Thermo-hydrodynamic Analysis of Misaligned Journal Bearing Considering Surface Roughness and Non-Newtonian Effects

Asst. Prof. Basim Ajeel Abass
E-mail: *drbajab62@yahoo.com*

Asst. Lec. Mustafa Mohammed Kadhim
engmustafa_mo@yahoo.com

Faculty member, *Mech. Eng. Department*, *University of Babylon*, *Babylon-Iraq*

ABSTRACT

This paper presents a numerical simulation for the combined effect of surface roughness and non-Newtonian behavior of the lubricant on the performance of misaligned journal bearing. The modified Reynolds equation to include the effect of non-Newtonian lubricant and bearing surface roughness has been formulated. The model accounts for the lubricant viscosity dependence on temperature and shear rate. In order to make a complete thermo-hydrodynamic analysis (THD) of rough surface misaligned journal bearing lubricated with non-Newtonian lubricant, the modified Reynolds equation coupled with the energy, heat conduction equations, the equation related the viscosity and temperature with appropriate boundary conditions have been solved simultaneously. The performance characteristics of the bearing were presented with different roughness parameter for the pressure, temperature, load carrying capacity, misalignment moment and friction force. The computer program prepared to solve the governing equations of the problem has been verified by comparing the results obtained through this work with that published by different workers. It has been found that the results are in a good agreement .The results obtained in the present work showed that the surface roughness characteristics of opposing surfaces and its orientation play an important role in affecting the performance parameters of the bearing. It has been shown that the load in rough aligned journal bearing is higher than that in rough misaligned journal bearing for all surface roughness patterns (γ). An increase in load has been calculated and found to be 29.5% for the bearing with moving roughness while it becomes 32% for the bearing with stationary roughness.

KEYWORDS: Thermo-Hydrodynamic Analysis, Surface Roughness, Journal Misalignment, Non-Newtonian Lubricant.

التحليل الثرموهيدروديناميكي للمسند المقعدي باعتبار تاثيرات المحاورالغير متحدة والتصرف اللانيوتوني

أ.م.د. باسم عجيل عباس
جامعة بابل / كلية الهندسة

م.م. مصطفى محمد كاظم
جامعة بابل / كلية الهندسة

الخلاصة:

يتضمن البحث تمثيلا عدديا للتاثير المشترك للخشونة السطحية والتصرف اللانيوتوني للزيت على اداء المساند المقعدية الغير متحدة المحاور. تم تهيئة معادلة رينولدز المحورة لتاخذ بنظر الاعتبار التاثير المشترك للخشونة السطحية والتصرف اللانيوتوني للزيت.ياخذ النموذج الرياضي الموضوع بنظر الاعتبار اعتماد لزوجة الزيت على درجة الحرارة ومعدل القص. لغرض اجراء تحليل حراري كامل للمساند المقعدية ذات السطوح الخشنة واللامتحدة المحاوروالمزيتة بالزيت اللانيوتوني التصرف فان معادلات رينولدز والطاقة ومعادلة انتقال الحرارة بالحمل اضافة الى معادلة اللزوجة والشروط الحدية

المناسبة حلت انيا. ان الصفات الادائية للمساند المقعدية مثل الضغط والحمل وعزم الانحراف وقوة الاحتكاك قد تم دراستها ولمعاملات خشونة سطحية مختلفة. تم اختبار البرنامج الحاسوبي المعد لحل المعادلات الحاكمة لهذه المسالة عن طريق مقارنة بعض النتائج المستحصلة في هذا البحث مع تلك المنشورة في الادبيات اللمتعلقة بالموضوع. اظهرت النتائج المستحصلة بان الخصائص السطحية للمساند المقعدية تلعب دورا مهما في اداء تلك المسا ند.

الكلمات الرئيسية: التحليل الحراري ، الخشونة السطحية، المحاور اللامتطابقة ،الزيوت ذات التصرف اللانيوتوني.

INTRODUCTION

Hydrodynamic bearings have been used in various ap- plications of mechanical industry since along time to support rotating shafts of heavy machines. They are con- sidered as a good choice due to their construction sim- plicity, reliability, efficiency and low cost. Hydrody- namic journal bearings are commonly found in internal combustion engines, for supporting the crank shaft, ce- ment mill and turbocharger applications. Any hydrody- namic lubrication process consists essentially of two surfaces (bearing and journal surfaces) in relative motion shearing a thin layer of viscous fluid between them. Oil film pressure and temperature inside the bearing affected by the applied conditions, operating parameters (such as, rotational speed), geometric factors (such as, radial clear- ance) and types of lubricant (Newtonian or non-Newto- nian lubricant). Oil viscosity as known varies substan- tially with the temperature leading to a large change in the bearing load capacity, consequently, thermal effect in hydrodynamic lubrication plays a crucial role in predict- tion of the bearing performance. Large number of re- searchers had been studied this subject using different methods of solutions as in **Pinkus and Bupara(1979), Majumder (1974), Mitsui et.al.(1983), Ferron et.al. (1983), Mitsui et.al.(1986), Mitsui (1987) , Oh et.al.(1998)**. Fluid film gap changes that occur in the axial direc- tion of the journal bearing are caused by a well known phenomenon called bearing misalignment which is defined as a non-parallelism between the axis of the bearing and that of the journal. This state has a considerable, and usually disadvantageous effect on the performance of the journal bearing. In engineering applications this misalignment can be identified with one, or combination of the following causes: each deflection of the journal or its support, thermal distortion of the shaft and bearing, error in manufacturing, or externally imposed misaligning moments. The above causes lead to two types of bearing misalignment namely, axial (vertical displacement) and twisting (horizontal displacement) or combination of them. Misalignment in hydrodynamic bearings has been recognized by several investigators as in **Oscar Pinkus et.al.(1979), P. H. Markhoet.al. (1979), Z. S. Safar (1984), M. O. A.Mokhtar et.al.(1985),D. Vijayraghavan et.al.(1998), S. M. Chun et.al.(2000), F. P. Brito et.al.(2007), L. Roy(2009).**

Journal bearing surfaces, in practice, are all rough. Surface roughness has a considerable effect on the functional characteristics of the bearing operating in hydrodynamic, and especially, in mixed lubrication regimes, this effect of surface roughness is greatly enhanced by increasing a common parameter of roughness called root mean square (RMS) or standard deviation (σ) of roughness. The operation parameters and geometric factors have also important role in enhancing the surface roughness effect. In recent years, a considerable amount of re- search activities have been devoted to the study of surface roughness effects on the performance of hydrodynamic journal bearing systems, and also the driving force behind these studies, is the frequent failure of tribological component due to metal to metal contact and the associated rise in the frictional heating in **Dyson(1976), Nadir Patir et.al.(1978), Safar (1988), Keith(1989), Banwait(2006), Yang et. al. (2008) , Sinha (2008).**

. Also the lubricant used in industrial applications seems to have non-Newtonian behavior rather than Newtonian because of the oil contamination. Many researchers dealt with this behavior using power law fluid model and another models as in **Nadir Patir et.al. (1979), Tripp (1983), Long Li(1998), Shi et. al. 1998, Wang, Shi et.al.(1998).**

The present work represents an attempt to study the effect of different parameters, namely, surface roughness, journal misalignment, non-Newtonian lubricant behavior and oil viscosity-temperature dependence on the performance of the hydrodynamic bearings. This is important in order to predict a realistic and accurate solution of the bearing performance and to avoid the probability of asperity contact in the space between two rough surfaces of the journal bearing which leads to a considerable increase of the maximum oil film temperature and reduced the bearing life.

