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# 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 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 complaint 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 (Sneck &Vohr 1983). The basic turbulent lubrication theory has been developed by different workers (Ng 1964 ,Taylor 1970,Costantinescu 1973, and Szrie 1974). . 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 (2004). Maneshian and Nasab (2009) presented a thermohydrodynamic analysis of turbulent flow in journal bearings based on computational fluid dynamic technique. All these workers treat the lubricant as incompressible pure oil. A relatively few published studies concerning the effect of air bubbles on hydrodynamic bearing performance. 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). . Hayward (1961) 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 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. Abdel–Latife et al.(1985) 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. . Nikolajsen (1999) 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. El-Butch (2001) 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. Song and Kim( 2002) 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. Chun (2002) studied the effect of aerated oil on high speed journal bearings using classical thermohydrodynamic lubrication theory coupled with analytical models for viscosity and density of air-oil mixture in fluid film bearing. . El-Butch and El-Tyabe (2006) 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. Goodwin et al. (2007) proposed theoretical model based on Reynolds equation modified to allow for the effect of aeration on lubricant viscosity and density. The hydrodynamic bearings used in rotating machinery are usually faced with a thin layer of Babbitt material. This alloy losses 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 [Kuznetsov et.al 2011, Thomsen and Klit 2012 and Cha et.al.2013) show that using compliant liners led to improvement of journal bearing operation.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 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 Fig.(1).
To generalize the study, consider the dimensionless coordinates defined by:

$$\theta = \frac{x}{R}, \quad \bar{y} = \frac{y}{C}, \quad \bar{z} = \frac{z}{L}, \; \bar{P} = \frac{P_g C^2}{\mu_0 U R}$$

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(Chun 2004):

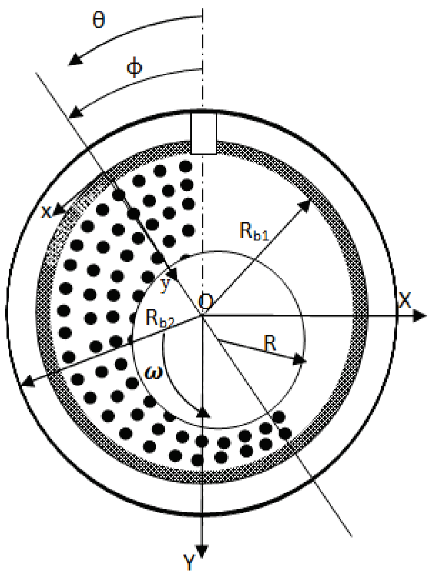



Fig.(1). Journal bearing geometry and coordinates

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

The following empirical equations proposed by (Sneck & Vohr 1983) can be used to evaluate the turbulence coefficients $G_x$ and $G_z$ for $1000 \leq Re \leq 30,000$ as follows:

$$G_x = \frac{1}{12 + 0.0136\,(R_e \cdot \bar{h}/\bar{\mu})^{0.9}} \qquad (2)$$

and

$$G_z = \frac{1}{12 + 0.0043 \cdot (\bar{h} \cdot {}^{U}/_{\nu})^{0.96}} \tag{3}$$

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 (Chun 2004) :

$$\frac{\bar{\rho}}{C_{p0}}\left\{\left(\frac{\bar{h}}{2} - \frac{\bar{h}^3}{\bar{\mu}} G_x \frac{\partial \bar{p}}{\partial \theta}\right)\frac{\partial (c_p T)}{\partial \theta} - \left(\frac{R}{L}\right)^2 \left(\frac{\bar{h}^3}{\bar{\mu}} G_z \frac{\partial \bar{p}}{\partial z}\right)\frac{\partial (c_p T)}{\partial z}\right\} = \frac{\tau_c \bar{\mu}}{\bar{h}} + \frac{\bar{h}^3}{\bar{\mu}}\left\{G_x\left(\frac{\partial \bar{p}}{\partial \theta}\right)^2 + \left(\frac{R}{L}\right)^2 G_z \left(\frac{\partial \bar{p}}{\partial z}\right)^2\right\} - (\bar{q}_{st} + \bar{q}_{bT}) \tag{4}$$

where

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

The increase in $\tau_c$ is considered by turbulent Couette shear stress factor $C_f$ as:

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

where $C_f$ can be evaluated as (Sneck & Vohr 1983):

$$C_f = 1 + 0.0012\left(\frac{U h}{\vartheta}\right)^{0.94}$$

The density and the dynamic viscosity of the aerated oil used in Reynolds equation are derived in (Nikolajsen 1999) which can be rewritten her in as:

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

where

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

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

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

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

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

a : is the constant depending on oil type, a value of 0.9070 is used through this work as in (Chun 2004).
$T_f$: 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 (Nikolajsen 1999):

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

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 (Nikolajsen 1999)

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

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(Chun 2004):

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

where

b,c : are constants depends on the lubricant type. Values of 9.8500 and 3.5180 adopted by (Chun 2004) for the constants b and c .

