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Effect of Bearing Elastic Deformation on the Turbulent Thermohydrodynamic Lubrication of Misaligned Plain Journal Bearings

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Abstract:

This paper investigates the effect of bearing shell elastic deformation on the steady state behavior of misaligned journal bearings. Performance trends of elastic and rigid bearings with different degree of misalignment (D_m) have been compared. Bearing shell elastic deformation, thermal effects, pressure and temperature variation for a bearing working under turbulent lubrication have been taken into consideration. It is found that misaligned bearings working at higher misalignment coefficients have higher load carrying capacity since the oil film thickness approaches to zero in this case. Considering the effect of bearing shell elastic deformation with the effect of oil film temperature causes lower values of bearing load. Some of the results obtained in this paper have been compared with that published by other workers and found to be in a good agreement.

Keywords: Turbulent flow, bearing misalignment, bearing shell Elastic deformation, Thermal effect.

Introduction

Journal Bearings consists of two cylinders rotating relative to each other. The main purpose of the journal bearing is to support the rotating machinery by providing sufficient lubrication to separate the moving parts and to minimize the friction due to rotation. Journal bearings are extensively used when rotating machinery operate at high speeds. Under certain conditions, bearings of large dimensions (large clearance) or that which work with fluid of low kinematic viscosity, turbulence may occur at moderate velocities currently used in the design of hydrodynamic bearing. Theoretical analysis of turbulent lubrication dated back to 1965when Constantinescu [1], developed a theory of turbulent lubrication. Coefficients for flow characteristics of a turbulent fluid had been introduced to modify the Reynolds equation. Number of papers appeared dealing the effect of turbulent lubrication on the performance of journal bearings [2-8]. Constantinscu and Galetues [2] utilized the possibilities of improving the accuracy of the evaluation of inertia forces in laminar and turbulent films. Robert and Hinton [3] applied the short plain journal bearing assumption to experimentally investigate the pressure distribution in a super laminar journal bearings. Capone et al. [4] examined the influence of turbulence on the shaft bearing in non-laminar lubrication regime. Gethin and Medwell [5] presented a description of the design of a high speed journal bearing test apparatus and some results recorded from a cylindrical bore bush fed by two axial grooves. Hashimoto and Wada [6] utilized a method to solve turbulent lubrication problems of sector-shaped, tilting-pad thrust bearings incorporating thermal and elastic deformation of the pad. Wilcock and Pinkus [7] calculated the dynamic coefficients of fluid film journal bearings under turbulence and viscosity variation conditions. Safar et al. [8] presented an analysis of the performance characteristics for a misaligned full journal bearing operating in turbulent regime. Kosasih and Tieu [9] modeled the Reynolds stress using the mixing length expression which is able to account for the effect of local shear stress gradient. The effect of mixing flow in a high speed journal bearing has been studied by Chun and Ha[10].The effect of viscosity variation on the performance of journal bearing has been appeared in many papers [11-15]. Ferron et al. [11] studied both theoretical and experimental thermo-hydrodynamic problem of a finite length journal bearing. Medwell and Gethin [12] deal with the evaluation of pressure and temperature fields which are generated in thin fluid films of varying thickness. Khonsari and Wang [13] analyzed the thermo-hydrodynamic of a full journal bearing with realistic model for the bushing made of two distinct layers. Bouard et al. [14] compared the results of three turbulent models with the experimental data. It had been shown that under high rotational speeds, thermal effects are important. Banwait and Chandrawat [15] solved a three dimensional energy and heat conduction equations for fluid and bush temperature and a one dimensional equation for journal temperature. Tanaka and Hatakenaka [16] derived a three dim-ensional turbulent thermohydrodynamic lubrication model based on isothermal turbulent lubrication model. Ghoneam and Strzelecki [17] stated that the operation of plain circular journal bearing at high speed is restricted by excessive temperature that is generated in the oil film and the loss of stability. The effect of misalignment problem in journal bearings has been studied extensively. Misalignment problems reported in the literature employ either misalignment angle or prescribed misalignment torque [18-21]. The present work is an attempt to study the combined effect of bearing misalignment and elastic deformation of the bearing shell on thermo-hydrodynamic perform-ance of journal bearing working under turbulent lubrication.

Theory

Figure (1) shows schematically a cylindrical journal bearing with longitudinal oil groove. The bearing undergoing combined eccent-ricity and misalignment. An x, y, z system frame is fixed to the non-rotating sleeve. The journal rotates at a constant angular velocity ($\boldsymbol{\omega}$) about the z-axis.

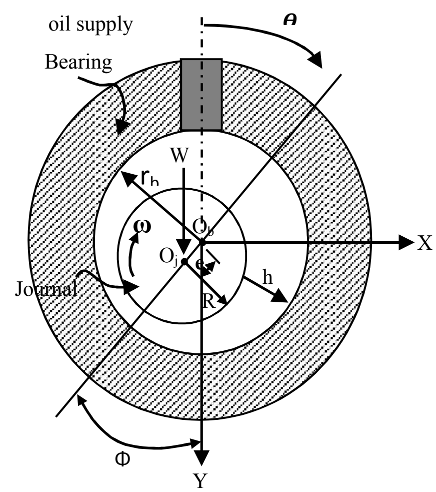

Figure (1): Schematic diagram of journal

Pressure distribution in the clearance space between journal and bearing is governed by the Reynolds equation. The Reynolds equation is modified to implement the effect of turbulent flow and the oil film temperature. [14]

$$\frac{\partial}{\partial\theta}(\bar{h}^3\bar{G}_X\frac{\partial\bar{p}}{\partial\theta})+\eta^2\frac{\partial}{\partial Z}(\bar{h}^3\bar{G}_Z\frac{\partial\bar{p}}{\partial Z})=\frac{\partial(\bar{h}\bar{G})}{\partial\theta}$$
……. (1)

Where

$$\eta = \frac{R}{L}$$

$$\overline{G}_x = \frac{12}{k_x}\int_0^1 \frac{\overline{y}}{\overline{\mu}(\overline{y})}\left[\overline{y} - \frac{\overline{F}_o}{\overline{F}_1}\right]d\overline{y}$$

$$\overline{F}_o = \int_0^1 \frac{d\overline{y}}{\overline{\mu}(\overline{y})}$$

$$\overline{F}_1 = \int_0^1 \frac{\overline{y}.d\overline{y}}{\overline{\mu}(\overline{y})}$$