GOVERNING EQUATIONS

A schematic diagram for the journal bearing used in the present analysis with the suitable coordinate has been shown in **Fig.1**. It can be shown in this figure the effect of surface roughness of both journal bearing surfaces and the effect of the shaft misalignment on the oil film thickness. The following are the governing equations adopted through this work:

Reynolds Equation

In the present work a modified Reynolds equation to include the effect of surface roughness and viscosity variation along and across the fluid film has been derived following the same procedure of Feron et.al.(1983) using Patir and Chengs postulates(1978),(1979).

$$\frac{\partial}{\partial\theta}\left\{\phi_x \bar{h}^3 \bar{F} \frac{\partial \bar{p}}{\partial\theta}\right\} + \left(\frac{R}{L}\right)^2 \frac{\partial}{\partial \bar{z}}\left\{\phi_z \bar{h}^3 \bar{F} \frac{\partial \bar{p}}{\partial \bar{z}}\right\} = \frac{\partial}{\partial\theta}\left\{\bar{G}\cdot \bar{h}_T\right\} + \frac{1}{\Lambda}\cdot\frac{\partial}{\partial\theta}\left\{\bar{G}\phi_s\right\} \qquad (1)$$

where:

$$\bar{F} = \frac{\int_0^1 \left\{\int_0^1 \frac{\bar{y}}{\bar{\mu}}\, d\bar{y}\cdot\int_0^{\bar{y}} \frac{1}{\bar{\mu}}\, d\bar{y} - \int_0^{\bar{y}} \frac{\bar{y}}{\bar{\mu}}\, d\bar{y}\cdot\int_0^1 \frac{1}{\bar{\mu}}\, d\bar{y}\right\} d\bar{y}}{\int_0^1 \frac{1}{\bar{\mu}}\, d\bar{y}} \qquad (2)$$

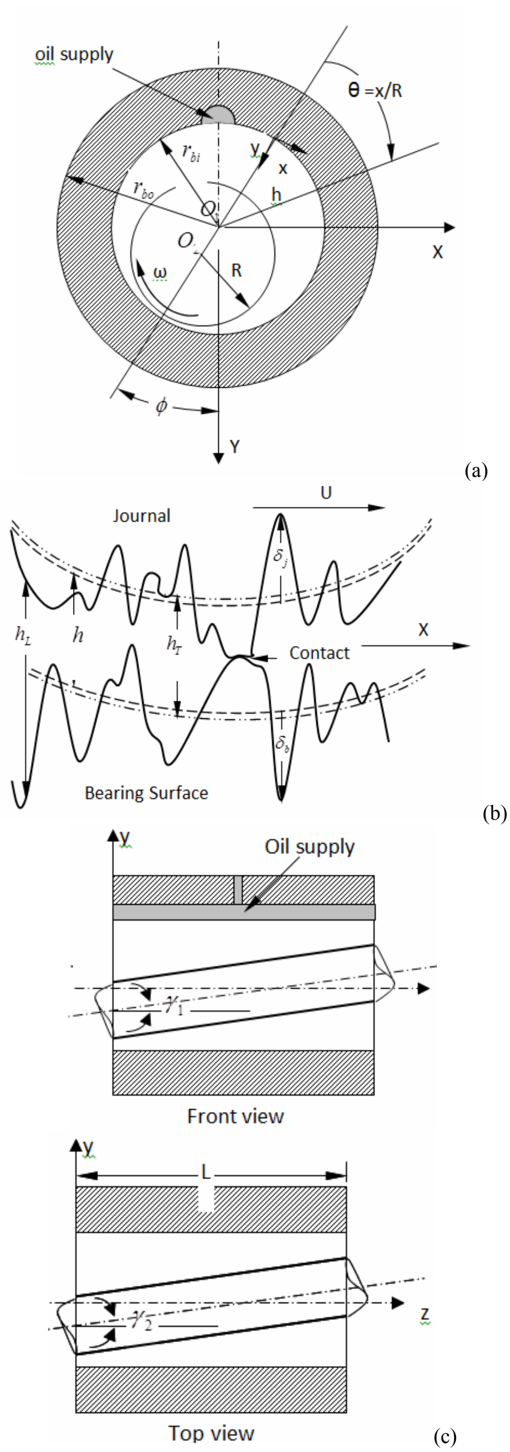

Fig.1 Geometrical Configuration of Grooved Misaligned Rough Journal Bearing System: (a) Journal Bearing Geometry and Coordinate System; (b) Oil Film Thickness and Surfaces Profile; (c) Bearing Misalignment.

$$\overline{G} = \frac{\int_0^1 \left\{ \int_0^{\overline{y}} \frac{1}{\overline{\mu}} d\overline{y} \right\} d\overline{y}}{\int_0^1 \frac{1}{\overline{\mu}} d\overline{y}} \tag{3}$$

$$\overline{h} = \frac{h}{c}, \quad \overline{h}_T = \frac{h_T}{c}, \quad \theta = \frac{x}{R}, \quad \overline{y} = \frac{y}{h_T}, \quad \overline{z} = \frac{z}{L}, \quad \Lambda = \frac{c}{\sigma}$$

$\overline{h}_T$ is the average film thickness (separation) between the rough surfaces as shown in **Fig.1(b)**, (ϕ_x, ϕ_z) are pressure flow factors, and ϕ_s is a shear flow factor.

The second term in the right side of average Reynolds equation represents the additional flow transport due to sliding in a rough bearing.

Oil Film Thickness

The local film thickness h_L is defined as shown in figure 1(b):

$$h_L = h + \delta_j + \delta_b \tag{4}$$

where h is the nominal film thickness (compliance) defined as the distance between the mean levels of the two surfaces. δ_j and δ_b are the random roughness amplitudes of the two surfaces (journal and bearing roughness amplitudes) measured from their mean levels, δ_j and δ_b assumed to have Gaussian distribution of heights with zero mean and standard deviations σ_j and σ_b respectively as was used in Patir and Chengs (1978),(1979). The combined roughness $\delta = \delta_j + \delta_b$ has a variance $\sigma = \sqrt{\sigma_j^2 + \sigma_b^2}$.

For the journal bearing system having Gaussian distribution of surface heights the expression for average fluid – film thickness ($\overline{h}_T$) in fully lubricated ($\Lambda\overline{h} \geq 3$) and partially lubricated ($\Lambda\overline{h} < 3$) regions of misaligned journal bearings can be expressed as follows.

$$\overline{h}_T = \overline{h} \quad \text{for} \quad \Lambda\overline{h} \geq 3$$

$$\overline{h}_T = \frac{\overline{h}}{2}\left[1 + erf\left(\frac{\Lambda\overline{h}}{\sqrt{2}}\right)\right] + \frac{1}{\Lambda\sqrt{2\pi}} e^{\frac{-(\Lambda\overline{h})^2}{2}} \Bigg\} \quad \text{for} \quad \Lambda\overline{h} < 3 \tag{5}$$

where $\overline{h}$ is the nominal fluid-film thickness (the fluid- film thickness of smooth journal bearing system) which can be introduced as in Jang et.al.(1987):

$$\overline{h} = 1 + \varepsilon\cos\theta \tag{6a}$$

For aligned journal bearing system

$$\overline{h} = \left\{(1 + \varepsilon\cos\theta) - \overline{z}\sigma_1^*\cos\theta + \overline{z}\sigma_2^*\sin\theta\right\} \tag{6b}$$

For misaligned journal bearing system where:

$$\sigma_1^* = 2\left(\frac{R}{c}\right)\left(\frac{L}{D}\right)\tan\gamma_1 \tag{6c}$$

$$\sigma_2^* = 2\left(\frac{R}{c}\right)\left(\frac{L}{D}\right)\tan\gamma_2 \tag{6d}$$

γ_1 and γ_2 are the tilting angles of the journal center line in vertical and horizontal direction respectively, as shown in figure 1(c).

FLOW FACTORS

The pressure flow factors ϕ_x and ϕ_z have been derived through numerical simulation as in Patir and Cheng (1978), while shear flow factor ϕ_s calculated as in Patir and Cheng (1979). The flow factors are determined in a function of the local average film thickness and in a function of the surface *r. m. s.* roughness (σ) and orientation (γ).

VISCOSITY-TEMPERATURE EQUATION

Energy equation is coupled to Reynolds equation through a nonlinear temperature viscosity relationship, since the viscosity of the lubricant was assumed to be variable across the film and a round the circumference. The dependence of viscosity on the temperature is given by the following equation as described by Ferron et.al.(1983):

$$\overline{\mu} = k_o - k_1\overline{T} + k_2\overline{T}^2 \tag{7}$$

k_o, k_1 and k_2 are viscosity coefficients, and $\overline{T}$ is a non-dimensional fluid temperature in the gap between two rough surfaces of the journal bearing system.