$T_r$ : oil film temperature in Rankin

$C_p$ is the specific heat (J/kg.$C^0$) of oil that may be correlated with Celsius Temperature $T_c$ as follow (Chun 2004)

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

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

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

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 :

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

The procedure used by Bouyer and fillon (2004) 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 .

$\varepsilon'_{max}$ is the maximum possible ε ' which can be computed from (Bouyer & Fillon 2004)

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

The elastic deformation on the working surface of compliant bearing can be evaluated as (Yang &Jeng 2001) :

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

where

$$C_d = \frac{\mu_o \omega \left(R/_c\right)^3}{E} \times \frac{(1+\acute{\upsilon})(1-2\acute{\upsilon})}{1-\acute{\upsilon}}$$

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

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

$P_g = P_g|_{supply}$ (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:

$$q_z = 0.0 \quad at \quad \bar{Z} = 0.0 \; and \; 1.0$$

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 (Chun 2004):

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

The heat transfer coefficient of the bush, adjusted for the reduction of wetting area in the cavitation region by the contraction ratio, as below (Chun 2004)

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

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

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

**3. Numerical procedure**

The Reynolds and the energy equations are differential equations governing the fluid pressure and temperature 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 and 15 nodes along the axial direction. 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:

$$error_{(P)} = \frac{\sum_{k=1}^{kk}\sum_{i=1}^{n}\left|P^{n}_{g)i,k} - P^{n-1}_{g)i,k}\right|}{\sum_{k=1}^{kk}\sum_{i=1}^{n}\left|P^{n}_{g)i,k}\right|} \le 10^{-4}$$

Once the pressure field converges, the velocities components and their derivatives are evaluated. The energy equation is then solved simultaneously with Reynolds equation , and the boundary conditions are incorporated in the iterative system. The iterations stopped when the following convergence criterion is achieved:

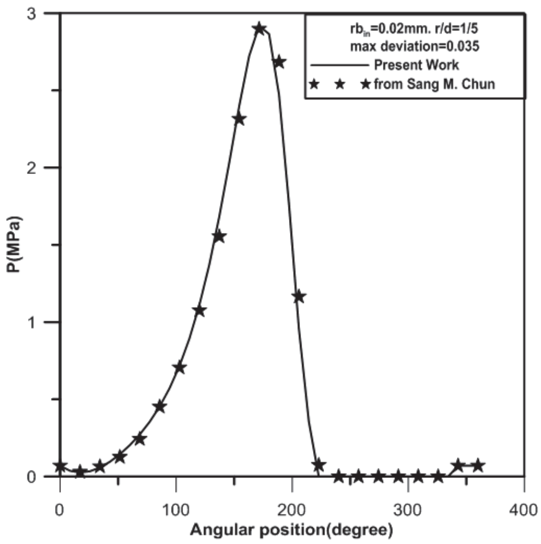
$$error_{(T)} = \frac{\sum_{k=1}^{kk}\sum_{i=1}^{n}\left|T^{n}_{i,k} - T^{n-1}_{i,k}\right|}{\sum_{k=1}^{kk}\sum_{i=1}^{n}\left|T^{n}_{i,k}\right|} \le 10^{-4}$$

**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 used in computer simulations are listed in table (1). A high speed operation considering convective condition on the bearing walls, elastic deformation of polymer liner, misalignment of the bearing shaft, the effect of bubble radius, aeration level and 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 2 to 5. The pressure and temperature distributions for a bearing lubricated with bubbly oil of bubble radius $rb_{in}$= 0.02mm and aeration level $rb_{in}/d_{in}$=1/5 have been obtained and compared with that published by(Chun 2002) as shown in figures 2 and 3. The bubbly oil dynamic viscosity obtained in the present work has been compared with that obtained by (Chun 2002) as shown in Fig.(4). 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 (Nikolajsen 1999) as shown in Fig.(5). 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 Fig.(6). 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. Fig.(7) shows the oil viscosity distribution through the bearing for different liner thickness. It can be deduced from this figure that the oil film viscosity increases as the elastic liner thickness increases. The effect of aeration level on maximum bearing pressure is presented in Fig.(8). 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. The decrease in maximum oil film pressure and load carrying capacity for a bearing with thicker elastic liner can be explained by referring to Fig.(9). The bearing power loss seems to increase when the bearing works with oil of higher aeration level (r/d) as shown in Fig.(10). 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 Fig.(11). 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 (12) 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. The effect of bubble radius ($rb_{in}$) on maximum oil film pressure can be shown in figure (13) . It is obvious that the oil film pressure decreases as the bearing lubricated with bubbly oil of larger bubble radius as shown in figure (14). This is can be attributed to the decrease of the surface tension force in this case. The liner elastic deformation increases for small ($rb_{in}$). This is can attributed to the higher oil film pressure in this case. Figure (15) 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. Figure (16) shows the attitude angle of the bearing increases for the bearing lubricated with bubbly oil of larger bubbles radius. Lower values of attitude angle have been noticed for the bearing with thicker elastic liner. A higher oil film temperature has been obtained for the bearing working at higher degree of misalignment as shown in figure (17). 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.

