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Research Paper

EFFECT OF SMALL ROTATION ON THE PERFORMANCE OF SHORT JOURNAL BEARING LUBRICATED WITH COUPLE STRESS FLUID

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The objective of this paper is to predict the performance of rotor bearing system lubricated with couple stress fluid accounting the cross effect of velocities. Modified Reynold's equation and the Stokes' Constitutive equation have been obtained. Narrow bearing approximations have been used to obtain the solutions in the closed form. The effects of couple stress, frame rotation and the Viscosity-density effects have been obtained numerically. The results so obtained are consistent with the physical situation of the problem.

Keywords: Couple Stress, Friction, Hydrodynamicm Journal Bearing, Thin Film

INTRODUCTION

Theory of squeeze film has, so far, been playing very important role in the field of engineering in practical situations such as in lubrication in machine elements, lubrication in human body at synovial joints etc., (U P Singh *et al*., 2012a, 2012b, 2012c). Thus, a considerable attention of scientists and researchers has been paid on it. (C W Allen and A A Mckillop, 1970) shown that there is a good agreement between the theoretical and experimental results for a Newtonian squeeze film behavior between fixed and rotating annulii. Afterwards, a large number of researchers studied and concluded on different types of fluids and their behavior as a squeeze film. (M B Banerjee *et al*., 1981) pointed out almost all the physical systems are under effect of rotation - though it may be very small and they have extended the classical theory of lubrication (O Reynolds, 1986). (M B Banerjee *et al*., 1982; and R S Gupta and M B Banerjee, 1985; and R S Gupta and P Kavita, 1986) shown that in certain situations the qualitative properties of the bearing system may be different and obtained a certain class of fundamental solutions of the generalized Reynolds equation which are not allowed in the classical Reynolds theory (O Reynolds, 1986). (R S Gupta and M B Banerjee, 1985) elaborated to show theoretically the effect of small rotations in squeeze film journal bearing and shown that the results obtained shows a good agreement with the practical results.

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On the other hand, the theory of non-Newtonian polar fluids has also been center of attention of the researchers for their increasing use in industrial machine elements due to the rheological behavior in nature because of the presence of additives, suspensions and long chain polymers. Among the polar fluid theories, the couple stress fluid theory developed by (V K Stokes, 1966), which considers couple stresses in addition to the classical Cauchy stress, has been of much interest to the researchers from a long time. It is the generalization of the classical fluid theory which allows for polar effects such as the presence of couple stresses and body couples. Linear shearing stress and shearing rate relationship doesn't exist for such lubricants. Stokes discussed the fluid theory in detail in his treatise (V K Stokes, 1984). Afterwards, the squeeze film lubrication of couple stress fluid has been studied by (J R Lin, 1997; and G Ramanaiah, 1979) and observed an increase in load carrying capacity.

J R Lin (1997) has recently investigated the effect of couple stress lubricant on static characteristic of rotor bearing and analyzed the problem under assumptions of negligible shear stress between rotor and bearing system which, in practice, has a measurable effect. Also the effect of small rotation on the performance of bearing system cannot be overlooked. Therefore, the journal bearing under squeezing film condition along with the shearing stress will behave differently and need to be investigated under realistic conditions.

Hence, in order to investigate the problem under the said realistic physical condition as per the classical theory developed by (O Reynolds, 1986) and extended by (M B Banerjee *et al*., 1982; and R S Gupta and M B Banerjee, 1985; and R S Gupta and P Kavita, 1986) and onward, the modified Reynolds equation has been obtained, using microcontinuum theory for lubricants containing substructures (T Ariman *et al*., 1973) The interaction of microcontinuum theory developed by (V K Stokes, 1966) for lubricants with polar effects that is couple stress, body couples and the non-symmetric tensors has been used to develop the generalized Reynolds equation. The constitutive equations of an incompressible fluid with couple stress and small rotations (V K Stokes, 1966); (S Chandrashekhar, 1970; (R C Sharma and M Sharma, 2004) are-

$$\nabla \bar{V} = 0 \qquad ...(1)$$

$$\rho \frac{D\bar{V}}{Dt} = -\nabla p + \rho \bar{F} + \frac{1}{2}\rho \nabla \times C + \mu \nabla^2 \bar{V} - \eta \nabla^4 \bar{V} + 2\rho\left(\bar{V} \times \bar{\Omega}\right) \qquad ...(2)$$

where the vectors $\bar{V}$, $\bar{F}$, C and $\bar{\Omega}$ represents the velocity, body force per unit mass, body couple per unit mass, and rotation respectively; ρ is the density of the fluid, p is the pressure, μ is the shear viscosity and η is the new material constant standing for couple stress fluid property.