ENERGY EQUATION

The temperature distribution in the circumferential and a cross fluid-film directions for incompressible, Newtonian and non – Newtonian lubricant can be determined from a steady-state energy equation. The following non-dimensional form of energy equation is adopted through this work, Ferron et.al.(1983), Yang et.al.(2008):

$$\lambda_1\left\{\bar{u}\frac{\partial\bar{T}}{\partial\theta}\right\}+\lambda_1\left\{\frac{\bar{v}}{\bar{h}_T}-\bar{u}\frac{\bar{y}}{\bar{h}_T}\cdot\frac{\partial\bar{h}_T}{\partial\theta}-\left(\frac{R}{L}\right)\bar{w}\frac{\bar{y}}{\bar{h}_T}\cdot\frac{\partial\bar{h}_T}{\partial\bar{z}}\right\}\frac{\partial\bar{T}}{\partial\bar{y}}$$
$$=\lambda_2\frac{1}{\bar{h}_T^2}\frac{\partial^2\bar{T}}{\partial\bar{y}^2}+\lambda_3\frac{\bar{\mu}}{\bar{h}_T^2}\left\{\left(\frac{\partial\bar{u}}{\partial\bar{y}}\right)^2+\left(\frac{\partial\bar{w}}{\partial\bar{y}}\right)^2\right\} \quad (8)$$

where:

$$\lambda_1=\frac{\rho C_o UR}{K_{oil}},\ \lambda_2=\left(\frac{R}{c}\right)^2,\ \lambda_3=\left(\frac{R}{c}\right)^2\frac{\mu_{in}U^2}{K_{oil}T_{in}},$$

and $\bar{T}=\frac{T}{T_{in}}$

$\bar{u},\bar{v}$ and $\bar{w}$ are the non-dimensional form of a velocity components of the lubricant flow. The terms on the left-hand side represent the energy transported by convection. The first term on the right-hand side represents the heat transfer by conduction and the last term on the RHS represents the energy generated by internal friction known as the viscous dissipation term. The heat generated is due to viscous shear of the lubricant.

FLUID-FILM VELOCITY COMPONENTS

The flow of lubricant between two rough surfaces can be modeled by an equivalent flow model. The equivalent flow model is defined as two smooth surfaces separated by a clearance equal to the average gap ($\bar{h}_T$). Based on this model, a new group of pressure flow factors (φ'_x,φ'_z) and a shear flow factor (φ'_s) are defined as follows:

$$\phi'_x=\frac{\bar{h}^3}{\bar{h}_T^3}\phi_x\ ;\ \phi'_z=\frac{\bar{h}^3}{\bar{h}_T^3}\phi_z\ \text{ and }\ \phi'_s=\phi_s \quad (9)$$

The non-dimensional velocity components in circumferential and axial directions are expressed as [Nagaraju et.al. 2007]:

$$\bar{u}=\frac{u}{U}$$
$$=\phi'_x\bar{h}_T^2\frac{\partial\bar{p}}{\partial\theta}\left\{\int_0^{\bar{y}}\frac{\bar{y}}{\bar{\mu}}d\bar{y}-\frac{\int_0^1\frac{\bar{y}}{\bar{\mu}}d\bar{y}.\int_0^{\bar{y}}\frac{1}{\bar{\mu}}d\bar{y}}{\int_0^1\frac{1}{\bar{\mu}}d\bar{y}}\right\} \quad (10)$$
$$+\frac{\int_0^{\bar{y}}\frac{1}{\bar{\mu}}d\bar{y}}{\int_0^1\frac{1}{\bar{\mu}}d\bar{y}}+\frac{\phi'_s}{\Lambda\bar{h}_T}\frac{\int_0^{\bar{y}}\frac{1}{\bar{\mu}}d\bar{y}}{\int_0^1\frac{1}{\bar{\mu}}d\bar{y}}$$

$$\bar{w}=\frac{w}{U}=\phi'_z\bar{h}_T^2\frac{\partial\bar{p}}{\partial\bar{z}}\left(\frac{R}{L}\right)\left\{\int_0^{\bar{y}}\frac{\bar{y}}{\bar{\mu}}d\bar{y}-\frac{\int_0^1\frac{\bar{y}}{\bar{\mu}}d\bar{y}.\int_0^{\bar{y}}\frac{1}{\bar{\mu}}d\bar{y}}{\int_0^1\frac{1}{\bar{\mu}}d\bar{y}}\right\} \quad (11)$$

While the non-dimensional velocity component across the fluid-film is obtained from the continuity equation and is expressed as [Nagaraju et.al. 2007]:

$$\bar{v}=\frac{v}{U}\left(\frac{R}{c}\right)=-\bar{h}_T\int_0^{\bar{y}}\left\{\begin{array}{l}\frac{\partial\bar{u}}{\partial\theta}+\left(\frac{R}{L}\right)\frac{\partial\bar{w}}{\partial\bar{z}}\\-\frac{\bar{y}}{\bar{h}_T}\cdot\frac{\partial\bar{h}_T}{\partial\theta}\frac{\partial\bar{u}}{\partial\bar{y}}\end{array}\right\}d\bar{y} \quad (12)$$

HEAT–CONDUCTION EQUATION

The temperature distribution through the solid bush can be evaluated by solving the heat-conduction equation. The steady state heat-conduction equation with no heat source in non-dimensional form can be expressed as in Ferron (1983), Shi (1998):

$$\frac{\partial^2\bar{T}_b}{\partial\bar{r}_b^2}+\frac{1}{\bar{r}_b}\frac{\partial\bar{T}_b}{\partial\bar{r}_b}+\frac{1}{\bar{r}_b^2}\frac{\partial^2\bar{T}_b}{\partial\theta^2}=0 \quad (13)$$

where:

$\bar{T}_b$ is non-dimensional bearing temperature,
$(\theta,\bar{r}_b)$ are the coordinates in circumferential and radial directions respectively.

NON-Newtonian Model

The constitutive relation of the power—law fluid model in non-dimensional form is expressed as in Nagaraju et.al.(2007):

$$\bar{\tau}=\bar{\mu}\left(\dot{\beta}\right)^{n_1} \quad (14)$$

where n_1 and $\bar{\mu}$ are the power law index and consistency index (Newtonian lubricant viscosity). For an incompressible non-Newtonian fluid, the shear strain rate $(\dot{\beta})$ is independent of direction and by considering it as a function of the second strain invariant of shear strain rate I_2 it can be expressed in non – dimensional form as:

$$\dot{\beta}=\sqrt{I_2}=\left[\left(\frac{1}{\bar{h}}\frac{\partial \bar{u}}{\partial \bar{y}}\right)^2+\left(\frac{1}{\bar{h}}\frac{\partial \bar{w}}{\partial \bar{y}}\right)^2\right]^{1/2} \quad (15)$$

The viscosity of a non-Newtonian lubricant is described by an apparent viscosity $(\bar{\mu}_{non})$ which is defined as a function of shear strain rate $(\dot{\beta})$.

$$\bar{\mu}_{non}=\frac{\bar{\tau}}{\dot{\beta}}=\bar{\mu}\left(\dot{\beta}\right)^{n_1-1} \quad (16)$$

BEARING PERFORMANCE CHARACTERISTICS

For obtaining the steady state characteristics for single axial grooved misaligned journal bearing, the load components along and perpendicular to the line of centers are found out from:

$$\bar{W}_r=\int_0^1\int_0^{2\pi}\bar{p}\cos\theta\cdot d\theta\cdot d\bar{z} \quad (17)$$

$$\bar{W}_t=\int_0^1\int_0^{2\pi}\bar{p}\sin\theta\cdot d\theta\cdot d\bar{z} \quad (18)$$

Therefore the total load carrying capacity of the rough misaligned journal bearing can be evaluated as:

$$\bar{W}=\sqrt{\bar{W}_r^2+\bar{W}_t^2} \quad (19)$$

The attitude angle (ϕ) between the load line and the line of centers can be expressed as Oscar Pinkus et.al.(1979):

$$\phi=\tan^{-1}\left(-\frac{\bar{W}_t}{\bar{W}_r}\right) \quad (20)$$

The Sommerfeled number (bearing number) can be expressed as:

$$So=\frac{\mu_{in}\omega LR}{\pi W}\left(\frac{R}{c}\right)^2=\frac{2\mu_{in}NLR}{60W}\left(\frac{R}{c}\right)^2 \quad (21)$$

$\omega=\frac{2\pi N}{60}=$ angular velocity of the shaft (rad/sec)

$W=UL\mu_{in}\left(R/c\right)^2\bar{W}=$ load carrying capacity(N).

