



Effect of Bearing Compliance on Thermo-hydrodynamic Lubrication of High Speed Misaligned Journal Bearing Lubricated with Bubbly Oil

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Abstract

In the present work the effect of bearing compliance on the performance of high speed misaligned journal bearing lined with a compliant PTFE liner lubricated with bubbly oil at high speeds has been studied. The effect of induced oil film temperature due to shearing effect has been implemented. Hydrodynamic effect of the compliant bearing and the influence of aerated oil have been examined by the classical thermohydrodynamic lubrication theory modified to include the effect of oil film turbulence and oil film temperature with suitable models for bubbly oil viscosity and density. The effect of liner elastic deformation has been implemented by using Winkler model. The effects of variable density and specific heat on the most important bearing parameters such as maximum pressure, maximum temperature, bearing load carrying capacity and power losses have been investigated. The results obtained show that the oil film pressure and load carrying capacity increased for the bearing lubricated with bubbly oil of higher aeration level and smaller size of air bubbles. Including the effect of elastic deformation of the bearing liner reduces the oil film pressure, load carrying capacity and frictional power loss for the misaligned bearing working at the same circumstances

Keywords: Journal bearings, hydrodynamic lubrication, THD, compliant effect, bubbly oil, misalignment effect.

1. Introduction

Hydrodynamic bearings are used in industry to support rotating shafts of machines. They are considered as a good choice due to their constructive simplicity, reliability, efficiency, and low cost. Many assumptions have been made to simplify the generalized Reynolds equation such as, the lubricant is pure and free of air bubbles, the bearing is rigid, aligned and working under laminar oil flow. This is not the case for all bearings. Air bubbles are often infiltrated into oil, which makes the bearing lubricated with bubbly oil. It cannot be avoided in practice and has a substantial effect on the performance of such bearings. In fluid film bearings the transition from laminar to turbulent flow is likely to occur at Reynolds number of 2000 [1]. The basic turbulent lubrication theory has been developed by different

workers [2,3,4,5]. Maneshian and Nasab [6] presented a thermohydrodynamic analysis of turbulent flow in journal bearings based on computational fluid dynamic technique. The effect of variable oil density and specific heat on maximum pressure, maximum temperature, bearing load, frictional losses and side leakage in high speed journal bearing have been studied by Chun [7]. All these workers treat the lubricant as incompressible pure oil. Relatively few published studies concerning the effect of air bubbles on hydrodynamic bearing performance have been noticed. When the oil supplied to the lubricated parts and the air entrained into the oil in the form of small bubbles the lubricant viscosity and the density found to be dependent on the air volume fraction (volume ratio of air to total mixture). Abdel-Latife et al. [8] studied the steady state bubbly thermohydrodynamic

behavior of rigid circular pad thrust bearing. The variation in the oil density and viscosity due to the presence of the bubbles as well according to the pressure differentials and temperature rise were considered. Hayward [9] measured a small increase in aerated oil viscosity experimentally. It has been shown that in low volume fraction range the viscosity of air-oil mixtures increases as the volume fraction increases. Smith [10] examining theoretically the effect of bubble entrainment on the performance of isothermal, plain inclined slider bearing subjected to steady loading. The bubbles are considered to influence the lubricating film by altering the lubricant's viscosity and density. Goodwin et al.[11] proposed theoretical model based on Reynolds equation modified to allow for the effect of aeration on lubricant viscosity and density. Nikolajsen [12] was the first to derive analytical models for the density and the viscosity of aerated oil. The results show that the oil aeration can double the load capacity of plain journal bearings. Song and Kim[13] theoretically analyzed the effect of air bubbles evenly distributed in lubricating oil on the bearing performance. The effects of surface tension and air bubble radius are taken into consideration. The rheological properties of bubbly oil are measured under relatively low to high shear rates using a rheometer by Yang et al.[14]. A constitutive equation including shear rate and temperature is constructed and used to develop the bubbly lubrication model of journal bearings, in which cavitation algorithm is applied.

El-Butch and El-Tyabe [15] studied the combined effect of surface roughness and bubbles content on the thermohydrodynamic performance of journal bearings lubricated with bubbly oil. It has been shown that the bearing load carrying capacity is higher at higher bubble content as a direct consequence of the higher pressure values attained. El-Butch [16] Analyzed the thermohydrodynamic performance of dynamically loaded tilting pad journal bearing lubricated with bubbly oil. A slight increase in oil maximum pressure has been noticed in this work. The hydrodynamic bearings used in rotating machinery are usually faced with a thin layer of Babbitt material. This alloy loses its strength and starts to creep at elevated temperature. This thermal constraints treated by the application of PTFE as a facing material. It has been shown that with a compliant liner the oil film pressure decreased while the oil film thickness increased. Many workers show that using compliant liners led to improve the journal bearing operation [17,18,19]. The motivation

behind the present work is to develop numerical thermohydrodynamic model to analyze the effect of bearing elastic deformation on the characteristics of misaligned journal bearing lined with PTFE compliant liner lubricated with bubbly oil working in turbulent regime.

2. Mathematical Formulation

The physical configuration and the coordinate system of the journal bearing lubricated with bubbly oil considered in this work is shown in figure(1). To generalize the study, the following dimensionless coordinates have been defined:

$$\theta = \frac{x}{R}, \quad \bar{y} = \frac{y}{c}, \quad \bar{z} = \frac{z}{L}, \quad \bar{P} = \frac{PC^2}{\mu_o UR}$$

The modified Reynolds equation, to include the effect of air bubbles and journal high speed, governing the oil film pressure is given in non-dimensional form as [22]:

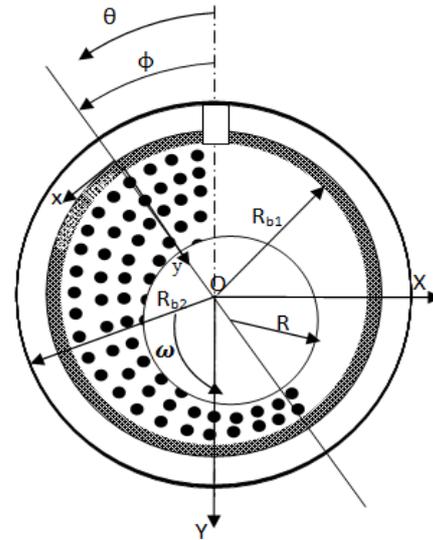


Fig. 1. Journal bearing geometry and coordinates.

