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Offset Journal Bearing Lubricated with Couple Stress Fluids and its Static Characteristics

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Abstract: *The aim of this paper is to study the effect of couple stress fluid on the static performance of an offset journal bearing. To find the pressure profile the Reynolds equation in non-dimensional form is solved using finite difference method. The static characteristics i.e. load carrying capacity, attitude angle and Sommerfeld number are determined and compared with Newtonian fluid. It was found that with the increase in couple stress parameter the load carrying capacity and attitude angle are enhanced and sommerfeld number is decreased. Effect of couple stress parameter on pressure profile is also carried out which shows that maximum increases with couple stress parameter.*

Keywords: *offset journal bearing, Reynolds, Sommerfeld number, Couple stress fluid.*

1.0 INTRODUCTION

Hydrodynamic journal bearing is a bearing operating with hydrodynamic lubrication, in which the bearing surfaces are separated from the journal surface by the lubricant film, generated by the journal rotation. The operation of a hydrodynamic journal bearing depends upon the shearing of a film of lubricant in the clearance space between the bearing and the journal. The load supporting pressure is generated within the film by continuous rotation of the journal. There are various types of journal bearing which are used for different working conditions and different fluid. To know what type of journal bearing can be used under which conditions, it is essential to determine their static and dynamic characteristic for different fluid and different working conditions. So, In this paper, static characteristics of offset journal bearing with couple stress fluid are determined.

To understand these characteristic a number of author have done their research work in this area. Tayal *et al* [1] determined static characteristics of elliptical hydrodynamic journal bearing for non-Newtonian (1981). They investigated the effect of ellipticity on the performance characteristics i.e. effect of eccentricity on load carrying capacity, attitude angle etc. of finite-width hydrodynamic journal bearings using the cubic shear stress law. These results for non-Newtonian lubricants are compared with those for Newtonian lubricants. The Athanasios Chasalevris [2] studied the static and dynamic characteristics of three lobe journal bearing for laminar regime. He showed the results for eccentricity ratio, attitude angle and loads carrying capacity for a range of Sommerfeld number and different cases of load orientation and compared with theoretical and experimental data from the literature. In 1995, Lin [3] determined effect of couple stress fluid on the static characteristics of rotor bearing system. According to him, the influence of couple stress effects on the characteristics of the system is apparent and not negligible. Comparing with the Newtonian case, the couple stress effects of fluids containing suspensions provide an enhancement in the load capacity, as well as a reduction in the attitude angle and the friction parameter. Crosby and Chetti

[4] are studied Static and dynamic characteristics of two-lobe journal bearings lubricated with couple-stress fluids. They determined load carrying capacity, the stiffness and damping coefficients, the non-dimensional critical mass, and the whirl ratio are determined for various values of the couple stress parameter. The results obtained are compared with the characteristics of two lobe bearings lubricated with Newtonian fluids. They found that the effect of the couple stress parameter is very significant on the performance of the journal bearing. Chetti [5] studied the effects of the couple stress parameter on the key performance of a four-lobe journal bearing. such as; the load carrying capacity, the friction force, side leakage, the stiffness and damping, the critical mass and whirl ratio are determined by him and he found that the presence of couple stresses improves the performance characteristics as compared to Newtonian fluid