Table(1): Parameters of Journal bearing and oil (Chun 2002)

| Properties | Values |
|---|---|
| Inlet lubricant viscosity $\mu_o$ , 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( $\varepsilon$) | 0.65 |
| inlet lubricant temper-ature ( $T_{in}$) , C$^o$ | 40 |
| bush temperature ($T_b$), C$^o$ | 45 |
| shaft temperature ( $T_s$), C$^o$ | 45 |
| lubricant density at inlet temperature ($\rho$), 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 |
| Axial groove width | 17.1$^o$ |
| Liner elastic coefficient ($C_d$) (Kuzentsov et.al.2011) | 34.65 |
| Liner thickness(t) , mm | 0.0-2.5 |
| Modules of elasticity (E) , GPa | 0.11 |
| Passion's ratio (ύ) | 0.46 |
| Bubble surface tension($\sigma$),N/m | 0.0365 |




.Fig.(2) A comparison between the oil film pressure distribution obtained in the present work with that published by (Chun 2002)

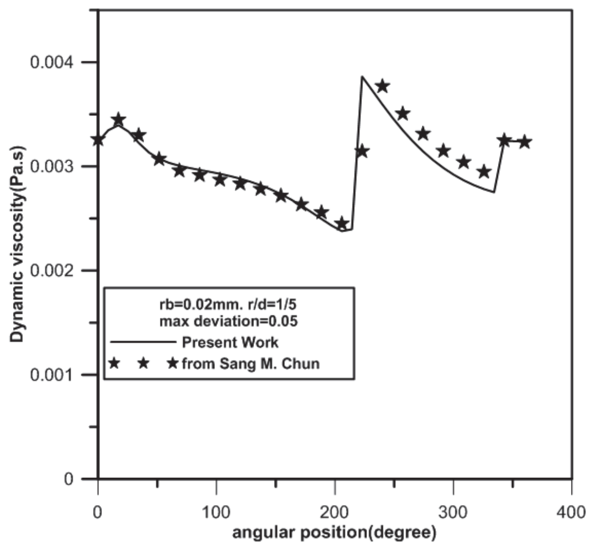



Fig.(4) A comparison between the oil viscosity distribution obtained in the present work with that published by(Chun 2002)

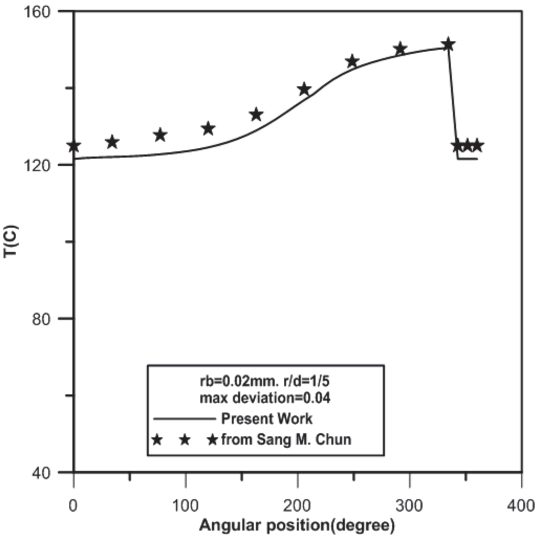



Fig.(3) Comparison between the oil film temperature distribution obtained in the present work with that published by(Chun 2002)

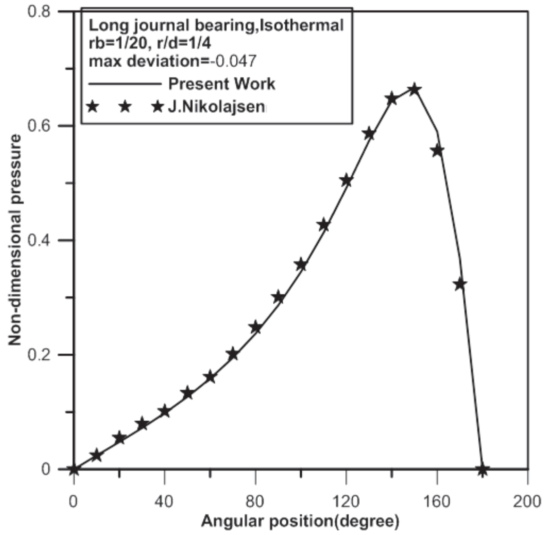



Fig.(5) Comparison between the oil film pressure distribution obtained in the present work with that published by (Nikolajsen 1999)