CONSTITUTIVE EQUATIONS AND BOUNDARY CONDITIONS

The physical configuration of a journal bearing is shown in the *Figure 1*. Consider a layer of fluid which is kept rotating at a constant rate. Let Ω be the angular velocity of frame rotation about y axis. The lubricant in the system is taken to be Stokes' couple stress fluid. The body forces and body couples are assumed to be absent. Under the assumptions of hydrodynamic lubrication applicable to thin film as used by (O Pinkus and B Sternlitch, 1961); (U P Singh, 2013), the field equations governing the motion of an incompressible fluid given in Cartesian co-ordinate system are-

$$\frac{\partial p}{\partial x} = 2p\Omega w + \mu\frac{\partial^2 u}{\partial y^2} - \eta\frac{\partial^4 u}{\partial y^4} \quad ...(3)$$

$$\frac{\partial p}{\partial y} = 0 \quad ...(4)$$

$$\frac{\partial p}{\partial z} = -2p\Omega u + \mu\frac{\partial^2 w}{\partial y^2} - \eta\frac{\partial^4 w}{\partial y^4} \quad ...(5)$$

These equations are solved under the following boundary conditions for u and w -

$u = 0$ at $y = 0$ and h ...(6)

$\mu\frac{\partial^2 u}{\partial y^2} = \frac{\partial p}{\partial x}$ at $y = 0$ and h ...(7)

$w = 0$ at $y = 0$ and h ...(8)

$\mu\frac{\partial^2 u}{\partial y^2} = \frac{\partial p}{\partial z}$ at $y = 0$ and h ...(9)

where u and w are the velocity components in x and z directions respectively and h is the film thickness between the journal bearing system.

MODIFIED REYNOLD'S EQUATION

A schematic iterative technique has been obtained to find the solutions of the equations (3) and (5) under the boundary conditions (6) through (9) for velocity components u and w. The step-wise procedure is as under.

From equations (3) and (5) the differential equations in u and w explicitly are:

$$\left(\frac{\eta^2}{2\rho\Omega}\right)\frac{\partial^8 u}{\partial y^8} - \left(\frac{\mu\eta}{\rho\Omega}\right)\frac{\partial^6 u}{\partial y^6} + \left(\frac{\mu^2}{2\rho\Omega}\right)\frac{\partial^4 u}{\partial y^4} + (2\rho\Omega u) = -\frac{\partial p}{\partial z} \quad ...(10)$$

$$\left(\frac{\eta^2}{2\rho\Omega}\right)\frac{\partial^8 u}{\partial y^8} - \left(\frac{\mu\eta}{\rho\Omega}\right)\frac{\partial^6 u}{\partial y^6} + \left(\frac{\mu^2}{2\rho\Omega}\right)\frac{\partial^4 u}{\partial y^4} + (2\rho\Omega w) = -\frac{\partial p}{\partial x} \quad ...(11)$$

These are 8-degree partial differential equations to be solved under the boundary conditions (6), and (7) for u and (8) and (9) for w.

Since, the 8-degree partial differential equation cannot be solved with only four known boundary conditions, in order to solve the equations (3) and (5) for u and w with these boundary conditions only, the well-established concept and procedure for solving the differential equations by (J B Scarborough, 1966) has been used.

METHODOLOGY

With the methodology mentioned in above technique for solving the differential equations, the stepwise solutions are given as–

Step-I

In the first iterative approximation, the solution for u and w are considered as under, in which only the second order velocity derivative has been considered while neglecting the cross effects.

$$u = \frac{1}{2\mu}\frac{\partial p}{\partial x}\left(y^2 - yh\right) \quad ...(12)$$

$$w = \frac{1}{2\mu}\frac{\partial p}{\partial z}\left(y^2 - yh\right) \quad ...(13)$$

Step-II

Using above velocities u and w as initial solution, the next iterative solution is as under, considering cross effects along with second order variation in velocity –

$$u = \frac{\partial p}{\partial x}\left[\frac{1}{2\mu}\left(y^2 - yh\right) + \frac{1}{\eta}\left(\frac{y^4 - 2y^3 + yh^3}{4!}\right)\right] + \frac{2\rho\Omega}{\mu\eta}\frac{\partial P}{\partial z}\left(\frac{y^6 - 3y^5h + 5y^3h^3 - 3yh^5}{6!}\right) \quad ...(14)$$

$$w=\frac{\partial p}{\partial z}\left[\frac{1}{2\mu}\left(y^2-yh\right)+\frac{1}{\eta}\left(\frac{y^4-2y^3+yh^3}{4!}\right)\right]-\frac{2\rho\Omega}{\mu\eta}\frac{\partial P}{\partial x}\left(\frac{y^6-3y^5h+5y^3h^3-3yh^5}{6!}\right)$$

...(15)

which appear to be a better solution than earlier.