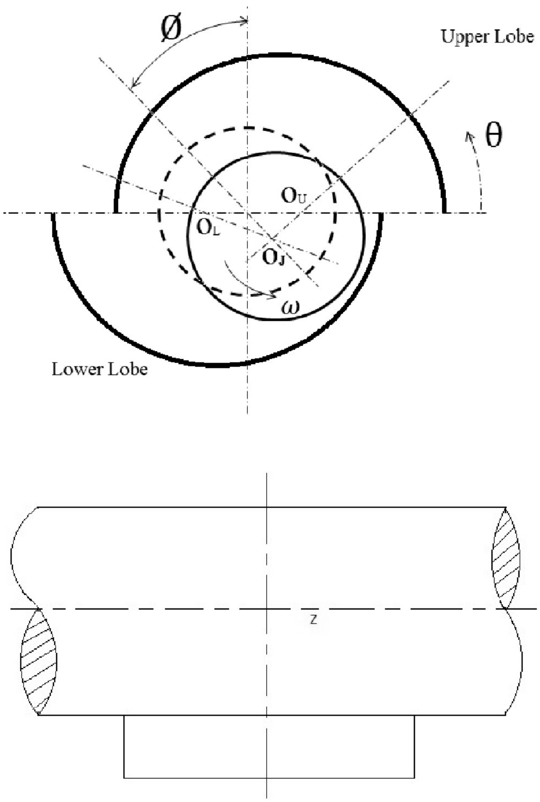

Fig.1 Physical configuration of an offset journal bearing

Physical configuration of offset journal bearing is shown in Fig.1. In offset bearing upper and lower lobe are displayed horizontally.

2.0 MATHEMATICAL ANALYSES

The geometric details of offset journal bearing configuration is shown in Fig. 1. Analysis of noncircular bearing involves solution of the governing equations separately for an individual lobe

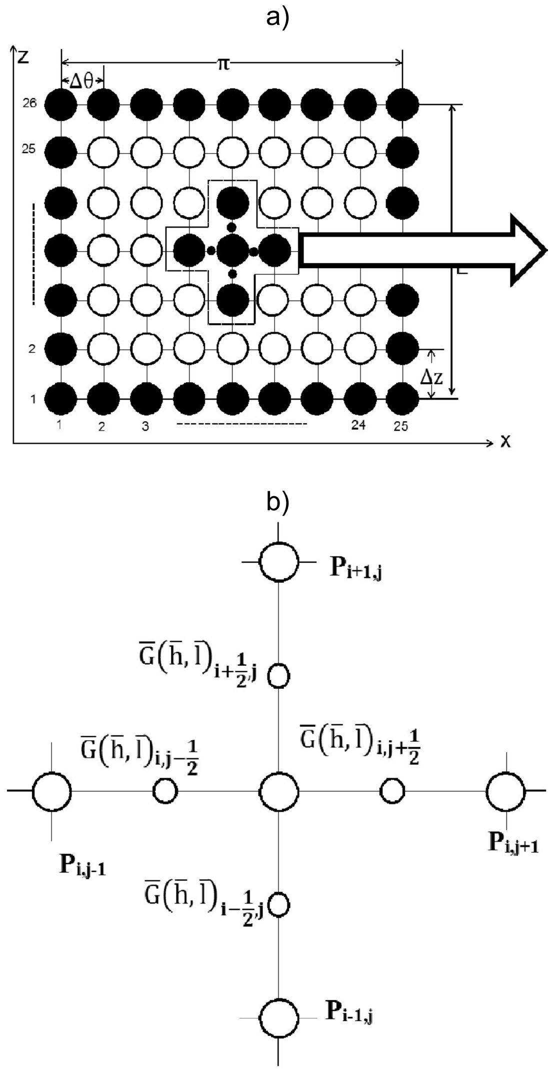

Fig. 2: Separation of the set of nodal points for the finite difference method from the surface of the bearing a) Schematic drawing showing the grid points used b) view of a single set of five adjacent points

of the bearing, treating each lobe as an independent partial bearing. To generalize the analysis for offset journal bearing, the film geometry of each lobe is described with reference to bearing fixed Cartesian axes (Fig. 1). The fluid film thickness expressed as:

$$\bar{h} = 1 + (\bar{X}_L - \bar{X}_J)\cos\theta + (\bar{Y}_L - \bar{Y}_J)\sin\theta \ldots (1)$$

The dimensionless form of the modified Reynolds equation is given below:

$$\frac{\partial}{\partial\theta}\left(\bar{G}(h,l)\frac{\partial\bar{p}}{\partial\theta}\right) + \frac{\partial}{\partial\bar{z}}\left(\bar{G}(h,l)\frac{\partial\bar{p}}{\partial\bar{z}}\right) = 6\frac{d\bar{h}}{d\theta} + 12\frac{d\bar{h}}{d\tau} \ldots (2)$$

