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Effect of Viscosity Variation on the Static Performance of Short Hydrodynamic Journal Bearing Operating with Couple Stress Fluids under Turbulent Flow Regime

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ABSTRACT: *A theoretical study of the combined effects of turbulence and viscosity variation in a hydrodynamic journal bearing lubricated with couple stress fluid is presented. To account for the couple stress effect arising from a lubricant blended with various additives Stokes microcontinuum constitutive equations are used. The viscosity-temperature relationship is replaced by a viscosity-oil film thickness relationship. Based on this modified Reynolds equation is derived for a bearing system, operating under turbulent flow condition. To obtain a closed form solution, the narrow bearing approximation is considered. According to the results obtained, the couple stresses and turbulence enhance the load carrying capacity and decrease the coefficient of friction. Where as the viscosity variation factor decreases the load carrying capacity and increases the friction parameter. The results are compared with the Newtonian fluid and laminar flow conditions.*
Keywords: *Hydrodynamic Journal Bearing, Couple Stress Fluid, Turbulence, Viscosity Variation.*

I. INTRODUCTION

Generally, studies of hydrodynamic behavior focus on the performance of bearings lubricated with a Newtonian viscous fluid. However recent experimental studies showed that the performance of lubricated contacts could be improved by blending the base oil with additives. Lubricants containing additives should be treated as non-Newtonian fluids. In the literature, two models are widely used to describe the non-Newtonian behavior of the lubricant in hydrodynamic condition. The first rheological model is the well-known power-law fluid model [1,2]. The second model consists of the couple stress model based on micro continuum theory derived by Stokes [3]. Stokes' theory is the simplest of fluids that allows for polar effects such as the presence of couple stress, body couples and non-symmetric tensors. This couple stress model is important for engineering and scientific applications of pumping fluids such as synthetic lubricants, colloidal fluids, liquid crystals and bio-fluids. Ramanaiah and Sankar [4] investigated the squeeze film behavior between rectangular plates lubricated by fluids with couple stress. Ramanaiah [5,6] predicted the effects of couple stress fluids on the squeeze film behavior between finite plates of various shapes and studied the performance of long slider bearing including the consideration of cavitation. Sinha and Singh [7] examined the effects of couple stresses on the lubrication of rolling contact bearings. J.R.Lin [8,9,10] analyzed the effect of couple stress on short bearing performance, on long partial bearing and on finite journal bearing characteristics. Abdullah and et al [11] estimated the eccentricity ratio and couple stress parameter for a given experimentally measured pressure distribution for finite journal bearings using an inverse solution. J.R.Lin and et al [12] analyzed the effect of couple stresses in the cyclic squeeze films of finite partial bearings. X.L.Wang and et al [13] studied numerically the performance of the dynamically loaded journal bearing lubricated with couple stress fluids. N.B. Naduvinamani, P.S. Hiremath and G.Gurubasevaraj [14,15,16] studied the rehological effects of couple stress fluids on the performance of squeeze film porous journal bearings, and the effect of surface roughness with couple stress fluids. Similarly J-R. Lin, R-F.Lin and T-B. Chang [17] derived the dynamic couple stress Reynolds equation of sliding - squeezing surfaces and applied to the plane inclined slider bearing.

A.A.Elshakawy [18] utilized the both porous media model and the couple stress model to study the effects of lubricant additives on the performance of hydro dynamically lubricated journal bearings.

It is well recognized that the steady - state characteristics under consideration in designing a bearing system, are with laminar flow assumption. However under certain operating conditions like high speeds or high clearance or when the viscosity of the lubricant is sufficiently low, the oil flow will change from laminar to turbulent. A turbulent fluid film affects the performance beatings in a number of ways. There are five distinct turbulence models based on (i) mixing length theory (ii) eddy viscosity (iii) the mid-channel velocity (iv) the bulk fluid theory and (v) the kinetic energy of turbulence are available to deal with the turbulent lubricant films. Several authors have reported numerous experimental and analytical studies of bearings operating beyond the laminar flow regime in the past few years. Vinay kumar [19] critically reviewed the literature on plain hydrodynamic journal bearings in the turbulent regime. G.Jayachandra reddy and et al., [20] studied the effect of couple stress on the performance of a short journal bearing system operating under turbulent flow regime.