Step-III

In order to further improve the solution obtained for u and w, substituting their values up to the second order term of velocity, the next improved solution is as–

$$\begin{aligned}u=&\frac{\partial p}{\partial x}\left[\frac{1}{2\mu}\left(y^2-yh\right)+\frac{\mu}{\eta^2}\left(\frac{y^6-3y^5h+5y^3h^3-3yh^5}{6!}\right)\right.\\&\left.-\frac{(2\rho\Omega)^2}{\mu\eta^2}\left(\frac{y^{10}-5y^9h+30y^7h^3-126y^5h^5+255y^3h^7-355yh^9}{10!}\right)\right]\\&+\frac{\partial P}{\partial z}\left[\frac{2\rho\Omega}{\mu\eta}\left(\frac{y^6-3y^5h+5y^3h^3-3yh^5}{6!}\right)\right.\\&\left.+\frac{4\rho\Omega}{\eta^2}\left(\frac{y^8-4y^7h+14y^5h^3-28y^3h^5+17yh^7}{8!}\right)\right]\end{aligned}$$

...(16)

$$\begin{aligned}u=&\frac{\partial p}{\partial z}\left[\frac{1}{2\mu}\left(y^2-yh\right)+\frac{\mu}{\eta^2}\left(\frac{y^6-3y^5h+5y^3h^3-3yh^5}{6!}\right)\right.\\&\left.-\frac{(2\rho\Omega)^2}{\mu\eta^2}\left(\frac{y^{10}-5y^9h+30y^7h^3-126y^5h^5+255y^3h^7-355yh^9}{10!}\right)\right]\\&+\frac{\partial P}{\partial x}\left[\frac{2\rho\Omega}{\mu\eta}\left(\frac{y^6-3y^5h+5y^3h^3-3yh^5}{6!}\right)\right.\\&\left.+\frac{4\rho\Omega}{\eta^2}\left(\frac{y^8-4y^7h+14y^5h^3-28y^3h^5+17yh^7}{8!}\right)\right]\end{aligned}$$

...(17)

Particular Cases

Case-I

If bearing system consist of Newtonian fluid (η = 0), the above said solution is identical with Gupta and Banerjee [8] as under-

$$u=\frac{1}{2\mu}\frac{\partial p}{\partial x}\left(y^2-yh\right)-\frac{2\rho\,\Omega}{\mu^2}\frac{\partial P}{\partial z}\left(\frac{y^4-2y^3h+yh^3}{4!}\right)+$$

...(18)

$$w=\frac{1}{2\mu}\frac{\partial p}{\partial z}\left(y^2-yh\right)-\frac{2\rho\,\Omega}{\mu^2}\frac{\partial P}{\partial x}\left(\frac{y^4-2y^3h+yh^3}{4!}\right)+$$

...(19)

Case-II

If bearing system consists of Stokes' couple stress fluid, the above solution is identical with (J R Lin, 1997) as under up to a good approximation to the usual solution obtained by Lin -

$$u=\frac{Uy}{h}-\frac{1}{\eta}\frac{\partial P}{\partial x}\left(\frac{y^4-2y^3h+yh^3}{4!}\right)-\frac{\mu}{\eta^2}\frac{\partial P}{\partial x}\left(\frac{y^6-3y^5h+5y^3h^3-3yh^5}{6!}\right)+$$

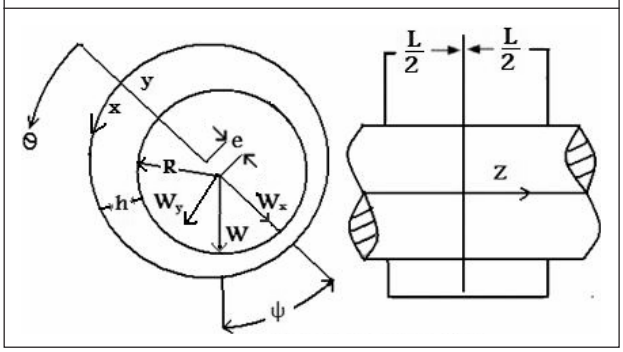
...(20)

$$w=-\frac{1}{\eta}\frac{\partial P}{\partial z}\left(\frac{y^4-2y^3h+yh^3}{4!}\right)-\frac{\mu}{\eta^2}\frac{\partial P}{\partial z}\left(\frac{y^6-3y^5h+5y^3h^3-3yh^5}{6!}\right)+$$

...(21)

Hence the technique so developed for obtaining the cross effects of this problem under the defined boundary conditions is logically and physically correct to a good approximation because the direct solution of the equations (3) and (5) cannot be obtained because of the lack of additional boundary conditions due to 8-degree partial differential equation in u and w.