Where

$$\theta = \frac{x}{R}, \bar{z} = \frac{z}{R}, \bar{h} = \frac{h}{C}, \bar{p} = \frac{pC^2}{\mu UR}, \tau = \omega t, \bar{\mu} = \frac{\mu}{\mu_o} = 1$$

and

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

Following the numerical solution of equation (2), i.e. after specifying the dependence of dimensionless pressure distribution upon dimensionless coordinates, obtained quantity was expressed back in the standard pressure forms. In order to facilitate the numerical implementation the surface of bearing has been spread and nodal points have been set aside in the surface area (m +1) * (n +1).

In the point with the index *i, j* (Fig. above) individual units of the Reynolds equation are brought closer, replacing the derivatives into differential quotients:

$$\frac{\partial}{\partial\theta}\left(\bar{G}(\bar{h},\bar{l})\frac{\partial\bar{p}}{\partial\theta}\right) = \frac{\bar{G}(\bar{h},\bar{l})_{i,j+\frac{1}{2}}\left(\frac{p_{i,j+1}-p_{i,j}}{\Delta\theta}\right) - \bar{G}(\bar{h},\bar{l})_{i,j-\frac{1}{2}}\left(\frac{p_{i,j}-p_{i,j-1}}{\Delta\theta}\right)}{\Delta\theta}$$

$$\frac{\partial}{\partial\bar{z}}\left[\bar{G}(\bar{h},\bar{l})\frac{\partial\bar{p}}{\partial\bar{z}}\right] = \frac{\bar{G}(\bar{h},\bar{l})_{i+\frac{1}{2},j}\left(\frac{p_{i+1,j}-p_{i,j}}{\Delta\bar{z}}\right) - \bar{G}(\bar{h},\bar{l})_{i-\frac{1}{2},j}\left(\frac{p_{i,j}-p_{i-1,j}}{\Delta\bar{z}}\right)}{\Delta\bar{z}}$$

$$6\frac{\partial\bar{h}}{\partial\theta} = 6\left(\frac{\bar{h}_{i,j+1}-\bar{h}_{i,j-1}}{2\Delta\theta}\right) = 3\left(\frac{\bar{h}_{i,j+1}-\bar{h}_{i,j-1}}{\Delta\theta}\right)$$

$$12\frac{\partial\bar{h}}{\partial\tau} = 12(-X_{Jd}\cos\theta - Y_{Jd}\sin\theta)$$

$$\bar{p}_{i,j+1}c_1 + \bar{p}_{i,j-1}c_2 + \bar{p}_{i+1,j}c_3 + \bar{p}_{i-1,j}c_4 - \bar{p}_{i,j}c_5 = s \ldots\ldots (3)$$

$$c_1 = \frac{\bar{G}(\bar{h},\bar{l})_{i,j+\frac{1}{2}}}{\Delta\theta^2} = \frac{\bar{G}(\bar{h},\bar{l})_{i,j} + \bar{G}(\bar{h},\bar{l})_{i,j+1}}{2\Delta\theta^2},$$

$$c_3 = \frac{\bar{G}(\bar{h},\bar{l})_{i,j} + \bar{G}(\bar{h},\bar{l})_{i+1,j}}{2\Delta\bar{z}^2}, c_4 = \frac{\bar{G}(\bar{h},\bar{l})_{i-1,j} + \bar{G}(\bar{h},\bar{l})_{i,j}}{2\Delta\bar{z}^2}.$$

$$c_5 = -(c_1 + c_2 + c_3 + c_4), s = 3\left(\frac{\bar{h}_{i,j+1} - \bar{h}_{i,j-1}}{\Delta\theta}\right)$$

Pressure $\bar{p}_{i,j}$ is a function of these constants and a function of the four nearby pressure data (in the grid). For n*m grid points (Fig. 4) n*m equations were obtained and further fixed in a program written in Matlab environment. Finally, the way of pressure distribution p = p (x, z) was obtained. There is no need to write the equation for the points lying on the edges of grid. Since, pressure at these points is known.

