Steady State Characteristics of Finite Oil Journal Bearingsconsidering Fluid Inertia effect and Influence of pressure dependent variable viscosity

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Abstract: *This theoretical work describes the influence of fluid-inertia effects on performance characteristics of finite journal bearingconsidering with pressure dependent viscosity. The theoretical analysis is intended to show the effect of fluid inertia on the journal bearing performance for three-dimensional bearing geometries.The average Reynolds equation is modified to include the fluid inertia effect and variable viscosity effect and is used to obtain pressure field in the fluid-film. The solutions of modified average Reynolds equations are obtained using finite difference method and appropriate iterative schemes. The effects of circumferential fluid-film pressure distribution, load carrying capacity of the bearing are studied by considering fluid-inertia effects.The steady state bearing performance analysis is done through parametric study of the various variables like modified Reynolds number, eccentricity ratio, slenderness ratio, attitude angle, Viscosity Parameter.The variation of bearing load carrying capacity, attitude angle, has been studied and plotted against various parameters.It has been found from the analysis that the steady state load carrying capacity increases with eccentricity ratio as well as with modified Reynolds number as viscosity parameter increases.It is also observed that load carrying capacity is higher for the effect of variable viscosity than Isoviscous lubricants.*

Keyword- *modified Reynolds number, slenderness ratio, attitude angle, sommerfeld number, eccentricity ratio, journal bearings, inertia, and Viscosity Parameter.*

I INTRODUCTION

The basic assumptions in the classical hydrodynamic theory include negligible fluid inertia forces in comparison to the viscous forces.Pinkus and Sterlincht[2], have shown that the fluid inertia effect cannot be neglected when the viscous and the inertia forces are of the same order of magnitude.In recent times synthetic lubricants, low viscosity lubricants, are used in industries andowing to high velocity it is possible to arrive at such a situation where flow is laminar but the fluid inertia effect cannot be neglected, the classical Reynolds equation is not valid in such case.

Keeping in view of the above,consideration of inertia effect of a lubricant flow maybe one of the areas of recent extension of the classical lubrication theory.Among the few studies related to fluid inertiaeffects ,Constatinescu and Galetuse[1]evaluated the momentum equations for laminar and turbulent flows by assuming the velocity profiles is not strongly affected by the inertia forces . Banerjee et.al [3] introduced an extended form of Reynolds equation to include the effect of fluid inertia,adopting an iteration scheme.Chen and Chen[7] obtained the steady-state characteristics of finite bearings including inertia effect using the formulation of Banerjee et al.[3].Tichy and Bou-said[4] and Kakoty and Majumdar [5] used the method of average inertia in which inertia terms are integrated over the film thickness to account for the inertia effect in their studies.The above studies were mainly based on ideally smooth bearing surfaces.

In the present work,a modified average Reynolds equation and a solution algorithm are developed to include fluid inertia and pressure depended variable viscosity effects in the analysis of lubrication problems. The solutions of modified average Reynolds equations are obtained using finite difference method with appropriate iterative schemes.The developed model is used to study the influence of fluid inertia and variable viscosity effects on the steady state characteristics such as circumferential pressure,load carrying capacity,attitude angle and side leakage of a hydrodynamic finite oil journal bearing.

II BASIC THEORY

The modified average Reynolds equation for fully lubricated surfaces is derived starting from the Navier-Stokes equations and the continuity equation with few assumptions. The non-dimensional form of the momentum equations and the continuity equation for a journal bearing may be written as (Figure.1)

$$R^*_e\left[\Omega\,\frac{\partial \bar{u}}{\partial \tau}+\bar{u}\,\frac{\partial \bar{u}}{\partial \theta}+\bar{v}\frac{\partial \bar{u}}{\partial \bar{y}}+\bar{w}\left(\frac{D}{L}\right)\frac{\partial \bar{u}}{\partial \bar{z}}\right]=-\frac{\partial \bar{p}}{\partial \theta}+\frac{\partial^2 \bar{u}}{\partial \bar{y}^2} \quad (1)$$