$$\frac{\partial}{\partial \theta} \left(\frac{\bar{\rho} \bar{h}^3}{\bar{\mu}} G_x \frac{\partial \bar{P}}{\partial \theta} \right) + \left(\frac{R}{L} \right)^2 \frac{\partial}{\partial \bar{z}} \left(\frac{\bar{\rho} \bar{h}^3}{\bar{\mu}} G_z \frac{\partial \bar{P}}{\partial \bar{z}} \right) = \frac{1}{2} \frac{\partial (\bar{\rho} \bar{h})}{\partial \theta} \quad \dots(1)$$

The following empirical equations proposed by [1] can be used to evaluate the turbulence coefficients G_x and G_z for $1000 \leq Re \leq 30,000$ as follows:

$$G_x = \frac{1}{12 + 0.0136 * (h * u / \nu)^{0.9}} \quad \dots(2)$$

and

$$G_z = \frac{1}{12 + 0.0043 * (h * u / \nu)^{0.96}} \quad \dots(3)$$

The steady state energy equation for incompressible fluid and the hypothesis that the film thickness is very small compared to other dimensions of the bearing can be written in dimensionless form as [22] :

$$\frac{\bar{\rho}}{c_{p0}} \left\{ \left(\frac{\bar{h}}{2} - \frac{\bar{h}^3}{\bar{\mu}} G_x \frac{\partial \bar{P}}{\partial \theta} \right) \frac{\partial (C_p \bar{T})}{\partial \theta} - \left(\frac{R}{L} \right)^2 \left(\frac{\bar{h}^3}{\bar{\mu}} G_z \frac{\partial \bar{P}}{\partial \bar{z}} \right) \frac{\partial (C_p \bar{T})}{\partial \bar{z}} \right\} = \frac{\bar{\tau}_c \bar{\mu}}{\bar{h}} + \frac{\bar{h}^3}{\bar{\mu}} \left\{ G_x \left(\frac{\partial \bar{P}}{\partial \theta} \right)^2 + \left(\frac{R}{L} \right)^2 G_z \left(\frac{\partial \bar{P}}{\partial \bar{z}} \right)^2 \right\} - (\bar{q}_{st} + \bar{q}_{bt}) \quad \dots(4)$$

where:

$$\bar{q}_{st} = \bar{H}_{st}(\bar{T} - \bar{T}_s) \text{ and } \bar{q}_{bt} = \bar{H}_{bt}(\bar{T} - \bar{T}_b)$$

The increase in τ_c is considered by turbulent Couette shear stress factor C_f as:

$$\tau_c = C_f \mu U / h$$

C_f can be evaluated as [1]:

$$C_f = 1 + 0.0012 \left(\frac{Uh}{\rho} \right)^{0.94}$$

The density and the dynamic viscosity of the aerated oil used in Reynolds equation are derived in [12] which can be rewritten her in as:

$$\bar{\rho} = \frac{\rho}{\rho_{oil}} = \frac{(1+\delta)[\bar{P}_{oil} + 2(\bar{\sigma}/\bar{r})]}{\delta + \bar{P}_{oil} + 2(\bar{\sigma}/\bar{r})} \quad \dots(5)$$

where:-

$$\delta = \frac{m_{air}}{m_{oil}} = \frac{(\bar{P}_{oil})_{in} + 2(\bar{\sigma}/\bar{r}_{in})}{(3/4\pi) (\bar{r}_{in}/\bar{d}_{in})^{-3} - 1}$$

\bar{r} : is the real root between 0 and \bar{r}_{in} of the following polynomial equation

$$\bar{P}_{oil} \bar{r}^3 + 2\bar{\sigma} \bar{r}^2 - [(\bar{P}_{oil})_{in} + 2(\bar{\sigma}/\bar{r}_{in})] \bar{r}^3_{in} = 0$$

The effect of oil film temperature on the oil density of the pure oil can be calculated as [22]:

$$\rho_{oil} = 0.0361(a - 0.000354T_f) \times 27,680$$

a : is the +constant depending on oil type, a value of 0.9070 is used through this work as in [22].

T_f : the Fahrenheit temperature of oil

Bubbly oil viscosity can be divided into two parts.

The first is the reduced oil viscosity which expected to diminish with increasing aeration rate due to the negligible internal viscosity of the entrained bubbles. It can be expressed as [12]:

$$\bar{\mu}_1 = \frac{\bar{\rho}}{1 + \delta}$$

In addition to the viscosity reduction discussed above, an increase in viscosity is expected due to the surface tension of the entrained bubbles. The final expression for the viscosity contribution due to surface tension can be written as [12]

$$\bar{\mu}_2 = \frac{\mu}{\mu_{oil}} = \Gamma \bar{h}_{in}^{3/2} \bar{r}^2 / \sqrt{\bar{h}}$$

where:

$$\Gamma = \frac{\pi^2 \sigma}{\sqrt{2} \mu_{oil} U r^3} \left(\frac{r_{in}}{d_{in}} \right)^3$$

The viscosity of lubricants is also a function of temperature. The kinematic viscosity can be evaluated for any temperature by following equation as in [22]:

$$\nu_{oil} = \frac{\mu_{oil}}{\rho_{oil}} = 10^{10[b-c \log_{10}(T_r)]} - 0.6$$

where:

b, c : are constants depends on the lubricant type. Values of 9.8500 and 3.5180 adopted by [22] for the constants b and c .

T_r : oil film temperature in Rankin

C_p is the specific heat (J/kg.C⁰) of oil that may be correlated with Celsius Temperature T_c as follow: [22]

$$C_p = 1796 + \frac{691}{160} T_c$$

The final expression for the overall viscosity of aerated oil is :

$$\bar{\mu} = \bar{\mu}_1 + \bar{\mu}_2 \quad \dots(6)$$

The oil film thickness for the bearing including the elastic deformation of a compliant bearing liner can be evaluated as:

$$\bar{H} = \bar{h} + \bar{\delta}$$

where :

(\bar{h}) : is the oil film thickness of aligned journal bearing which can be evaluated as :

$$\bar{h} = 1 + \varepsilon \cos(\theta - \phi) \quad \dots(7)$$

The procedure used by Bouyer and Fillon [20] has been followed in the present work to study the effect of journal misalignment on the journal bearing performance.

$$h = c + e \cos(\theta - \phi)$$

$$+ e' \left(\frac{z}{l} - \frac{1}{2} \right) \cos(\theta - \alpha - \phi)$$

where

e' : is the magnitude of the projection of the axis of the misaligned journal on the mid plane of the bearing .The dimensionless form of the film thickness can be written as:

$$\bar{h} = 1 + \varepsilon \cos(\theta - \phi) + \varepsilon' \left(\bar{z} - \frac{1}{2} \right) \cos(\theta - \alpha - \phi)$$

ε' : is the misalignment eccentricity ratio which can be evaluated as:

$$\varepsilon' = D_m \varepsilon'_{max}$$

where

D_m is the degree of the misalignment in value from 0 to 1 .