The mean viscous friction force of grooved rough misaligned journal bearing in dimensional form is described by the following equation as in Nadir Patir and Cheng(1979):

$$fr=\int_0^L\int_0^{2\pi R}\tau\, dx dz=\int_0^L\int_0^{2\pi R}\left(\mu\frac{U}{h}\left(\phi_f-\phi_{fs}\right)+\varphi_{fP}\frac{h}{2}\frac{\partial p}{\partial x}\right)dx dz \quad (22)$$

The non-dimensional form of the above equation is:

$$\bar{fr}=\frac{fr}{\mu_{in}UL\left(R/c\right)}=\int_0^1\int_0^{2\pi}\left(\frac{\bar{\mu}}{\bar{h}}\left(\phi_f-\phi_{fs}\right)+\varphi_{fP}\frac{\bar{h}}{2}\frac{\partial\bar{p}}{\partial\theta}\right)d\theta d\bar{z} \quad (23)$$

where $\left(\phi_f,\phi_{fs},\phi_{fP}\right)$ are empirical shear stress factors used to explain the effect of surface roughness on the friction force of journal bearing as in Nadir Patir and Cheng (1979).

The required moment for steady state operation is calculated directly from the fluid film pressure distribution. Two components of the moment vector are required to maintain the two assigned angles γ_1 and γ_2. The non- dimensional form of these components are expressed as in Buckholz et.al. (1986):

$$\bar{M}_r=\int_0^1\int_0^{2\pi}\bar{p}\bar{z}\cos\theta\cdot d\theta\cdot d\bar{z} \quad (24)$$

$$\bar{M}_t=\int_0^1\int_0^{2\pi}\bar{p}\bar{z}\sin\theta\cdot d\theta\cdot d\bar{z} \quad (25)$$

and the total moment in non-dimensional form can be calculated as follows:

$$\bar{M}=\sqrt{\bar{M}_r^2+\bar{M}_t^2} \quad (26)$$

BOUNDARY CONDITIONS

The following boundary conditions are used to gather with the governing equations to analyze the problem of thermohydrodynamic performance of grooved misaligned journal bearing with surface roughness.

A) Lubricant flow field

1) at the oil supply groove: $\theta=2\pi-\phi$; $\bar{p}=\bar{p}_{in}$

2) at the journal bearing edges: $\bar{z}=0$ & $\bar{z}=1$
$\bar{p}=\bar{p}_{atm}=0$

3) at the cavitation zone: $\partial\bar{p}/\partial\theta=0$ and

$\bar{p} = 0$

B) Thermal field

1) the temperature across the oil film in the groove zone known as mixed temperature $(\bar{T}_{mix})$ is assumed to be constant and can be estimated as in Banwait (2006):

$$\bar{T}_{mix} = \frac{Q_{rec}\bar{T}_r + Q_{in}\bar{T}_{in}}{Q_{rec} + Q_{in}} \tag{27}$$

where $\bar{T}_r =$ recirculation temperature;
$\bar{T}_{in} =$ inlet oil temperature;
$Q_{in} =$ supply oil flow rate (m^3/sec); $Q_{rec} =$ recirculation flow rate (m^3/sec) and is expressed as follow:

$$Q_{rec} = LUc\int_0^1 \bar{u}\bar{h}d\bar{y} \tag{28}$$

2) at the oil-bush interface and at the oil-shaft interface the matching temperatures are

$$\bar{T} = \bar{T}_b \text{ at } \bar{y} = 0 \text{ and } \bar{T} = \bar{T}_{sh} \text{ at } \bar{y} = 1$$

3) the heat flux continuity on the surface between the bush and the oil film interface which yield to the following as in Ferron et.al. (1983),Yang and Jeng(2008):

$$\left.\frac{\partial \bar{T}_b}{\partial \bar{r}_b}\right|_{\bar{r}_b=1} = -\frac{K_{oil}}{K_b}\frac{r_{bi}}{c}\frac{1}{\bar{h}_T}\left.\frac{\partial \bar{T}}{\partial \bar{y}}\right|_{\bar{y}=0} \tag{29}$$

where $\bar{r}_b = r_b/r_{bi}$ and K_{oil} is the thermal conductivity of the fluid which is constant.

4) The boundary condition which is referred to the loses heat by free convection [Ferron et.al. 1983]:

$$\left.\frac{\partial \bar{T}_b}{\partial \bar{r}_b}\right|_{\bar{r}_b=\bar{r}_{bo}} = -\frac{h_{conv}}{K_b} r_{bi}\left(\bar{T}_{bo} - \bar{T}_a\right) \tag{30}$$

COMPUTATIONAL TECHNIQUE

The governing equations, Reynolds, energy, heat conduction and the cross film viscosity incorporating non Newtonian effect equations are solved for pressure and temperature distribution satisfying appropriate boundary conditions. A computer program was developed to determine the performance characteristics of the bearing. The finite difference scheme was used. The solution field was discretized into rectangular meshs of (m-1) sections in ($\boldsymbol{\theta}$) direction and (n-1) sections in (Z) direction to give mesh size of (m*n). The following iterative procedure was used to calculate the bearing performance characteristics:

1- Input the bearing data and initial values of different variables to the computer program.
2- Calculate the initial value of the attitude angle.
3- Assume initial pressure distribution.
4- Calculate the oil film thickness with suitable misalignement and roughness models.
5- Calculate the pressure distribution by solving Reynolds equation.
6- Repeat steps (4) and (5) until convergance occur.
7-Calculate the temperature distrbutio through the oil film.
8-Repeat steps (4-7) until cpnvrgent solution is obtained.
9- Calculate the load components (W_r) and (W_t), hence calculate the attitude angle.
10- Compare the new value of attitude angle with old one. If there is no convergence, a new pressure field is computed until the difference between the last values of attitude angle reached less than one degree.
Calculate The performance parameters of the bearing.

It should be noted that negative pressure values are replaced by zero.

The iterations are repeated until the pressure satisfies the following convergence criterion:

$$\text{error}_{(p)} = \frac{\sum\sum\left|\bar{p}^{\,n}_{(i,k)} - \bar{p}^{\,n-1}_{(i,k)}\right|}{\sum\sum\left|\bar{p}^{\,n}_{(i,k)}\right|} \le 10^{-4}$$

And the oil film temperature satisfies the following convergence criterion:

$$\text{error}_{(\bar{T})} = \frac{\sum\sum\left|\bar{T}^{\,n}_{(i,k)} - \bar{T}^{\,n-1}_{(i,k)}\right|}{\sum\sum\left|\bar{T}^{\,n}_{(i,k)}\right|} < 10^{-6}$$

VALIDATION

To validate the present numerical scheme, the pressure distribution, temperature distribution, and the maximum load carrying capacity of the lubricant film compared with earlier experimental results for smooth, aligned bearing lubricated with Newtonian lubricant as shown in **Fig.s 2 to 4** . The numerical results agree quite well with Ferron's experimental work (1983). Also the numerical results for the maximum oil film pressure and the load carrying capacity agrees quite well with Roy's experimental work (2009) as shown in **Fig. 5** and **6.**

RESULTS AND DISCUSSION

The following numerical results are found for the combined effect of bearing surface roughness and non-Newtonian effect. The journal bearing variables used in computer program are listed in **Table 1**. **Figure 7** shows the effect of bearing surface roughness on the load carrying capacity of the bearing lubricated with Newtonian and non- Newtonian lubricant for different values of (h/σ) .It is clear from this figure that the load carrying capacity of the bearing with moving surface roughness decreases with increasing values of (h/σ). For the bearing with longitudinal roughness, the load carrying capacity of such bearing is greater than that of the bearing with smooth surfaces. Since the bearing with longitudinally oriented asperities permits only a small amount of side flow which enhances the flow in the direction of flow.