Earlier theories were based on the assumption that the viscosity μ was constant although it is a function of both pressure and temperature. The viscosity of all liquids, and particularly of hydrocarbon lubricants, decrease with increasing temperature. This variation in viscosity with temperature is important in many practical applications where lubricants are required to function over a wide range of temperature [21]. There is no fundamental mathematical relationship, which will accurately predict the variation in the viscosity of oil with temperature. The formulae proposed for defining the viscosity – temperature relationship are purely empirical, and for accurate calculations the lubrication engineer requires experimental data. Generally, it is assumed that thermal equilibrium exits and that the viscosity varies with the temperature according to a given law. For practical application of the law, the temperature at each point should be known and this would require a complete thermal calculation. However, a viscosity-temperature relationship can be replaced by a viscosity-film thickness relationship as it has been verified experimentally that the highest temperature occurs in zones where the film thickness is lowest [22]. When the viscosity $\mu_1 \; at \; h = h_1$ (oil inlet condition) is known, then

$$\mu = \mu_1 \left(\frac{h}{h_1} \right)^Q$$

Where Q, usually lies between 0 and 1. (According to the nature of the lubricant.). It is further assumed that the new material constant η for couple stress fluids also vary similar to the μ [23]. P.sinha and et.al [24] studied the effect of viscosity variation in journal bearings lubricated with micropolar fluids. D.F.Wilcock and O.Pinkus [25] examined the influence of turbulence and variable viscosity on the dynamic properties of journal bearings by assuming that the viscosity is to vary exponentially with the temperature. G.jayachandra reddy and et.al [26] analyzed the effect of viscosity variation on the static performance of a narrow journal bearing operating with couple stress fluids. The present paper predicts theoretically the effect of viscosity variation on static characteristics of a short bearing operating with couple stress fluid under turbulent flow regime.

II. THEORETICAL ANALYSIS

Fig.1.represents the physical configuration of a short hydrodynamic journal bearing. The film thickness is expressed by the following continuous function

$$h = c + e\cos\theta \qquad (1)$$

where c=radial clearance ,

e= eccentricity and θ = circumferential angle of the bearing.

Modified Reynolds equation for couple stress fluid bearing operating under turbulent flow regime is [14]

$$\frac{\partial}{\partial x}\left[\frac{h^3}{K_X} f(h,l)\frac{\partial p}{\partial x}\right] + \frac{\partial}{\partial z}\left[\frac{h^3}{K_Z} f(h,l)\frac{\partial p}{\partial z}\right] = \frac{U\mu}{2}\frac{dh}{dx} \qquad (2)$$

$K_X = 12 + 0.0260 \; Re^{0.8625}$

$K_Z = 12 + 0.0198 \; Re^{0.741}$

Re = Reynolds Number = $\dfrac{\rho U h}{\mu}$

For laminar flow $K_X = 12$, $K_Z = 12$ i.e., Re=0

Where h = the fluid film thickness.

$l = \left(\frac{\eta}{\mu}\right)^{\frac{1}{2}}$ = The characteristic length of additives in a Newtonian lubricant.

η is the material constant for the couple stress , μ is the viscosity coefficient, and

U = Journal velocity.

$$f(h,l) = 1 - 12\frac{l^2}{h^2} + 24\frac{l^3}{h^3}\tanh\left(\frac{h}{2l}\right) \qquad (3)$$

Now it is assumed that, the Newtonian viscosity μ is varying along the fluid film thickness h according to [22]

$$\mu = \mu_1 \left(\frac{h}{h_1}\right)^Q \qquad (4)$$

where μ_1 is the inlet viscosity at $h = h_1 = c(1+\varepsilon)$. The exponent Q may be determined using the relation:

$$Q = \frac{\log\left(\mu_1 / \mu_2\right)}{\log\left(h_1 / h_2\right)} \qquad (5)$$

Where μ_2 is the outlet viscosity with film thickness h_2.