Initial condition:

$$p\left(\theta, \pm\frac{\lambda}{2}\right) = 0, X_{Jd} = 0, Y_{Jd} = 0$$

The coefficients c_1, c_2, c_3, c_4, c_5 are elements different from zero in the matrix C, the s coefficient is the element of one-column vector RHS. As a result a system of linear equations is given in the form of unknowns $\bar{p}$

By solving numerically set of equations according to the formula pressure values were obtained.:

$$\{\bar{p}\} = [C]^{-1} * \{S\} \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

The fluid film reaction components in 'X' and 'Y' directions are given

$$\bar{F}_X = \int_{\theta_1}^{\theta_2}\int_{-\lambda/2}^{\lambda/2}\bar{p}\cos\theta\, d\bar{z}d\theta,$$

$$\bar{F}_Y = \int_{\theta_1}^{\theta_2}\int_{-\lambda/2}^{\lambda/2}\bar{p}\sin\theta\, d\bar{z}d\theta \ldots (5)$$

$$F_x = \frac{\bar{F}_X\mu\omega R^4}{c^2}, F_y = \frac{\bar{F}_Y\mu\omega R^4}{c^2}, \emptyset = \tan^{-1}\frac{F_x}{F_y},$$

$$F_{NET} = \sqrt{F_x^2 + F_y^2}, S = \frac{\mu NDL}{F_{NET}}\left(\frac{R}{c}\right)^2$$

1.0 3.0 RESULTS AND DISCUSSION

Fig. (3-7) shows the effects of couple stress fluid on the static characteristic of offset journal bearing. Figure 3 shows the effect of couple stress fluid on the attitude angle at different value of couple stress parameter. It is clear from figure that attitude angle increases with the eccentricity and couple stress parameter.

