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Elastohydrodynamic Analysis of Elliptical Bearing with Micropolar Fluid

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Abstract: **This research is concerned with the analysis of noncircular journal bearings taking deformability of the bearings liner and variation of viscosity due to the presence of various additives in the lubricant. A survey of literature shows that a few investigations have been carried out on circular bearings operating with micropolar lubricants. Literatures are available on static analysis of such bearings, but literature on dynamic analysis is scarce. Literature survey also shows that no work has been carried out on EHD analysis of circular and non circular bearings operating with micropolar lubricants. So it is felt that there is a need to compute static and *dynamic characteristics of elastohydodynamic journal bearings operating with micropolar lubricants.***

Keywords: **Introduction, Bearingbehaviour with micropolar fluid, Reynolds equation, performance characteristics.**

I. INTRODUCTION

When bearing is subjected to heavy loads the bearing shell deforms. The deformation of the bearing shell modifies the film thickness and this in turn affects the performance characteristics of the bearing. Therefore elastohydrodynamic analysis is considered to recompute the performance characteristics of circular and non circular bearings. In hydrodynamic journal bearing, the load supporting high pressure fluid film is created due to the shape and relative motion between the two surfaces. The moving surface pulls the lubricant into a wedge-shaped zone, at a velocity sufficiently high to create the high pressure film necessary to separate the two surfaces against the load.

Hydrodynamic bearings are generally used in cases when the relative velocity are high enough as a result of continuous increase in the sizes and speeds of the rotating machinery or due to use of fluids having kinematic viscosity, the oil film close in the bearings frequently becomes turbulent. As the magnitude of hydrodynamic pressure depends upon the relative tangential or normal velocity, sometimes the developed pressure may not be sufficient for obtaining the desired load carrying capacity, particularly at the low relative velocity and there by demands external supply of pressurized lubricants. This type of lubrication is called externally pressurized or hydrostatic lubrication. These bearings are also known as hybrid bearings as their operational principle includes both self-acting and external pressurization.

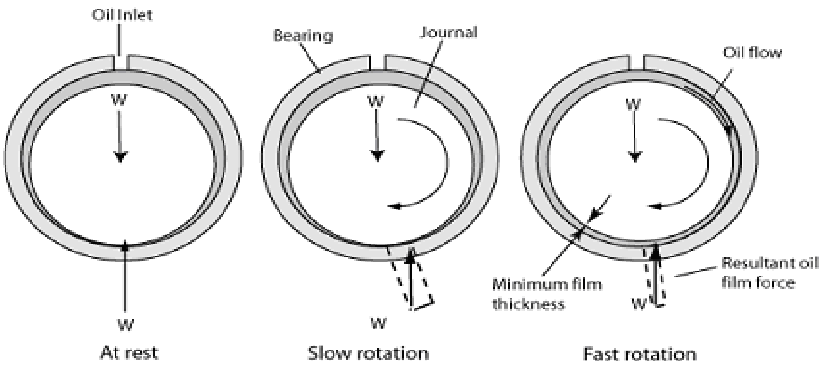

Fig 1: Principle of hydrodynamic journal bearing

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II. BEARINGS BEHAVIOUR WITH MICROPOLAR LUBRICATION

Theory of micropolar lubrication the first application of the theory was presented by Eringenhimself for the steady motion of micropolar fluids in a circular channel in which the profiles for the velocity, microrotational velocity, shear stress difference and the couple stress on the fluid surface adjacent to the wall were presented graphically. The velocity profile was found to lose its parabolic nature and was smaller than that of classical Navier-Stokes fluid. Though the shearing stress remained the same as that determined by classical theory, the surface shear was found to be reduced by an amount equivalent to the effect of the distributed couples aroused on the fluid surface in a thin layer adjacent to the surface thus, indicating the development of a boundary layer phenomenon not present in the Navier-Stokes theory.

Balaram presented an analysis of micropolar squeeze films between two rectangular plates of infinite length along with the expressions for the pressure, the load carrying capacity of the squeeze film and the relationship of film thickness with time. The results showed an improvement in load carrying capacity with the increase in the density of the macromolecular volume but a decrease in load capacity with an increase in substructure particle size.