The parameter Q $(0 \le Q \le 1)$ depends on the particular lubricant used: for perfect Newtonian fluids $Q = 0$, whereas for perfect gasses $Q = 1$. for mathematical simplicity the couple stress parameter l is assumed to be independent of viscosity variation. This can be done by assuming that η is varying in the same way as μ.

III. NARROW OR SHORT JOURNAL BEARING ANALYSIS:

In order to simplify the problem and to obtain a closed form solution for the fluid pressure a narrow bearing approximation is assumed. Since $\lambda^2 << 1$ for narrow bearing approximation, the circumference variations of pressure can be neglected as compared to the axial variation. Then the modified Reynolds equation reduces to

$$\frac{\partial}{\partial z}\left[\frac{h^3}{K_Z} f(h,l)\frac{\partial p}{\partial z}\right] = \frac{U\mu}{2}\frac{dh}{dx} \qquad (6)$$

and the boundary conditions are

$$p = 0 \quad at \quad z = \pm L/2$$
$$\frac{dp}{dz} = 0 \quad at \quad z = 0 \qquad (7)$$

The first boundary condition represents the symmetric pressure distribution in the z-direction. The second boundary condition is for the ends of the bearing.

Substituting $x = R\theta$, $\varepsilon = \frac{e}{c}$, $h = c(1+\varepsilon\cos\theta)$,

$$\mu = \mu_1 \left(\frac{h}{h_1}\right)^Q, \text{ and } h_1 = c(1+\varepsilon) \quad i.e., h \text{ at } \theta = 0 \qquad (8)$$

and integrating the above equation(6) and applying the boundary conditions

$$p = \frac{-k_z \mu_1 U \varepsilon \sin\theta}{4R c^2 (1+\varepsilon)^Q (1+\varepsilon\cos\theta)^{3-Q} f(l,h)} \left[\frac{L^2}{4} - z^2\right] \qquad (9)$$

Introducing the non dimensional variables

$$z^* = \frac{z}{L} \quad , \lambda = \frac{L}{2R} \quad , \quad l^* = \frac{l}{c} \quad , \quad p^* = \frac{pc^2}{\mu_1 UR} \qquad \text{and}$$

$$f^*(l^*, h^*) = 1 - 12\frac{l^{*2}}{h^{*2}} + 24\frac{l^{*3}}{h^{*3}}\tanh(\frac{h^*}{2l^*}) \qquad (10)$$

the non dimensional fluid film pressure becomes

$$p^* = \frac{-k_z \lambda^2 \varepsilon \sin\theta}{4(1+\varepsilon)^Q (1+\varepsilon\cos\theta)^{3-Q} f^*(l^*, h^*)} \left[\frac{1}{4} - z^{*2}\right] \qquad (11)$$

at the middle of the bearing i.e., Z^*=0.

$$p^* = \frac{k_z \lambda^2 \varepsilon \sin\theta}{4(1+\varepsilon)^Q (1+\varepsilon\cos\theta)^{3-Q} f^*(l^*, h^*)} \qquad (12)$$

where $k_z = 12 + 0.0198 R_e^{0.741}$

IV. BEARING STATIC CHARACTERISTICS:

Once the fluid pressure is known from the above equation, the bearing static Characteristics can be obtained.

4.1 Load carrying capacity: The load carrying capacity can be calculated by integrating the film pressure acting on the journal surface, along and perpendicular to the line of centers. The components W_r and W_t are given respectively by

$$W_r = W\cos\phi = -2R \int_{\theta=0}^{\theta=\pi} \int_{z=0}^{z=1/2} p\cos\theta \, dz \, d\theta \qquad (13)$$

$$W_t = W\sin\phi = 2R \int_{\theta=0}^{\theta=\pi} \int_{z=0}^{z=1/2} p\sin\theta \, dz \, d\theta \qquad (14)$$