ε'_{max} is the maximum possible ε' which can be computed from [20]

$$\varepsilon'_{max} = 2\left(\sqrt{1 - \varepsilon_0^2 \sin^2 \alpha} - \varepsilon_0 |\cos \alpha|\right)$$

The elastic deformation on the working surface of compliant bearing can be evaluated as [21] :

$$\bar{\delta} = C_d \cdot \bar{P} \cdot \bar{t}$$

where :

$$C_d = \frac{\mu_0 \omega (R/c)^3}{E} \times \frac{(1 + \dot{v})(1 - 2\dot{v})}{1 - \dot{v}}$$

Eq.(1) is a differential equation whose main unknown parameter is the pressure, the boundary conditions are:

$$P(\theta, 0) = P(\theta, L) = 0$$

$$P = P_s \text{ (At the oil supply groove)}$$

$$P(\theta_c, z) = \frac{\partial P}{\partial \theta} = 0.0$$

The following assumptions are used to define the temperature boundary conditions as:

At the ends of the bearing, it is reasonable to assume that no heat will be transferred to the surroundings in the axial direction. That is, the temperature of the oil having come out to the surroundings is assumed to be the same as of that at the end of the bearing, so:

$$q_z = 0.0 \text{ at } \bar{z} = 0.0 \text{ and } 1.0$$

The oil temperature is assumed as the mixing temperature at the bearing groove as below

$$T_{mix} = \frac{(Q_{in})T_{in} + L_c Q_{rec} T_{rec}}{(Q_{in}) + L_c Q_{rec}}$$

where L_c the contraction ratio of the oil film which is defined as [7]:

$$L_c(\theta) = \frac{\int_0^L \int_0^{h(\theta^*, z)} u(\theta^*, z) dy dz}{\int_0^L \int_0^{h(\theta, z)} u(\theta, z) dy dz}$$

The heat transfer coefficient of the bush, adjusted for the reduction of wetting area in the cavitation region by the contraction ratio as below[7]

$$H_b = L_c H_{bo}^* + (1 - L_c) H_{bg}^*$$

where

H_{bo}^* is the mixed heat transfer coefficient of aerated oil that is adjusted with the air volume ratio (V) as :

$$H_{bo}^* = H_{bo}(1 - V) + H_{bg}^* V$$

The non-dimensional load parameters \bar{W}_y , and \bar{W}_x , parallel and normal to the line of centers, respectively, are given by:

$$\bar{W}_y = - \int_0^{2\pi} \int_0^1 \bar{P} \cos \theta d\bar{z} d\theta$$

$$\bar{W}_x = \int_0^{2\pi} \int_0^1 \bar{P} \sin \theta d\bar{z} d\theta$$

The total load parameter, \bar{W} , can be evaluated as

$$\bar{W} = \sqrt{\bar{W}_y^2 + \bar{W}_x^2}$$

The attitude angle of the bearing can be calculated as

$$\phi = \tan^{-1} \left(\frac{\bar{W}_y}{\bar{W}_x} \right)$$

3. Numerical Procedure

The Reynolds and the energy equations are differential equations governing the fluid pressure and temperature of the oil film interlinked through viscosity and density temperature dependence. These equations are discretized using central finite difference scheme. The physical domain is divided into 43 nodes along circumferential direction, 15 nodes along the axial direction and 6 nodes across the oil film. The initial oil film pressure and attitude angle have been assumed and the Reynolds equation is solved iteratively. The iterative procedure is stopped when the following convergence criterion is achieved:

$$error_{(\bar{P})} = \frac{\sum_{k=1}^{kk} \sum_{i=1}^n |\bar{P}_{i,k}^n - \bar{P}_{i,k}^{n-1}|}{\sum_{k=1}^{kk} \sum_{i=1}^n |\bar{P}_{i,k}^n|} \leq 10^{-4}$$

Once the pressure field converges, the velocities components and their derivatives are evaluated.

The energy equation is then solved simultaneously with Reynolds equation. The boundary conditions are incorporated in the iterative system. The temperature iterations stopped when the following convergence criterion is achieved:

$$error_{(\bar{T})} = \frac{\sum_{k=1}^{kk} \sum_{i=1}^n |\bar{T}_{i,k}^n - \bar{T}_{i,k}^{n-1}|}{\sum_{k=1}^{kk} \sum_{i=1}^n |\bar{T}_{i,k}^n|} \leq 10^{-4}$$

A computer program written in FORTRAN 90 has been used to solve the governing equations, the iterative procedure is shown in figure(2).

4. Results and Discussion

The results obtained for the performance of finite length journal bearing lubricated with bubbly oil are presented herein. The journal bearing variables as well as the mechanical and thermal properties of the PTFE liner used in computer simulations are listed in table (1). The effect of high speed operation considering convective condition on the bearing walls, elastic deformation of PTFE liner, misalignment of the bearing shaft, the effect of bubble radius, aeration level, bubble surface tension on the performance of the bearing are considered. To validate the approach used through the present work, a comparison between some of the results obtained in this work have been compared with that obtained by other workers as shown in figures (3 to 6). The pressure and temperature distributions for a bearing lubricated with bubbly oil of bubble radius $rb_{in} = 0.02\text{mm}$ and aeration level $rb_{in}/d_{in} = 1/5$ have been obtained and compared with that published by [22] as shown in figures (3) and (4). The bubbly oil dynamic viscosity obtained in the present work has been compared with that obtained by [22] as shown in figure(5). The maximum deviation between the results has been calculated and found to be (5%). The oil film pressure for infinitely long bearing lubricated with bubbly oil has been obtained and compared with that published by [12] as shown in figure(6). The maximum deviation between the results is calculated and found to be (4.7%). The effect of bearing liner with different thicknesses on the oil film pressure generated can be shown in figure (7). It can be deduced from this figure that the rigid bearing shows higher oil film pressure. The oil film pressure decreases as the thickness of the bearing liner increases. This is can be attributed to the elastic deformation of the bearing liner which increases as the liner thickness increases and hence increases the oil film thickness. It is well known that the oil film pressure is inversely proportional to the oil film thickness. Figure (8) shows the oil viscosity distribution through the bearing for different liner thickness. It can be deduced from this figure that the oil film viscosity is slightly increases as the elastic liner thickness increases. This figure explains that the mechanical deformation has the dominant effect on the oil film pressure than the oil viscosity. The effect of aeration level on maximum bearing pressure is presented in figure (9). It can be seen from this figure that maximum oil film pressure increases for the bearing lubricated with bubbly oil of higher aeration level and retains the highest values

