distribution in the bearing, it is necessary to obtain the velocity field of the fluid, which will be used later in the energy equation.

The velocities in the longitudinal, radial and axial direction (u, v e w) are determined from the Navier-Stokes equations and the continuity equation as follow:

$$u = r_e \cdot \omega \cdot \frac{y}{h} - y \cdot \left(\frac{h-y}{2 \cdot \eta} \right) \cdot \frac{\partial p}{\partial x} \tag{2}$$

$$v = \frac{\partial h}{\partial x} \cdot \left[\frac{y}{h} \cdot u + r_e \cdot \omega \cdot \left(\frac{y}{h} \right)^2 \cdot \left(1 - \frac{y}{h} \right) \right] \tag{3}$$

$$w = -y \cdot \left(\frac{h-y}{2 \cdot \eta} \right) \cdot \frac{\partial p}{\partial y} \tag{4}$$

Where $\partial h / \partial z$ has been vanished.

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 source of heat generation. 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. Thus, the conduction through of the oil film is the most significant. For this reason, the term $\partial^2 T / \partial z^2$ can be vanished.

In the most of the 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 energy equation is given by the following expression:

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

$$\Phi = 2 \cdot \left[\left(\frac{\partial u}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial y} \right)^2 \right] - \frac{2}{3} \cdot \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} \right)^2 + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 + \left(\frac{\partial w}{\partial y} \right)^2 + \left(\frac{\partial w}{\partial x} \right)^2 \tag{6}$$

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:

$$\eta = \eta_i \cdot \exp(-C_1 \cdot (T - T_i) + C_2 \cdot p) \tag{7}$$

Where η_i is the reference viscosity, T_i is the reference temperature and C_1 and C_2 are the lubricant parameters.

This sequence (steps of solution) is repeated until the convergence, which occurs when the difference between the temperatures of two consecutive steps for the same point of analysis is less than a specific error.

However, this solution 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, which solution is shown in the next section.

2.1. Solution of the Reynolds' Equation by finite difference method

From the Reynolds' Equation for hydrodynamic lubrication, many researches have developed numerical methods to solve the equation for radial bearings. However, the analytic solutions are limited in specific cases, such as the Sommerfeld's Solution and Ocvirk's Solution. Consequently, numerical methods have been better developed due to the improvement of the computational resources, facilitating the solution of this equation.

In order to apply the finite difference method in the Reynolds' Equation, it is convenient to reduce the number of variables of the problem applying a dimensionless set. With this purpose, the following dimensionless set is proposed:

$$\bar{x} = \frac{x}{r_e} \quad (8) \qquad \bar{z} = \frac{z}{r_e} \quad (9) \qquad \bar{h} = \frac{h}{C} = 1 + \varepsilon \cdot \cos\theta \quad (10)$$

$$\overline{\eta} = \frac{\eta}{\eta_0} \quad (11) \qquad \overline{p} = \frac{p}{6 \cdot \eta_0 \cdot \omega \cdot \left(\frac{r_e}{C}\right)^2} \quad (12)$$

Where C is the radial clearance and ε is the dimensionless eccentricity.
Substituting the dimensionless set in Eq. (1) the dimensionless Reynolds' Equation is obtained.

$$\frac{\partial}{\partial \overline{x}}\left(\frac{\overline{h}^3}{\overline{\eta}} \cdot \frac{\partial \overline{p}}{\partial \overline{x}}\right) + \frac{\partial}{\partial \overline{z}}\left(\frac{\overline{h}^3}{\overline{\eta}} \cdot \frac{\partial \overline{p}}{\partial \overline{z}}\right) = \frac{\partial \overline{h}}{\partial \overline{x}} \quad (13)$$