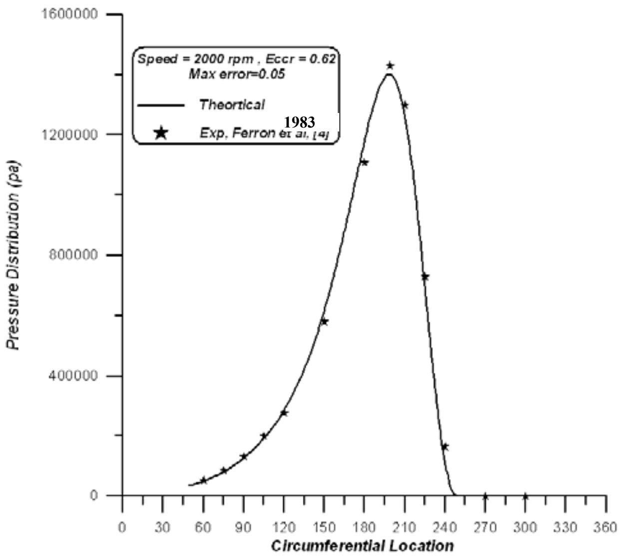

Fig.2 Oil film pressure distribution for smooth journal bearing

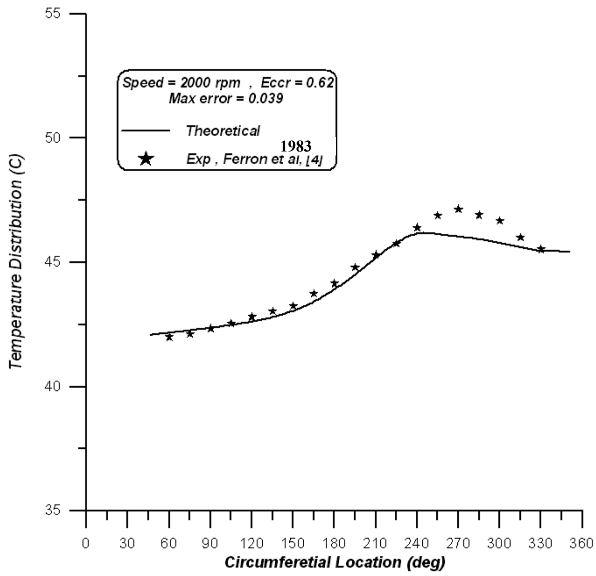

Fig.3 Oil Film temp. distribution for smooth journal bearing.

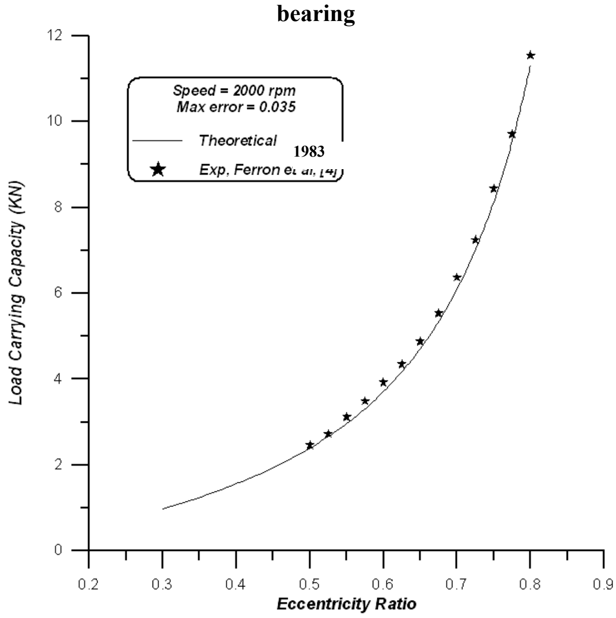

Fig. 4 Load Carrying Capacity versus Eccentricity Ratio for Smooth Journal Bearing

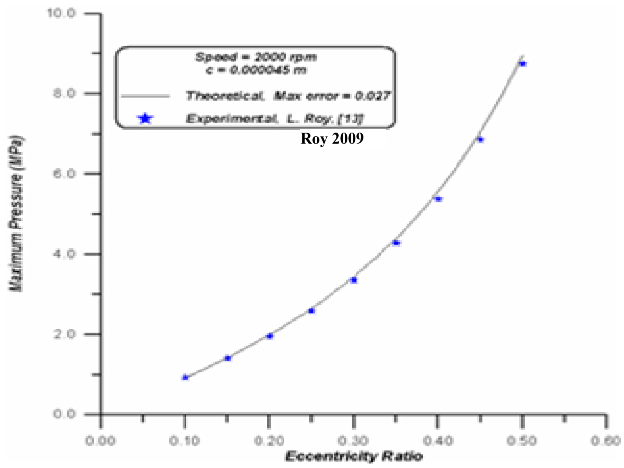

Fig.5 Maximum Pressure versus Eccentricity Ratio for Smooth Journal Bearing [Roy 2009].

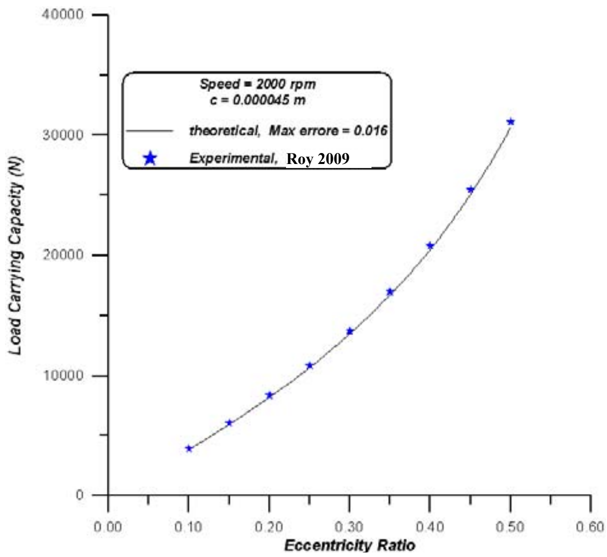

Fig.6 Load Carrying Capacity versus Eccentricity Ratio for Smooth Journal Bearing [Roy 2009].

different values of (h/σ) .It is clear from this figure that The bearing with isotropic and transverse roughness shows lower load carrying capacity than that of the bearing with smooth surfaces. This is expected since the decrease in (γ) results in small valley lengths which increased the side flow thereby decreasing the main flow which reduces the flow factors. The load carrying capacity of the bearing for all of the above cases become higher as the bearing lubricated with non- Newtonian lubricant of higher power index (n). The load carrying capacity of the bearing with stationary surface roughness shows different behavior than the bearing with moving surface roughness as can be seen from **Fig.**8.

Table 1. Geometric and operation parameters of the bearing.

