Static and Dynamic Characteristics of Plain Journal Bearing Lubricated With Couple Stress Fluid-A Review

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Abstract— A bearing is a machine element which has relative motion with another moving element known as journal. Due to the relative motion between the contacting surfaces friction and wear are present between contacting surfaces. In order to reduce the wear and friction, a layer of lubricants is used. Lubricants show strong influence on the performance of bearing. In order to improve the desired characteristics of bearing additives are added to the lubricant. In this paper various static and dynamic performance characteristics of plain journal bearing lubricated with couple stress fluid is studied.

Key words: Journal Bearing, couples stress fluid, static characteristics, dynamic characteristics

I. INTRODUCTION

Bearing is used to prevent friction between parts during relative movement. Plain journal bearings consist of a shaft or journal which rotates freely in a supporting metal sleeve or shell called as bearing. The primary function of a bearing is to carry load between a rotor and the case with as little wear as possible. In order to reduce the wear, frictional resistance and frictional heat generated, a layer of fluid (known as lubricant) may be provided. The lubricants used are mineral oils which are obtained by refining silicon oil, grease, petroleum etc. Lubricants show the strong influence on the performance of bearing. A lubricant without additives does not serve satisfactorily under general operating conditions. Hence additives are added to lubricant to improve its desired characteristics. Couple stress in the fluid arises due to presence of additives causing the working fluid to behave as non-Newtonian polar fluid. Polar effect of molecules results in magnetic spin of molecules and generates stress tensors within the fluid. Modified Reynolds equation is derived for the couple stress fluid on the basis of Stokes microcontinnum theorem. Finite difference method is used for the solution of modified Reynolds. Static characteristics include attitude angle, bearing pressure and bearing load. Dynamic characteristics include stiffness coefficient, damping coefficient, threshold speed, critical mass and whirl frequency.

II. LITERATURE SURVEY

R. Sinhasan and K. C. Goyal [1] did computer-aided study for static and dynamic performance characteristics of an elastohydrodynamic journal bearing with a non-Newtonian lubricant. Deformation in bearing shell was obtained by solving the three-dimensional elasticity equations. On the basis of results obtained they culminated that, critical mass, stiffness and damping coefficient can be improved with the help of non-Newtonian fluid. Minimum fluid film thickness, attitude angle and power loss decreases with increase in critical mass, stiffness and damping coefficient while side flow increases.

R. Sinhasan and K. C. Goyal [2] did computer-aided study of the transient response of a circular journal bearing with non-Newtonian lubricant. They applied finite element method to solve momentum and continuity equations flow field. The journal centre motion trajectories were obtained by solving non-linear equations of motion and predicted that the journal bearing system becomes unstable even for slight perturbation in reference critical mass. Observation shows that the linearized equations of motion gives a limiting cycle when the journal mass equal to the critical mass. They carried out their study for Newtonian as well non-Newtonian lubricants. The non-linear equation of motion results in lower value of stability margin than that predicted by linearized system.

J. R. Lin [3] derived modified Reynolds equation on the basis of Stokes microcontinnum theory and studied the influence of the couple stress fluid on the performance of a rotor bearing system. He concluded that, the influence of couple stress effects improves load carrying capacity, as well as a reduction in the attitude angle and the friction parameter as compared to that of the Newtonian lubricant case.

J. R. Lin [4] carried out theoretical investigation of rheological effects on couple stress fluids for finite journal bearings. On the accounts of the Stokes microcontinnum theory, the effects of couple stresses on the static characteristics of finite journal bearings were presented. In accordance of his investigation he concluded that couple stresses provide an enhancement in the load carrying capacity whereas the attitude angle and the friction parameter get reduced.

Hsiu-Lu Chiang, et al. [5] inspected the performance characteristics of finite journal-bearing systems under the effects of couple stress fluid. According to the results they stated that couple stress effect raises the film pressure of the lubricant fluid, improves the load carrying capacity and reduces the friction parameter, at high eccentricity ratio. The surface roughness effect dominates the long bearing approximation.

III. ANALYSIS

A. *Basic Equations*

Schematic figure of plain circular bearing is shown in Figure 1 below. Wedge film is formed due to curved surface of journal and bearing. For present study working lubricant is assumed to be incompressible couple stress fluid. Modified Reynolds equation is derived for couple stress fluid on the basis of Stokes microcontinnum theorem.

Continuity equitation for three dimensional fluid flows is given as,

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho V) = 0 \tag{3.1}$$

Modified Navier-stokes equation for incompressible couple stress fluid can be given as

$$\rho A = -\nabla p + \rho F^* + \frac{1}{2}\, curl(\rho T) + \mu \nabla^2 V - \eta \nabla^4 V \tag{3.2}$$

where V, A, $F^*$and T are the velocity vector, acceleration, body force per unit mass and body couple per unit mass respectively. ρ is the density, p is the hydrodynamic pressure, μ is the shear viscosity and η is a new material viscosity defining the couple stress property having the units of momentum.