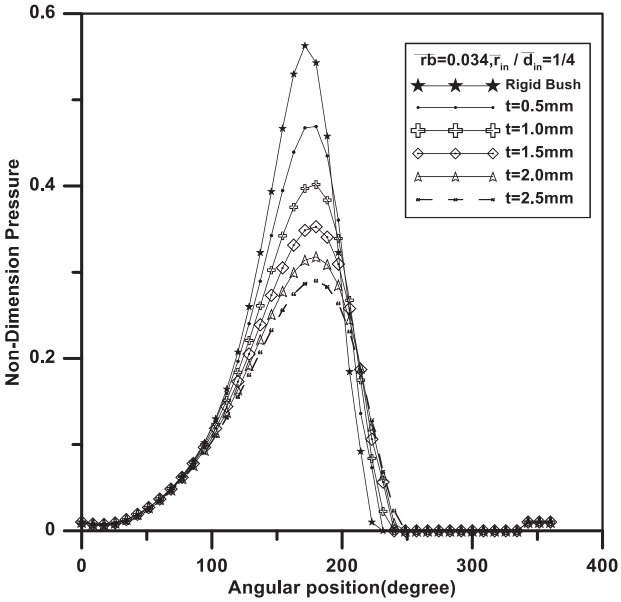



Fig.(6): Pressure distribution curves for a bearing with different Liner thickness

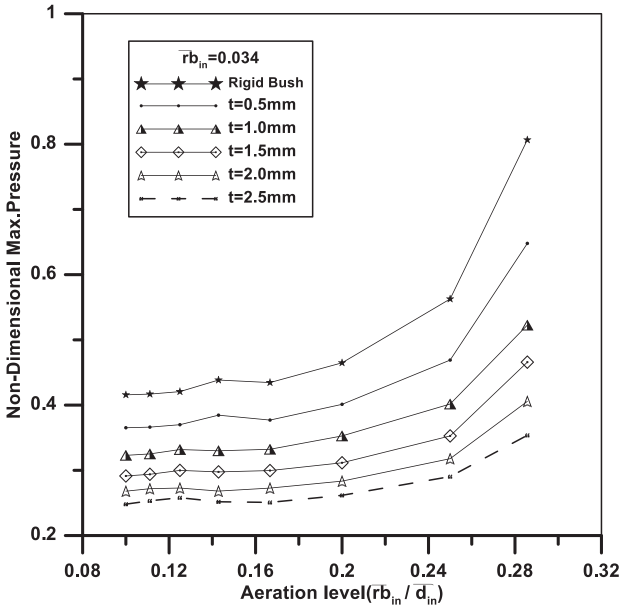



Fig.(8):Effect of bearing liner elastic deformation on the non-dimensional maximum pressure