$$\frac{\partial \bar{p}}{\partial \bar{y}}=0 \quad (2)$$

$$R^*_e\left[\Omega\,\frac{\partial \bar{w}}{\partial \tau}+\bar{u}\,\frac{\partial \bar{w}}{\partial \theta}+\bar{v}\frac{\partial \bar{w}}{\partial \bar{y}}+\bar{w}\left(\frac{D}{L}\right)\frac{\partial \bar{w}}{\partial \bar{z}}\right]=-\left(\frac{D}{L}\right)\frac{\partial \bar{p}}{\partial \bar{z}}+\frac{\partial^2 \bar{w}}{\partial \bar{y}^2} \quad (3)$$

$$\frac{\partial \bar{u}}{\partial \theta}+\frac{\partial \bar{v}}{\partial \bar{y}}+\left(\frac{D}{L}\right)\frac{\partial \bar{w}}{\partial \bar{z}}=0 \quad (4)$$

Where, $\bar{z}=\frac{z}{L/2}$, $\bar{y}=\frac{y}{c}$, $\theta=\frac{x}{R}$, $\tau=\omega_p.t$, $\Omega=\frac{\omega_p}{\omega}$,

$\bar{u}=\frac{u}{\omega R}$, $\bar{v}=\frac{v}{c\,\omega}$ $\bar{w}=\frac{w}{\omega R}$ $\bar{p}=\frac{p c^2}{\eta\,\omega R^2}$ and

$$R_e^*=R_e.\frac{c}{R}=\frac{\rho\,\omega c^2}{\eta}$$

The fluid film thickness can be given as

$$h=c+e\cos\theta \quad (5)$$

$$\bar{h}=1+\varepsilon\cos\theta \quad (6)$$

where, $\bar{h}=\frac{h}{c}$, $\varepsilon=\frac{e}{c}$,

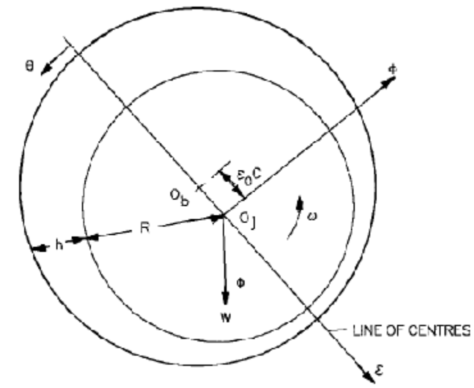

Fig.1 The schematic diagram of oil Journal Bearing

Here the variation in the density with time is considered to be negligible. The momentum equations may be presented in the following form using equation of continuity.However,the second momentum equation is not used any further because there is no variation in pressure across the film.After Constantinescu and Galetuse [1] the velocity components are approximated by the parabolic profiles. The velocity components may be expressed in non-dimensional form as follows:

$$\bar{u}=\left[\frac{\bar{y}}{\bar{h}}+Q_\theta\left(\frac{\bar{y}^2}{\bar{h}^2}-\frac{\bar{y}}{\bar{h}}\right)\right] \quad (7)$$

$$\bar{w}=\left[Q_z\left(\frac{\bar{y}^2}{\bar{h}^2}-\frac{\bar{y}}{\bar{h}}\right)\right] \quad (8)$$

Q_θ and Q_z are dimensionless flow parameter in θ and $\bar{z}$ direction respectively.Substituting these two into momentum equations and integrating give

$$Q_\theta=\frac{\bar{h}^2}{2}\left(\frac{\partial \bar{p}}{\partial \theta}\right)+R_e^*\times I_X \quad (9)$$

$$Q_z=\frac{\bar{h}^2}{2}\left(\frac{D}{L}\right)\left(\frac{\partial \bar{p}}{\partial \bar{z}}\right)+R_e^*\times I_Z \quad (10)$$