A. Governing equations:

The field equations for micropolar fluids in vectorial form are **:**

Principle of conservation of mass:
$$\frac{\partial \rho}{\partial t} + \nabla \cdot \left(\rho \vec{V}\right) = 0 \qquad (1)$$

Conservation of linear momentum:
$$(\lambda + 2\mu)\nabla\left(\nabla \cdot \vec{V}\right) - \frac{(2\mu + \chi)}{2} \nabla \times \nabla \times \vec{V} + \chi \nabla \times \vec{v} - \nabla \pi^{*} + \rho F_B = \rho \frac{D\vec{V}}{Dt} \qquad (2)$$

Conservation of angular momentum:
$$(\alpha + \beta + \gamma)\nabla\left(\nabla \cdot \vec{v}\right) - \gamma\nabla \times \nabla \times \vec{v} + \chi \nabla \times \vec{V} - 2\chi \vec{v} + \rho C_B = \rho j \frac{Dv}{Dt} \qquad (3)$$

Where$\rho$is the mass density, $V_r$is the velocity vector and $v_r$is the microrotational velocity vector.π^* is the thermodynamic pressure and is to be replaced by the hydrodynamic film pressure, p, since $\pi^* = -\left[\frac{\partial E}{\partial \rho^{-1}}\right] = p$ where E is the internal energy and p is to be determined by the boundary conditions. μ and are the familiar λ viscositycoefficients of the classical fluid mechanics, whileα, β *and* $\gamma$are the new viscosity coefficients derived as the combinational effects of the gyro viscosities for micropolar fluid as defined by Eringen.

B. REYNOLDS EQUATION:

The Reynolds Equation is a partial differential equation governing the pressure distribution of thin viscous fluid films in Lubrication theory. It should not be confused with Osborne Reynolds other namesakes, Reynolds number and Reynolds-averaged Navier–Stokes equations. It was first derived by Osborne Reynolds in 1886. The classical Reynolds Equation can be used to describe the pressure distribution in nearly any type of fluid film bearing; a bearing type in which the bounding bodies are fully separated by a thin layer of liquid or gas.

The general Reynolds equation is:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial y}\right) = \frac{\partial}{\partial x}\left(\frac{\rho h\left(u_a + u_b\right)}{2}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h\left(v_a + v_b\right)}{2}\right) + \rho\left(w_a - w_b\right) - \rho u_a \frac{\partial h}{\partial x} - \rho v_a \frac{\partial h}{\partial y} + h\frac{\partial \rho}{\partial t} \qquad (4)$$

Where:

- p is fluid film pressure.
- x and y are the bearing width and length coordinates.
- z is fluid film thickness coordinate.
- h is fluid film thickness.
- μ is fluid viscosity.

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- ρ is fluid density.
- u, v, w are the bounding body velocities in x, y, z respectively.
- a, b are subscripts denoting the top and bottom bounding bodies respectively.

The equation can either be used with consistent units or nondimensionalized.

The Reynolds Equation assumes:

- The fluid is Newtonian.
- Fluid viscous forces dominate over fluid inertia forces. This is the principal of the Reynolds number.
- Fluid body forces are negligible.
- The variation of pressure across the fluid film is negligibly small (i.e. $\frac{\partial p}{\partial z} = 0$
- The fluid film thickness is much less than the width and length and thus curvature effects are negligible. (i.e. $h \ll l$ and $h \ll w$

The basic assumptions in micropolar lubrication to a journal bearing include the usuallubrication assumptions in deriving Reynolds equation and the assumptions to generalize the micropolareffects :

1. The flow is incompressible and steady, *i.e.* **ρ** constant and $\frac{\partial \rho}{\partial t} = 0$

2. The flow is laminar *i.e.* free of vortices and turbulences.

3. Body forces and body couples are negligible, *i.e.* ***F***B = 0 and ***C***B = 0.

4. The film is very thin in comparison to the length and the span of the bearing. Thus, the curvature effect of the fluid film may be ignored and the rotational velocities may be replaced by the translatory velocities.

5. No slip occurs at the bearing surfaces.

6. Bearing surfaces are smooth, non-porous and rigid *i.e.* no effects of surface roughness or porosity and the surface can withstand infinite pressure and stress theoretically without having any deformation.