Figure 1: Physical Configuration



Now, replacing the velocity components for and in equations (16) and (17) respectively in the continuity equation (1) and integrating with respect to with the boundary conditions, equation (22-23)

$v = 0$ at $y = 0$ ...(22)

$v = -\dfrac{dh}{dt}$ at $y = h$ ...(23)

the modified Reynold's equation is finally derived in equation (24).

$$\frac{\partial}{\partial x}\left[\left\{h^3+\frac{17}{1680}\frac{\mu^2}{\eta^2}h^7-\frac{13.086364}{30240}\frac{(2\rho\Omega)^2}{\eta^2}h^{11}\right\}\frac{\partial P}{\partial x}\right]$$
$$+\frac{\partial}{\partial z}\left[\left\{h^3+\frac{17}{1680}\frac{\mu^2}{\eta^2}h^7-\frac{13.086364}{30240}\frac{(2\rho\Omega)^2}{\eta^2}h^{11}\right\}\frac{\partial P}{\partial z}\right]$$
$$+\frac{\partial}{\partial x}\left[\left\{\frac{17}{28}\frac{(2\rho\Omega)}{6!\mu\eta}h^7-\frac{31}{9}\frac{(4\rho\Omega)}{8!\eta^2}h^9\right\}\frac{\partial P}{\partial z}\right]$$
$$+\frac{\partial}{\partial z}\left[\left\{\frac{17}{28}\frac{(2\rho\Omega)}{6!\mu\eta}h^7-\frac{31}{9}\frac{(4\rho\Omega)}{8!\eta^2}h^9\right\}\frac{\partial P}{\partial x}\right]=-12\mu\frac{dh}{dt} \quad ...(24)$$

Under assumptions of short bearing, $\frac{\partial}{\partial x}<<\frac{\partial}{\partial z}$, the equation (24) reduces to-

$$\frac{\partial^2 P}{\partial z^2}\left[h^3+\frac{17}{1680}\frac{\mu^2}{\eta^2}h^7-\frac{13.086364}{30240}\frac{(2\rho\Omega)^2}{\eta^2}h^{11}\right]$$
$$+\frac{\partial P}{\partial z}\left[\frac{17}{240}\frac{(2\rho\Omega)}{\eta}h^6-\frac{31}{1680}\frac{(2\rho\Omega)}{\eta^2}\mu h^8\right]\frac{dh}{dx}=-12\mu \quad ...(25)$$

$$A\frac{\partial^2 P}{\partial z^2}-B\frac{\partial P}{\partial z}=-48\lambda^2\cos\theta \quad ...(26)$$

where the superscript '*' has been dropped for simplicity, and

$$A=h^3+\frac{17}{1680}\frac{h^7}{l^4}-\frac{13.086364}{30240}\frac{M^2h^{11}}{l^4} \quad ...(27)$$

$$B=-\left(\frac{31}{10080}\frac{Mh^3}{l^4}-\frac{17}{120}\frac{Mh^6}{l^2}\right)\lambda\varepsilon\sin\theta \quad ...(28)$$

Solving the equation (25) for pressure P with the condition of zero pressure at the bearing ends, i.e. $P = 0$ at $z=\pm\frac{1}{2}$

The non-dimensional film pressure is obtained as under;

$$P=-\frac{c}{2B}[1+2z]+\frac{c}{2B}\left[\frac{Exp\left(\frac{Bz}{A}\right)-Exp\left(-\frac{B}{2A}\right)}{\sinh\left(\frac{B}{2A}\right)}\right] \quad ...(29)$$

where,

$$C=-48\lambda^2\cos\theta \quad ...(30)$$

Since, the rotational parameter M is small, ignoring the third and higher power of M and using narrow journal bearing approximation to calculate the pressure, the simplified pressure P is,

$$P=\frac{C}{2A}\left[\left(z^2-\frac{1}{4}\right)+\frac{B}{3A}\left(z^3+\frac{1}{8}\right)+\frac{B^2}{12A^2}\left(z^4-\frac{1}{16}\right)\right] \quad ...(31)$$