where,

$$I_X=\frac{\bar{h}}{2}\left[\begin{array}{l}\frac{1}{2}\Omega\left(1-\frac{1}{3}Q_\theta\right)\frac{\partial \bar{h}}{\partial \tau}-\frac{1}{6}\Omega\,\bar{h}\,\frac{\partial Q_\theta}{\partial \tau}-\frac{1}{3}\left(1-\frac{1}{2}Q_\theta+\frac{1}{10}Q_\theta^{\,2}\right)\frac{\partial \bar{h}}{\partial \theta}+\\ \frac{1}{3}\bar{h}\left(\frac{1}{5}Q_\theta-\frac{1}{2}\right)\frac{\partial Q_\theta}{\partial \theta}+\frac{1}{30}\left(\frac{D}{L}\right)\bar{h}\,Q_z\frac{\partial Q_\theta}{\partial \bar{z}}+\frac{1}{6}\left(\frac{D}{L}\right)\bar{h}\left(\frac{1}{5}Q_\theta-\frac{1}{2}\right)\frac{\partial Q_z}{\partial \bar{z}}\end{array}\right] \quad (11)$$

$$I_Z=\frac{\bar{h}}{2}\left[\begin{array}{l}-\frac{1}{6}\Omega\,Q_z\frac{\partial \bar{h}}{\partial \tau}-\frac{1}{6}\Omega\bar{h}\frac{\partial Q_z}{\partial \tau}-\frac{1}{6}Q_z\left(\frac{1}{5}Q_\theta-\frac{1}{2}\right)\frac{\partial \bar{h}}{\partial \theta}-\\ \frac{1}{6}\bar{h}\left(\frac{1}{5}Q_\theta-\frac{1}{2}\right)\frac{\partial Q_z}{\partial \theta}+\frac{1}{30}\bar{h}Q_z\frac{\partial Q_\theta}{\partial \theta}+\frac{1}{15}\left(\frac{D}{L}\right)\bar{h}\,Q_z\frac{\partial Q_z}{\partial \bar{z}}\end{array}\right] \quad (12)$$

Considering case of variable viscosity it has been observed oils viscosity increases with pressure and the following relationship is assumed similar to Majumdar et.al. [9],

$\eta = \eta_0 e^{\alpha p}$ (13)

Where α = piezo viscosity co-efficient, η_0 Viscosity of oil at the inlet condition

Assuming modified pressure function q we get

$q = \frac{1}{\alpha}(1 - e^{-\alpha p})$ (14)

Neglecting time depended terms one can obtain the following form of modified Reynold's equation for steady state conditionconsidering fluid inertia effect.

$$\frac{\partial}{\partial\theta}\left(\bar{h}_0^{\,3}\frac{\partial \bar{q}_0}{\partial\theta}\right) + \left(\frac{D}{L}\right)^2 \frac{\partial}{\partial \bar{z}}\left(\bar{h}_0^{\,3}\frac{\partial \bar{q}_0}{\partial \bar{z}}\right) =$$

$$6\frac{\partial \bar{h}_0}{\partial\theta} - 2\times R_e^* \times \left[\frac{\partial}{\partial\theta}\left(\bar{h}_0 \times I_X\right) + \left(\frac{D}{L}\right)\frac{\partial}{\partial \bar{z}}\left(\bar{h}_0 \times I_z\right)\right] \quad (15)$$

Where, $\bar{q}_0 = \frac{q_0 c^2}{\eta_0 \omega R^2}$ and

$$Q_\theta = \frac{\bar{h}^2}{2}\left(\frac{\partial \bar{q}_0}{\partial\theta}\right) + R_e^* \times I_X \quad (16)$$

$$Q_z = \frac{\bar{h}^2}{2}\left(\frac{D}{L}\right)\left(\frac{\partial \bar{q}_0}{\partial \bar{z}}\right) + R_e^* \times I_Z \quad (17)$$