According to the mesh shown in the Fig. 2, it is possible to obtain the discretized solution for the Reynolds' Equation:

$$\frac{\partial}{\partial \overline{x}}\left(\frac{\overline{h}^3}{\overline{\eta}} \cdot \frac{\partial \overline{p}}{\partial \overline{x}}\right) = \frac{\frac{h_R{}^3}{\eta_R} \cdot p_R + \frac{h_L{}^3}{\eta_L} \cdot p_L - \left(\frac{h_R{}^3}{\eta_R} + \frac{h_L{}^3}{\eta_L}\right) \cdot p_N}{\Delta x^2} \quad (14)$$

$$\frac{\partial}{\partial \overline{z}}\left(\frac{\overline{h}^3}{\overline{\eta}} \cdot \frac{\partial \overline{p}}{\partial \overline{z}}\right) = \frac{\frac{h_T{}^3}{\eta_T} \cdot p_T + \frac{h_B{}^3}{\eta_B} \cdot p_B - \left(\frac{h_T{}^3}{\eta_T} + \frac{h_B{}^3}{\eta_B}\right) \cdot p_N}{\Delta z^2} \quad (15)$$

$$\frac{\partial \overline{h}}{\partial \overline{x}} = \frac{h_R - h_L}{\Delta x} \quad (16)$$

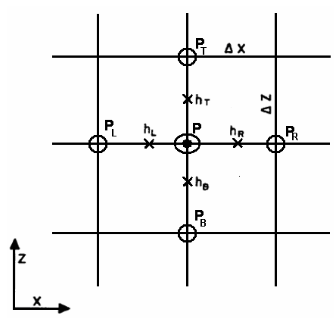


Figure 2. Computational mesh of the model.

Substituting the previous expressions in Eq. (13), the pressure p_N is written as:

$$p_N = \frac{\frac{h_R - h_L}{\Delta x} + \frac{h_T{}^3}{\eta_T \cdot \Delta z^2} \cdot p_T + \frac{h_R{}^3}{\eta_R \cdot \Delta x^2} \cdot p_R + \frac{h_B{}^3}{\eta_B \cdot \Delta z^2} \cdot p_B + \frac{h_L{}^3}{\eta_L \cdot \Delta z^2} \cdot p_L}{\frac{h_T{}^3}{\eta_T \cdot \Delta z^2} + \frac{h_R{}^3}{\eta_R \cdot \Delta x^2} + \frac{h_B{}^3}{\eta_B \cdot \Delta z^2} + \frac{h_L{}^3}{\eta_L \cdot \Delta z^2}} \quad (17)$$

This equation can also be represented as:

$$p_N = c_0 + c_1 \cdot p_T + c_2 \cdot p_R + c_3 \cdot p_B + c_4 \cdot p_L \quad (18)$$

Where c_0, c_1, c_2, c_3 and c_4 are constants. Therefore, a mesh of N points results in N algebraic equations, which can be to resolved by methods of linear systems resolution or iterative methods.

From the pressure distribution, it is possible to evaluate the hydrodynamics forces in the oil film, which can be done integrating the pressure distribution, according to the following equations.

$$F_v = \sum_{n=1}^{n} p_N \cdot \cos\theta_N \cdot (\Delta x) \cdot (\Delta z) \qquad (19) \qquad F_h = \sum_{n=1}^{n} p_N \cdot \sin\theta_N \cdot (\Delta x) \cdot (\Delta z) \qquad (20)$$

Where F_v and F_h are the hydrodynamic forces in vertical and horizontal directions, respectively.

2.2. Solution of the Energy Equation by finite difference method

Initially, a coordinate transformation in the problem domain is necessary. After the coordinate transformation, it is possible to work in a rectangular domain, where a rectangular mesh is applied to obtain the thermohydrodynamic behavior. In this way, it is possible to write the Energy Equation in the transformed domain as:

$$u \cdot h \cdot \frac{\partial T}{\partial \xi} + \frac{\partial h}{\partial x} \cdot r_e \cdot \omega \cdot \eta^2 \cdot (1-\eta)\frac{\partial T}{\partial \eta} = \frac{k}{\rho \cdot Cp} \cdot \frac{1}{h} \cdot \left(\frac{\partial^2 T}{\partial \xi^2} + \frac{\partial^2 T}{\partial \eta^2}\right) + \frac{\mu}{\rho \cdot Cp} \cdot \frac{1}{h} \cdot \Phi_T \qquad (21)$$