As in Bouard et al. [14], Constantinescu turbulence model has been utilized through this work. Turbulent coefficients K_x and K_z can be evaluated as:

$$k_x = 12 + 0.0136 R_{eh}^{0.9} \qquad \text{......... (3)}$$

$$k_Z = 12 + 0.00432 . R_{eh}^{0.96}$$

The oil film thickness can be expressed as:

$$\overline{G}_z = \frac{12}{k_z}\int_0^1 \frac{\overline{y}}{\overline{\mu}(\overline{y})}\left[\overline{y} - \frac{\overline{F}_o}{\overline{F}_1}\right]d\overline{y}$$

$$\overline{G} = 1 - \frac{\overline{F}_o}{\overline{F}_1} \qquad \text{………….. (2)}$$

$$h = \overline{h}(\theta, Z) + \delta \qquad \text{…………. (4)}$$

The film thickness equation given by Jang and Khonsari [22], for misaligned journal bearing is used her in; see figure (2)

$$\overline{h} = 1 + \epsilon_o \, Cos\,(\theta - \phi_o) + \epsilon' \,(\overline{Z} - 0.5).\, Cos\,(\theta - \alpha - \phi_o) \qquad \text{……….. (5)}$$

$$\epsilon'_{max} = 2*[(1 - \epsilon_o^2 \sin^2 \alpha)^{1/2} - \epsilon_o |cos\alpha|].. \text{ (6)}$$

While ϵ' can be expressed in terms of ϵ'_{max} and the parameter which represents the degree of the misalignment (D_m),as follows:

$$\epsilon' = D_m * \epsilon'_{max} \qquad \text{……….. (7)}$$

where D_m takes values of 0 to 1.

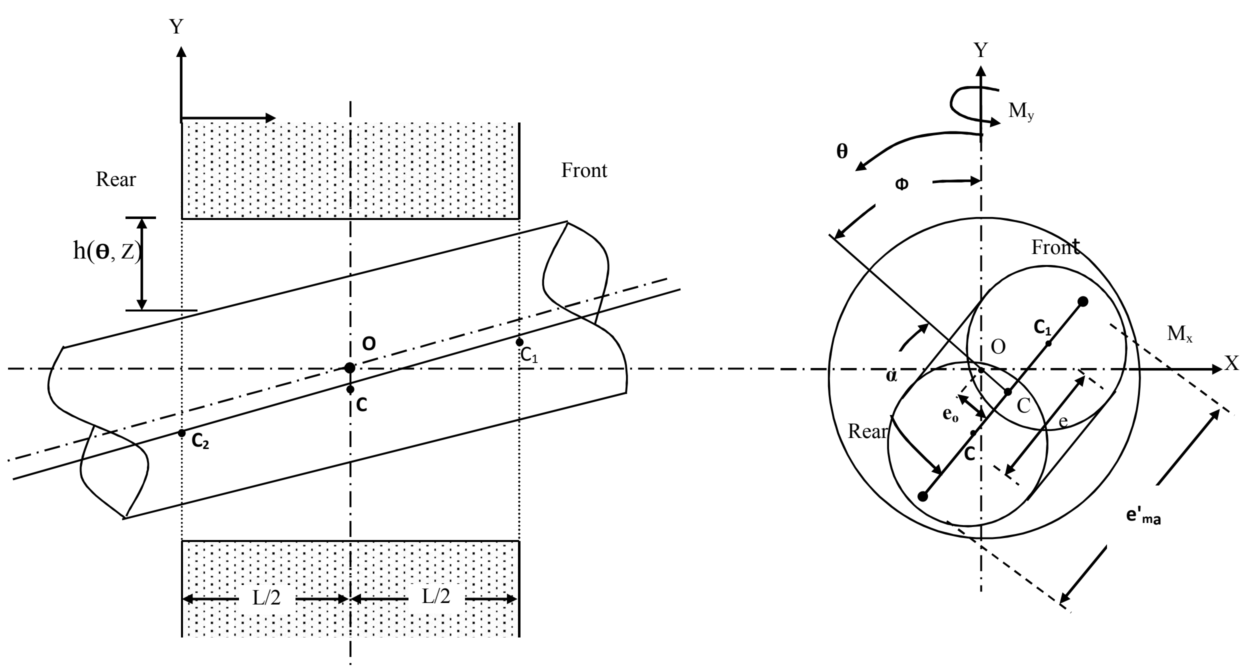

Figure (2): Scheme of a misaligned journal bearing

δ: Bearing shell elastic deformation which can be evaluated as in [23].

$$\delta = [\frac{\mu.\omega.R.r_b.t.(1-\nu^2)}{C^3.E}].P = C_o P \quad ...(8)$$

Pressure boundary condition

The following boundary conditions are used to solve the Reynolds equation.

(a). Pressure at the edges of bearing is assumed to be atmospheric

$$P(\theta,0)=0 \quad .....(9)$$

$$P(\theta,1)=0$$

(b). the well-known Reynolds boundary conditions have been adopted through this work.

$$P(\theta \geq \theta_c, Z) = (\frac{\partial P}{\partial \theta})\Big|_{\theta=\theta c} = 0 \quad .....(10)$$

Where θ_c is the angle at which oil film rupture occurs, i.e. the flow becomes liquid gas.

(c). the inlet supply pressure P_s is constant and assumed to be atmospheric.

Energy equation:

The temperature induced in oil film due to shearing effect can be determined by solving the energy equation. Considering steady state lubrication, an incompressible fluid and the hypothesis that the film thickness is very small compared to other dimensions of the journal bearing. This equation can be written as [14].

$$\bar{u}\frac{\partial \bar{T}}{\partial \theta} + \left\{\bar{v} - \frac{1}{P_e}\frac{1}{\bar{h}^2}\frac{\partial}{\partial \bar{y}}(1+\bar{\mu}\frac{\in_m}{v}.\frac{P_r}{P_r^t})\right\}\frac{\partial \bar{T}}{\partial \bar{y}} =$$
$$\frac{1}{P_e}\frac{1}{\bar{h}^2}(1+\bar{\mu}\frac{\in_m}{v}.\frac{P_r}{P_r^t})\frac{\partial^2 \bar{T}}{\partial \bar{y}^2} + \frac{N_e}{Re}.\frac{\bar{\mu}}{\bar{h}^2}$$
$$(1+\frac{\in_m}{v})(\frac{\partial \bar{u}}{\partial \bar{y}})^2 \quad ....(11)$$

Where

$\in_m$: Eddy viscosity for momentum.