$$I_X = \frac{\bar{h}_0}{2}\left[\begin{array}{l} -\frac{1}{3}\left(1 - \frac{1}{2}Q_\theta + \frac{1}{10}Q_\theta^2\right)\frac{\partial \bar{h}_0}{\partial\theta} + \frac{1}{3}\bar{h}_0\left(\frac{1}{5}Q_\theta - \frac{1}{2}\right)\frac{\partial Q_\theta}{\partial\theta} + \\ \frac{1}{30}\left(\frac{D}{L}\right)\bar{h}_0 Q_z \frac{\partial Q_\theta}{\partial \bar{z}} + \frac{1}{6}\left(\frac{D}{L}\right)\bar{h}_0\left(\frac{1}{5}Q_\theta - \frac{1}{2}\right)\frac{\partial Q_z}{\partial \bar{z}} \end{array}\right] \quad (18)$$

and

$$I_Z = \frac{\bar{h}_0}{2}\left[\begin{array}{l} -\frac{1}{6}Q_z\left(\frac{1}{5}Q_\theta - \frac{1}{2}\right)\frac{\partial \bar{h}_0}{\partial\theta} - \frac{1}{6}\bar{h}_0\left(\frac{1}{5}Q_\theta - \frac{1}{2}\right)\frac{\partial Q_z}{\partial\theta} + \\ \frac{1}{30}\bar{h}_0 Q_z \frac{\partial Q_\theta}{\partial\theta} + \frac{1}{15}\left(\frac{D}{L}\right)\bar{h}_0 Q_z \frac{\partial Q_z}{\partial \bar{z}} \end{array}\right] \quad (19)$$

Boundary conditions for equation (15) are as follows

1. The pressure at the ends of the bearing is assumed to be zero (ambient): $\bar{q}_0(\theta, \pm 1) = 0$
2. The pressure distribution is symmetrical about the mid-plane of the bearing: $\frac{\partial \bar{q}_0}{\partial \bar{z}}(\theta, 0) = 0$
3. Cavitation boundary condition is given by: $\frac{\partial \bar{q}_0}{\partial\theta}\left(\theta_2, \bar{z}\right) = 0$ and $\bar{q}_0\left(\theta, \bar{z}\right) = 0$ for $\theta \geq \theta_2$

Where θ_2 is the angular coordinate at which the film cavitates.

Now from equation (14) the p_0 can be written in terms of q_0 as $p_0 = -\frac{\ln(1 - \alpha q_0)}{\alpha}$

After non-dimensionalised the relation can be given as

$\bar{p}_0 = \frac{\ln(1 - B\bar{q}_0)}{B}$ (20)

Where, viscosity parameter, $B = \frac{\eta_0 \alpha \omega R^2}{c^2}$

The equations(15), (16), (17), (18) and (19) are first expressed in finite difference form and solved using Gauss-Siedel method in a finite difference scheme.

III METHOD OF SOLUTION

To find out steady-state pressure all the time derivatives are set equal to zero and Equations. (15), (16), (17), (18) and (19) are solved simultaneously.For $\varepsilon \leq 0.2$ the pressure distribution and flow parameters $Q_\theta$and Q_z are evaluated from inertia less (i.e, $\mathrm{Re}^* = 0$) solution, i.e., solving classical Reynold's equation. These values are then used as initial value of flow parameters to solve Eqs.(16) and (17) simultaneously for $Q_\theta$and Q_zUsing Gauss-Siedel method in a finite difference scheme. Then update I_x & I_z and then calculate $Q_\theta$and Q_z for use to solve Eq.(15) with particular surface roughness pattern $(\gamma)$and surface roughness parameter $(\Lambda)$for new pressure p with inertia effect by using a successive over relaxation scheme. The latest values of Q_θ, $Q_z$and p are used iteratively to solve the set of equations until all variables converges. The convergence criterion adopted for pressure is $[1-(\Sigma p_{new}/\Sigma p_{old})] \leq 10^{-5}$ and also same criterion for $Q_\theta$and Q_z. For higher eccentricity ratios ($\varepsilon > 0.2$) the initial values for the variables are taken from the results corresponding to the previous eccentricity ratios. Very small increment in ε is to be provided as Re^* increases. The procedure converges up to a value of $\mathrm{Re}^*=1.5$ which should be good enough for the present study. After getting $\bar{q}_0$ solving the above equations simultaneouslythe $\bar{q}_0$ is then converted to $\bar{p}_0$ using equation (20) to calculate fluid film pressures.