Where W is the load carrying capacity of the bearing and is given by

$$W = \sqrt{\left[W_r^2 + W_t^2\right]} \qquad (15)$$

and let the non-dimensional form can be

$$W_r^* = W^* \cos\phi$$

$$W^* = \frac{W c_H^2}{\mu U L R^2} \qquad = -2\lambda^2 \int_{\theta=0}^{\theta=\pi} \frac{(1+0.00615\, R_e^{\,0.741})}{(1+\varepsilon)^Q\,(1+\varepsilon\cos\theta)^{(3-Q)}}\,\frac{\varepsilon\sin\theta}{f^*(h^*,l^*)}\cos\theta\; d\theta$$

(16)

$$W_t^* = W^* \sin\phi$$

$$= 2\lambda^2 \int_{\theta=0}^{\theta=\pi} \frac{(1+0{,}00615 R_e^{\,.741}}{(1+\varepsilon)^Q\,(1+\varepsilon\cos\theta)^{(3-Q)}}\,\frac{\varepsilon\sin\theta}{f^*(h^*,l^*)}\sin\theta\; d\theta$$

(17)

4.2.Attitude angle*:* It is the angle between the line passing through the centers and the

load acting direction, and is given by

$$\phi = \tan^{-1}\left[\frac{W_t^*}{W_r^*}\right] \qquad (18)$$

4.3.*Friction parameter*: The friction force acting on the journal can be obtained by

Integrating the shear stress around the journal [27]

$$F' = 2R \int_{z=0}^{z=1/2} \int_{\theta=0}^{\theta=\pi} \tau\, d\theta\, dz \qquad (19)$$

Where the shear stress

$$\tau = \mu \frac{\partial u}{\partial y}\Big|_{y=h} - \eta \frac{\partial^2 u}{\partial y^2}\Big|_{y=h} \qquad (20)$$

and the friction parameter is given by

$$f\,\frac{R}{c} = \frac{2\pi}{\left[1-\varepsilon^2\right]^{1/2} W^*} \qquad (21)$$

The load carrying capacity in eq(16), eq(17) cannot be obtained by direct integration, but it can be numerically evaluated by Gaussian quadrature [28]. Then the attitude angle of the bearing and the friction parameter can be calculated.

V. RESULTS AND DISCUSSION

Selection of design parameters: (i) Length to diameter ratio (λ): For narrow bearing approximation $\lambda^2 \leq 1$. The value of this length to diameter ratio in the present analysis is 0.25. (ii) Eccentricity ratio(ε): The eccentricity ratio range in practice is 04.-0.6.
Minimum eccentricity ratio 0.2 and Maximum is 0.8 in steps 0f 0.2 has been taken for analysis. (iii) Couple stress parameter (l^*): The couple stress fluid is characterized by this non-dimensional parameter l^*. It may be defined as the characteristics of the interaction of the fluid with the bearing geometry. The value of this couple stress parameter depends upon the characteristic material length of the polar suspensions l or the radial

clearance c. As the couple stress parameter approaches to zero, the problem reduces to the non-Newtonian case. The values of l^* are 0.0,0.2,0.4, and 0.6 are taken in the present analysis.(iv) Viscosity variation factor (Q): It is usually lies between 0 and 1 according to the nature of the lubricant. Numerical values of 0.0,0.2,0.4,0.6,0.8 and 1.0 are assumed for the Q in order to discuss the effects of viscosity variation in the present analysis.(v)Reynolds number (Re) in real bearings a combination of couette and poiseuille flow. To study the performance of bearings in turbulent flow regime, Reynolds number varying from zero to 10,000 has been selected for analysis purpose. Putting zero to Reynolds number refer to the laminar flow condition.

Fig.2. Shows the dimensionless fluid pressure p^* as a function of circumferential coordinate θ at the mid-plane of the bearing i.e. $z^* = 0.0$. for different Reynolds number. The operating conditions are, eccentricity ratio as 0.6, couple stress parameter as 0.6 and the viscosity variation parameter as 0.6. The broken curve refers to the Newtonian case and the Reynolds number zero refers to the laminar flow condition. As the Reynolds number increases the dimensionless pressure also increases. The magnitude is significantly high compared to the Newtonian fluid case.