The local turbulent viscosity $\frac{\in_m}{v}$ can be evaluated as follows [24]:

For $0 \leq y \leq h/2$

$$\frac{\in_m}{v} = K\left[\frac{y}{v}\sqrt{\frac{\tau}{\rho}} + \delta_l.\tanh(\frac{y}{v.\delta}\sqrt{\frac{\tau}{\rho}})\right]$$

For $h/2 \leq y \leq h$ the expression becomes

... (12)

$$\frac{\in_m}{v} = K\left[\frac{(h-y)}{v}\sqrt{\frac{\tau}{\rho}} + \delta_l.\tanh(\frac{(h-y)}{v.\delta}\sqrt{\frac{\tau}{\rho}})\right]$$

The values of K and δ used in ref [24] has been adopted in the present work

K=0.4, δ_l=12.075

$$\tau = \sqrt{\tau_{xy}^2 + \tau_{zy}^2}$$

$$\tau_{xy} = \mu.(1+\frac{\in_m}{v}).\frac{\partial u}{\partial y} \quad ......(13)$$

$$\tau_{zy} = \mu(1+\frac{\in_m}{v}).\frac{\partial w}{\partial y}$$

In these equations,, is the lubricant dynamic viscosity considered as a function of temperature, defined as in [25].

$$\mu = \mu_o . e^{\beta_o (T - T_o)} \quad ......(14)$$

It can be rewritten in dimensionless form as follows

$$\bar{\mu} = \frac{\mu}{\mu_o} = e^{\beta_o (\bar{T} - T_o)} \quad ......(15)$$

Where

$$\bar{T} = \frac{T - T_o}{T_r} \quad ......(16)$$

Since T_r is reference temperature which can be defined as:

$$T_r = \frac{R^2 . \mu_o . \omega}{\rho . C_p . C^2} \quad .....(17)$$

$$P_e\text{: Peclet number} = \frac{\rho . C_p . \omega . C^2}{k} \quad .....(18)$$

$$P_r\text{:Prandtl number} = \frac{P_e}{R_e} = \frac{\mu_o . C_p}{k} \quad ....(19)$$

$$N_e\text{: Eckert number} = \frac{R^2 . \omega^2}{C_p . T_o} \quad ....(20)$$

$$P_r^t \text{ : Turbulent Prandtl} = \frac{\in m}{\in H} \quad ....(21)$$

The turbulent Prandtl number is assumed to be constant and equal to 1.0 as in [15].

Heat conduction equation

The Fourier heat conduction equation given below is solved to obtain the temperature distribution in the bush with assumed thermal properties [4].

$$\frac{\partial^2 \bar{T}_b}{\partial \bar{r}^2} + \frac{1}{\bar{r}} . \frac{\partial \bar{T}_b}{\partial \bar{r}} + \frac{1}{\bar{r}^2} . \frac{\partial^2 \bar{T}_b}{\partial \theta^2} = 0 \quad ....(22)$$

Where

$\bar{T}_b = \frac{T_b}{T_o}$, T_b is bearing bush temperature and T_o is oil inlet temperature

Temperature Boundary Conditions

The following boundary conditions have been adopted to solve the energy equation.

(a). The oil temperatures at the oil – bush and at the oil shaft interfaces are matching temperatures.

$T=T_b$ at $y=0$ and $T=T_s$ at $y=h$

Where T_b is bearings surface temperature and T_{sh} is shaft surface temperature

(b). the temperature boundary conditions at the inlet groove are given as:

$$T\big|_{\theta = -\phi} = T_{mix}$$

Where T_{mix} is mixing temperature of the re-circulating oil which can be evaluated as [10].

$$T_{mix} = \frac{Q_{in} \cdot T_o + L_c Q_{rec} T_{rec}}{Q_{in} + L_c Q_{rec}} \quad .....(23)$$

Where

$$Q_{rec} = L \int_0^h u . dy \quad ......(24)$$

It can be rearranged as follows:

$$Q_{rec} = L.U.C.\int_0^1 \bar{u}.\bar{h}.d\bar{y} \qquad \text{...... (25)}$$

L_c: is the contraction ratio of the oil film which can be defined as:

$$L_c(\theta) = \frac{\int_{-(L/2)}^{(L/2)} . \int_0^{h(\theta^*,z)} u(\theta^*, Z).dy.dz}{\int_{-(L/2)}^{(L/2)} . \int_0^{h(\theta,z)} u(\theta, Z).dy.dz} \qquad \text{..... (26)}$$

Where θ^*: is the angle of the beginning of the cavitation region.

(c). At the oil bush interface, the heat flux continuity condition can be utilized in non-dimensional parameter as [11].

$$\left.\frac{\partial \bar{T}}{\partial \bar{r}}\right|_{\bar{r}=1} = -\frac{K_{oil}}{K_b} . \frac{r_{bin}}{c} . \frac{1}{\bar{h}_T} . \left.\frac{\partial \bar{T}}{\partial \bar{y}}\right|_{\bar{y}=0} \qquad \text{....... (27)}$$

Where:

$$\bar{r} = \frac{r_{bo}}{r_{bin}}$$

K_{oil}: the oil thermal conductivity which is constant in the active zone (non-cavitated zone)

(d). for the outer circumference of the bush, ($r = r_{bo}$), the free convection hypothesis using non-dimensional parameters gives [11].

$$\left.\frac{\partial \bar{T}}{\partial \bar{r}}\right|_{\bar{r}=\bar{r}_{bo}} = -\frac{h_{conv.}}{K_b} . r_{bin} . (\bar{T}_{bo} - \bar{T}_a) \qquad \text{...... (28)}$$

Bearing Characteristics

As the oil film pressure of the bearing is obtained, the most important bearing performance characteristics can be evaluated. The load carrying capacity along the line of centers and its perpendicular can be found out from.

$$\bar{W}_r = \int_0^1 \int_0^{2\pi} \bar{P}Cos(\theta).d\theta.d\bar{Z} \qquad \text{......... (29)}$$

$$\bar{W}_t = \int_0^1 \int_0^{2\pi} \bar{P}Sin(\theta).d\theta.d\bar{Z} \qquad \text{......... (30)}$$

Tthe total load carrying capacity can be evaluated as:

$$\bar{W} = \sqrt{\bar{W}^2_r + \bar{W}^2_t} \qquad \text{......... (31)}$$

The attitude angle (ϕ) between the load line and the line of centers can be expressed as:

$$\phi = \tan^{-1}\left(\frac{\bar{W}_t}{\bar{W}_r}\right) \qquad \text{....... (32)}$$

Sommerfeld Number can be mathematically represented as:

$$S_m = \frac{\mu.\omega.L.R}{\pi W}\left(\frac{R}{C}\right)^2 \qquad \text{........... (33)}$$

For a given pressure distribution, the pressure gradient $\frac{\partial P}{\partial \theta}$ can be evaluated numerically and hence the viscous shear stress (τ_v) can be calculated as:

$$\tau_v = \mu.\left(\frac{U}{h}\right).\tau_c + \left(\frac{h}{2}\right).\frac{\partial P}{\partial X} \qquad \text{........... (34)}$$