Since the bearing is symmetrical about its central plane ($\bar{z}$=0),only one half of the bearing needs to be considered for the analysis, Once the pressure distribution is evaluated fluid film forces and the load bearing capacity $\bar{W}_o$ and attitude angle (φ) are calculated as follows:

A. *Fluid film forces*

The non-dimensional fluid film forces along line of centres and perpendicular to the line of centres are given by

$$\bar{F}_{r_0} = \int_0^1 \int_{\theta_1}^{\theta_2} \bar{p} \cos\theta \; d\theta \, d\bar{z} \quad (21)$$

$$\bar{F}_{\phi_0} = \int_0^1 \int_{\theta_1}^{\theta_2} \bar{p} \sin\theta \; d\theta \; d\bar{z} \quad (22)$$

Where, $\bar{p} = \dfrac{pc^2}{\eta_0 \, \omega \, R^2}$, $\bar{z} = \dfrac{z}{L/2}$, $\bar{F}_{r_0} = \dfrac{F_{r_0} \; c^2}{\eta_0 \; \omega \; R^3 \; L}$,

and $\bar{F}_{\phi_0} = \dfrac{F_{\phi_0} \; c^2}{\eta_0 \; \omega \; R^3 \; L}$, and θ_1 and θ_2 are angular coordinates

at which the fluid film commences and cavitates respectively.

B. *Steady state load*

The steady state non-dimensional load and attitude angle are given by

$$\bar{W}_0 = \sqrt{\left(\bar{F}_{r_0}{}^2 + \bar{F}_{\phi_0}{}^2 \right)} \quad (23)$$

$$\phi_o = \tan^{-1}\left(\frac{-\bar{F}_{\phi_0}}{\bar{F}_{r_0}} \right) (24)$$

Since the steady state film pressure distribution has been obtained at all the mesh points, then fluid film forces can be evaluated by integrating equations(21) and (22) numerically by using Simpson's 1/ 3 rd. rule to get $\bar{F}_r$ and $\bar{F}_\phi$. The steady state load $\bar{W}_0$ and the attitude angle ϕ_0 are calculated using equations (23) and (24).

The present steady state results (considering only fluid inertia effect and Iso viscous flluid) are compared to the results of Kakoty et.al., [5] and Chen & Chen [7] (for L/D = 1.0) as given in Table 1. These three results are in good agreement.

The theoretical study has been done considering Inertia effects variable viscosity effect.The results have been compared with available data of researchers.

Table 1: Comparison of Steady -state characteristics of a finite Iso viscousoil journal bearings for L/D=1 with fluid Inertia Effect.