Where Φ_T is the viscous dissipation in the transformed domain, written as:

$$\Phi_T = 2 \cdot \left[\left(h \cdot \frac{\partial u}{\partial \xi}\right)^2 + \left(\frac{\partial v}{\partial \eta}\right)^2\right] - \frac{2}{3} \cdot \left(h \cdot \frac{\partial u}{\partial \xi} + \frac{\partial v}{\partial \eta}\right)^2 + \left(\frac{\partial u}{\partial \eta} + h \cdot \frac{\partial v}{\partial \xi}\right)^2 + \left(\frac{\partial w}{\partial \eta}\right)^2 + \left(h \cdot \frac{\partial w}{\partial \xi}\right)^2 \qquad (22)$$

Therefore, as presented before in the Reynolds' Equation, it is necessary to obtain the Energy Equation in the dimensionless form, as follow:

$$\bar{\xi} = \frac{\xi}{r_e} \quad (23) \qquad \bar{z} = \frac{z}{r_e} \quad (24) \qquad \bar{h} = \frac{h}{C} = 1 + \varepsilon \cdot \cos\theta \quad (25) \qquad \bar{\mu} = \frac{\mu}{\mu_0} \quad (26)$$

$$\bar{u} = \frac{u}{r_e \cdot \omega} = \eta - 3 \cdot \frac{\bar{h}^2}{\bar{\mu}} \cdot \frac{\partial \bar{p}}{\partial \bar{\xi}} \cdot \eta \cdot (1-\eta) \qquad (27)$$

$$\bar{v} = \frac{v}{r_e \cdot \omega} = \frac{\partial \bar{h}}{\partial \bar{\xi}} \cdot \frac{C}{r_e} \cdot \eta \cdot \left[\bar{u} + \eta^2 \cdot (1-\eta)\right] \qquad (28)$$

$$\bar{w} = \frac{w}{r_e \cdot \omega} = -3 \cdot \frac{\bar{h}^2}{\bar{\mu}} \cdot \frac{\partial \bar{p}}{\partial \bar{z}} \cdot \eta \cdot (1-\eta) \qquad (29)$$

Substituting this dimensionless set in the Eq. (21), it obtains:

$$\bar{u} \cdot \frac{\partial T}{\partial \bar{\xi}} + \frac{1}{\bar{h}} \cdot \frac{\partial \bar{h}}{\partial \bar{x}} \cdot \eta^2 \cdot (1-\eta)\frac{\partial T}{\partial \eta} = Pe^* \cdot \frac{1}{\bar{h}^2} \cdot \left(\frac{\partial^2 T}{\partial \xi^2} + \frac{\partial^2 T}{\partial \eta^2}\right) + \frac{\mu_0 \cdot \omega}{\rho \cdot Cp} \cdot \left(\frac{r_e}{C}\right)^2 \cdot \frac{\bar{\mu}}{\bar{h}} \cdot \overline{\Phi_T} \qquad (30)$$

Where $Pe^* = \dfrac{k}{\rho \cdot Cp \cdot C^2 \cdot \omega}$ is the inverse of the Peclet number and $\overline{\Phi_T}$ is the dimensionless viscous dissipation, written as:

$$\overline{\Phi_T} = 2\cdot\left[\left[\left(\frac{C}{r_e}\right)\cdot\bar{h}\cdot\frac{\partial\bar{u}}{\partial\xi}\right]^2+\left(\frac{\partial\bar{v}}{\partial\eta}\right)^2\right]-\frac{2}{3}\cdot\left[\left(\frac{C}{r_e}\right)\cdot\bar{h}\cdot\frac{\partial\bar{u}}{\partial\xi}+\frac{\partial\bar{v}}{\partial\eta}\right]^2+ \\ +\left[\frac{\partial\bar{u}}{\partial\eta}+\left(\frac{C}{r_e}\right)\cdot\bar{h}\cdot\frac{\partial\bar{v}}{\partial\xi}\right]^2+\left(\frac{\partial\bar{w}}{\partial\eta}\right)^2+\left[\left(\frac{C}{r_e}\right)\cdot\bar{h}\cdot\frac{\partial\bar{w}}{\partial\xi}\right]^2\Bigg] \tag{31}$$