The value of τ_c can be estimated according to Constantinescu's model [26] as:

$$\tau_c = 1 + 0.0023.(Re_h)^{0.855} \quad \text{.......... (35)}$$

Integration of such shear stress along the journal circumferences gives directly the frictional drag as follows:

$$F = \int_0^L \int_0^{\theta_c} \tau_v .d\theta.dz \quad \text{....... (36)}$$

The coefficient of friction can then be evaluated as:

$$f = \frac{F}{W}.(r / c) \quad \text{........ (37)}$$

The amount of flow that needs to be replenished to maintain a full film can be expressed as :

$$Q = \frac{2}{\mu} \int_0^{\theta_c} \frac{h^3}{k_z} . \frac{\partial P}{\partial Z}\bigg|_{Z=L} .d\theta \quad \text{.......... (38)}$$

The journal bearing configuration in figure (2) requires the application of external moments for steady - state operation. The required misalignment moments for steady – state operation are calculated directly from the pressure distribution. The moments about axes shown in figure (2) are defined in dimensionless form as [22].

$$\overline{M}_x = \int_0^1 \int_0^{2\pi} \overline{P}.(\overline{Z} - \frac{1}{2}).Cos(\theta).d\theta.d\overline{Z} \quad \text{..... (39)}$$

$$\overline{M}_y = \int_0^1 \int_0^{2\pi} \overline{P}.(\overline{Z} - \frac{1}{2}).Sin(\theta).d\theta.d\overline{Z} \quad \text{..... (40)}$$

The total misalignment moment in dimensionless form can be evaluated as:

$$\overline{M} = \sqrt{\overline{M}^2_x + \overline{M}^2_y} \quad \text{....... (41)}$$

Computational technique:

The governing equations, namely, Reynolds equation modified to include the effect of oil film turbulence, the energy equation, the heat conduction equation and equation which relates the oil viscosity to the oil temperature are solved for pressure and temperature satisfying the pressure and temperature boundary conditions. A computer program was developed to determine the performance characteristics of the bearing. The finite difference approximation of the governing equations was developed and the solution of the resulting set of simultaneous equations was obtained using direct iterative procedure with successive over relaxation. Cavitation is modeled by setting all calculated negative pressure to zero. The iterations are repeated until the pressure and the temperature satisfy the following convergence criteria:

$$\mathbf{error}_{(\bar{P})} = \frac{\sum\sum\left|\bar{P}^{n}_{(i,j)} - \bar{P}^{n-1}_{(i,j)}\right|}{\sum\sum\left|\bar{P}^{n}_{(i,j)}\right|} \le 10^{-4} \quad ,$$

$$\mathbf{error}_{(\bar{T})} = \frac{\sum\sum\left|\bar{T}^{n}_{(i,j)} - \bar{T}^{n-1}_{(i,j)}\right|}{\sum\sum\left|\bar{T}^{n}_{(i,j)}\right|} \le 10^{-3}$$

Where (n) and (n-1) represent the present and the previous iterations respectively . On achieving the pressure convergence, the attitude angle is evaluated as in equation (32) which was compared with assumed value until the convergence obtained when the difference between two successive values become less than one degree. Static performance characteristics such as load carrying capacity, friction force, lubricant flow rate, bearing misalignment moments, and bearing maximum temperature are evaluated after achieving the convergence of pressure, temperature and attitude angle. The flow chart of the computer program can be shown in figure (3).

Validation

To validate the present numerical scheme, some of the results obtained in the present work have been compared with that published by other workers. Figure (4) shows a comparison for the results of Sommerfeled number for different eccentricity ratios obtained by the present work with that obtained by Capone et al. [4]. The nondimensional friction coefficient for different degrees of misalignment obtained through this work has been compared to that obtained by Chun and Ha [10] as shown in figure (5). Figure (6) shows a comparison between the nondimensional side leakage flow obtained in this work with that obtained by Constantinescu [2]. The maximum deviation between the results is 1.25%. The temperature distribution for the oil film in laminar flow condition (k_x , k_z=12) with speed (S=4000 rpm) has been obtained in this work and compared with the experimental results obtained by Ferron et al.[11] as shown in figure (7). The maximum deviation between the results has been calculated and found to be (1.85%). It is clear from the above comparisons that the results obtained through this work are in a reasonable agreement with that obtained by other workers.

Results and discussion

Figure (8) illustrates the effect of journal misalignment on the temperature distribution of the bearing oil film. A bearing with elastic shell of elastic coefficient (C_o=0.05) and different misalignment coefficients (D_m=0, 0.2, and 0.4) is studied in this case. It is obvious from this figure that misaligned bearings having higher values of misalignment coefficients have higher values of oil film temperature. An increase of (8%) in oil film temperature for the bearing with misalignment coefficient of (0.5) has been obtained in comparison with the aligned one. This is can be attributed to the higher shear rate of the oil film in this case. The effect of bearing misalignment on the oil film pressure distribution for a bearing working under certain conditions (R_e=10000, C_o=0.05 and ϵ=0.3) can be seen in figure (9). It is clear from this figure that a higher oil film pressure is obtained for the bearing with higher misalignment coefficients. It can be noticed from this figure that a value of non-dimensional pressure of (0.399) for a bearing with misalignment coefficient of (0.4) has been obtained in comparison with (0.14) for aligned bearing (i.e. D_m=0). This can be attributed to the lower values of oil film thickness in the bearing with higher misalignment coefficient. It is well known that the oil film pressure is very sensitive to the variation in the oil film thickness. A slight decrease in oil film temperature can be observed when the bearing with higher elastic coefficient is used as shown in figure (10). The effect of journal misalignment and the elastic deformation of the bearing shell on the oil film maximum pressure is shown in figure (11). It is obvious from this figure that the maximum pressure of the oil film increases as the degree of bearing misalignment increases and decreases as the elasticity coefficient increases. Since it is well known that the misalignment coefficient is an indication to how the journal surface is close to the bearing surface and hence the higher values of (D_m) means thinner oil film thickness and hence higher pressure value while the increase of the elasticity index(C_o)

means an increase in oil film thickness which causes in lower pressure values. Figure (12) shows that oil film pressure increases when the bearing works at higher Reynolds number. Figure (13) illustrates the effect of elastic deformation of the bearing shell on the values of the maximum temperature through the oil film thickness. It can also be shown from this figure that a lower value of induced temperature has been obtained when the bearing shell deformation has been taken into consideration. This can be attributed to the increase of oil film thickness with the increase of the bearing shell elastic deformation which causes a lower oil film shear rate and hence lower oil film temperature. Figure (14) shows the effect of journal misalignment on the values of the maximum temperature through the oil film thickness. It is obvious from this figure that journal misalignment affects the oil film temperature. The higher values of misalign-nment coefficient means a thinner oil film causing a higher shearing rate of the oil film and hence higher oil film temperature. Figure (15) shows the relationship between attitude angle and degree of misalignment at different values of coefficient of elastic deformation (Co=0, 0.02 and 0.05). It is obvious from this figure that a slight decrease in attitude angle has been noticed for the bearing having an elastic shell with lower coefficient of elastic deformation. A higher misalignment moment was obtained for the bearing working at higher degree of misalignment (bearing with higher values of D_m) as shown in figures (16) and (17) . This can be attributed to the higher oil film pressure obtained in this case. It can be seen from these figures that the bearing misalignment moment increases by (30%) as the bearing working at a misalignment coefficient of (0.5) rather than (0.25) and has an elastic bearing shell with elastic coefficient (0.05). The calculations have been performed for a bearing working at eccentricity ratio of (0.7).