Parameter	Symbol	Value and Unit
Bearing length	L	0.08 m
External bearing radius	r_{bo}	0.1 m
Journal radius	R	0.05 m
Radial clearance	C	0.000152 m
Eccentricity ratio	E	0.62
Misalignment angles in vertical and horizontal direction	γ_1, γ_2	(0.0002 and 0.00005 rad)
Surface roughness parameter	Λ	8 – 19
Surface pattern parameter for transverse, isotropic and longitudinal roughness pattern	Γ	1/6, 1.0, 6.0 respectively
Variance ratio	Vr_j	1.0, 0.0
Inlet lubricant temperature	T_m	40˚C
Ambient temperature	T_a	40˚C
Inlet lubricant pressure	p_m	70000 pa
Rotation speed	N	2000 rpm
Lubricant density at inlet temperature	ρ	860 Kg/m^3
Lubricant viscosity at inlet temperature	μ_{in}	0.0277 pa.sec
Viscosity coefficients	$(k_o, k_1,, k_2)$	3.287, 3.064 and 0.777

Lubricant specific heat	C_o	2000 J/Kg.˚C
Lubricant thermal conductivity	K_{oil}	0.13 W/m.˚C
Bush thermal conductivity	K_b	250 W/m.˚C
Bush convection heat transfer coefficient	h_{conv}	80 W/m^2.˚C
Air thermal conductivity	K_a	0.025 W/m.˚C
Groove angle	α_g	18 deg
Power law index of non-Newtonian Lubricant	N	1.2 and 0.8

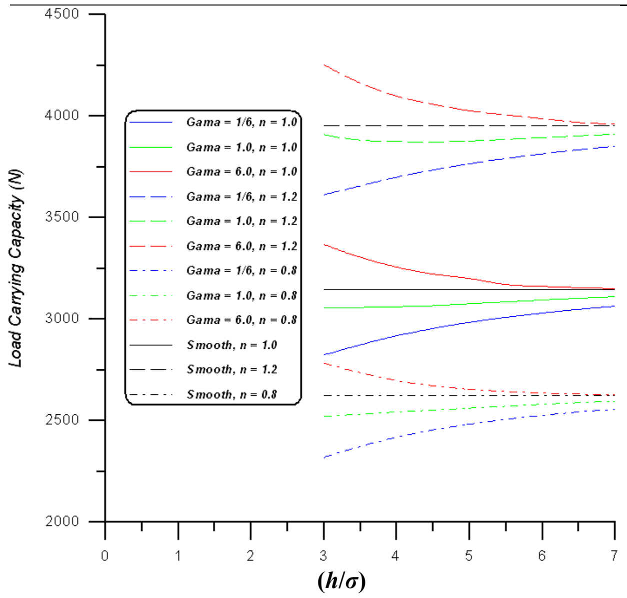

Fig.7 Hydrodynamic Load versus (h/σ) for Moving Surface Roughness in Misaligned Bearing using Newtonian (n = 1) and Non-Newtonian Lubricant (n = 1.2 and 0.8).

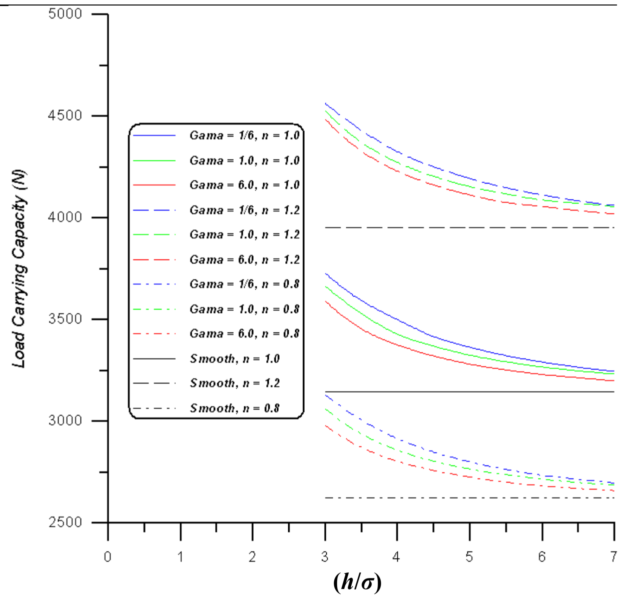

Fig.8 Hydrodynamic Load versus (h/σ) for Stationary Surface Roughness in Misaligned Bearing using Newtonian (n = 1) and Non-Newtonian Lubricant (n = 1.2 and 0.8).

The load carrying capacity increases with decreasing values of (h/σ), since decrease in (h/σ) is accompanied by a large increase in the mean gap, which enhances the flow factors. The bearing with transverse and isotropic roughness shows higher load carrying capacity than the bearing having longitudinally surface roughness. This is attributed to the fact that the asperities on the stationary rough surface acts a barriers in restricting the flow which results in a decreased mean flow as reveled by a negative shear flow factor. The load carrying capacity become higher when the bearing lubricated with a non Newtonian lubricant of higher index (n).

Figures 9 and 10 show the variation of maximum pressure against (h/σ) for the bearings with moving and stationary roughness. As expected the behavior of the maximum pressure is similar to that described for the bearing load carrying capacity. **Fig.11** shows the variation of the maximum oil film temperature with (h/σ) for a bearing with moving surface roughness. It can be seen from this figure that the oil film temperature increases as (h/σ) decreases obtained for the bearing with transverse surface roughness. It can be deduced from this figure that lower maximum oil film temperature was obtained for the bearing with longitudinal surface rough- ness and it becomes higher for lower values of (γ). This can be attributed to the fact that the longitudinally oriented asperities permits only a lower values of shear flow factor while the transverse oriented asperities shows the highest value of shear flow factor. It is clear that the oil film temperature increases as the bearing lubricated with non –Newtonian lubricant of higher power index.

This is due to the higher viscosity in this case. The bearing with stationary surface roughness shows lower oil film temperature than that obtained for the bearing with smooth surfaces as shown in **Fig.12**.This is expected since it is well known that the asperities on the stationary rough surface act as a barriers in restricting the flow. This results in a decreased mean flow as reveled by a negative shear flow factor. It is clear that the oil film temperature decreases as (h/σ) decreases. The bearing with surfaces having transverse roughness shows lower temperature. This is attributed to the maximum

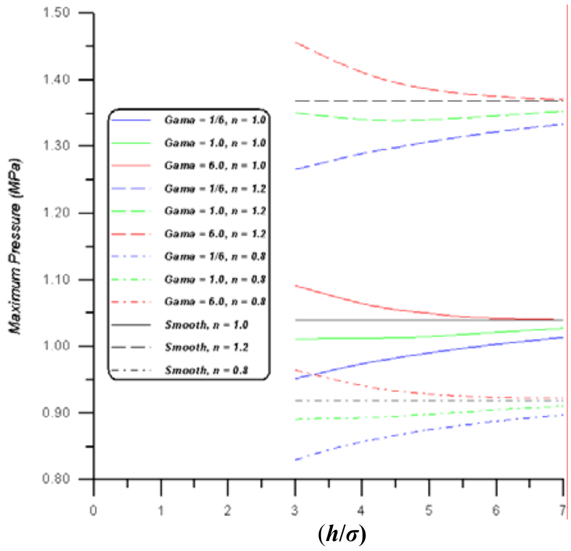

Fig.9 Hydrodynamic Maximum Pressure versus (h/σ) for Moving Surface Roughness in Misaligned Journal Bear- ing using Newtonian and Non-Newtonian Lubricant (n = 1.2 and 0.8).

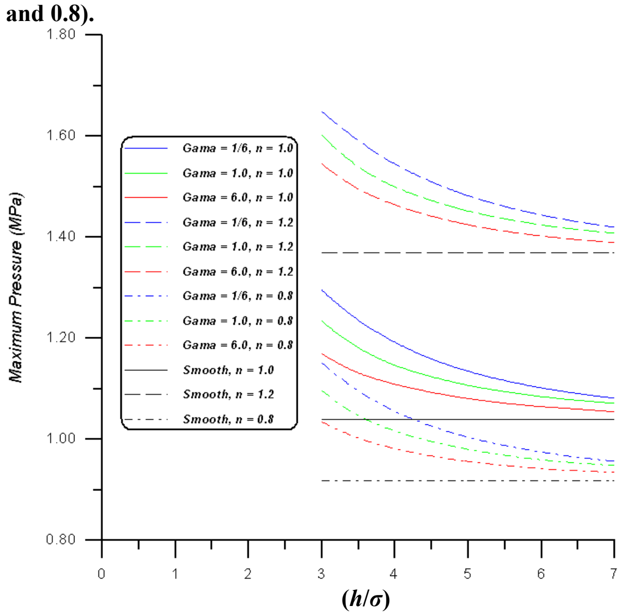

Fig.10 Hydrodynamic Maximum Pressure versus (h/σ) for Stationary Surface Roughness in Misaligned Journal Bearing using Newtonian and Non-Newtonian Lubricant.

resistance to the main flow offered by the transverse roughness thereby increasing the side flow, which reduces the flow factor ($\phi_x < 1$). Bearing lubricated with non –Newtonian lubricant of higher power law index shows a higher oil film temperature due to the higher viscosity and shearing effect.