Finally, the discretization by finite difference can be applied and, using central difference, the partial derivatives are obtained as:

$$\frac{\partial T}{\partial\xi}=\frac{T_R-T_L}{2\cdot\Delta x}\ (32)\qquad \frac{\partial T}{\partial\eta}=\frac{T_T-T_B}{2\cdot\Delta y}\ (33)\qquad \frac{\partial^2 T}{\partial\xi^2}=\frac{T_R+T_L-2\cdot T_N}{\Delta x^2}\ (34)\qquad \frac{\partial^2 T}{\partial\eta^2}=\frac{T_T+T_B-2\cdot T_N}{\Delta y^2}\ (35)$$

Introducing these equations in the dimensionless Energy Equation, it is possible to obtain:

$$T_N=\frac{\frac{\mu_0\cdot\omega}{\rho\cdot Cp}\cdot\left(\frac{r_e}{C}\right)^2\cdot\frac{\bar{\mu}}{\bar{h}}\cdot\overline{\Phi_T}+\left(\frac{Pe^*\cdot\frac{1}{\bar{h}^2}}{\Delta y^2}-\frac{\frac{1}{\bar{h}}\cdot\frac{\partial\bar{h}}{\partial\bar{x}}\cdot\eta^2\cdot(1-\eta)}{2\cdot\Delta y}\right)\cdot T_T+\left(\frac{Pe^*\cdot\frac{1}{\bar{h}^2}}{\Delta x^2}-\frac{u}{2\cdot\Delta x}\right)\cdot T_R}{2\cdot Pe^*\cdot\frac{1}{\bar{h}^2}\cdot\left(\frac{1}{\Delta x^2}+\frac{1}{\Delta y^2}\right)}+ \\ +\frac{\left(\frac{Pe^*\cdot\frac{1}{\bar{h}^2}}{\Delta y^2}+\frac{\frac{1}{\bar{h}}\cdot\frac{\partial\bar{h}}{\partial\bar{x}}\cdot\eta^2\cdot(1-\eta)}{2\cdot\Delta y}\right)\cdot T_B+\left(\frac{Pe^*\cdot\frac{1}{\bar{h}^2}}{\Delta x^2}+\frac{u}{2\cdot\Delta x}\right)\cdot T_L}{2\cdot Pe^*\cdot\frac{1}{\bar{h}^2}\cdot\left(\frac{1}{\Delta x^2}+\frac{1}{\Delta y^2}\right)} \tag{36}$$

Where T_N is the temperature of the central difference, T_L is the temperature of the left difference, T_R is the temperature of the right difference, T_T is the temperature of the top difference and T_B is the temperature of the bottom difference.

3. RESULTS

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

Table 1. Operation Conditions

Bearing Diameter	D = 87 mm
Bearing Width	L = 29.1 mm
Radial Clearance	C = 38 μm
Bearing Load	W = 100 KN
Lubricant Density	ρ= 860 Kg/m³
Lubricant Thermal Conductivity	k = 0.13 W/m.°C
Shaft Thermal Conductivity	ks = 50 W/m.°C
Reference Viscosity	ηi = 0.018 Pa.s
Reference Temperature	Ti = 50 °C
Shaft Temperature	Te = 80 °C

In order to accomplished 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 bearing operating temperature. So, the adiabatic condition was adopted at the bearing.

Since it is difficult to precisely determine the average viscosity of the lubricant film, the shaft temperature is adopted to a lubricant viscosity for the isothermal analyses.

Hence, Table 2 presents the obtained results from the data showed in Table 1, considering the rotation speeds of 2400 RPM, 4000 RPM and 6000 RPM.

Table 2. Obtained Results from the numerical simulation.