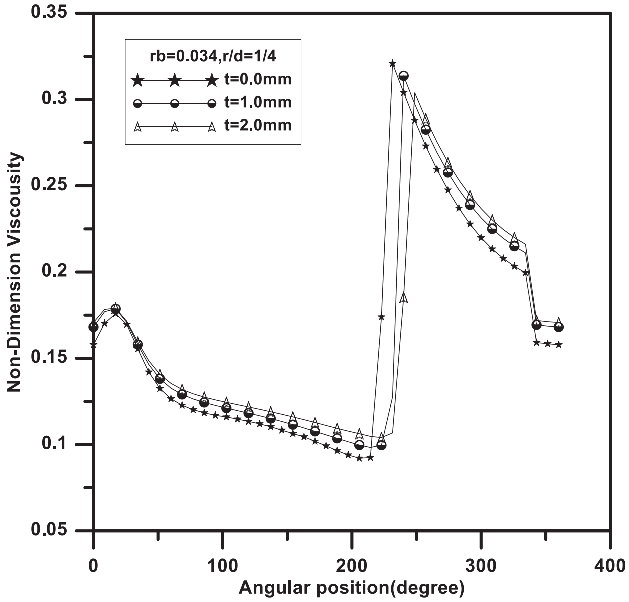



Fig.(7): Effect of elastic deformation on the bubbly oil viscosity distribution

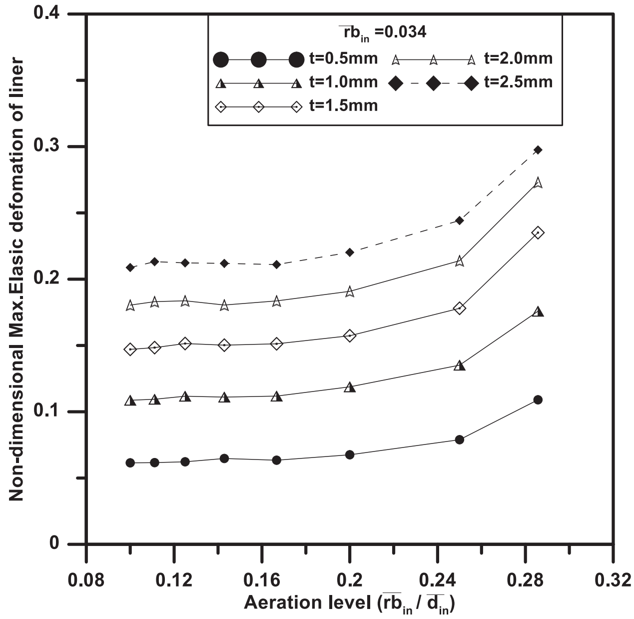



Fig.(9): Effect of aeration level on the maxi mum elastic deformation for different liner thickness

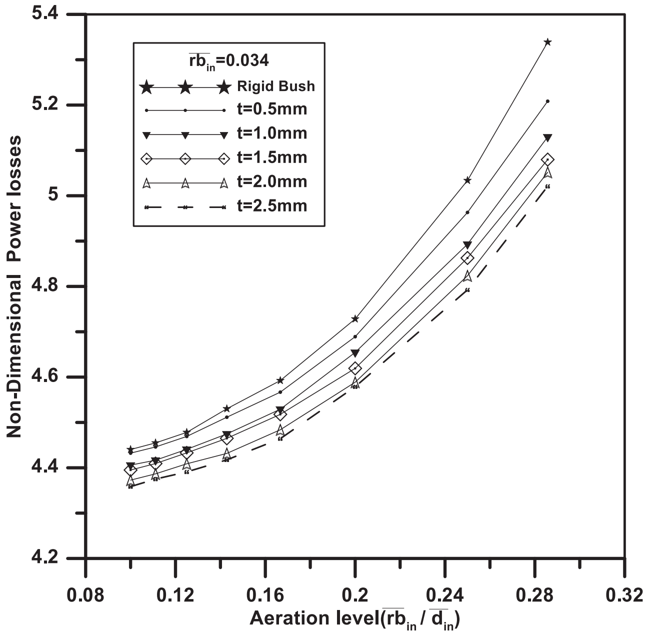



Fig.(10): Effect of aeration level non-dimensional power losses for different bearing liner thickness

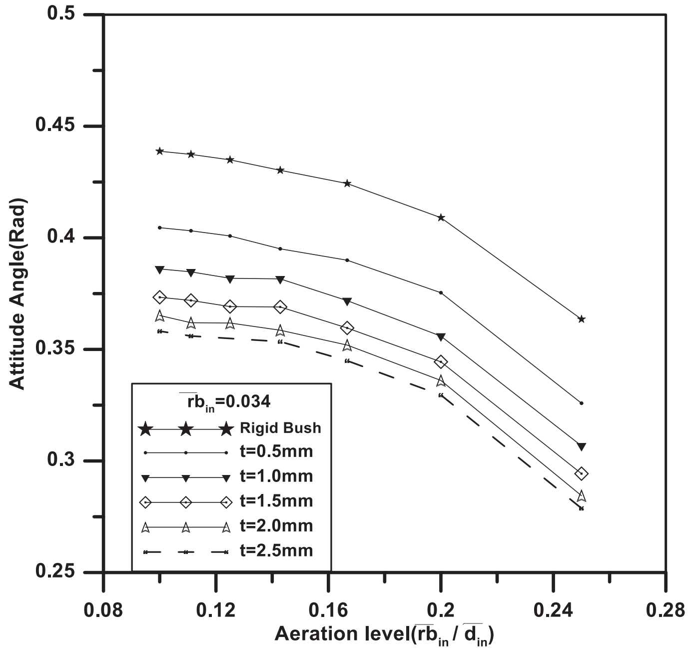



Fig.(12): Effect of aeration level on the bearing attitude angle for different bearing liner thickness