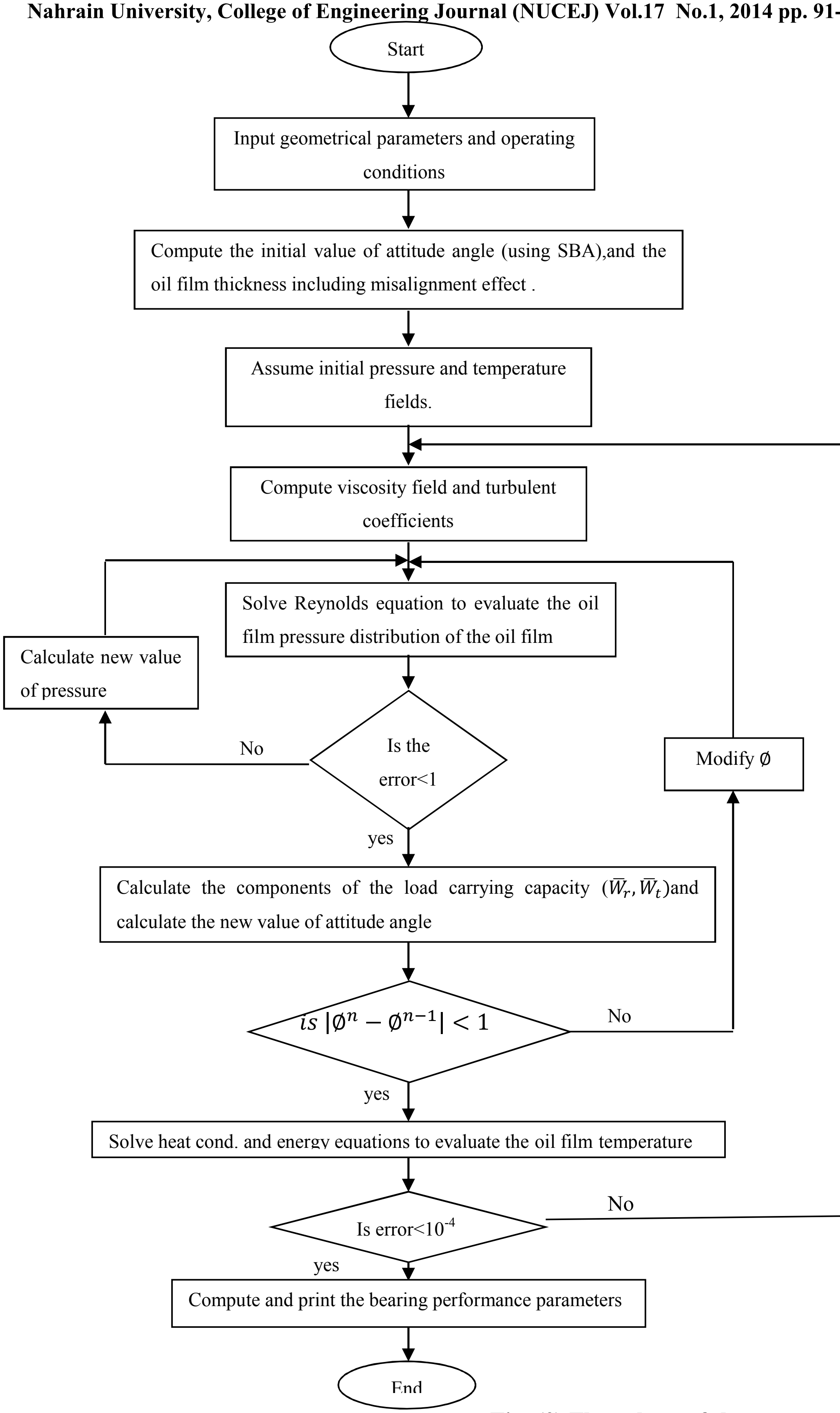

Fig. (3) Flow chart of the computer program

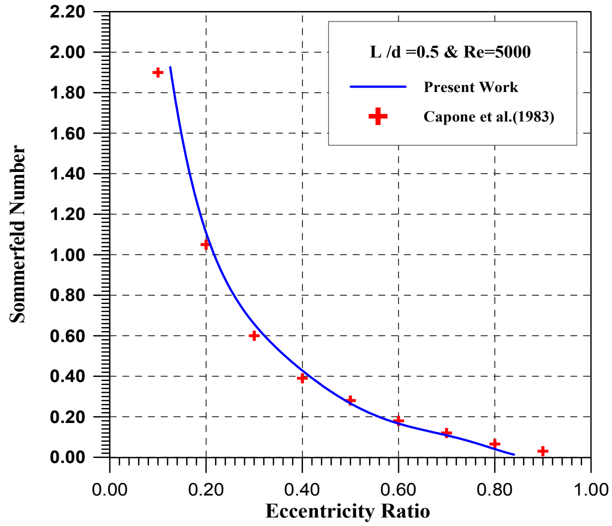

Fig. (4) Sommerfeld number against eccentricity ratio comparison with Capone results [4]

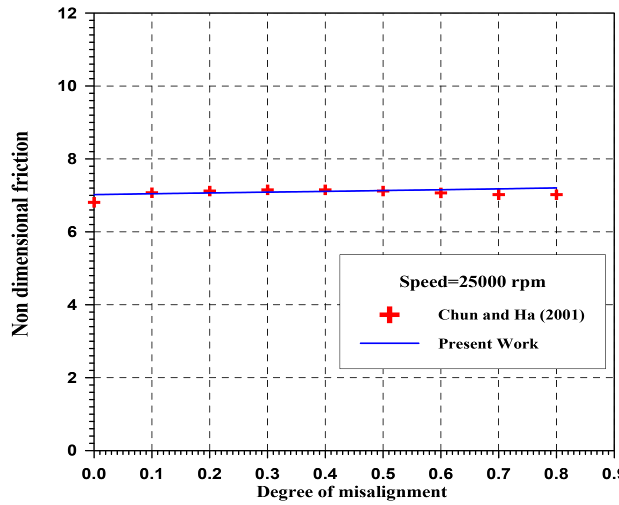

Fig (5) Non dimensional friction force against degree of misalignment comparison with Chun and Ha [10]