	THD Model		Isothermal Model	
Rotation velocity (RPM)	ε	φ (degrees)	ε	φ (degrees)
2400	0,95203	16,064759	0,93296	18,48226
4000	0,93849	17,836201	0,91212	21,04391
6180	0,92602	19,46424	0,88546	23,96212

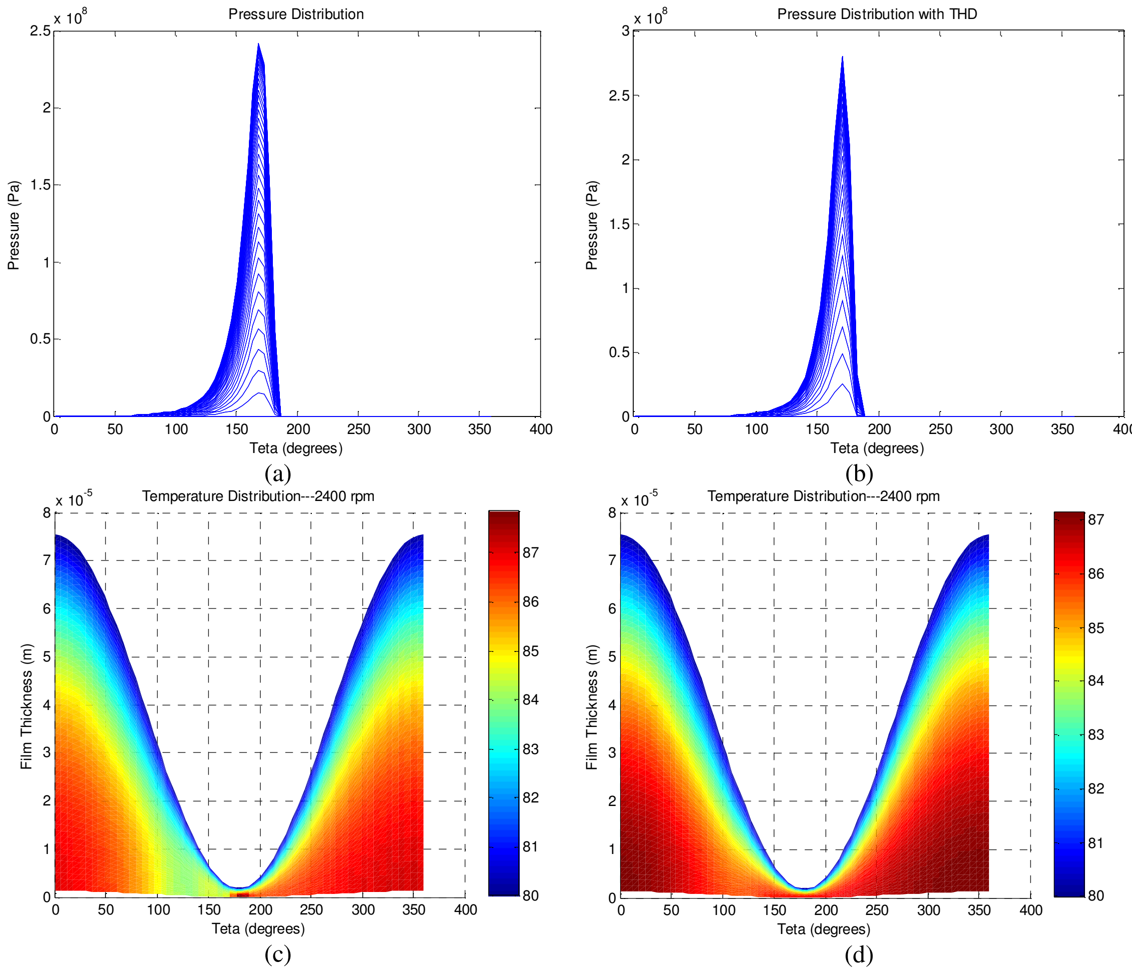


Figure 3. Numerical results at 2400 RPM. (a) Pressure distribution for isothermal model. (b) Pressure distribution for THD model. (c) Temperature distribution at the bearing's middle plane. (d) Temperature distribution at the bearing's extremity.