BEARING CHARACTERISTICS

Once the film pressure is determined from equation (31), the bearing characteristics can now be obtained as follows:

Load capacity

The load capacity can be calculated integrating the film pressure acting on the journal rotor. The component of load along (W_x) and in (W_y) the perpendicular to the center line and load carrying capacity W are given by-

$$W_x=W.\cos\psi=-R\int_{\theta=0}^{\theta=\pi}\int_{z=-\frac{L}{2}}^{z=\frac{L}{2}}P\cos\theta dz d\theta \quad ...(32)$$

$$W_y=W.\sin\psi=R\int_{\theta=0}^{\theta=\pi}\int_{z=-\frac{L}{2}}^{z=\frac{L}{2}}P\sin\theta dz d\theta \quad ...(33)$$

$$W=\sqrt{W_x^{\,2}+W_y^{\,2}} \quad ...(34)$$

where ψ is the altitude angle defined by

$$\psi = \tan^{-1}\left(W_y / W_x\right)$$

Taking non-dimensional load capacity

$$W^* = \frac{WC^2}{(d\varepsilon / dt)\mu R^3 L}$$

the components of load carrying capacity in non-dimensional form can be expressed as:

$$W_x = W^*.\cos\psi = -\int_{\theta=0}^{\theta=\pi}\int_{z=-\frac{1}{2}}^{z=\frac{1}{2}} P^* \cos\theta \, dz^* d\theta \quad ...(35)$$

$$W_y = W^*.\sin\psi = \int_{\theta=0}^{\theta=\pi}\int_{z=-\frac{1}{2}}^{z=\frac{1}{2}} P^* \sin\theta \, dz^* d\theta \quad ...(36)$$

The non-dimensional load capacity W^* can now be evaluated as-

$$W^* = \sqrt{W_x^{*2} + W_y^{*2}} \quad ...(37)$$

RESULTS AND DISCUSSION

In this paper, to account the effect of couple stress and frame rotation, the modified generalized Reynolds equation is derived. The effects of couple stress and rotation have been considered simultaneously to study the variations on various bearing performance properties e.g. pressure, load carrying capacity, friction parameter etc. of bearing system. Earlier researchers (J R Lin, 1997); (G Ramanaiah, 1979) have not considered the effect of rotation while discussing the effects of couple stress on the bearing performance while (R S Gupta and M B Banerjee, 1985) considered the rotation in their investigation but without the couple stress effect.

In the present study, an emphasis has been made to investigate the problem considering the simultaneous effect of small rotation and couple stress together with the variation in pressure along squeezing direction on the performance of short journal bearing. To discuss the effect of rotation, a dimensionless parameter M has been introduced (R S Gupta and P Kavita, 1986). Since, M is a function of rotation Ω, bearing clearance c, fluid density ρ and the fluid viscosity μ, it can be identified as the interaction of the fluid property, bearing geometry and bearing performance. For a particular fluid, a small value of M may be either due to small rotations or small bearing clearance or both and a larger value of M depends similarly on the rotation as well as bearing clearance; but, for no rotation, the value of M is being taken as zero. To study the effect of couple stress, the parameter l - a fluid property dependent has been used. Since, l has dimension of length, the dimensionless parameter $l^*(=l / c)$ - a fluid and bearing property dependent have been introduced.

In this process numerical results have been obtained from the pressure equation (31), load capacity equation (34) to (36), friction parameter equation (40). Since, in practice, the length to diameter ratio (λ) of a short journal bearing is preferred to be small, the value for λ is suitably taken as 0.3 throughout the discussion. The numerical results for the bearing properties under discussion have been obtained for couple stress parameter l^*=0.2, 0.4, 0.6, 1.0 and eccentricity ratio ε =0.2, 0.4, 0.6, 0.8 shown in figures 4.1 to 4.3 for pressure, figures 4.3 to 4.6 for Load capacity, 4.7 and 4.8 for altitude angle and figures 4.9 to 4.10 for friction parameter.

To establish the results for the effect of couple stress, the variation of film pressure and load carrying capacity of bearing due to couple stress, have been discussed without

rotation. Further, to analyze the performance and behavior of the bearing under small rotation, the results have been considered for various values of rotation parameter M varying from 0 to 0.15.