for rigid bearing. The increase of the oil film pressure with the aeration level can be attributed to the higher oil viscosity due to the higher surface tension of the bubbly oil in this case. It can also be seen from this figure that decrease in maximum oil film pressure has been obtained for the bearing with thicker PTFE liner. This is can be attributed to the mechanical deformation of the PTFE liner. Decrease in load carrying capacity for the bearing with thicker elastic liner can be explained by referring to figure(10). It can be shown from this figure that the elastic deformation of the bearing liner increases when the bearing lubricated with bubbly oil of higher aeration level. The bearing power loss seems to increase when the bearing works with oil of higher aeration level (r/d) as shown in figure(11). This can be attributed to the higher surface tension force in this case. The effect of the oil aeration level on the maximum oil film temperature can be shown in figure(12). It can be shown from this figure that the maximum oil film temperature increases for the bearing lubricated with bubbly oil of higher aeration level due to the higher bubbly oil viscosity in this case. Figure (13) shows that the attitude angle decreases for the bearing working with bubbly oil of higher aeration level (rb_{in}/d_{in}). A bearing with elastic liner shows lower values of attitude angle in comparison with the rigid bearing. The decrease in attitude angle becomes negligible for a bearing with elastic liner of higher liner thickness. This indicates that using a PTFE liner for the bearing improves its performance since the bearing works with lower attitude angle shows better stability. The effect of bubble radius (rb_{in}) on maximum oil film pressure can be shown in figure (14). It is obvious that the oil film pressure decreases as the bearing lubricated with bubbly oil of larger bubble radius. Figure (15) shows that the maximum elastic deformation of the PTFE layer decreases when the bearing lubricated with bubbly oil of larger bubble radius. This is can be attributed to the decrease of the surface tension force in this case. The liner elastic deformation increases when the bearing lined with thicker PTFE liner. This is can attributed to the higher oil film pressure in this case. Figure (16) shows that the power loss decreases as the bearing lubricated with bubbly oil of larger bubble radius. It becomes lower for a bearing with elastic liner of higher thickness. This is can be explained by knowing that the oil film pressure decreases when the bearing lined with thicker PTFE liner. Figure (17) shows the attitude angle of the bearing increases for the bearing lubricated with bubbly oil of larger bubbles radius

which can be attributed to the increase in oil film pressure in this case. Lower values of attitude angle have been noticed for the bearing with thicker elastic liner. It is well known that the oil film pressure decreases for the bearing lined with thicker PTFE liner which causes in lower values of the load carrying capacity components and hence lower values of attitude angle. A higher oil film temperature has been obtained for the bearing working at higher degree of misalignment as shown in figure (18). It can also be shown from this figure that a higher oil film temperature is obtained for a bearing lubricated with a bubbly oil of higher aeration level. This indicates that the shear rate of the oil film increases when the bearing working at higher degree misalignment. The increase in maximum temperature becomes higher for the bearing lubricated with lubricant of higher aeration level.

5. Conclusions

The effect of bearing liner elastic deformation on thermohydrodynamic lubrication of a finite length misaligned journal bearing lubricated with bubbly oil working at turbulent flow regime has been studied in the present work. From the results obtained it is clear that the oil film pressure and load carrying capacity seem to be significantly affected when the bearing lubricated with oil contained air bubbles. The oil film pressure and load carrying capacity increase when the bearing lubricated with bubbly oil of higher aeration level and smaller size of air bubbles. The oil film temperature seems to be slightly increased for the bearing lubricated with bubbly oil of a higher aeration level while the oil film temperature decreases for the bearing lubricated with bubbly oil of smaller size of air bubbles. All the above results obtained in expense of power loss, which seems to be higher for a misaligned bearing lubricated with bubbly oil of higher aeration level and smaller air bubble size. However including the effect of elastic deformation of the bearing liner reduces the oil film pressure, load carrying capacity and frictional power loss for the misaligned bearing working at the same circumstances mentioned above. Complaint liner inserted in the bearing seems to improve the operational safety and prevent the metal to metal contact of the bearing components.

**Table 1,
Parameters of the oil and the PTFE liner.**

Properties	Values
Parameters of the oil [22]	
Inlet lubricant viscosity μ_o , Pa.s	0.026
journal radius (R), m	0.0368
External bearing radius (r_{bout}), m	0.1
Bearing length (L) m	0.0368
Radial clearance (C), m	0.0001466
Rotation speed (N), rpm	40000
Inlet lubricant pressure (p_s), N/m ²	70000
Eccentricity ratio(ϵ)	0.65
inlet lubricant temper-ature (T_{in}) °C	40
bush temperature (T_b), °C	45
shaft temperature (T_s), °C	45
lubricant density at inlet temperature (ρ), kg/m ³	869.53
lubricant specific heat (C_p), J/kg.C ^o	1968.75
Bush and shaft convection heat transfer (H_{st}), W/m ² .C	7700
air convection heat transfer (H_{bg}), W/m ² .C	2400
Bubble surface tension(σ), N/m	0.0365
Axial groove width	17.1 ^o
Mechanical properties of PTFE [17]	
Liner elastic coefficient (C_d)	34.65
Liner thickness(t), mm	0.0-2.5
Modules of elasticity (E), GPa	0.11
Passion's ratio (ν)	0.46