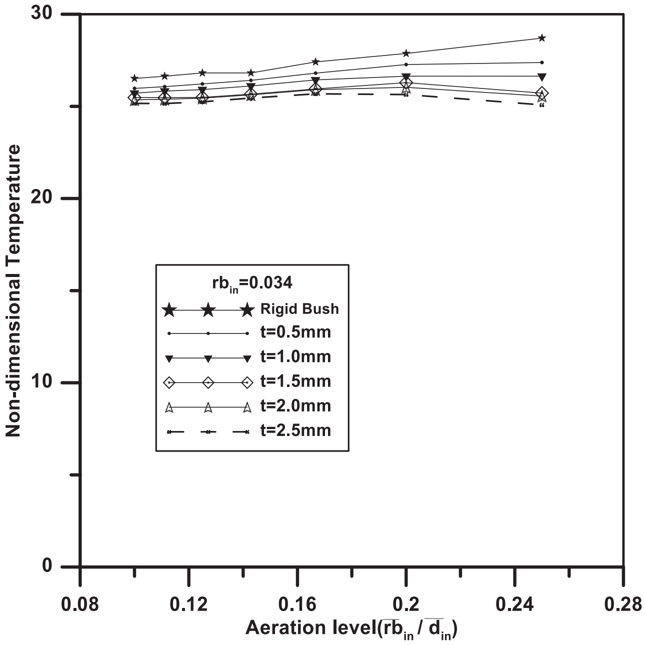



Fig.(11): Effect of aeration level non-dimensional max. temperature for different bearing liner thickness

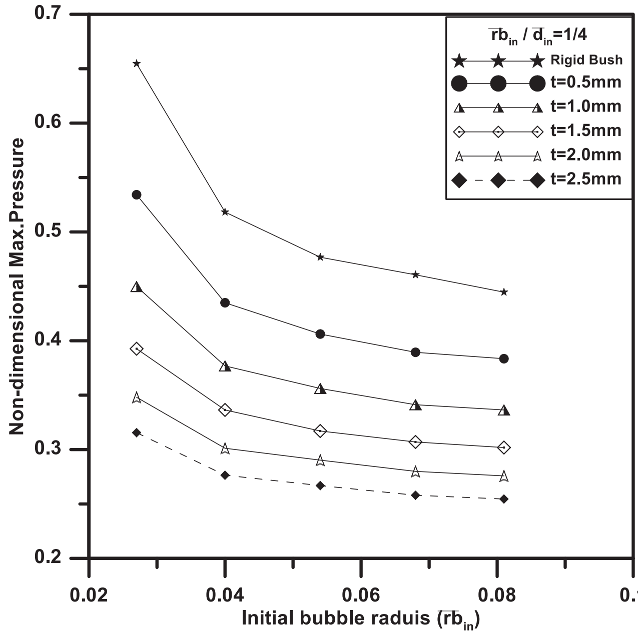



Fig.(13): Effect of initial bubble radius on the non-dimensional max. oil film pressure for different bearing liner thickness

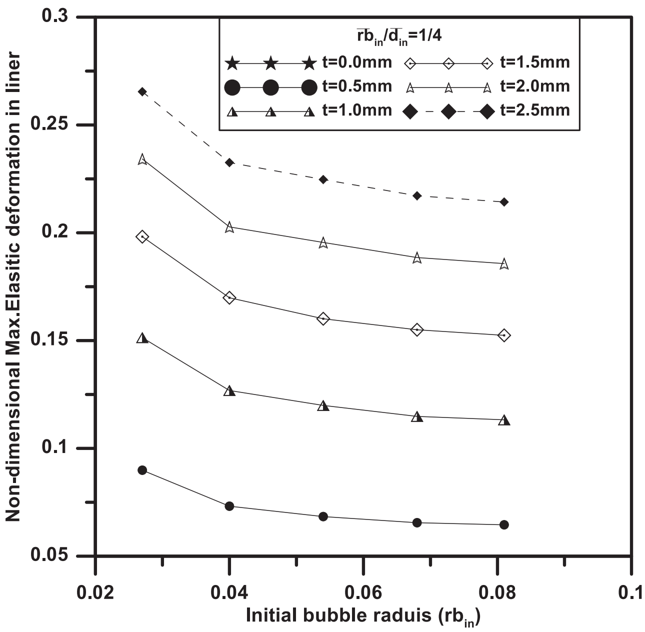



Fig.(14): Effect of initial bubble radius on the liner elastic deformation for different bearing thickness