Considering thin incompressible fluid film between journal bearing which leads to absence of body couple, body force and inertia force equation (2) can be written as,

Fluid flow along X-direction,

$$\eta \frac{\partial^4 u}{\partial z^4} - \mu \frac{\partial^2 u}{\partial z^2} = -\frac{\partial p}{\partial x} \tag{3.3}$$

For fluid flow along Y-direction,

$$\eta \frac{\partial^4 v}{\partial z^4} - \mu \frac{\partial^2 v}{\partial z^2} = -\frac{\partial p}{\partial y} \tag{3.4}$$

For fluid flow along Z-direction,

$$\frac{\partial p}{\partial z} = 0 \tag{3.5}$$

The no slip boundary conditions for the velocity distributions are

$$\left.\begin{array}{l} \text{At } y = 0;\ u = 0;\ w = 0; \\ \text{At } y = h;\ u = U;\ w = 0; \end{array}\right\} \tag{3.6}$$

Velocity due to squeeze action of fluid film,

$$v(x,0,z) = 0, v(x,h,z) = V \tag{3.7}$$

Couple stress vanishes at the boundary,

$$\left.\frac{\partial^2 u}{\partial z^2}\right|_{z=0} = \left.\frac{\partial^2 v}{\partial z^2}\right|_{z=0} = \left.\frac{\partial^2 u}{\partial z^2}\right|_{z=h} = \left.\frac{\partial^2 v}{\partial z^2}\right|_{z=h} = 0 \tag{3.8}$$

Velocity component obtained are:

$$u = U + U\frac{y}{h} + \frac{1}{2\mu}\frac{\partial p}{\partial x}\ y^2 - \left\{ y^2 - yh + 2l^2 \left[1 - \cosh\left(\frac{2y-h}{2l}\right) X \left(\cosh\left(\frac{h}{2l}\right)\right)^{-1} \right]\right\} \tag{3.9}$$

$$w = \frac{1}{2\mu}\frac{\partial p}{\partial z}\ y^2 - \left\{ y^2 - yh + 2l^2 \left[1 - \cosh\left(\frac{2y-h}{2l}\right) X \left(\cosh\left(\frac{h}{2l}\right)\right)^{-1} \right]\right\} \tag{3.10}$$

Here $l = \sqrt{\frac{\eta}{\mu}}$ and U, V are tangential and normal velocity components of journal.

q_x and q_z are along the flow fluxes along X and Z directions. They are given as:

$$q_x = \int_0^h u dy = \frac{U}{2}\, h - \frac{h^3}{12\mu} f(l,h) \frac{\partial p}{\partial x} \tag{3.11}$$

$$q_z = \int_0^h w dy = -\frac{h^3}{12\mu} f(l,h) \frac{\partial p}{\partial z} \tag{3.12}$$

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

Modified Reynolds equation governing the pressure distribution on the film is given as,

$$\frac{\partial}{\partial x}\left\{ f(l,h)\frac{\partial p}{\partial x}\right\} + \frac{\partial}{\partial z}\left\{ f(l,h)\frac{\partial p}{\partial z}\right\} = 6\mu U \frac{\partial h}{\partial x} + 12\mu \frac{\partial h}{\partial t} \tag{3.13}$$

B. Non-Dimensional Form of Modified Reynolds Equation

The non- dimensional form of modified Reynolds equation for an incompressible fluid with couple stress is:

$$\frac{\partial}{\partial \theta}\left(f(\bar{l},\bar{h})\frac{\partial \bar{p}}{\partial \theta}\right) + \frac{\partial}{\partial \bar{z}}\left(f(\bar{l},\bar{h})\frac{\partial \bar{p}}{\partial \bar{z}}\right) = 6\frac{\partial \bar{h}}{\partial \theta} + 12\frac{\partial \bar{h}}{\partial \bar{t}} \tag{3.14}$$

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

C. Film Thickness

Following equation gives non-dimensional fluid film formed at the location from positive Y axis:

$$\bar{h} = 1 - \bar{X_j}\cos(90 + \emptyset + \theta) - \bar{Y_j}\sin(90 + \emptyset + \theta) \tag{3.15}$$

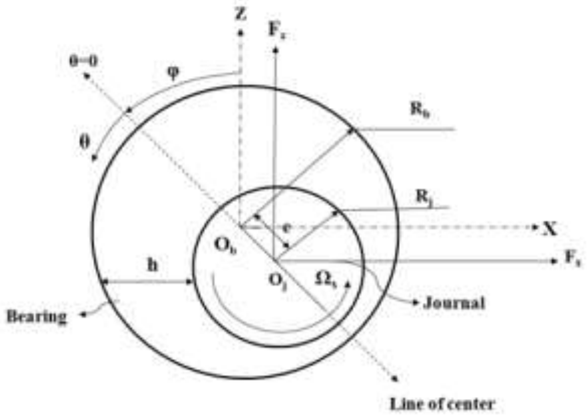

Fig. 1: Schematic diagram of a plain circular journal bearing

D. Load Carrying Capacity

Following expressions gives vertical and horizontal components of load,

$$\bar{F_x} = \int_0^{L/R_j}\int_0^{\theta_2} \bar{p}\cos(\theta + \emptyset)\, d\theta\, d\bar{z} \tag{3.16}$$

$$\bar{F_z} = \int_0^{L/R_j}\int_0^{\theta_2} \bar{p}\sin(\theta + \emptyset)\, d\theta\, d\bar{z} \tag{3.17}$$