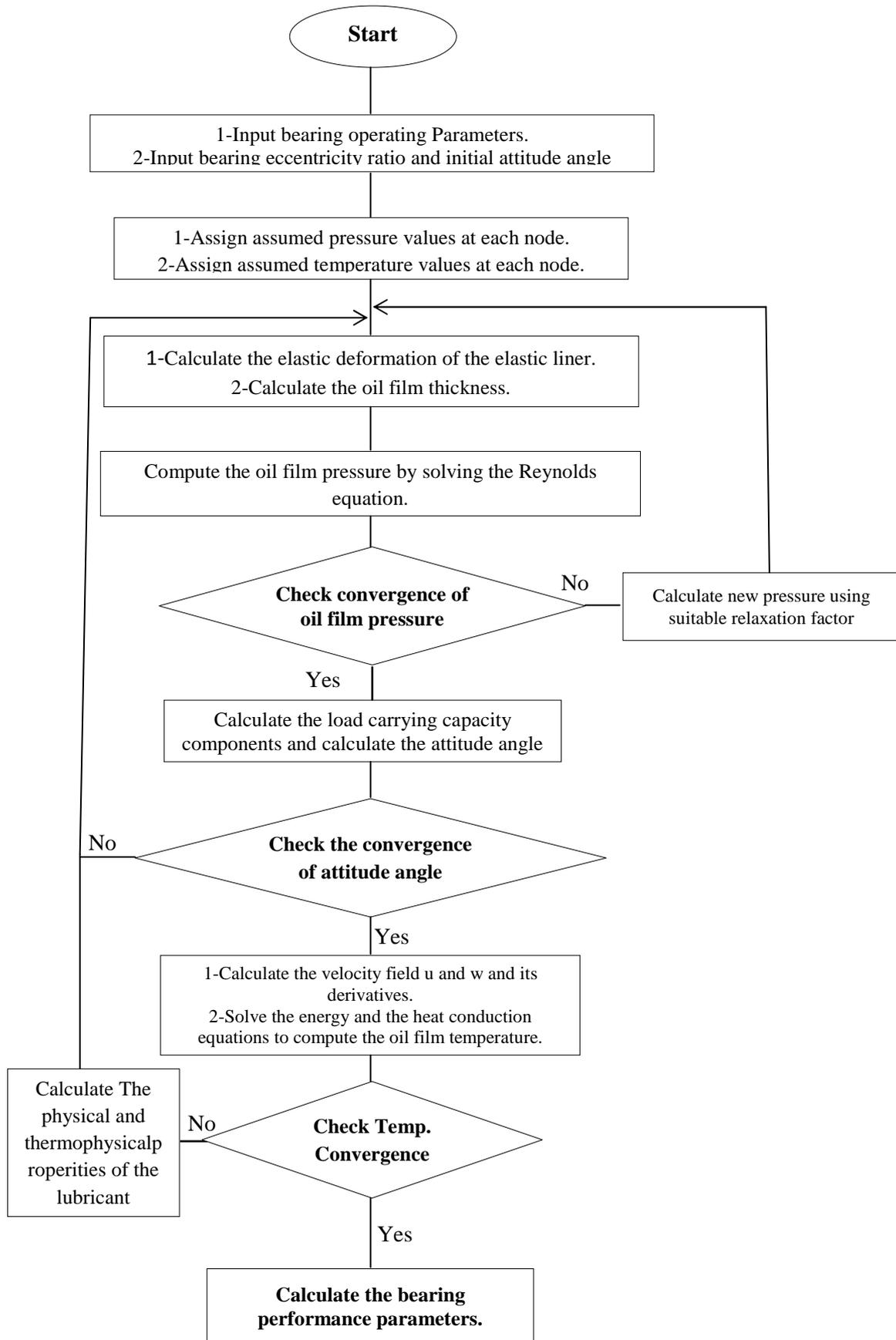


Fig. 2. Flow diagram of solution algorithm.

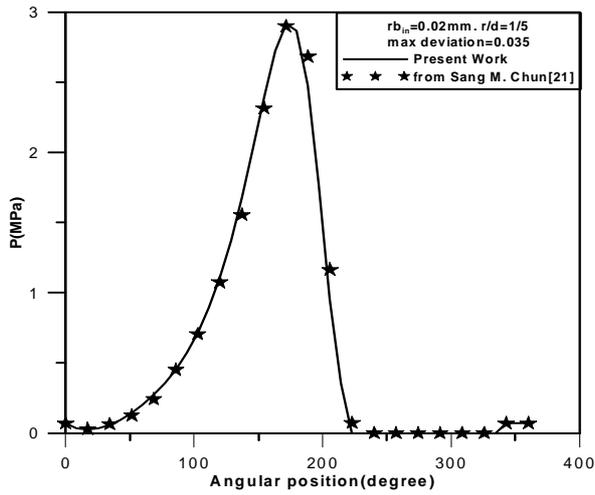


Fig. 3. Comparison with the results in Ref.[22]. Pressure distribution.

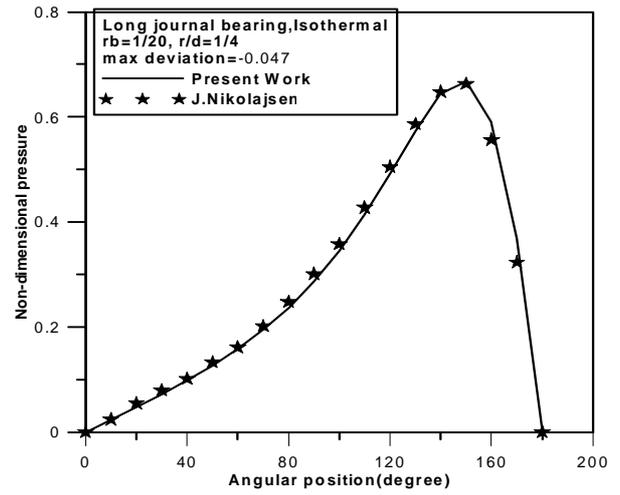


Fig. 6. Comparison with the results in Ref.[12]. Pressure distribution.

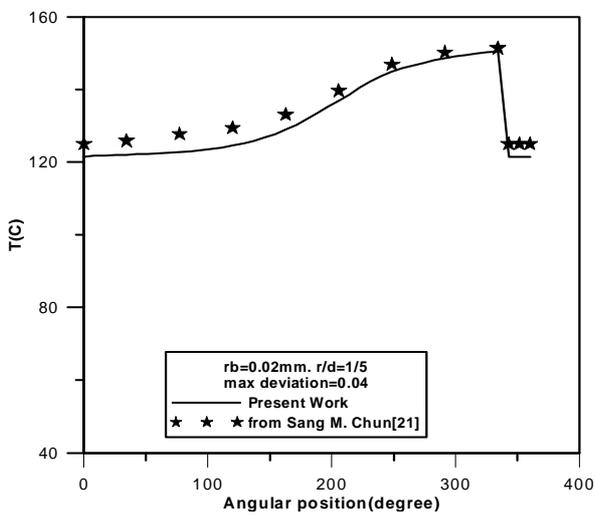


Fig. 4. Comparison with the results in Ref.[22]. Temperature distribution.