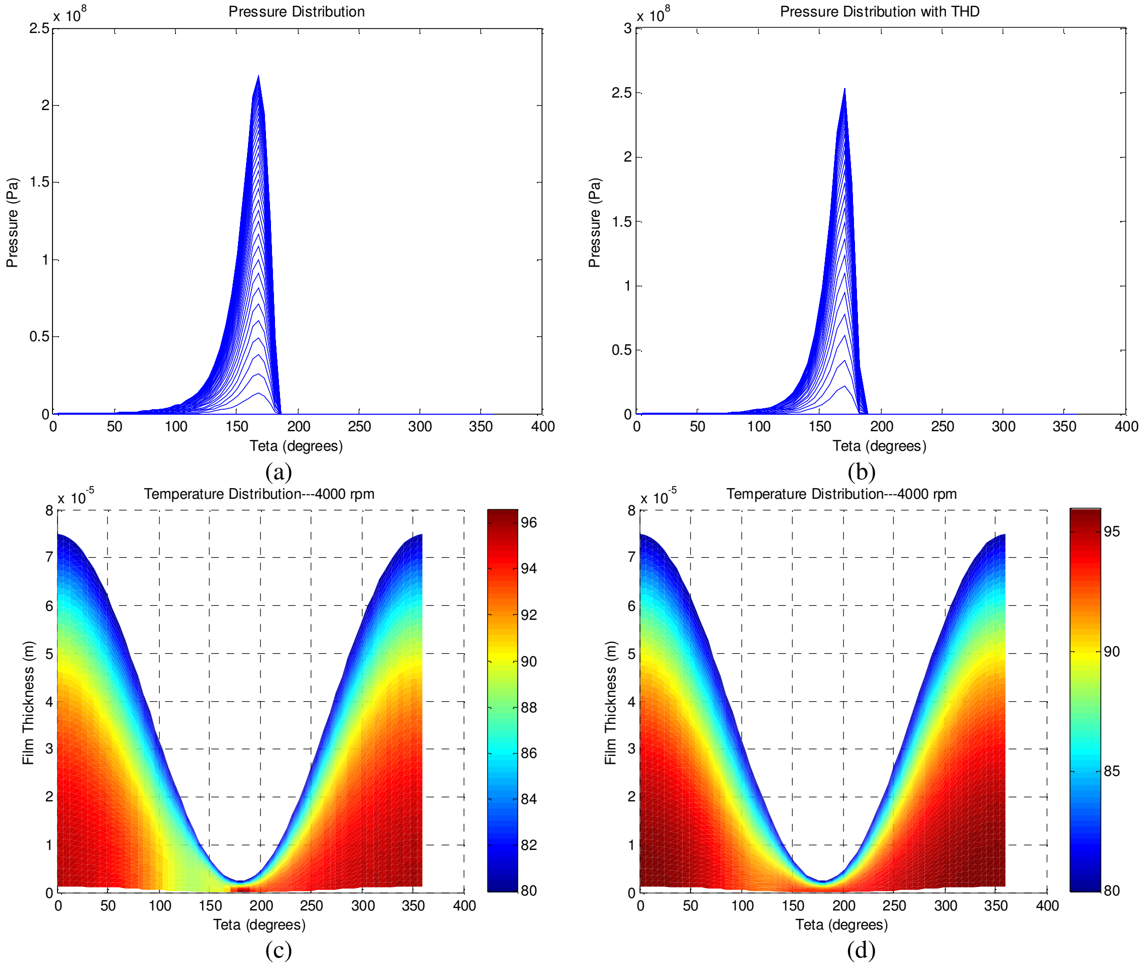
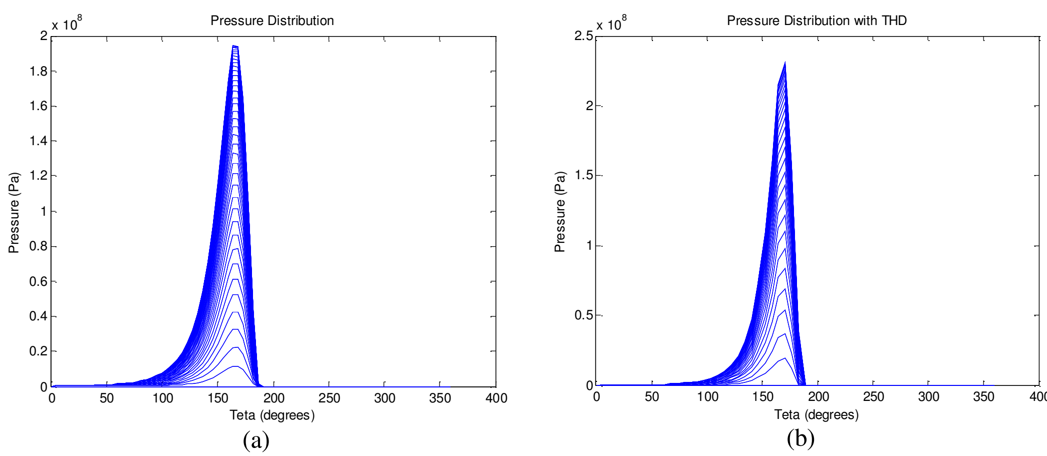
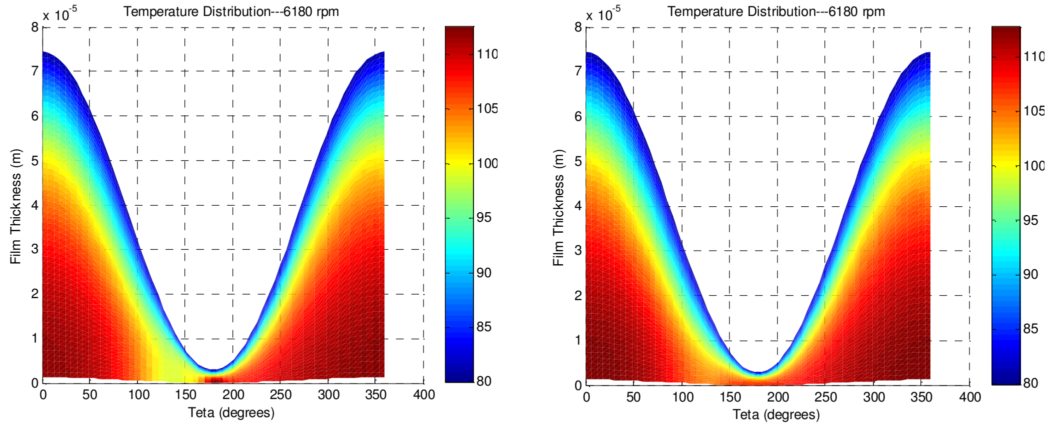