Figures 13 and **14** show the variation of Sommerfeled number with roughness parameter (h/σ). These Figures confirms the results have been previously discussed, related the load carrying capacity since Sommerfeled number proportional to the reciprocal of the load. The friction force produced in the bearing increased for the bearing lubricated with non-Newtonian lubricant with higher power law index as shown in **Fig.15**. This is due to the higher oil viscosity in this case. It can also be shown from this figure that the surface of moving roughness with transverse and isotropic roughness surfaces have a lower friction force than that in bearing with smooth surfaces with reducing the values of (h/σ). This is can be attributed to the positive shear stress factor and the correction factor of pressure (which always lower than 1) causes to decrease the friction force. The variation of friction force with roughness parameter (h/σ) for a bearing with stationary surface roughness can be shown in **Fig.16**. It is clear from this figure that the force increases with decreasing values of roughness parameter (h/σ). This can be explained by knowing that for the rough surface with stationary roughness , the valleys can be considered as stagnant pockets produce a large resistance to the fluid flow which leads to increase the shear stress and hence the friction force. The bearing with longitudinal, transverse and isotropic roughness surfaces shows higher friction force than that of smooth surface. Bearing with surfaces having transverse roughness shows higher friction force than that with isotropic or longitudinal roughness which indicates the increase of the shear stress factor in this case. The effect of non-Newtonian behavior of the oil on the moment of rough misaligned bearing with stationary and

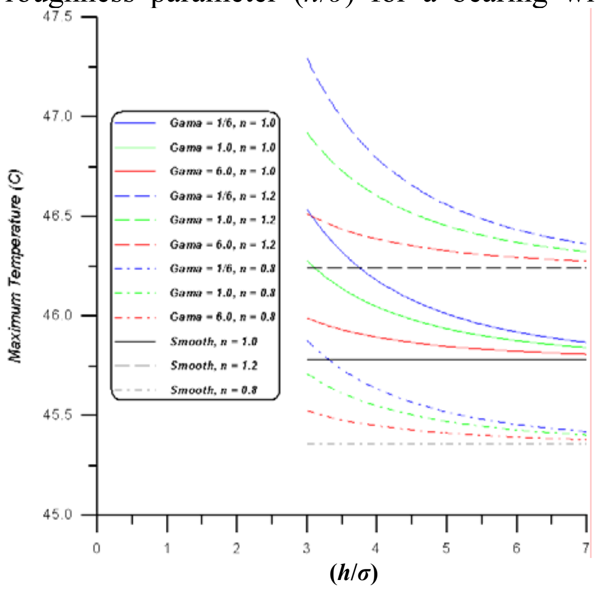

Fig.11 Hydrodynamic Maximum Temperature versus (h/σ) for Moving Surface Roughness in Misaligned Journal Bearing using Newtonian (n = 1) and Non-Newtonian

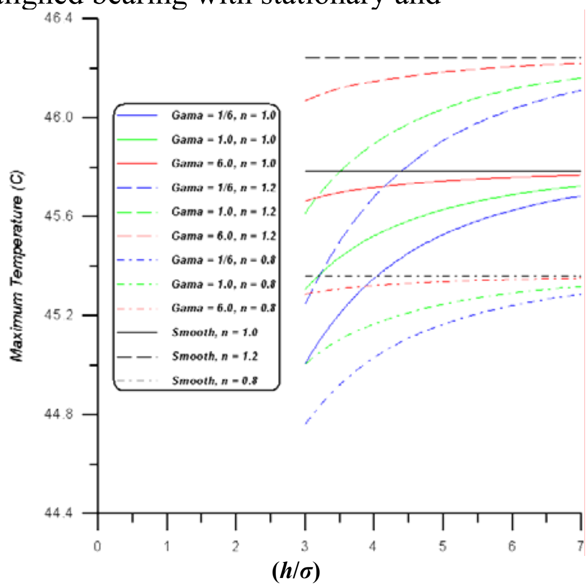

Fig.12 Hydrodynamic Maximum Temp. distribution for Stationary Surface Roughness in Misaligned Journal Bearing using Newtonian and Non-Newtonian Lubricant (n= 1.2 and 0.8)

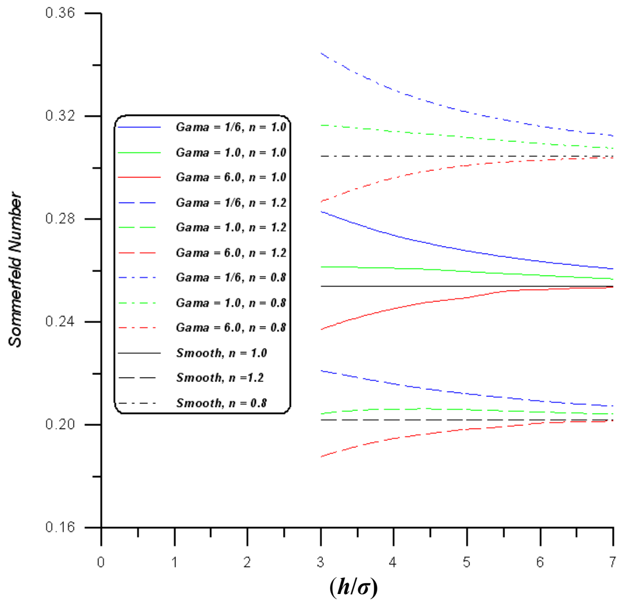

Fig.13 Sommerfeld Number versus (h/σ) for Moving surface roughness.

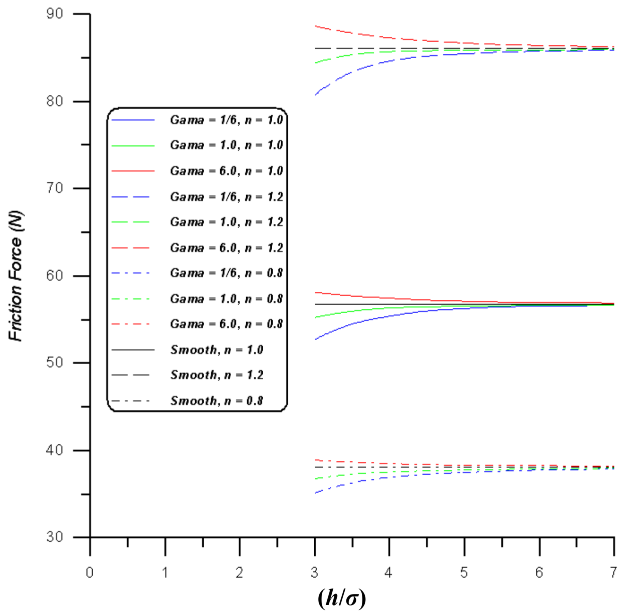

Fig.15 Friction Force versus (h/σ) for Moving Surface Roughness in Misaligned Journal Bearing using Newtonian and Non-Newtonian Lubricant.

moving roughness can be shown in **Fig.s(17) and (18)**. It can be shown from these figures that the bearing moment increases with bearing parameter for the bearing with moving surface roughness while it decreases for the bearing with stationary surface roughness.