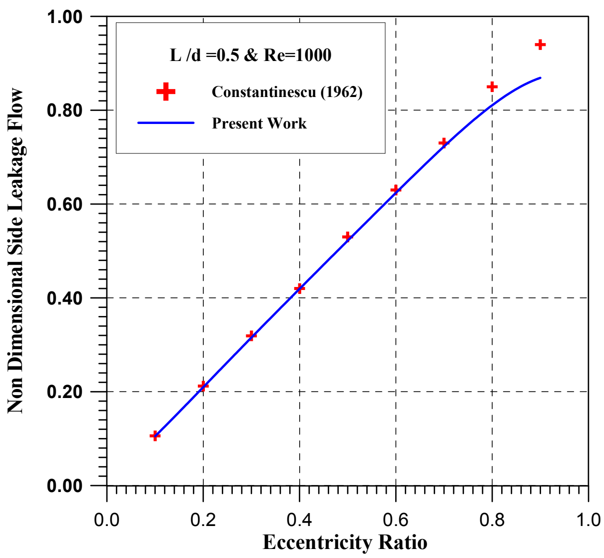

Fig.(6) Non dimensional side leakage flow against eccentricity ratio comparison with the results of ref.[1]

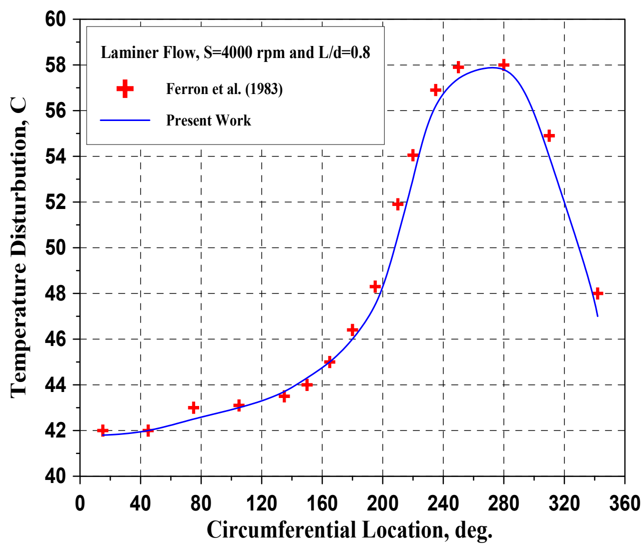

Fig.(7)temperature distribution against circumferential location comparison with ferron et.al.[11]

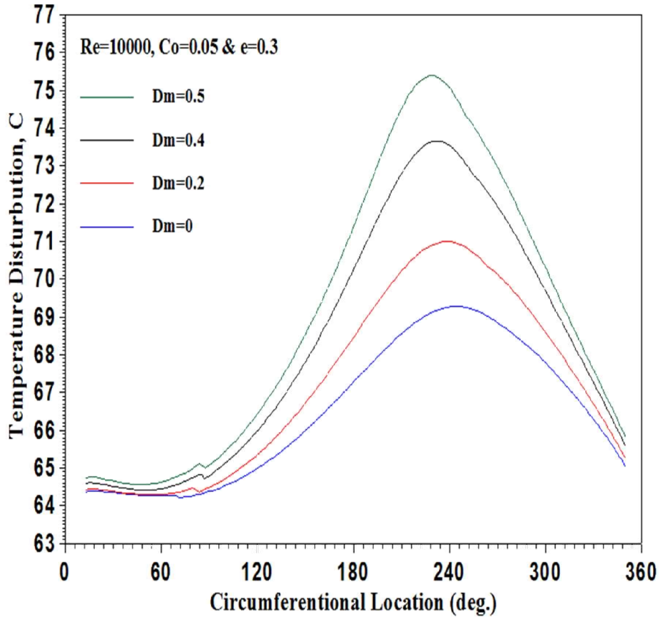

Fig. (8) Temperature distribution curves against circumferential location for different values of misalignment degree.

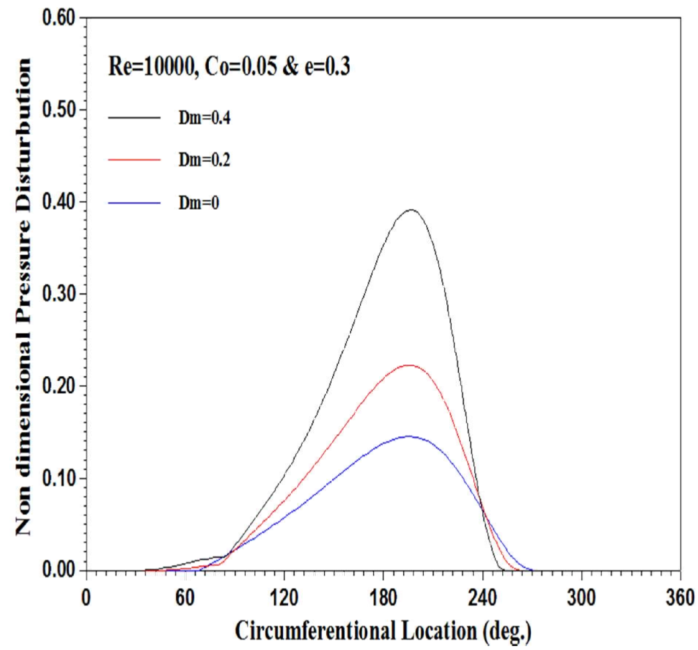

Fig. (9) Pressure distribution curves against circumferential location for different values of misalignment degree.

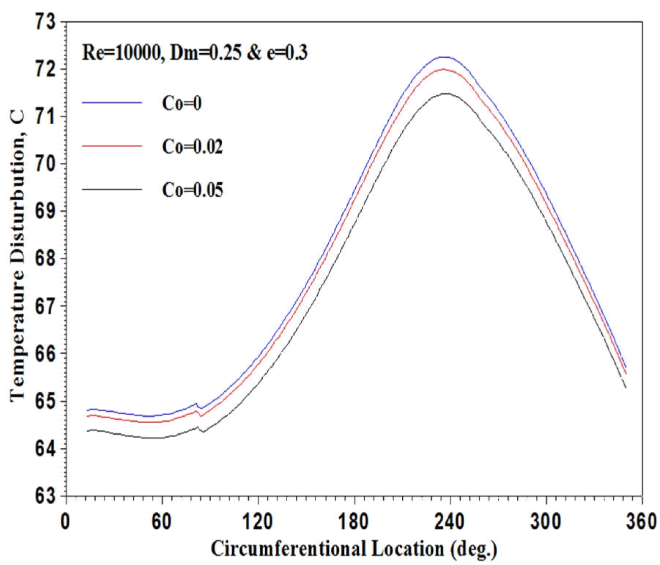

Fig. (10) Temperature distribution curves for different elastic coefficients.