Figure 2 shows the variation of normalized pressure $\wp\left[=P^*/P_l\right]$ as variation with respect to circumferential angle θ varying from 90° to 180° for couple stress parameter l^* = 0.2, 0.4, 0.6 considering the bearing without rotation at the mid plane $z^* = 0$ and eccentricity ratio. $\varepsilon = 0.6$ It is observed that the normalized pressure Π increases with the circumferential angle θ and reaches maximum at $\theta = 180^0$. The pressure below $\theta = 90^0$ has not been considered because of its small value. Moreover, the normalized pressure increases with increase in couple stress at a particular angle θ which is consistent to (J R Lin, 1997).

Figure 2: Variation of Normalized Pressure as Variation of θ for Different Values of Couple Stress Parameter

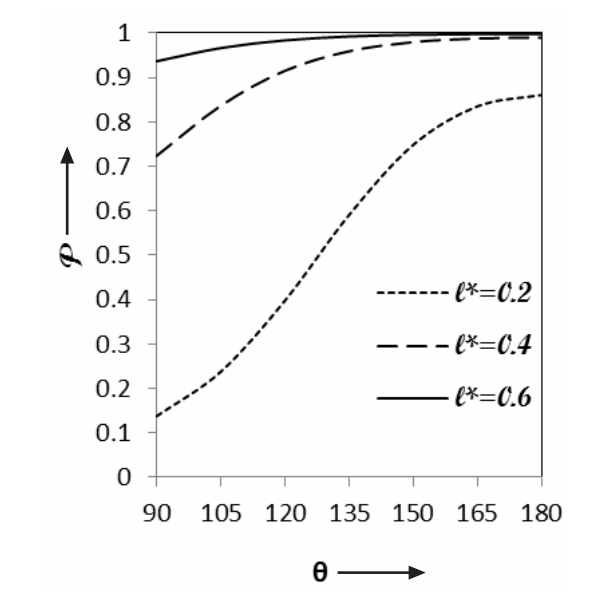


Figure 3: Variation of Normalized Pressure as Variation of θ for Different Values of Rotation Parameter at No Rotation

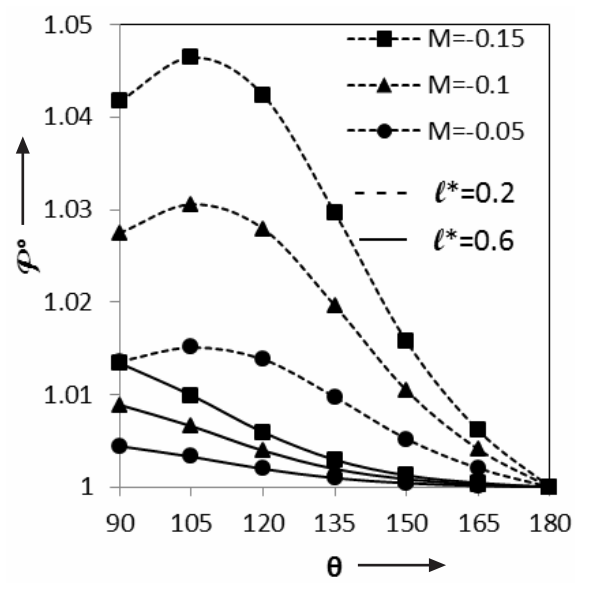


Figure 3 shows the variation of relative pressure $\wp^o\left[=P^*/P_m\right]$ with respect to circumferential angle θ for different values of rotation parameter M and couple stress parameter l^* = 0.2, 0.6 at z^* = 6 and eccentricity ratio $\varepsilon = 0.6$. It is observed that the normalized film pressure is higher for higher values of M and for each nonzero value of M, the normalized film pressure is higher in comparison to without rotation. It is also observed that the effect of rotation is more dominant for lower values of l^* and decreases with the increase in the value of l^*.

Figure 4 shows the variation of normalized pressure $\wp^o\left[=P^*/P_m\right]$ versus dimensionless bearing coordinate $z^*[-4,4]$ for different values of rotation parameter M and couple stress parameter l^* at eccentricity ratio $\varepsilon = 0.6$. and $\theta = 120^0$. Again it is observed that the normalized film pressure is higher for the higher values of M whereas the nature of the curves shows the relative variation of pressure to the pressure in absence of rotation because as M tends to 0, $\wp^o$ tends to 1.

Figure 5 shows the normalized load carrying capacity W $(=W^*/W_l)$ as a function of bearing eccentricity ratio ε for different values of couple stress parameter for different values of couple stress parameter $l^* = 0.2$, 0.6 in absence of rotation. It is observed that as a result of increase in pressure, the load capacity increases with the increase in couple stress which agrees with the results obtained by (J R Lin, 1997).