Re*	ε_0	$\bar{W}_0$	$\bar{W}_0$	$\bar{W}_0$	ϕ_0	ϕ_0	ϕ_0
		Present	**Kakoty**	**Chen-Chen**	**Present**	**Kakoty**	**Chen-Chen**
0	**0.2**	**0.5031**	**0.5042**	**0.501**	**77.541**	**73.710**	**73.900**
	0.5	**1.7592**	**1.7903**	**1.779**	**58.398**	**56.640**	**56.800**
	0.8	**7.0854**	**7.4597**	**7.146**	**36.345**	**34.660**	**36.200**
	0.9	**16.891**	**17.713**	**16.982**	**26.175**	**23.900**	**26.400**
0.28	**0.2**	**0.5043**	**0.5055**	**0.504**	**77.927**	**73.750**	**74.200**
	0.5	**1.7940**	**1.7980**	**1.785**	**57.725**	**56.720**	**57.000**
	0.8	**7.1942**	**7.4837**	**7.151**	**36.310**	**34.720**	**36.300**
	0.9	**17.043**	**17.761**	**16.993**	**26.240**	**23.930**	**26.400**
0.56	**0.2**	**0.5055**	**0.5070**	**0.505**	**78.311**	**73.790**	**74.500**
	0.5	**1.8296**	**1.8058**	**1.790**	**57.114**	**56.790**	**57.200**
	0.8	**7.3013**	**7.5081**	**7.159**	**36.291**	**34.780**	**36.400**
	0.9	**17.191**	**17.8090**	**17.002**	**26.308**	**23.970**	**26.400**
1.4	**0.2**	**0.5092**	**0.5112**	**0.5086**	**79.4491**	**73.9500**	**75.3000**
	0.5	**1.9345**	**1.8307**	**1.5870**	**55.4596**	**57.0500**	**58.0000**
	0.8	**7.6098**	**7.5852**	**7.1870**	**36.2559**	**35.0200**	**36.7000**
	0.9	**17.6245**	**------**	**17.0300**	**26.5092**	**----**	**26.6000**

Table 2: Comparison of Steady -state characteristics of an oil journal bearings for L/D=1 considering fluid Inertia Effect and pressure dependent viscosity effect in present study.[L/D=1.0, B=0.06]

Re*	ε_0	$\bar{W}_0$	$\bar{W}_0$	ϕ_0	ϕ_0
		Iso viscous	Variable Viscosity	Iso viscous	Variable Viscosity
0	0.2	0.5031	0.5081	77.5411	77.5008
	0.5	1.7592	1.8299	58.3980	58.0626
	0.8	7.0854	9.2746	36.3454	33.8620
0.28	0.2	0.5043	.5093	77.9273	77.8989
	0.5	1.7940	1.8678	57.7251	57.3893
	0.8	7.1942	9.4622	36.3106	33.8058
0.56	0.2	0.5055	.5105	78.3115	78.2950
	0.5	1.8296	1.9064	57.1143	56.7793
	0.8	7.3013	9.6482	36.2919	33.7639
1.4	0.2	0.5092	.5142	79.4491	79.4683
	0.5	1.9345	2.0210	55.4596	55.1264
	0.8	7.6098	10.2002	36.2559	33.6495

Effect of modified Reynold's number(Re*) on load carrying capacity.

The present steady state results with respect to load carrying capacity and attitude angle with fluid inertia effect only for Iso viscous fluid are compared to the results of Kakoty et.al., [5] and Chen & Chen [7] (for

L/D = 1.0) as given in Table 1. These three results are in good agreement A slight decrease in load capacity with modified Reynolds number (Re*) is observed in the present study. In the present study it is observed attitude angle increases slightly for all eccentricity ratios.

The effect of pressure dependent viscosity studied and compared also with the case of Iso viscous fluid as shown in Table 2. and observe that the load carrying capacity increasesmore with the effect of pressure dependent viscosity parameter.

IV RESULTS AND DISCUSSIONS

Steady state Analysis

A*Effect of viscosity parameters (B) with eccentricity ratio*

Figure 2.Variation of load carrying capacity with ecentricity ratio including inertia effect considering effect of variable viscosity

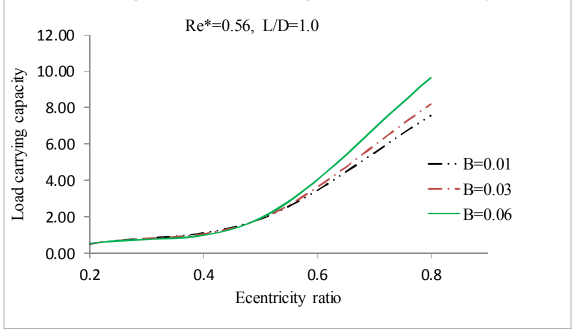

Figure 2 shows the variation of steady state load carrying capacity with eccentricity ratio for variable viscosity. The load carrying capacity increases with the increase of viscosity parameter. It also shows load carrying capacity increases with eccentricity ratio. The variation in insignificant at lower eccentricity ratio $\varepsilon_0 \leq 0.5$.