Figure 4. Numerical results at 4000 RPM. (a) Pressure distribution for isothermal model. (b) Pressure distribution for THD model. (c) Temperature distribution at the bearing's middle plane. (d) Temperature distribution at the bearing's extremity.



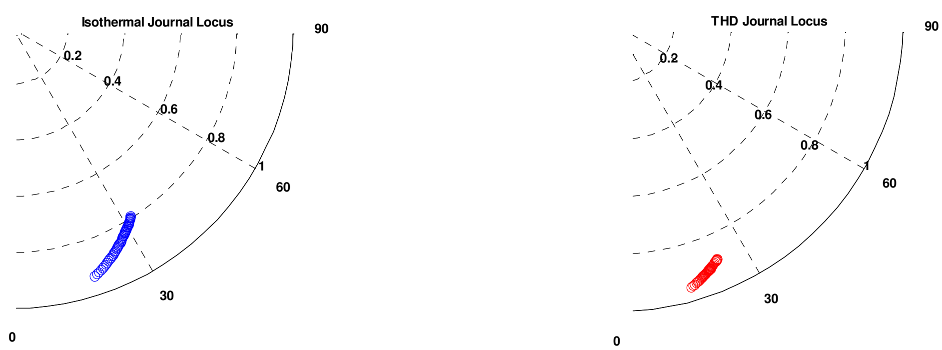


(c) (d)

Figure 5. Numerical results at 6180 RPM. (a) Pressure distribution for isothermal model. (b) Pressure distribution for THD model. (c) Temperature distribution at the bearing's middle plane. (d) Temperature distribution at the bearing's extremity.