Fig.3. Represents the dimensionless fluid film pressure verses bearing circumferential angle with different viscosity variation for different flows. In both laminar and turbulent flow conditions, as the viscosity variation parameter increases the dimensionless pressure decreases. The decrement is more pronounced in turbulent flow condition.

Fig.4. Fig.5 and Fig.6 refer to the load carrying capacity verses eccentricity ratio, couple stress parameter and Reynolds number respectively, with viscosity variation. As the eccentricity ratio increases dimensionless load also increases. The viscosity variation parameter has no effect at the low eccentricity ratios. The load carrying capacity decreases with the increase in the viscosity variation parameter due to the decrement in the pressure. Couple stress parameter increases the dimensionless load. As the Reynolds number increases the dimensionless load also increases, but at the higher numbers the value is almost same.

Fig.7, Fig.8. and Fig 9. Represent the attitude angle and eccentricity ratio, couple stress parameter and Reynolds number respectively, with viscosity variation. In all the cases as the viscosity variation parameter increases the attitude angle also increases. As the eccentricity ratio increases the attitude angle decreases. The increase in the couple stress decreases the attitude angle. Reynolds number has no effect on the attitude angle.

Fig.10, Fig.11. and Fig.12. Show the friction parameter verses eccentricity ratio, couple tress parameter and Reynolds number respectively with viscosity variation. The friction parameter decreases with the increase in the eccentricity ration or couple stress parameter or Reynolds number. The viscosity variation has no effect on the friction parameter at the low eccentricity ratios. At the higher Reynolds number the decrement in the friction parameter is less. As the viscosity variation parameter increases the friction parameter also increases

VI. Conclusions

According to the results discussed above the conclusions can be drawn as follows:

1.Couple stress parameter has a significant effect on the bearing static characteristics. There is an enhancement in the load carrying capacity and reduction in the attitude angle and friction parameter.

2.Increasing the value of Viscosity variation factor signify a decrease in viscosity, which may be a consequence of temperature rise. Hence as temperature increases load carrying capacity decreases and the coefficient of friction increases.

3.Turbulent flow withstands more loads and reduces the friction than laminar flow.

4.The net viscosity variation effect in bearing operating with couple stress fluids under turbulent flow regime is that there is an enhancement in the load carrying capacity and decrease in the attitude angle and coefficient of friction as compared to the Newtonian fluids.

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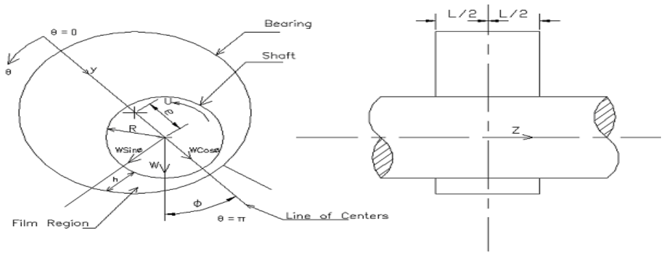

Fig : 1. Journal Bearing Physical Configuration

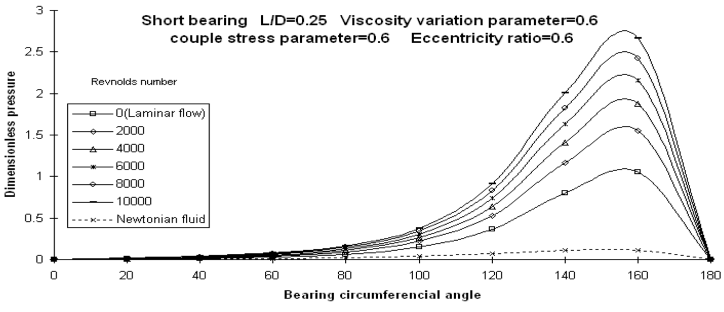

Fig.2 Bearing circumferencial angle vs Dimensionless pressure foe different flows