7. No fluid flow exists across the fluid film, *i.e.* the lubrication characteristics areindependent of *y*-direction.

8. The micropolar properties are also independent of *y*-direction. The velocity vector,the microrotational velocity vector and the fluid film pressure are given as:

$$\begin{aligned} \vec{V} &= \left[\vec{V}_x(x,y,z), \vec{V}_y(x,y,z), \vec{V}_z(x,y,z)\right] \\ \vec{v} &= \left[\vec{v}_1(x,y,z), \vec{v}_2(x,y,z), \vec{v}_3(x,y,z)\right] \\ p &= p(x,y,z) \end{aligned} \tag{5}$$

Note that for $\alpha = \beta = \gamma = \chi = 0$ and for negligible body couple per unit mass equation yields and so, $\vec{v} = 0$ equation reduces to the classical Navier-Stokes equation. For = 0 the veloχty vector and the microrotational velocity vector are uncoupled and the global motion of the fluid becomes free of the microrotation and their effects.

III. PERFORMANCE CHARACTERISTICS

The static and dynamic performance characteristics of both rigid and flexible bearings are calculated from the computed nodal pressures. The expressions for static and dynamic performance characteristics are derived as the functions of the nodal pressures, and the shape functions. The static characteristics include the load capacity, attitude angle, power loss and end leakage. The dynamic performance characteristics are studied in terms of fluid film stiffness and damping

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coefficients, threshold speed and damped frequency of whirl. The expressions for the static and dynamic performance characteristics of journal bearings are given in the following sections. Using the linearized equations of disturbed motion of the journal centre and Routh 7s criteria, the expressions for the critical mass and the threshold speed are derived.

The angle between the line of the centers and the load line is known as attitude angle and is given by

$$\phi = \tan^{-1}\frac{W_Y}{W_X} \tag{6}$$

Attitude angle is an important parameter. Although it is considered as a static characteristic, it has implications on the stability of the bearing system.

For bearing designed to carry vertical loads only (for example the gravity load) the relationship between eccentricity ratio ε and journal attitude angle, Φ, may be determined by investigating different values of Φ for a given value of ε untilthe value of F_h is found to be zero. For example, once we choose an arbitrary value of ε, Φ then corresponding filmthickness can be obtained (since ε, Φ determines the position of the shaft with respect to the bearing bore).

IV. CONCLUSION

1. For micropolar lubricants, the value of threshold speed obtained are less than that obtained for Newtonian lubricant at any value of ϵ.

2. For all the three rigid bearing configuration, the damped frequency of whirlincreases with increase in λc_r for any value of ϵ when mass transfer of additives is present.

3. The stability of rigid circular and non circular (two lobe and three lobe) bearings obtained for micropolar lubricants is less than that obtained for Newtonian lubricant at any value of ϵ.

4. At any value of λc_r, and ϵ, the threshold speed decreases but damped frequency of whirl increases when mass transfer rate of additives increases. For any ϵ, the peak pressure developed in the fluid film of circular bearing and each lobe of non-circular bearings, the load capacity, attitude angle, endand frictional force decrease with increase in F and significant reduction occurs when v is large. Reduction in the end leakage with increase in v is a favourable design condition.

5. For micropolar lubricant, at any $\overline{\psi}$, the values of load carrying capacity, end leakage, attitude angle and frictional force are more than those obtained with Newtonian lubricants for all the three bearing configurations studied.

6. For all the bearings studied, appreciable changes in all the dynamic coefficients are obtained with increase in ($\overline{\psi}$especially when v is high for both Newtonian and micropolar lubricants.

7. For both Newtonian and micropolar lubricants at small ϵ, the threshold speed increase with increase in for all the three bearing configurations. For large ϵ and if is small the circular bearing system is always stable whereas the three lobe bearing system remains stable when the value of $\overline{\psi}$exceeds 0.048 in the case of Newtonian lubricant and 0.049 in the case of micropolar lubricant (λc_r, = 0.4, K, = 0.4).

8. For micropolar lubricant the value of threshold speed obtained is less thanthat obtained for Newtonian lubricant at any ϵ and $\overline{\psi}$for all the bearings.

9. For all the bearing configurations and lubricants studied, the damped frequency of whirl decreases with increase in$\overline{\psi}$ at small ϵ. For large ϵ, at small values of v in the case of circular bearing and at any value of greater than 0.049 in the case of three lobe bearing, the journal when it is disturbed, returns to its equilibrium position without whirl.

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