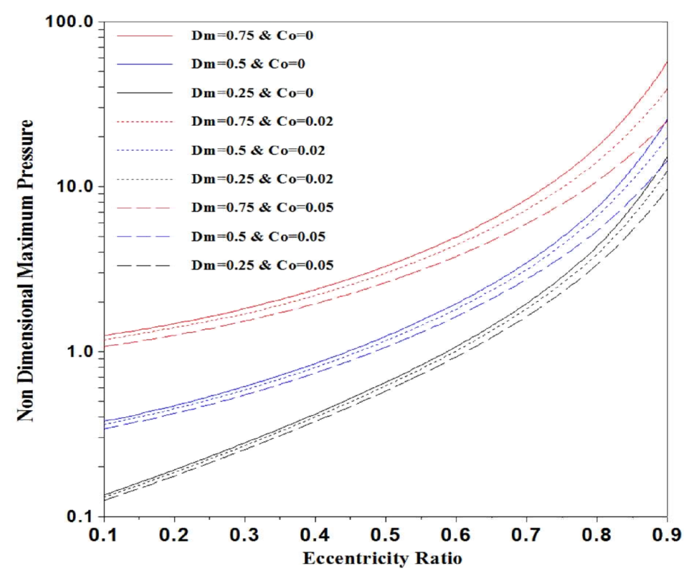

Fig. (11) Max. Pressure against eccentricity ratio for different values of C_o and D_m.

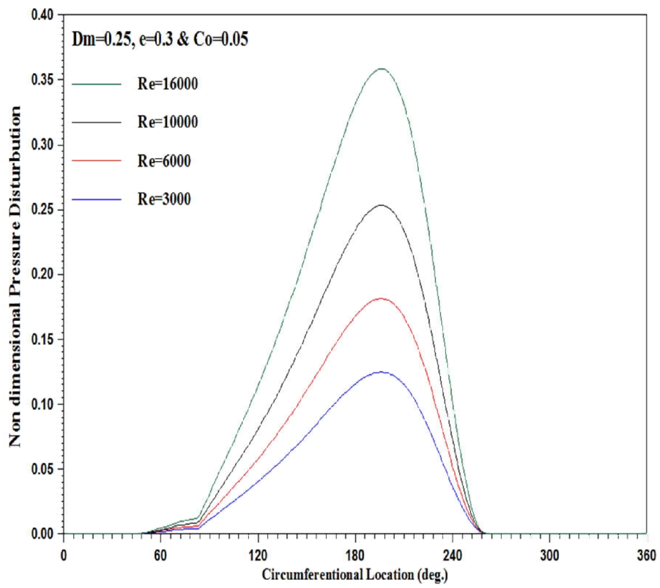

Fig. (12) Nondimensional oil film pressure distribution for different Reynolds number and elastic coefficient C_o=0.05

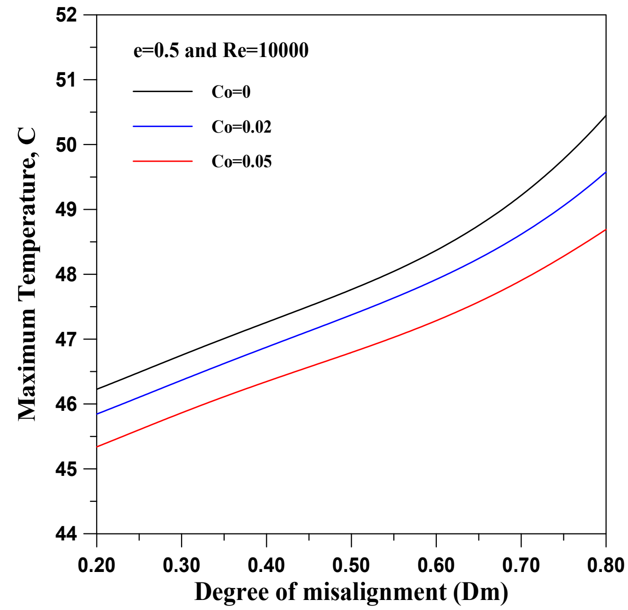

Fig. (13) Maximum temperature against degree of misalignment for different values of C_o

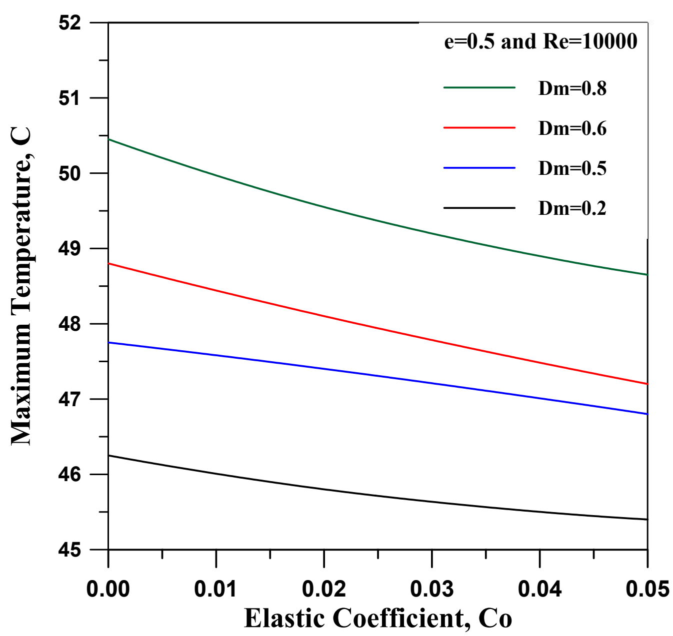

Fig. (14) Maximum temperature against elastic coefficient coefficients for different values of Dm

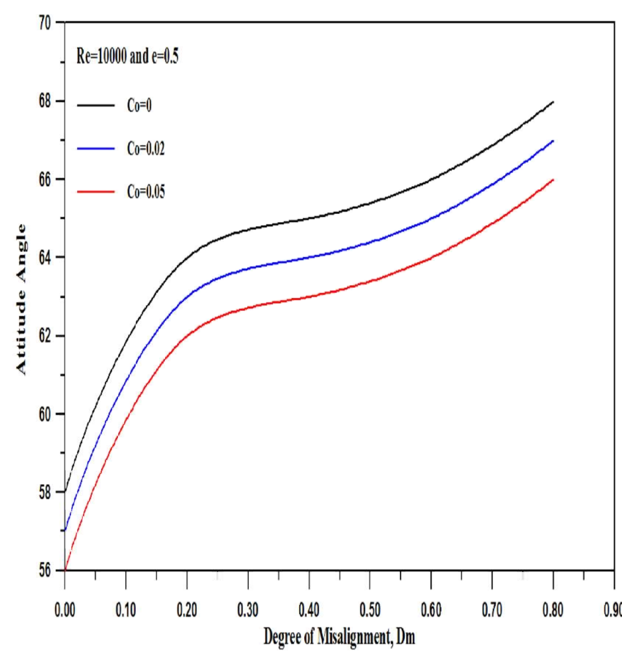

Fig (15) Attitude angle against degree of misalign-ment for different values of C_o