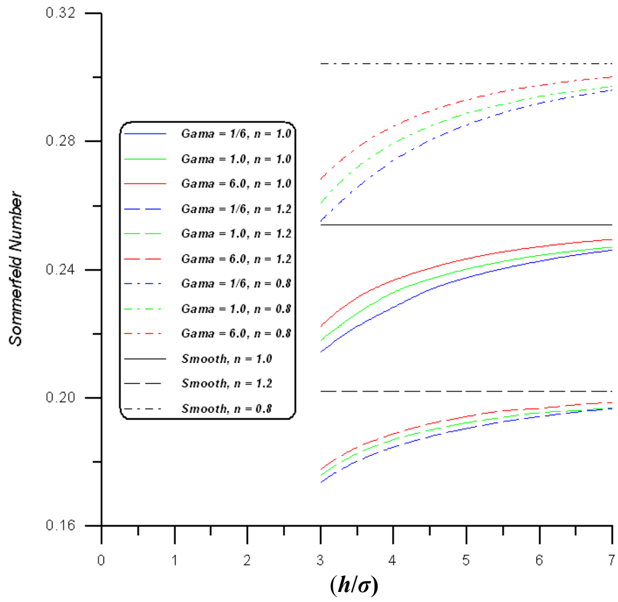

Fig.14 Sommerfeld Number versus (h/σ) for Stationary Surface Roughness in Misaligned Bearing using Newtonian (n = 1) and Non-Newtonian Lubricant (n = 1.2 and 0.8).

CONCLUSIONS

Thermohydrodynamic analysis of misaligned journal

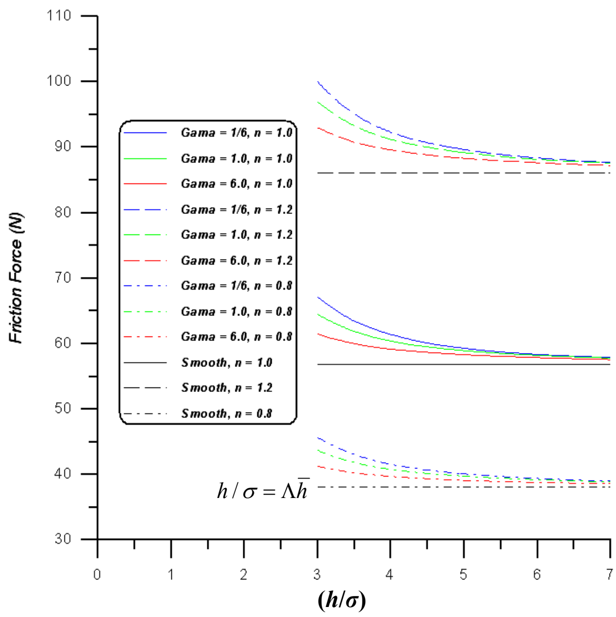

Fig.16 Friction Force versus (h/σ) for Stationary Surface Roughness in Misaligned Journal Bearing using Newtonian and Non-Newtonian Lubricant.

bearing with rough surface when lubricated with non-Newtonian lubricant has been implemented.The results obtained through this work leads to the following conclusions can be remarked:

1) The surface roughness characteristics of opposing surfaces (*i.e.*, variance ratio, Vr_j) and roughness orientations (γ) play an important role in reducing or increasing the performance parameters of the journal bearing.

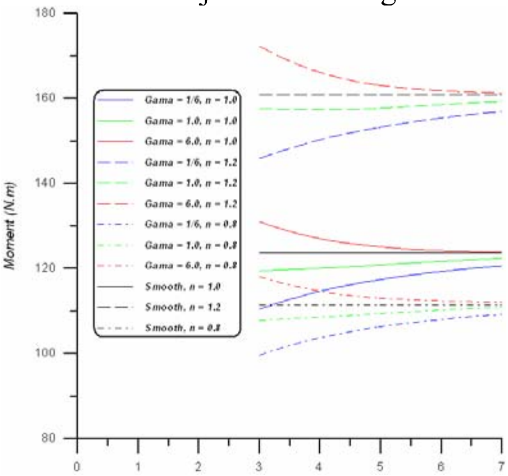

Fig.17 Moment versus (h / σ) for Moving Surface Roughness in Misaligned Journal Bearing using Newtonian ($n = 1$) and Non-Newtonian Lubricant ($n = 1.2$ and 0.8)

2) Combination of stationary roughness with a transverse surface roughness pattern provides maximum reduction in fluid – film temperature while it produces maximum increase in fluid – film pressure and load carrying capacity of the bearing. Contrary happens with moving surface roughness.

3) The combined effect of bearing misalignment and surface roughness in journal bearing leads to decrease each of load, pressure, temperature, friction force and mo- ment as compared with that of aligned journal bearing .

4) The non-Newtonian effect of lubricant has a significant effect on the thermohydrodynamic performance of journal bearing. Lubricant with higher values of power index ($n > 1.0$) causes to increase each of the values of bearing characteristics except of the sommerfeld number. These values decrease when the lower viscous lubricant index ($n < 1.0$) is used.

5) The effect of surface roughness on the performance parameters is more pronounce with decrease the value of roughness variable (h/σ). This effect is greatly affected by the rotational speed, non-Newtonian behavior and shaft misalignment.

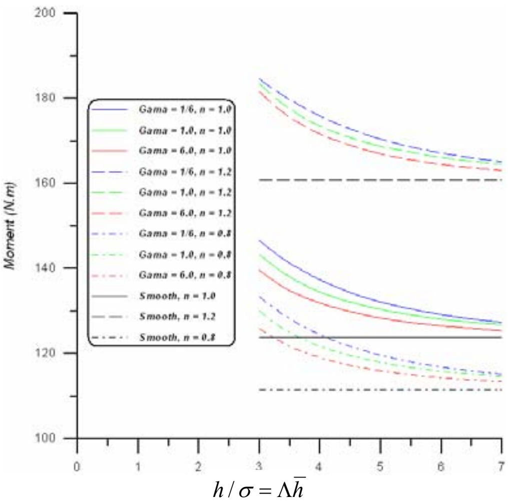

Fig.18 Moment versus (h / σ) for Stationary Surface Roughness in Misaligned Journal Bearing using Newtonian ($n = 1$) and Non-Newtonian Lubricant ($n = 1.2$ and 0.8)

6) There exists an interactive influence between surface roughness, thermal, journal isalignment and non- Newtonian behavior of lubricant effects on thermohydrodynamic performance of the bearing, and hence, these effects should be studied to gather.

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NOMENCLATURE

D : Diameter of journal (m)

f_r : Friction force (N)

i, j, k: indexes along x,y,z axes respectively.

M :moment of the bearing. (N.m)

n : power law index.

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

$p = p/\mu_{in} U \cdot c^2 / R$

S_O: Sommerfeld number

T : oil film temperature (C^o)

$\bar{T} = T/T_{in}$

T_s: Journal (shaft) temperature (C^o)

T_r: Recirculation oil temperature (C^o)

T_{mix}: Mixing oil temperature (C^o)

U : journal (shaft) speed (m/s)

Vr_J : journal rough surface variance ratio $= (\sigma_j/\sigma)^2$

Vr_b : bearing rough surface variance ratio $= (\sigma_b/\sigma)^2$

Vr : Combined variance ratio $= \left[(V_{rJ})^2 + (V_{rb})^2 \right]^{0.5}$

GREEK SYMBOLS

γ : Surface roughness parameter $= \lambda_{0.5x}/\lambda_{0.5z}$

γ_j, γ_b : Surface roughness parameter for journal and bearing surfaces respectively.

δ_j, δ_b : Random roughness amplitude of the journal and the bearing surfaces respectively. (μm)

Δ : Combined roughness height (μm)

ε : eccentricity ratio.

$\lambda_{0.5x}, \lambda_{0.5z}$ = 0.5 correlation lengths of the x and z profile (μm).

μ : Lubricant viscosity (Pa.sec)

μ_{in} : Inlet viscosity (Pa.sec)

$\bar{\mu}$:Dimensionless lubricant viscosity $= \mu/\mu_{in}$

μ_{non} : Dimensionless viscosity of non-Newtonian lubricant.

σ_j, σ_b : Standard deviation of δ_j, δ_b for rough surfaces (μm).

σ : standard deviation of the combined roughness $= \sqrt{\sigma_j^2 + \sigma_b^2}$

τ : Hydrodynamic shear stress N/m^2