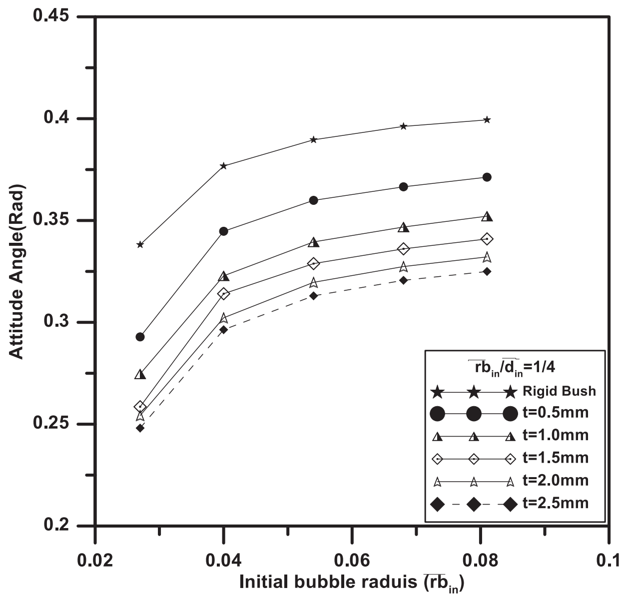



Fig.(16): Effect of initial bubble radius on the attitude angle for different bearing liner thickness

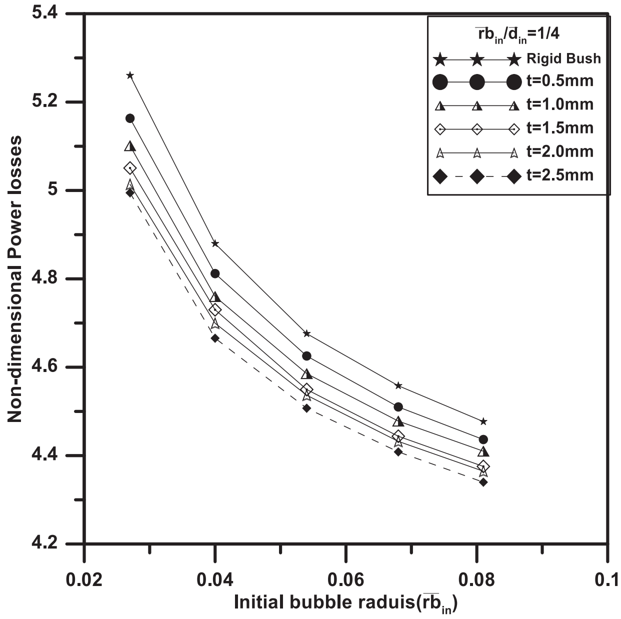



Fig.(15): Effect of initial bubble radius on the non-dimensional power losses for different bearing liner thickness

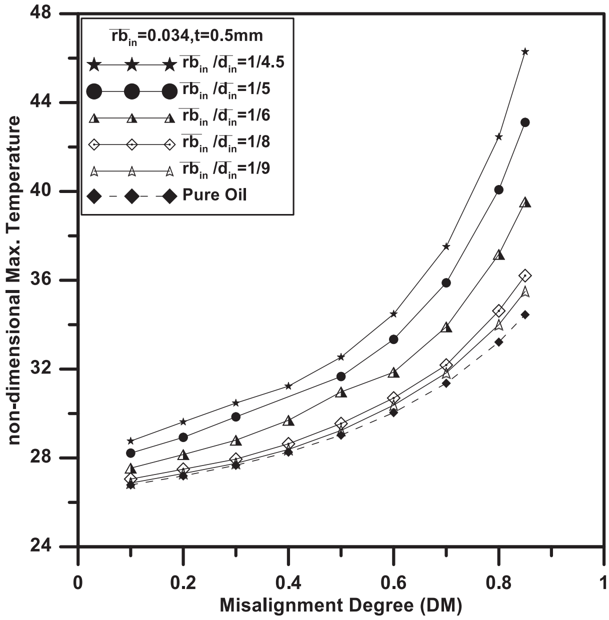



Fig.(17): Effect of journal misalignment on the non-dimensional max. temperature for different aeration level

**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.

**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 |
| $\bar{F}_t$ | 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) |
| $p_g$ | 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= $\frac{F_t}{L\mu_o U^2}\left(\frac{c}{R}\right)$ |
| $Q_s$ | Side leakage Flow Rate (m$^3$/s) |
| $Q_{rec}$ | Recirculation 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 bubble=$\frac{rb_{in}}{c}$ |
| T | Oil film Temperature (°C) |
| $\bar{T}$ | Dimensionless Oil Film Temperature $=\frac{\rho C_p c^2 (T-T_{in})}{\mu_o UR}$ |
| $T_a$ | 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) |
| $\bar{t}$ | Non-dimensional Thickness of bearing shell=$\frac{t}{R}$ |
| U | Journal (Shaft) Speed (m/s) |