It's known that the fluid viscous resistance increases with the deformation rate, i.e., the force required for a fast flux is greater than for slow flux, and also, the viscosity decreases with the temperature increasing, because the greater thermal energy allows the molecules to break, i.e., minor external forces are necessary to separate the molecular links. In this way, checking the results obtained from the THD model and the isothermal model, it can be easily noticed that the maximum pressure and the dimensionless eccentricity are greater in the THD model to compensate the lower viscosity, resulting in the same supporting force. According to Table 2, it's possible to verify that this tendency is easier to see in higher rotation speeds.

Figures 3c, 4c, and 5c show the temperature distribution in the lubricant film. The oil heats up when the journal begins to rotate inside the bearing. In this situation, fluid particles that are in contact with the shaft have the same tangential speed, and fluid particles that are in contact with the bearing surface are static. Being so, there is relative displacement between the atomic plans, i.e., lubricant shearing, which breaks its chemical bonds and releasing energy as heat. Therefore, in higher speeds, the fluid shearing is greatest and, consequently, the energy released is also highest, resulting in a highest generation of heat. It can be verified that the highest temperature is on the bearing wall and the lower temperature is on the journal surface. This is caused due to the boundary condition problem, which considers that the bearing is modeled as adiabatic and the shaft as isothermal. Moreover, the highest temperature is located where the fluid thickness is lower. It occurs because the shear is inversely dependent of the film thickness variation. Consequently, for the same velocity variation, a reduction in the fluid film thickness results in the shear increasing in this region.



(a) (b)

Figure 6. Locus of the journal center: (a) isothermal model; (b) THD model.

As the bearing studied here is quite similar to those applied in an automotive motor, the velocity variation related to the operation of a typical car is used in the simulation, considering the maximum rotation at 7000 RPM and the standard rotation around 2000 RPM. The computational simulation are performed using rotation speeds of 2400, 4000 and 6180 RPM respectively. From the numerical simulation, it is possible to compare the temperature distribution at the central region and at the bearing extremity. Therefore, it can be verified that the temperature distribution obtained at the central region and at the extremity of the bearing are different, although the heat transfer is neglected in the axial direction. This phenomenon can be explained because the viscosity depends also on the pressure and in short bearings, the axial pressure distribution is approximately parabolic. This fact leads to different temperature distribution for the two regions. Due to the pressure parabolic form in the axial direction, the bearing extremity presents lower pressure gradients when compared with the bearing's central region. Therefore, the bearing extremity presents more homogeneous temperature distributions.

Figure 6 shows the locus for the journal center in the bearing. As expected, for the same range of rotating velocities (2580 to 17280 rpm) and external loading (100 KN), both models, isothermal and THD, shows the same centralizing tendency, but it is clear that in the THD model the shaft eccentricity is higher and the attitude angle is lower than the isothermal model, due to the lowest supporting forces for highest temperatures. This means that the shaft centralizing effect is more significant in the isothermal model than in the THD model, what can lead to uncertain prediction in this kind of application.

4. CONCLUSIONS

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

- The results obtained from the THD model reveled that the dimensionless eccentricity is higher when compared with isothermal model for the same load, resulting in a higher pressure to support the journal.
- The maximum temperature is located at the minimum film thickness, because of the high shear stress in the lubricant in this region.
- Due to the neglecting of the heat transfer in the axial direction, the bearing's central and extremity regions have similar temperature values.
- In critical conditions, the high pressure gradients have significant influence in the temperature distribution, affecting the homogeneity of the distributions.

The finite difference solution proposed for this problem seems to be quite viable and the results present a very acceptable consistence.

5. ACKNOWLEDGEMENTS

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