Figure 6 shows the normalized load carrying capacity $W^a(=W^*/W_M)$ as a function of eccentricity ratio ε for different values of rotation parameter M at $l^* = 0.2$, 0.4. It is observed that the variation in the load capacity with the rotation follow the same pattern as that of variation of pressure with rotation shown in Figure 3. As a consequence of the variation in the pressure, the load capacity increases with the increase in the value of rotation and for each value of M. Further, the load carrying capacity of bearing is higher than the load capacity when bearing is operated without rotation.

Figure 7 shows normalized load capacity. $W^a(=W^*/W_M)$ as a function of couple stress parameter l^* for various values of rotation parameter M at eccentricity ratio $\varepsilon = 0.2$, 0.4. It is observed that for each value of ε and l^*, the variation of normalized load capacity with rotation again agrees the same as discussed earlier. It is also observed from both the

Figure 4: Variation of Normalized Pressure Variation of Z* for Different Values of Couple Stress Parameter

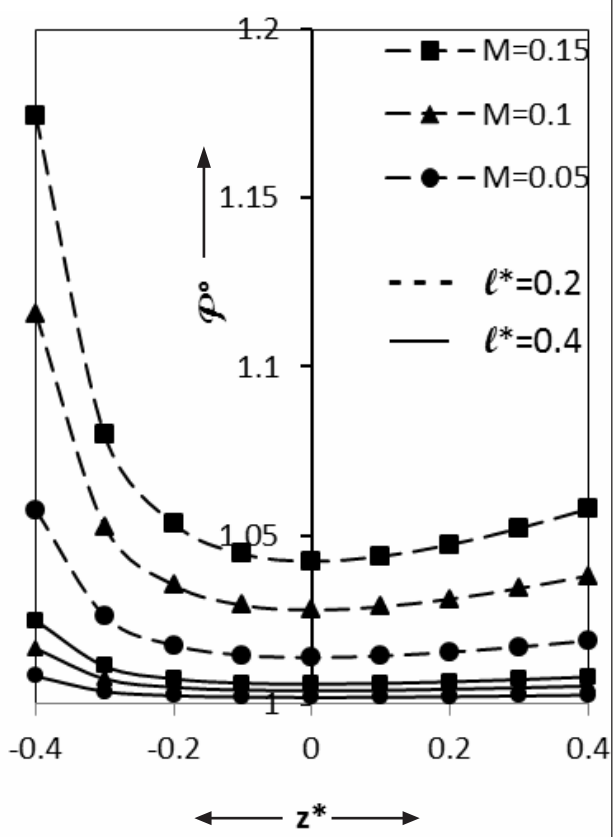


Figure 5: Variation of Normalized Load as Variation Couple Stress Parameter at No Rotation for Different Eccentricity Ratios

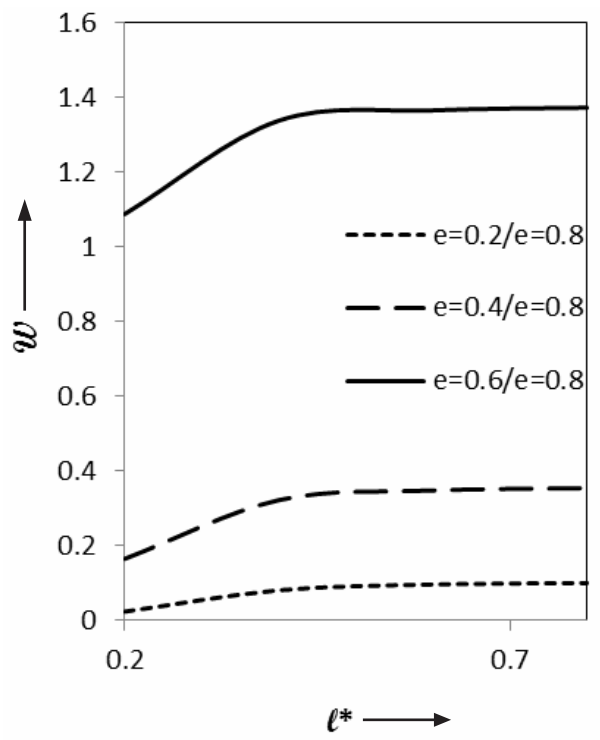


Figure 6: Variation of Normalized Load Capacity as Variation of Eccentricity Ratio for Different Values of Rotation Parameter