B *Effect of viscosity parameters (B)with Modified Reynold's number (Re*)*

Figure 3.Variation of load carrying capacity with modified Reynolds number considering the effect of variable viscosity

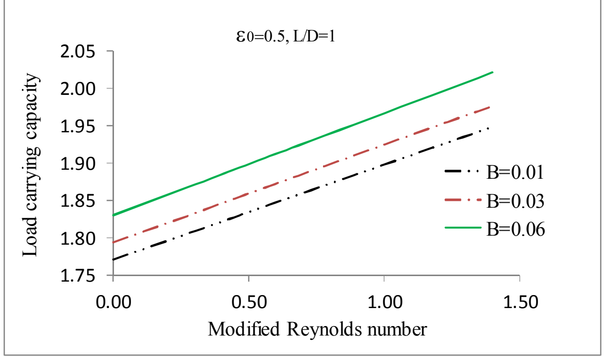

Figure 3 shows variation of steady state load carrying capacitywith modified Reynolds number for variable viscosity.The load carrying capacity increases with the increase of viscosity parameter.Also load carrying capacity increaseslinearly with Reynolds number.

C *Effect of viscosity parameters (B) with Attitude angle*

Figure 4.Variation of Attitude ange with ecentricity ratio including inertia effect and considering the effect of variable viscosity

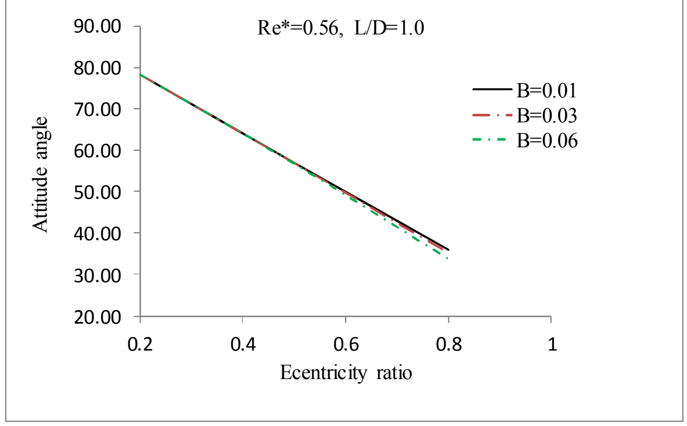

Figure 4 shows the variation of Attitude angle with eccentricity ratio for variable viscosity. Attitude angle decreases linearly with increase of eccentricity ratio and Attitude angle decreases as viscosity parameter increases.

D *Effect of viscosity parameters (B) with Slenderness ratio*

Figure 5. Variation of load carrying capacity with visosity parameter for different slenderness ratio

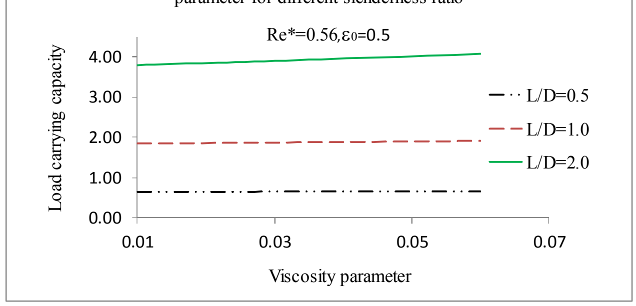

Figure 5.shows variation of steady state load carrying capacity with viscosity parameter for different slenderness ratio. It shows load carrying capacity increases with slenderness ratio but remains more or less constant for a specific slenderness ratio.

E Compare the *Effect of pressure dependent viscosity parameters(B) with the effect of ISO Viscous fluid*.