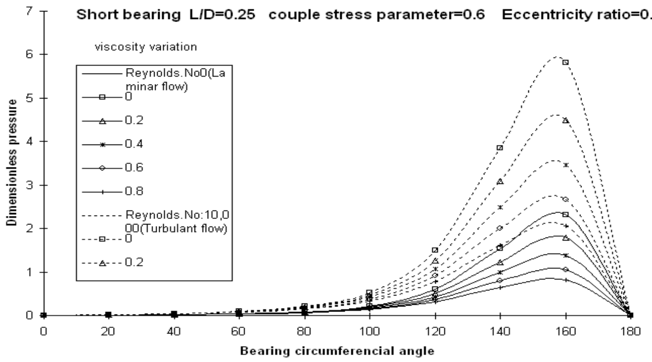

Fig.3 Bearing circumferencial nagle vs Dimensionless pressure with different viscosity variation for different flow rates

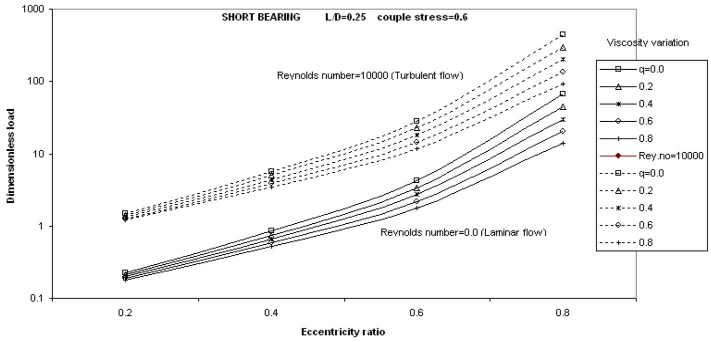

Fig.4 Eccentricity ratio vs Dimensionless load with viscosity variation for different flows

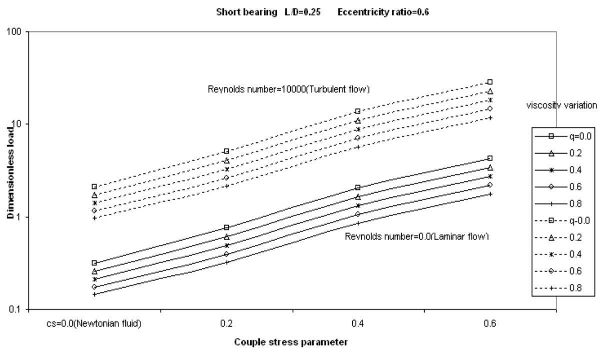

Fig.5 Couple stress parameter vs Dimensionless load with different viscosity variation at different flow regimes

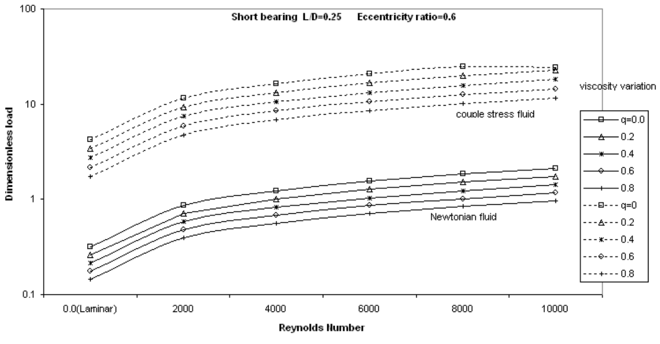

Fig.6 Reynolds number vs Dimensionless load with viscosity variation

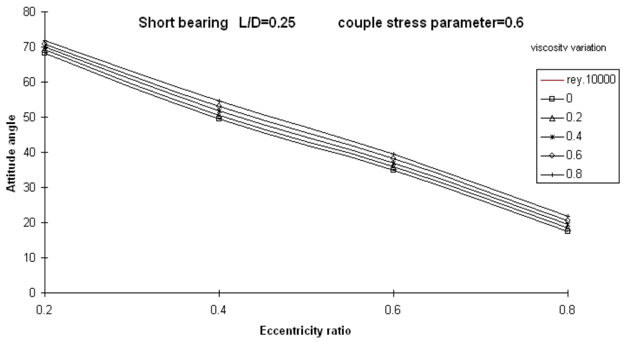

Fig.7 Eccentricity ratio vs Attitude angle with viscosity variation for different Reynolds number