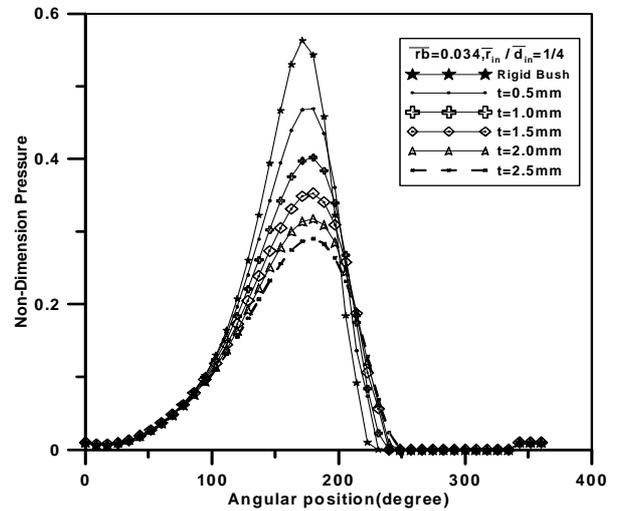


Fig.7. Circumferential distribution of oil film pressure (t=0-2.5mm).

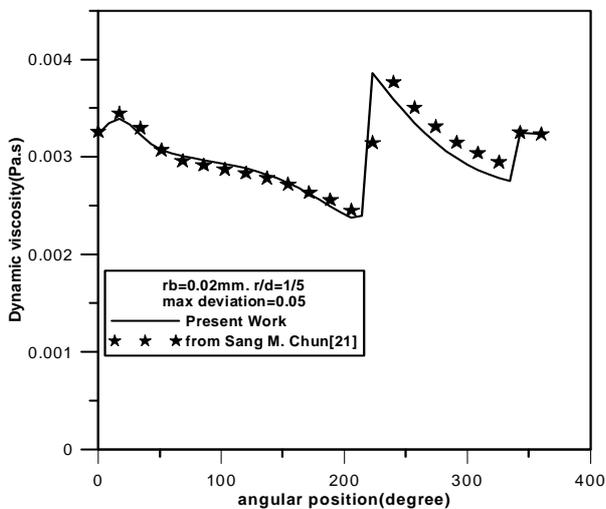


Fig. 5. Comparison with the results in Ref.[22]. Dynamic viscosity distribution.

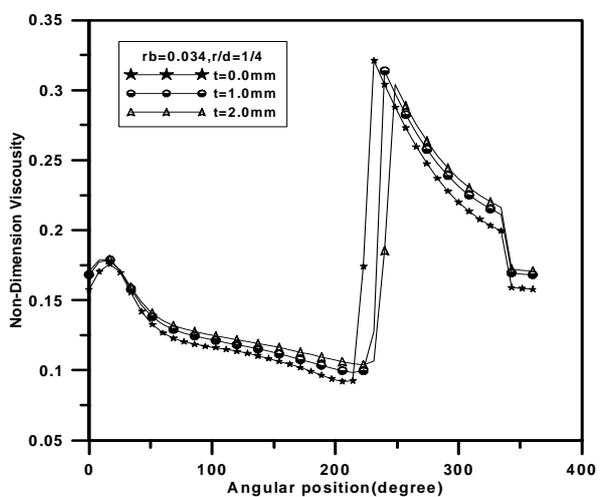


Fig. 8. Circumferential distribution of bubbly oil viscosity under different liner thickness (t=0-2mm).

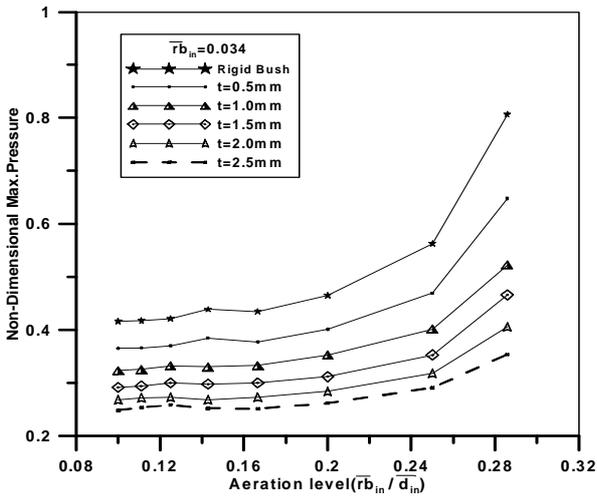


Fig. 9. Maximum pressure versus aeration level under different liner thickness.

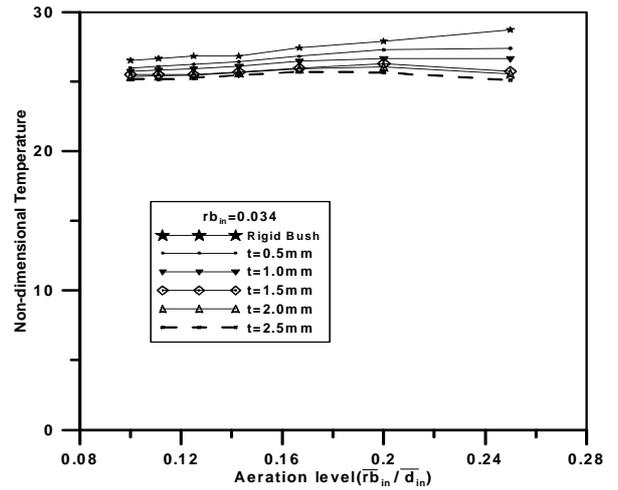


Fig. 12. Maximum oil film temperature versus aeration level under different PTFE liner thickness

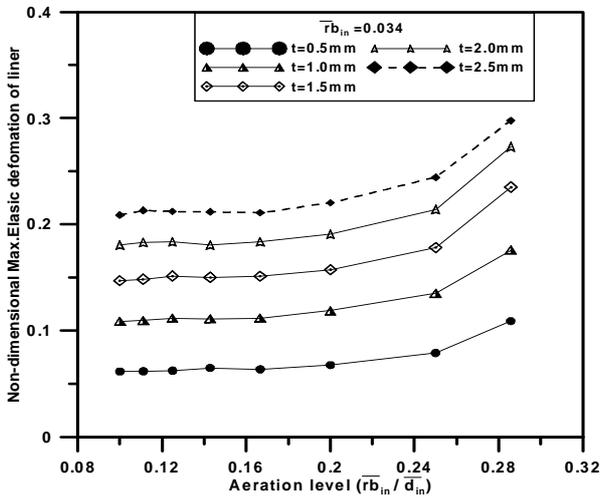


Fig. 10. Elastic deformation of PTFE liner versus aeration level under different liner thickness.