Figure 6.Comparison of load carrying capacity with ecentricity ratio for isoviscous and pressure dependent viscosity effect

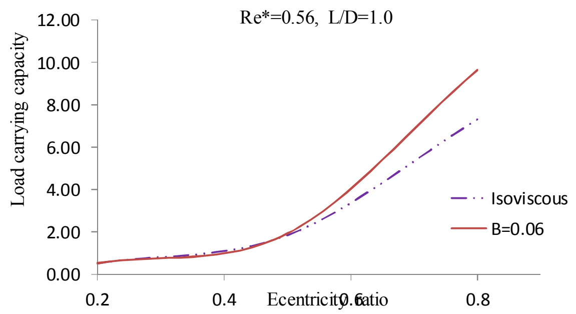

Figure 6 shows comparison of steady state load carrying capacity with eccentricity ratio for Isoviscous and the effect of pressure dependent variable viscosity. Steady state load carrying capacity increases with effect of pressure dependent variable viscosity and it is greater than the effect of Isoviscous fluid.

V CONCLUSIONS

(i)The steady state load carrying capacity increases as the viscosity parameter increases and also load carrying capacity increases as eccentricity ratio increases.

(ii) The steady state load carrying capacity increases as the viscosity parameter increases and load carrying capacity increases as modified Reynolds number increases.

(iii)Attitude angle decreases linearly as the viscosity parameter increases and also Attitude angle decreases when eccentricity ratio increases.

(iv)The steady state load carrying capacity increases as slenderness ratio increases. The variation of load carrying capacity is more or less constant with the increase of viscosity parameter for particular slenderness ratio.

(v)With the increase of variable viscosity parameter the steady state load carrying capacity increases more than Isoviscous lubricants.

VI NOMENCLATURE

c = Radial clearance (m)

D = Diameter of Journal (m)

e = eccentricity (m)

F_{r_0}, F_{ϕ_0} = Steady state hydrodynamic film forces (N).

$\bar{F}_{r_0}, \bar{F}_{\phi_0}$ = Dimensionless steady state hydrodynamic film forces

h = Film thickness, (m)

$\bar{h}$ = Dimensionless film thickness, h/c

L = Length of the bearing in m

p = Film pressure in Pa

$\bar{p}$ = dimensionless film pressure $\left(\frac{pc^2}{\eta\omega R^2}\right)$

R = radius of journal in m

R_e = Reynolds number, $\frac{\rho c R\omega}{\eta}$

R^*_e = Modified Reynolds number, $\left(\frac{c}{R}\right)R_e$

t = time in s

$u, v\ w$ = velocity components in x, y, z directions in m/s

$\bar{u}, \bar{v}, \bar{w}$ = dimensionless velocity components

$W0$ = steady-state load bearing capacity in N

$\bar{W}_0$ = Dimensionless steady-state load $\frac{W_0 c^2}{\eta\omega R^3 L}$

$x, y\ z$ = coordinates

$\theta, \bar{Y}, \bar{Z}$ = Dimensionless coordinates, $\frac{x}{R}, \frac{y}{c}, \frac{z}{L/2}$

$\varepsilon, \varepsilon_0$ = Eccentricity ratio e/c (dimensionless), steady-state eccentricity ratio

ρ = Density of the lubricant (kg m^{-3})

ω = Angular velocity of journal(rad s^{-1})

ω_p = Angular velocity of whirl (rad s^{-1})

Ω = Whirl ratio, $\frac{\omega_p}{\omega}$

η_0 = Absolute viscosity of lubricating Film at inlet condition (N s m^{-1})

ϕ = Attitude angle

Q_θ = Dimensionless flow parameter in θdirection

Q_z = Dimensionless flow parameter in $\bar{z}$ direction

$\bar{Q}$	=	Dimensionless side leakage
θ_1, θ_2	=	Angular coordinates at which film commences and cavitates.
$B = \frac{\eta_0 \alpha \omega R^2}{c^2}$		Viscosity Parameter

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