Resultant load is given as:

$$\bar{F} = \sqrt{\bar{F_x}^2 + \bar{F_z}^2}$$

E. Stiffness Characteristics

To find stiffness characteristics of the bearing journal is given perturbation (displacement) from its equilibrium position. Given displacement leads to variation in fluid film thickness and it causes variation in the static equilibrium forces and instantaneous fluid film forces. This variation is used to calculate stiffness coefficients and these are obtained as:

$$\begin{bmatrix} \bar{K_{xx}} & \bar{K_{xz}} \\ \bar{K_{zx}} & \bar{K_{zz}} \end{bmatrix} = \begin{bmatrix} -\frac{\partial}{\partial \bar{x}} \\ -\frac{\partial}{\partial \bar{z}} \end{bmatrix} [\bar{F_x}\, \bar{F_z}\,] \tag{3.18}$$

F. Damping Characteristics

To obtain damping characteristics journal perturbation is given in the form of velocity. This disturbance results change in film thickness and force variation in observed between static equilibrium forces and instantaneous forces. These differences in these forces are used to calculate damping coefficient. Hence damping coefficient is given by following equation:

$$\begin{bmatrix} \bar{C_{xx}} & \bar{C_{xz}} \\ \bar{C_{zx}} & \bar{C_{zz}} \end{bmatrix} = \begin{bmatrix} -\frac{\partial}{\partial \dot{\bar{x}}} \\ -\frac{\partial}{\partial \dot{\bar{z}}} \end{bmatrix} [\bar{F}_x\, \bar{F}_z] \tag{3.19}$$

G. Critical Mass

Critical mass of the system is function of dynamic characteristics of bearing. As per Routh-Hurwitz stability criteria critical mass is calculated by using the stiffness and damping coefficients and is expressed as:

$$\overline{M}_c = \frac{a_0}{b_0 - c_0} \tag{3.20}$$

Where $a_0 = \left(\overline{C_{xx}} * \overline{C_{yy}} - \overline{C_{yx}} * \overline{C_{xy}}\right)$

$$b_0 = \frac{(\overline{C_{xx}}+\overline{C_{yy}})*(\overline{K_{xx}}*\overline{K_{yy}}-\overline{K_{yx}}*\overline{K_{xy}})}{(\overline{K_{xx}}*\overline{C_{yy}}+\overline{C_{xx}}*\overline{K_{yy}}-\overline{K_{yx}}*\overline{C_{xy}}-\overline{K_{xy}}*\overline{C_{yx}})}$$

$$c_0 = \frac{\overline{K_{xx}}*\overline{C_{xx}}+\overline{C_{yy}}*\overline{K_{yy}}+\overline{K_{yx}}*\overline{C_{xy}}+\overline{K_{xy}}*\overline{C_{yx}}}{(\overline{C_{xx}}+\overline{C_{yy}})}$$

If applied journal mass $\overline{M_j}$ is less than the critical mass $\overline{M}_c$ i.e., $\overline{M_j} \leq \overline{M}_c$ the system shows stability converse to this statement if $\overline{M_j} > \overline{M}_c$ then the system shows instability. However for any negative values of critical mass system always shows stability.

H. Threshold Speed

It can be defined as the speed at which rotor becomes unstable. Threshold speed depends upon stiffness and damping coefficient. It can be given as:

$$\overline{\Omega_s} = \sqrt{\frac{\overline{F}*\overline{K_{eq}}}{\overline{M_c}}} \tag{3.21}$$

Where, $\overline{K_{eq}} = \frac{\overline{K_{xx}}*\overline{C_{zz}}+\overline{C_{xx}}*\overline{K_{zz}}-\overline{K_{zx}}*\overline{C_{xz}}-\overline{K_{xz}}*\overline{C_{zx}}}{\overline{C_{xx}}+\overline{C_{zz}}}$

I. Whirl Frequency Ratio

It is the ratio of the rotor whirl to the rotor onset speed at instability. Whirl frequency ratio $(\overline{f})$ depends on the dynamic bearing characteristics. Negative value of $\overline{f}$ implies absence of whirl.

$$\overline{f} = \sqrt{\frac{(\overline{K_{xx}}-\overline{K_{eq}})*(\overline{K_{yy}}-\overline{K_{eq}})-\overline{K_{yx}}*\overline{K_{xy}}}{(\overline{C_{xx}}*\overline{C_{yy}}-\overline{C_{yx}}*\overline{C_{xy}})}} \tag{3.22}$$

IV. Discussion and Conclusion

The objective of this work was to study various static and dynamic characteristics of the plain journal bearing lubricated with couple stress fluid theoretically. Lubricants containing additives shows great influence on the static and dynamic characteristics of journal bearing so in order to improve bearing life it is necessary to study static and dynamic characteristics of journal bearing.

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