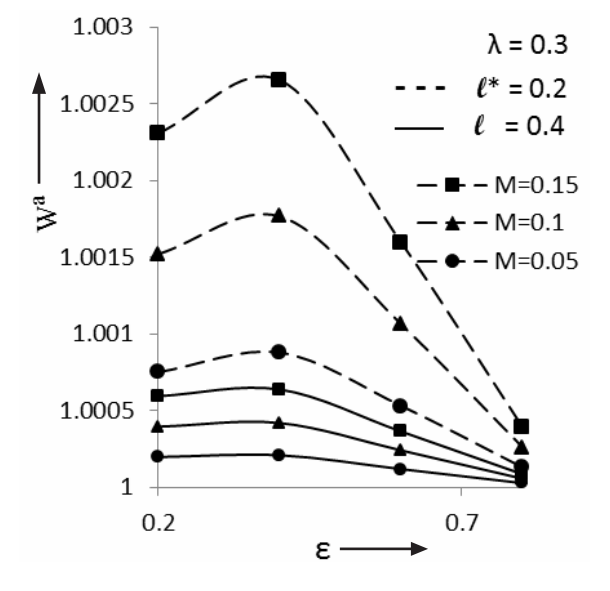
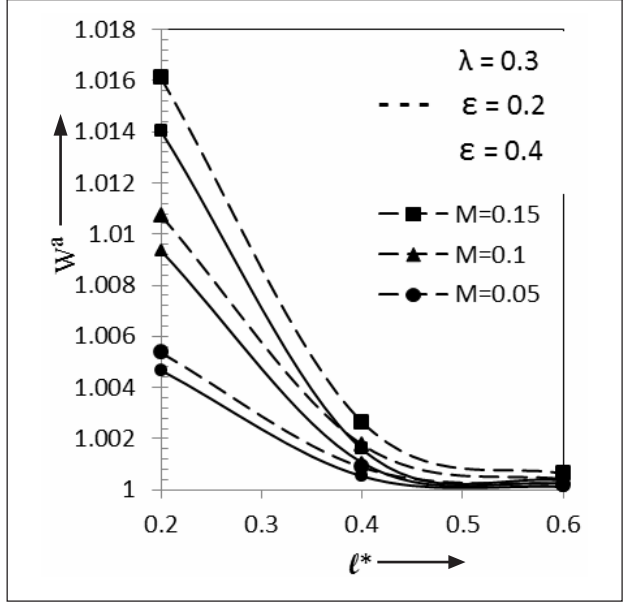


Figure 7: Variation of Normalized Load Capacity as Variation of Couple Stress Parameter for Different Values of Rotation Parameter



Figures 6 and 7 that the effect of rotation on the load carrying capacity bearing is more dominant for lesser values of l^*.

CONCLUSION

In the present theoretical study, the cross effect due to small rotation leads to much nearer to the realistic situation in the analysis as well as its performance. This will increase the safety factor while designing such bearing.

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Nomenclature

c	Bearing clearance
F_r	Friction parameter $\frac{fR}{c}$.
F_{r_m}	Friction paramete $M = 0$ r at.
F_{r0}	Normalized Friction parameter $\frac{F_r}{F_{r_m}}$.
h^*	$\frac{h}{c} = 1 + \varepsilon \cos\theta$.
ℓ	Couple stress parameter $\sqrt{\frac{\eta}{\mu}}$.
ℓ^*	Dimensionless couple stress parameter $\frac{\ell}{c}$.
L	Bearing length.
M	Frame rotation parameter $\left(\frac{2\rho\omega c^2}{\mu}\right)$.
P^*	Dimensionless pressure $\left(\frac{Pc^2}{\mu R^2 (d\varepsilon / dt)}\right)$.
P_ℓ	Dimensionless pressure at $\ell^* = 1$.
P_M	Dimensionless pressure at $M = 0$.
$\wp, \wp^o$	Normalized pressure; $\frac{P^*}{P}$, $\frac{P^*}{P_M}$.
R	Radius of Journal.
W^*	Dimensionless load capacity $\left(\frac{Wc^2}{\mu R^3 \mathrm{L}(d\varepsilon / dt)}\right)$.
W_ℓ	Dimensionless load capacity at $\ell^* = 1$.
W_M	Dimensionless load capacity at $M = 0$.
W	Normalized load capacity $\left(\frac{W^*}{W_\ell}\right)$.
W^a	Normalized load capacity $\left(\frac{W^*}{W_M}\right)$.
z^*	$\frac{z}{L}$.
ε	$\frac{e}{c}$.
η	Material constant representing couple stress property of fluid.
λ	$\frac{L}{2R}$.
μ	Viscosity of the fluid.
Ω	Rotational velocity.
ρ	Density of the fluid.
θ	Circumferential angle.