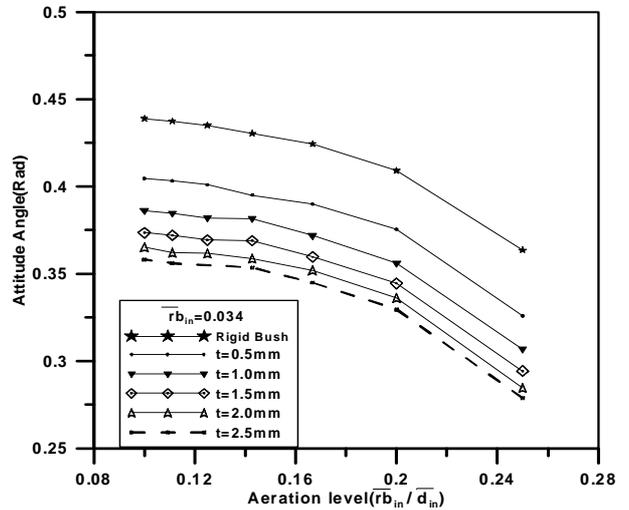


Fig. 13. Effect of aeration level on the bearing attitude angle for different bearing liner thickness

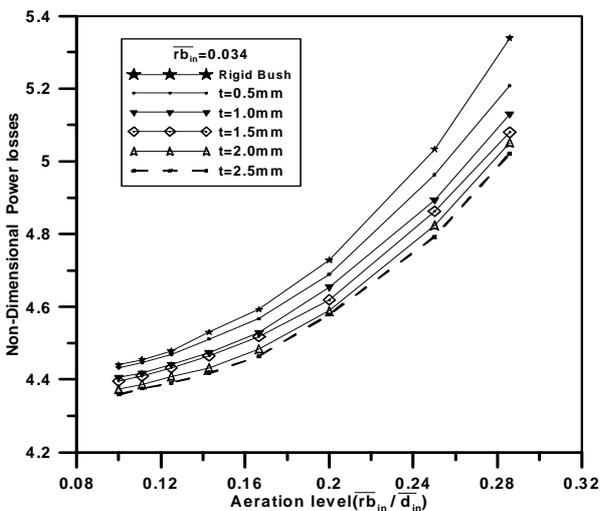


Fig. 11. Power losses versus aeration level under different PTFE liner thickness.

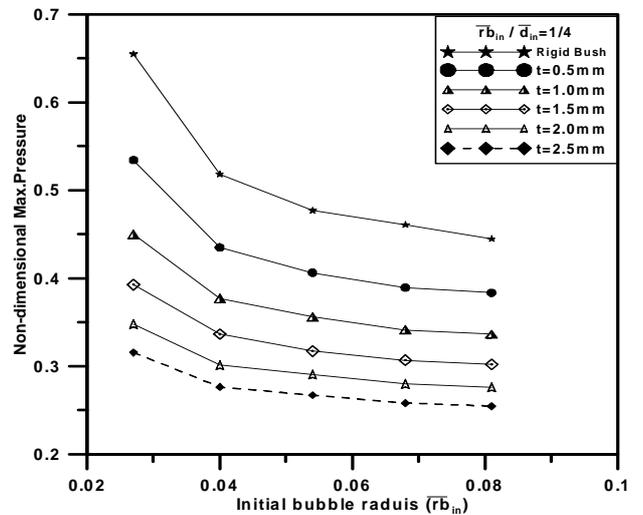


Fig. 14. Maximum oil film pressure versus initial bubble radius under different PTFE liner thickness.

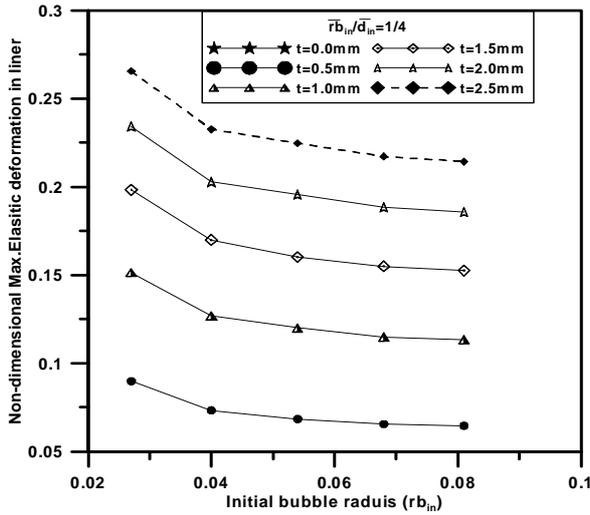


Fig. 15. Maximum elastic liner deformation versus bubble radius under different PTFE liner thickness.

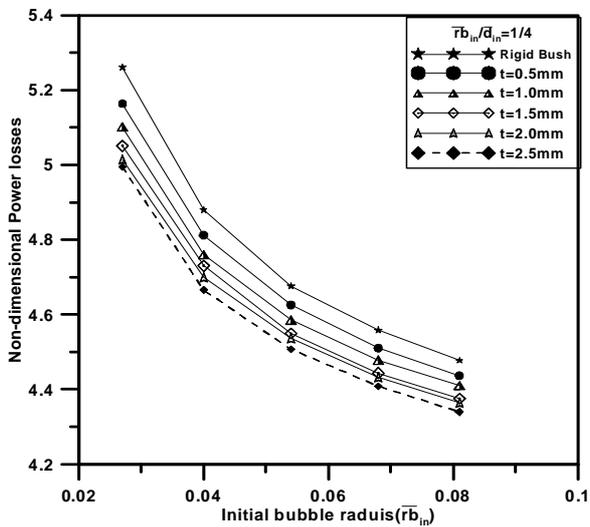


Fig. 16. Power losses versus bubble radius under different PTFE liner thickness.

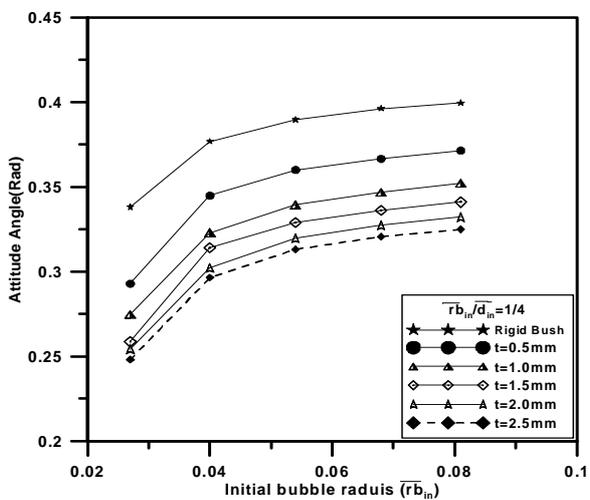


Fig. 17. Attitude angle versus bubble radius under different PTFE liner thickness.