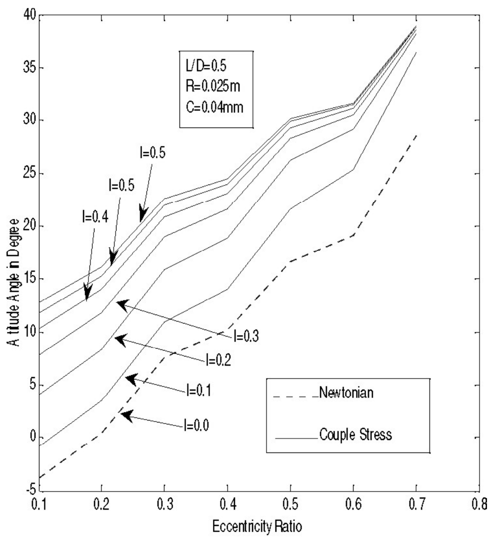

Fig.3 Attitude angle vs. eccentricity at different couple stresses parameter

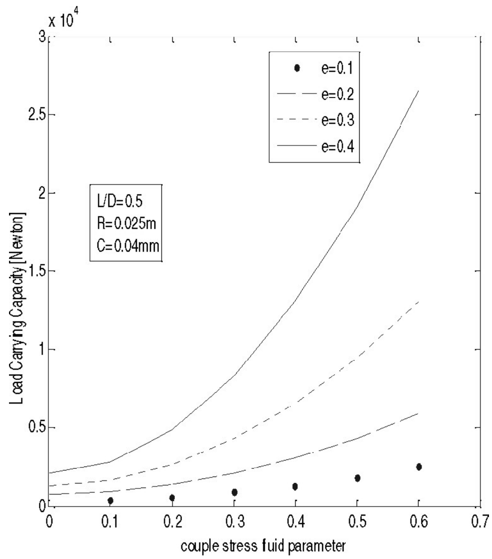

Fig.4 Load carrying capacity vs. non-dimensional couple stresses parameter

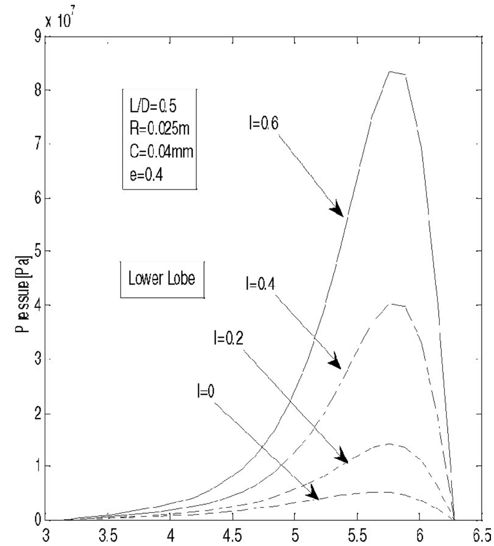

Fig.5 Effect of couple stress fluid parameter on pressure profile of the lower lobe of offset journal bearing

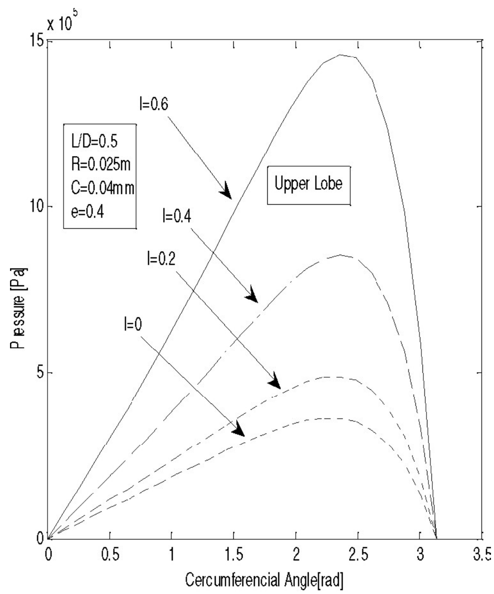

Fig. 6 Effect of couple stress fluid parameter on pressure profile of the upper lobe of offset journal bearingFig.4 shows the dimensionless load capacity as a function of couple stress parameter for different values of eccentricity ratio for aspect ratio=0.5. It can be seen from this figure that load carrying capacity is enhanced with increase in couple stress parameter and eccentricity ratio. Hence, journal can sustain more loads for couple stress fluid than Newtonian fluid.

Figures 5 and figure 6 depict the effect of couple stresses fluid on pressure as a function of circumferential angle for the journal operating at eccentricity ratio e = 0.4 in upper lobe and lower lobe. It can be seen that for high value of couple stress parameter the hydrodynamic film pressure is higher

As seen in the figure, the couple stress effects increase the hydrodynamic pressure; and the high the value of couple stress parameter is, the more the couple stresses affect the hydrodynamic pressure. As the couple stress parameter decreases, behavior of pressure curve tends to Newtonian. It is clear from. It is clear from figure that there is measurable effect of couple stress fluid in lower lobe as compare to upper lobe.

4.0 REFERENCES

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5.0 NOMENCLATURE

–	Represents the non-dimensional form
R	Radius of journal
L	Bearing length
e	Eccentricity, distance between the bearing eometrical center and journal center
ϕ	Attitude angle
ù	Angular speed of rotor
O_U	Geometrical center of upper lobe
O_L	Geometrical center of lower lobe
O_J	Geometrical center of journal
5Øß	Shear viscosity
U	Velocity of journal
C	Radial clearance
h	Fluid film thickness
p	Fluid film pressure
X_J, Y_J	Coordinates of the journal center
z	Axial coordinates
τ	Dimensionless time
F_X, F_Y	Non-dimensional hydrodynamic forces in x and y direction
5Øß$_1$	Location of leading edge of hydrodynamic film
5Øß$_2$	Location of trailing edge of hydrodynamic film
5Øß	Aspect ratio
S	Sommerfeld number
l	Couple stress parameter
N	speed of journal in RPM