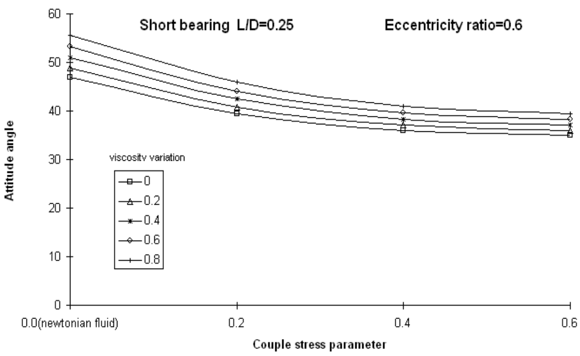

Fig.8 Couple stress parameter vs Attitude angle with viscosity variation

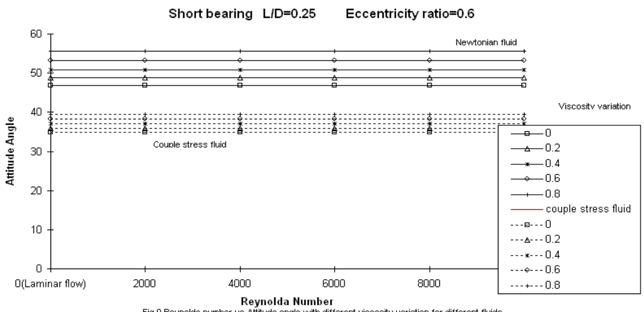

Fig.9 Reynolds number vs Attitude angle with different viscosity variation for different fluids

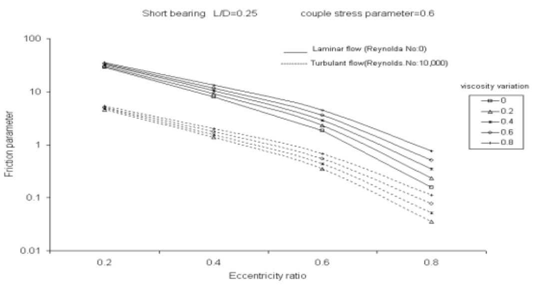

Fig.10 Eccentricity ratio vs Friction parameter with different viscosity variation for diffferent flow rates

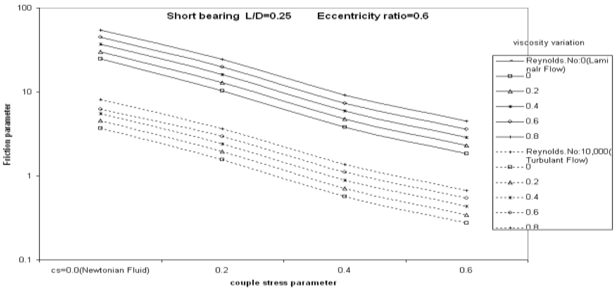

Fig.11 Couple stress parameter vs Friction parameter with viscosity variation for different flows

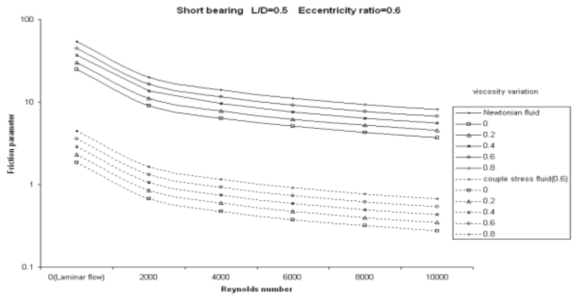

Fig.12 Reynolds number vs Friction parameter with different viscosity variation for different fluids