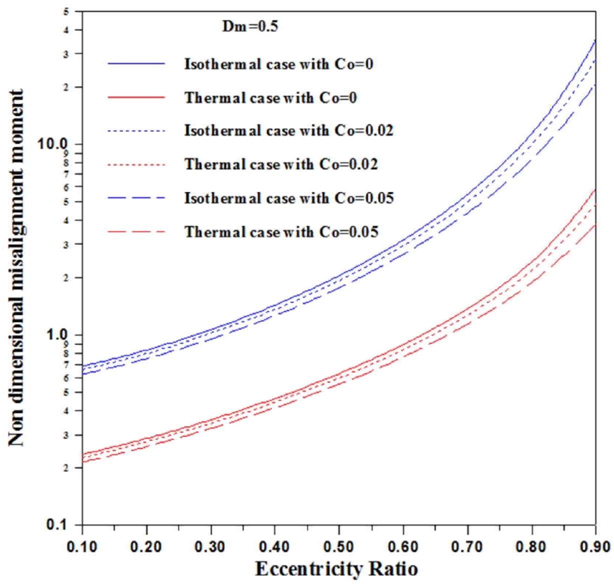

Fig. (16) Variation of bearing misalignment Moment with the eccentricity ratio for thermal And isothermal cases for D_m=0.5

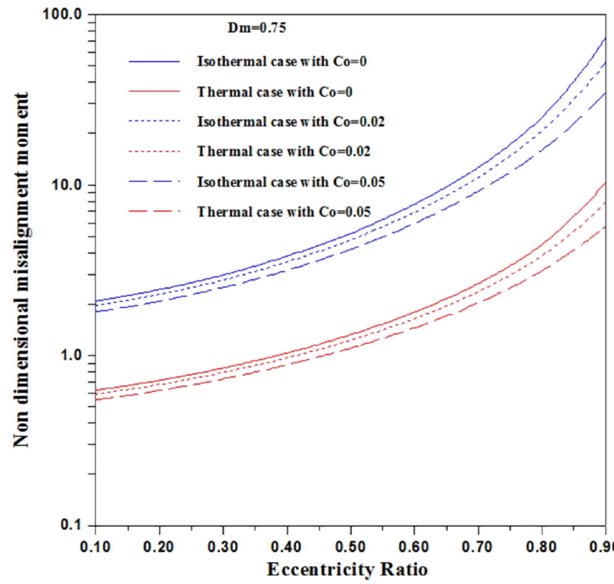

Fig. (17) Variation of bearing misalignment moment with the eccentricity ratio for thermal and isothermal cases D_m=0.75

Conclusions

A two-dimensional thermohydrodynamic analysis in turbulent flow regime is performed to study the combined effect of bearing shell elastic deformation and bearing misalignment on the performance of journal bearings. A Constantinsque turbulent theory has been adopted through this work. Satisfactory agreement has been obtained between the results obtained through this work when compared with that published by other workers. It is clear from the previous discussion that the effect of the bearing shell elastic deformation is to reduce the oil film pressure, temperature, and bearing misalignment moment. A higher reduction was obtained for the bearing with higher elasticity index (C_o). This effect cannot be neglected in bearing design especially when the bearing works at higher eccentricity ratios. It is also clear that the oil film pressure and temperature increasing when the bearing working at increasing degree of shaft misalignment (D_m).

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Nomenclature

C = radial clearance (m)

C_o = elastic coefficient

C_p = lubricant specific heat (J/kgC^o)

E=Youngs modulus of elasticity (N/m^2)

h = nominal oil film thickness (m)

$\bar{h}$ = (h/C) = non-dimensional oil film thickness

M=misalignment moment of the bearing

P = oil film pressure (N/m^2)

$\bar{P}$= nondimensional oil film thickness

R = journal radius (m)

r_b = bearing radius (m)

R_{eh} =Uh/v

t = bearing shell thickness (m)

T= oil film temperature (C^o)

$\bar{T}$ = nondimensional oil film temperature

u = oil velocity in x-direction (m/sec)

v= oil velocity in y- direction (m/sec)

w= oil velocity in z- direction (m/sec)

μ = lubricant viscosity (Pa.sec)

$\bar{\mu}$ = dimensionless lubricant viscosity.

μ_o = inlet lubricant viscosity (Pa.sec)

ρ = lubricant density (kg/m^3)
ω = journal rotational speed (rad/sec)

τ = local shear stress (N/m^2)

τ_v = viscous shear stress (N/m^2)

τ_c=Couettesurface shear stress (N/m^2)
τ_{xy} = shear stress component in x-y plane.
τ_{zy}= shear stress component in z-y plane
ϕ = attitude angle (deg)
$\boldsymbol{\upsilon}$ = Possion's ratio.

ϵ' = the magnitude of the projection of the complete journal center – line on the mid-plain.

ϵ'_{max}= the maximum possible value of ϵ' (for which the journal and the bearing will come into metal to metal contact).

α = the angle between the journal rear centerline projection and the eccentricity vector ϵ_o.

β_o = oil viscosity coefficient.

دراسة تاثير التشوه المرن لقشرة المساند المقعدية ودرجة حرارة الزيت على المساند الغير متطابقة المحاور والتي تعمل بالجريان الاضطرابي

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الخلاصة:

يتضمن هذا البحث دراسة تاثير التشوه المرن لقشرة المساند المقعدية على تصرف المساند المقعدية ذات المحاور غير المتطابقة في الحالة المستقرة. تم اجراء دراسة مقارنة بين المساند ذات القشرة الجسئة وتلك المرنة ولدرجات لاتمركز مختلفة.

تم الاخذ بنظر الاعتبار تاثير التشوه المرن لقشرة المسند وتاثير درجة حرارة الزيت وتغير الضغط ودرجة الحرارة على اداء المساند التي تعمل في حالة الجريان المضطرب. لقد لوحظ حدوث ارتفاع في تحمل المساند المقعدية ذات المحاور غير المتطابقة للحمل المسلط نظر لصغر سمك طبقة الزيت في هذه الحالة. ان اخذ التشوه المرن لقشرة المسند القعدي مع تاثير درجة حرارة الزيت بنظر الاعتبار سبب في انخفاض قيمة الحمل الذي يحمله المسند المقعدي. لقد تم مقارنة بعض النتائج المستحصلة في هذا البحث مع تلك المنشورة في مصادر اخرى ووجدت في حالة تطابق جيد .