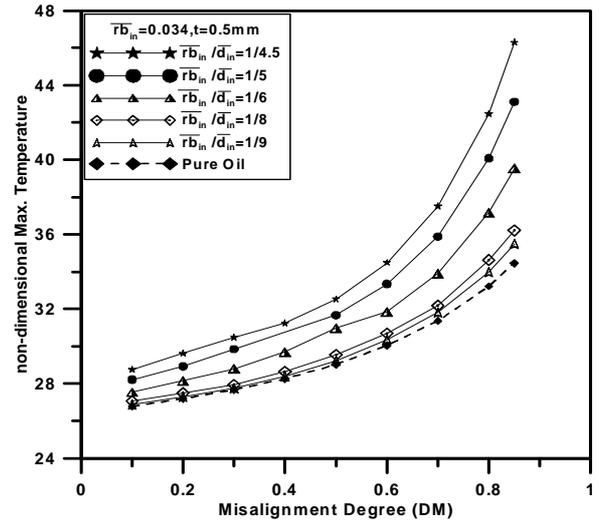


Fig. 18. Maximum oil film temperature versus misalignment degree under different aeration level.

List of Symbols

Symbol Definition

C	Radial Clearance (m)
D	Journal Diameter (m)
d _{in}	Distance between center to center bubble (m)
E	Young modulus of elasticity (N/m ²)
F _t	Friction Force (N)
\bar{F}_t	Dimensionless friction force= $\frac{F_t}{L} \left(\frac{c}{R} \right) / \mu_o U$
H _{st}	Bush Convection Heat Transfer Coefficient (W/m ² .K)
\bar{H}_{st}	Non-dimensional Bush Convection Heat Transfer Coefficient= $\frac{H_{st}R}{\rho_o C_p U C}$
L	Bearing Length (m)
L _c	Contraction Ratio
N	Journal Rotational Speed (rpm)
p	Oil Film Pressure (N/m ²)
\bar{P}	Non – dimensional Atmospheric Pressure
P _s	Oil Supply Pressure (N/m ²)
P _t	Frictional power losses (W)
\bar{P}_t	Non-dimensional frictional power losses= $\frac{P_t}{L \mu_o U^2} \left(\frac{c}{R} \right)$
Q _s	Side leakage Flow Rate (m ³ /s)
Q _{rec}	Recirculation Oil Flow Rate (m ³ /s)
\bar{R}	Gas constant for air (J/kg.K)
R _e	Local Reynolds Number
R _b	Bearing (Bush) Radius (m)
rb _{in}	Radius of inlet bubble (m)
$\bar{r}_{b_{in}}$	Non-dimensional Radius of inlet

	$\text{bubble} = \frac{r_{\text{bin}}}{c}$
T	Oil film Temperature (°C)
	Dimensionless Oil Film
\bar{T}	Temperature = $\frac{\rho C_p c^2 (T - T_{in})}{\mu_o U R}$
T_a	Ambient Temperature (°C)
T_b	Bearing Temperature (°C)
T_{in}	Inlet Oil Temperature (°C)
T_{mix}	Mixing Oil Temperature (°C)
T_s	Journal (Shaft) Temperature (°C)
T	Thickness of bearing shell (m)
	Non-dimensional Thickness of
\bar{t}	bearing shell = $\frac{t}{R}$
U	Journal (Shaft) Speed (m/s)

Greek Symbols

E	Eccentricity Ratio
μ	Lubricant Viscosity (Pa.s)
–	Dimensionless Lubricant
$\bar{\mu}$	Viscosity = $\frac{\mu}{\mu_o}$
μ_o	Inlet Lubricant Viscosity ((Pa.s)
ρ	Lubricant Density (kg/m ³)
	Non-dimensional Lubricant
$\bar{\rho}$	Density = $\frac{\rho}{\rho_o}$
ρ_o	Inlet Lubricant density
Ω	Journal Rotational Speed (rad/s)
τ_s	Local Shear Stress (N/m ²)
τ_c	Couette Surface Shear Stress (N/m ²)
δ	Elastic deformation of bearing shell
Φ	Attitude Angle (deg)
Υ	Possion's Ratio
ν	Kinematic Viscosity (m ² /s)
Σ	Surface tension of bubble (N/m)
	Non-dimensional Surface tension of
$\bar{\sigma}$	bubble = $\frac{\sigma}{\rho_o \bar{R} T c}$
$\bar{\delta}$	Non-dimensional elastic deformation

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تأثير التشوه المرن على التزيت الهيدروديناميكي الحراري للمساند المقعدية اللامتركة المحاور والمزيتة بزيت حاوي على الفقاعات الهوائية

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الخلاصة

يعالج العمل الحالي تأثير التشوه المرن على الخصائص الادائية للمساند المقعدية المبطنه بمادة مرنة (التفلون) واللامتوازية المحاور التي تعمل عند سرعات عالية عندما تزيت بزيت يحتوي على نسبة من الفقاعات الهوائية. تضمن البحث ايضا دراسة تأثير درجة الحرارة المستحثة في طبقة الزيت والنتيجة عن قص طبقات الزيت نتيجة لدوران المحور المسند. تمت معالجة المسألة موضوعة البحث باستخدام نظرية التزيت الثرموهيدروديناميكية المدلة لتأخذ بنظر الاعتبار الاضطراب في طبقة الزيت ودرجة حرارة طبقة الزيت وباستخدام موديلات رياضية مناسبة لدراسة تأثير وجود الفقاعات الهوائية على كثافة ولزوجة الزيت. تم تضمين تأثير التشوه المرن للطبقة المبطنه باستخدام موديل وينكلر. تم دراسة تأثير كثافة الزيت وسعته الحرارية المتغيرة مع درجة الحرارة على عوامل الاداء الاكثر اهمية للمسند وهي الضغط ودرجة الحرارة الاعظم للزيت وقابلية تحمل المسند للاحمال وضباع القدرة. اظهرت النتائج المستحصلة في هذا البحث زيادة في الضغط وقابلية التحميل للمسند المقعدي المزيت بزيت فقاعي ذي نسبة تهوية عالية وحجم فقاعات صغير. كما اظهرت النتائج انخفاض قيمة ضغط الزيت وقابليته على التحميل وفقدان القدرة الناتجة عن الاحتكاك عند الاخذ بنظر الاعتبار تأثير التشوه المرن للطبقة المبطنه للمسند لامتركة المحاور والذي يعمل تحت الظروف نفسها اعلاه.