**Greek Symbols**

| | |
|---|---|
| $\varepsilon$ | Eccentricity Ratio |
| $\mu$ | Lubricant Viscosity (Pa.s) |
| $\bar{\mu}$ | Dimensionless Lubricant Viscosity= $\frac{\mu}{\mu_o}$ |
| $\mu o$ | Inlet Lubricant Viscosity ((Pa.s) |
| $\rho$ | Lubricant Density (kg/m$^3$) |
| $\bar{\rho}$ | Non-dimensional Lubricant Density= $\frac{\rho}{\rho_o}$ |
| $\rho_o$ | Inlet Lubricant density |
| $\Omega$ | Journal Rotational Speed (rad/s) |
| $\tau_s$ | Local Shear Stress (N/m$^2$) |
| $\tau_c$ | Couette Surface Shear Stress (N/m$^2$) |
| $\delta$ | Elastic deformation of bearing shell |
| $\Phi$ | Attitude Angle (deg) |
| $\bar{\nu}$ | Possion's ratio |
| $\nu$ | Kinematic Viscosity (m$^2$/s) |
| $\sigma$ | Surface tension of bubble (N/m) |
| $\bar{\sigma}$ | Non-dimensional Surface tension of bubble= $\frac{\sigma}{\rho_o \bar{R} Tc}$ |
| $\bar{\delta}$ | Non-dimensional elastic deformation |

**References**

A.M.A.El-Butch and N. El-Tayeb "surface roughness effects on thermo-hydrodynamic lubrication of journal bearings lubricated with bubbly oil" , Lubrication science , vol.18 ,pp49-61, 2006.

A.M.A. EL-Butch " The Effect of Bubbly Oil on the THD Lubrication of Tilting-pad Journal Bearings Subjected to Rotating Unbalance Load", Meccanica vol. 36, PP 717–729, 2001.

B. Maneshian and S.A. Gandjalikhan Nassab, "thermodynamic analysis of turbulent flow in journal bearings running under different steady conditions" , IMechE Vol.223 Part J: engineering tribology , 2009.

Constantinescu, V.N., "Basic relationships in turbulent lubrication and their extension to include thermal effects", J. Lubr. Technol., Trans. ASME, Vol.95, 1973.

Evgeny Kuznetsov, Sergei Glavatskih, Michel Fillon “ THD analysis of compliant journal bearings considering liner deformation”, Tribology International, 44,(2011)1629-1641.

E.H. Smith "The Influence of surface Tension on bearings lubricated with bubbly liquids", journal of lubrication technology, Vo.100.

Hayward "the viscosity of bubbly oil" national engineering laboratory , fluid Report No. 99 , glasow, 1961.

H. J. Sneck and J. H. Vohr, "Hand book of lubrication" , Vol. II,1983 PP 69-90.

Jorgen L. Nikolajsen "viscosity and density models for aerated oil in fluid – film bearing", tribology transactions vol.42, pp186-191,1999.

J. Bouyer and M. Fillon, "thermodynamic behavior of misaligned plain journal bearing : theoretical and experimental approaches", tribology transaction , vol. 47 ,pp594-604 , 2004.

Kim Thomsen and Peder Klit "Improvement of journal bearing operational heavy misalignment using bearingflexibility and compliant liners" IMechE Part J:J Engineering Tribology vol.226,pp 651–660,2012.

Lotfie Ahmed Abdel-Latif, et al,"thermodynamic analysis of thrust-bearing with circular pads running on bubbly oil" , transactions of the ASME , vol. 107, October 1985.

Mj Goodwin,D.Dong, et al " Theoretical and experimental investigation of the effect of oil aeration on the load-carrying capacity of a hydrodynamic journal bearing", IMechE Vol. 221 Part J: J. Engineering Tribology,July 2007.

Matthew Cha, Patrik Isaksson b, SergeiGla vatskih “ Influence of pad compliance on nonlinear dynamic characteristics of tilting pad journal bearings” Tribology International 57 (2013) 46–53

Ng,C. W., “Fluid Dynamic Foundation of Turbulent Lubrication Theory,” STLE Transactions, Vol. 7, pp. 311-321., 1964.

Safar and Szeri , "Thermodynamic Lubrication in Laminar and Turbulent Regimes" , Transactions of the ASME, January 1974, PP48-57.

Sang Myung Chun "thermohydrodynamic lubrication analysis of high – speed journal bearing considering variable density and variable specific heat" , tribology international, Vol.37,pp405-413, 2004

Song Chop and Kyung Woong Kim "Analysis of bubbly lubrication in journal bearing", JSME international journal , vol. 45, No. 3 ,PP802-808,2002.

Sang Myung chun "A parametric study on bubbly lubrication oh high-speed journal bearings" , tribology international, Vol.37,pp1-13, 2002.

Taylor CM. ," turbulent lubrication theory applied to fluid film bearing design" , Proc. Inst. Mech. Engrs., part 3L , vol.184 , 1970.

Yung-Kuang Yang, Ming-Chang Jeng , "analysis of thermal effects on the misaligned hydrolic servo- cylinder" tribology international vo l.34, pp95-106, 2001